Journal of Heat and Mass Transfer Research 8 (2021) 163- 171

Semnan University

Journal of Heat and Mass Transfer Research

Journal homepage: http://jhmtr.semnan.ac.ir

# Effect of Fin Geometry on the Performance of Tubular-Fin Heat Exchangers: a Computational Fluid Dynamics Study

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**PAPER INFO**

***Paper history:***
Received: 2020-10-07
Revised: 2021-06-20
Accepted: 2021-06-21

***Keywords:***
Heat exchanger;
Fin geometry;
Heat transfer coefficient;
Pressure drop.

**ABSTRACT**

Tubular-Fin heat exchangers are a type of compact heat exchangers with prominent features like high levels of exchanged heat and less space occupancy. These heat exchangers are commonly used for exchanging heat between gas and liquid. In this study, for a tubular-fin heat exchanger, the heat transfer and pressure drop for circular and serrated fins with the triangular arrangement are numerically calculated. A three-dimensional numerical study with the Reynolds mean-averaged Navier-Stokes (k-ε) model for turbulence is conducted. In the Reynolds range of 6000 to 25000, the performance of four types of fin geometry (serrated, semi-serrated, circular and semi-circular) are compared. The results show that the circular fin has the highest heat transfer rate, while the serrated fin has the highest reduction in the gas temperature. It is also found that the semi-circular fin has the highest thermal enhancement factor and the semi-serrated fin has the highest heat transfer coefficient. The results of the present study can be beneficial in the selection of optimal fins in a heat exchanger from both practical and economic aspects.

DOI; 10.22075/JHMTR.2021.21528.1309



## 1. Introduction

One of the concerns of the heat exchanger industry is the design and selection of suitable heat exchangers from the economical and performance aspects. Experimental investigation of heat exchangers is desirable but not always cost-effective. Nevertheless, many experimental studies have been performed to determine the heat transfer, pressure drop and friction factor of a heat exchanger. To compensate for the low heat transfer coefficient of the gas fluids and increase the exchanged heat in the gas-liquid heat exchangers, the use of heat exchangers with tubular-fin is common. Shepherd et al. studied the plate heat exchanger with circular tubes[1]. Rocha et al. studied tubular-fin heat exchangers with circular and elliptical tubes and experimentally determined the heat transfer coefficients[2]. Kundu and Das calculated the optimal dimensions for a tubular-fin heat exchanger with square and triangular arrangements[3]. Abu Mahdi et al. investigated the effect of different geometrical parameters of round tube and plate finned heat exchangers on the heat transfer rate[4]. Chi and Wang have experimentally investigated the effect of the number of rows of pipes, fins spacing and also the diameter of the pipes on the heat transfer and pressure drop of a flat plate heat exchanger[5]. Tutar et al. have studied various heat exchangers, including tubular-fin heat exchangers by numerical methods[6]. Eric et al. investigated a single-row tubular-fin heat exchanger for different geometrical parameters by using numerical methods. They have also numerically investigated the effect of the distance between the fins, the distance between the tubes, the length of the fins and the thickness of the tubes on the heat transfer and pressure drop of the heat exchanger[7]. Engelman and Haught using the finite element method investigated velocity and temperature fields for tubular-fin heat exchangers[8]. Wong et al. conducted a series of

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experiments to investigate the relationship between heat transfer and pressure drop in geometrical parameters of 15 types of tubular-fin heat exchangers to determine the effect of fin spacing and the number of tubes on the friction coefficient[9]. Kayansayan investigated 10 different forms of tubular-fin heat exchangers in Reynolds of 100 to 30,000 to determine the effect of fins geometry on the heat transfer[10]. Lemouedda et al. presented the results of numerical calculations of spiral-finned tubes and compared them with the results of bare tubes [11]. Also in numerical methods, some researchers reported their results of studies on tubular-fin heat exchangers [12-21]. They studied different heat exchangers with various types of fin geometries and reported the changes in some key effective parameters. Min et al. investigated that for the plain finned tube heat exchanger, the heat transfer coefficient, on the whole, tends to decrease with increasing the serial number of tube rows, although the decreasing rate is quite small [22]. Al-Jaberi et al. performed experiments on circular fins with slanted blades attached to the copper tube surface to reduce the thermal boundary layer. They examined the effects of the number of slanted blades and the Reynolds number on the heat transfer characteristics. Their study shows that the Nusselt number for slanted fins is 20% to 27.5% higher than that of circular fins. However, in that study, the pressure drop in the tube bundle is not examined [23]. Recently, some researchers assumed a constant heat-transfer coefficient across the tube bundle as the heat transfer boundary condition [24-27]. Yan et al. recently published an article about an experimental study on the air-side heat transfer performance of the perforated fin-tube heat exchangers under the frosting conditions[28]. Genić et al. investigated the pressure drop of plate finned tube heat exchangers [29]. They reported a new correlation that is comprehensive and statistically superior to any other pressure drop correlation. In the optimization discussion, various methods have been implemented, including genetic algorithms as well as methods of changing geometries to improve thermodynamic conditions by reducing the entropy [30, 31].

Other effective parameters for studying tube bundles and heat exchangers such as a pressure drop coefficient and heat transfer coefficient had long been considered by researchers [32-36]. In the past years, experimental studies have also been carried out by researchers on heat transfer and pressure drop coefficients on tubular heat exchangers with serrated fins in different geometries. [37-40]. In the area of numerical analysis for serrated fins, one can also refer to the studies of Mcilwain[41]. They reported temperature contours and stream lines on the tube bundle. Muhič et al. investigated the effect of the needle and circular fins on the Nusselt number, in which needle fins had a better result[42]. Nada and Said investigated free convection heat transfer in a horizontal annulus with and without fins by numerical method for annular and longitudinal rectangular fins. They calculated the effects of the Rayleigh number, annulus depth, fins numbers and dimensions on the heat flow pattern. Over the past decades, computational fluid dynamics techniques have been rapidly developed to simulate the flow, especially the numerical simulation and solution of turbulent flow[43]. A review of all the methods applied today in computational fluid dynamics was given by Hosain and Fdhil[44]. Qiu et al. theoretically studied the heat transfer characteristics of a plain fin in the finned-tube evaporator assisted by solar energy[45]. One of the best methods to simulate turbulent flow is the Jones and Launder model (k-ε model) introduced in 1972[46]. Among the various types of tubular-fin heat exchangers, circular and serrated fins are more widely used than others due to their simplicity in construction and cheaper process due to less space occupancy and surface heat exchange.

Despite extensive studies on the performance of circular and serrated fins, the simultaneous performance of heat transfer and pressure drop of these fins has not been investigated so far. The present study investigates the effects of four types of fins including serrated, semi-serrated, circular, and semi-circular on the performance of a tubular-fin heat exchanger. Heat transfer rate, pressure drop, temperature change, Nusselt number, thermal enhancement factor (TEF) and heat transfer coefficient at different Reynolds numbers are calculated and compared for each type of fins.

## 2. Modeling details

In the present study, k-ε turbulence model is considered in COMSOL Multiphysics software. The k-ε model is popular for industrial applications due to its good convergence rate and relatively low memory requirements and performs well for most of the flow problems around complex geometries. It can deliver accurate enough result with good convergence rate for most of applications.

Heat transfer and fluid flow equations are then coupled together by the non-isothermal flow multiphysics. In order to verify mesh independence, five different meshes were created and it is found that a mesh of total 455367elements is fine enough to resolve important physical phenomena accurately in a reasonable amount of time. Figure 1 shows the sample mesh and mesh indepency structure used in the present study.

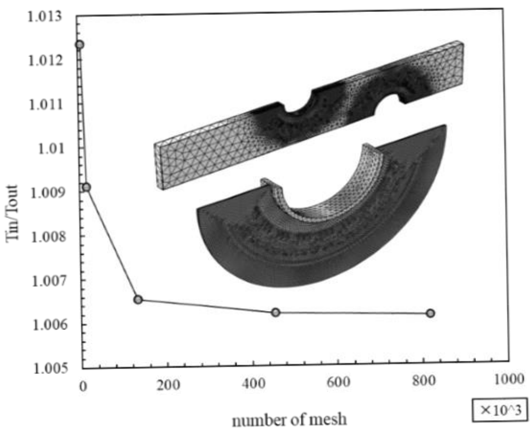


**Figure 1.** Sample of Mesh that used in the numerical modeling and mesh independence

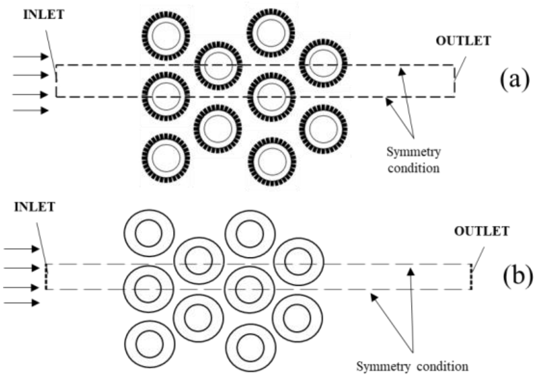


**Figure 2.** Side view of the studied tubular-fin heat exchanger with (a) serrated fins (b) circular fins

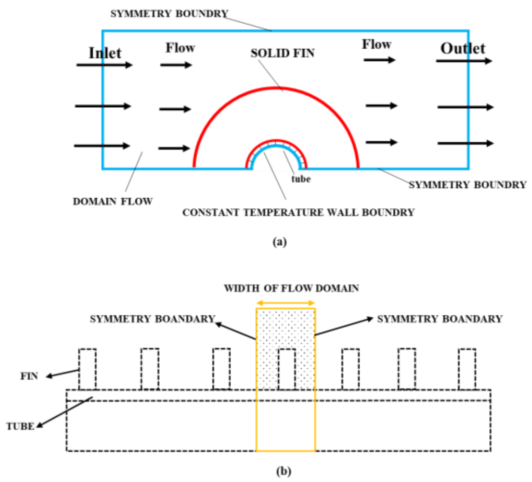


**Figure 3.** Boundary conditions for the studied geometry from (a) side view and (b) front view.

Figure 2 illustrates a schematic of the arrangement of a tube bundle with circular and serrated fins. In the current study, the gas passing through the pipe is nitrogen with inlet temperature at 498 K. The fins are positioned on pipes containing a liquid such as water, with a constant surface temperature of 298 K. Figure 3 shows a schematic of the boundary conditions governing the studied heat exchanger. Figure 4 shows the dimensions and sizes of the serrated, semi-serrated, circular and semi-circular fins.

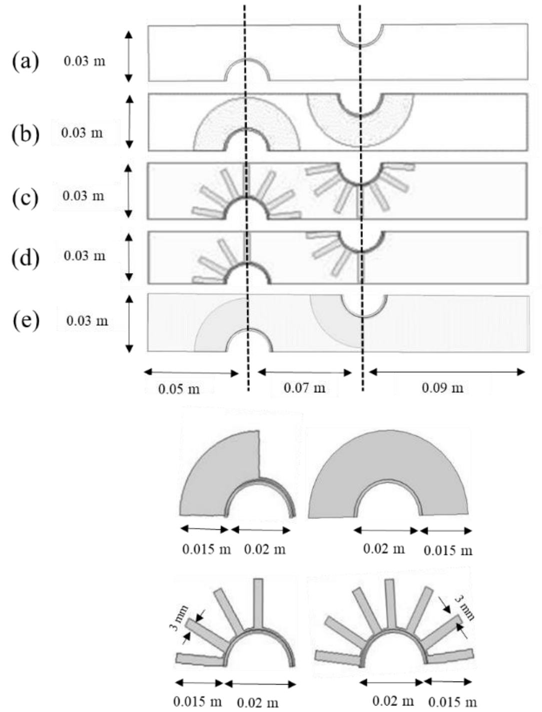


**Figure 4.** The geometrical dimensions of tubular-fin heat exchangers (a) without fin (b) circular fin (c) serrated fin (d) semi-serrated fin and (e) semi-circular fin.

## 3. Governing equations

The Navier-Stokes equations provide a mathematical model for the hydrodynamic and thermal description of the fluid. Because of the complexity of the full form of Navier-Stokes, three-dimensional analytical solutions are not feasible, so computer-aided numerical methods are the best option for solving these equations. The rapid advances in computer technology in recent decades have led to the widespread use of computational fluid dynamics in the numerical solution of fluid flow problems. Since all solvers of the Navier-Stokes equations require a lot of processing time and memory, so some simplifications in solving these equations are needed to reduce the computational costs. The simplified Navier-Stokes equations are described as the following.

### *3.1. Continuity equation*

For compressible fluids, the continuity equation is as follows[47]:

$$\frac{\partial \rho}{\partial t} + \vec{\nabla}.(\rho \vec{V}) = 0 \tag{1}$$

where $\vec{V}$, ρ and t represent vector of velocity, density and time respectively. For incompressible flow, continuity equation reduces to:

$$(\vec{\nabla}.\vec{V}) = 0 \tag{2}$$

### 3.2. Momentum equation

Due to the incompressible flow and assuming a constant viscosity coefficient, the momentum equations are written as follows[47].

$$\rho\frac{DV}{Dt} = \rho F - \nabla P + \mu\nabla^2 V \tag{3}$$

in which $\vec{V}$, $P$ , $F$ and $\mu$ respectively represent vector of velocity, pressure, volumetric forces and viscosity. $\frac{D}{Dt}$ represents a material derivative and is defined as $\left(\frac{D\varphi}{Dt} = \frac{\partial\varphi}{\partial t} + \vec{V}\cdot\nabla\varphi\right)$. Since the current flow in this study is turbulent, it is necessary to study the form of the equations in the turbulent flow. The continuity equation for turbulent flow for compressible flow is as follows:

$$\frac{\partial}{\partial x_i}(\overline{\rho\, u_\iota}) + \frac{\partial}{\partial x_i}(\overline{\rho'\, u_\iota'}) = 0 \tag{4}$$

here $\overline{u_\iota}$ and $\overline{u_\iota'}$ are average speed and the oscillatory component of the velocity respectively. Since $\rho' = 0$ for the incompressible flow, Equation 4 will be simplified to:

$$\frac{\partial \bar{u}_i}{\partial x_i} = 0 \tag{5}$$

The momentum equations for turbulent flow are as follows:

$$\rho\left[\frac{\partial \bar{u}_i}{\partial t} + \bar{u}_j\frac{\partial \bar{u}_i}{\partial x_j}\right] = \\ \bar{B}_\iota - \frac{\partial \bar{p}}{\partial x_i} + \frac{\partial}{\partial x_j}\left[\mu\frac{\partial \bar{u}_i}{\partial t} - \rho\overline{u_i' u_j'}\right] \tag{6}$$

which $\bar{B_\iota}$ is the volume force in the direction $i = (x.y.z)$ and $\rho\overline{u_i' u_j'}$ is called the turbulent stress or Reynolds stress, which is modelled with k-ε turbulence model.

The differential equation that describes the heat transfer in gas domain and solves for the temperature field is: $\rho C_p u.\nabla T + \nabla.q = Q$ where, $q$ [W/m$^2$] is the heat flux vector and is expressed as: $q = -k\nabla T$. $T$ [K] is the temperature, $C_p$ [J/(kg.K)] is the heat capacity at constant pressure, $k$ [W/(m.K)] is the thermal conductivity, and $Q$ [W/m$^2$] is the heat source or sink.

The heat transfer in the fin and tube domain is dominated by conduction and hence governs by Fourier's Law of heat conduction. It is to note that due to small temperature difference and lower emissivity, heat transfer through radiation is neglected. The heat transfer equation in the fin and tube domains also account for additional heat source or sink term and is expresses in the following form of energy conservation: $\rho C_p u.\nabla T + \nabla.q = Q$

The first term represents contribution of translational motion to the heat transfer in solids, which is negligible in this case, hence neglected. Second term is the conductive heat flux vector, which is expressed as: $q = -k\nabla T$ where, $k_s$[W/(m.K)] is the thermal conductivity of fin and tube material.

## 4. Results and discussion

### 4.1. Validation

For validation, we compared our numerical results with the results of Ref [42] done by Stojkov et al. For this purpose, the gas outlet temperature at three different input speeds (Uin) was computed. It can be concluded from Figure 5 that our results are in good agreement with Ref.[42]. However, there is a slight difference at the low inlet velocity which can be attributed to the different methods of turbulence modeling. We have employed the k-ε turbulence model and the k-ω turbulence model was utilized in Ref. [42]. Figure 6 also shows the contour of temperature for the present study and Ref.[42]. It clearly shows that the distribution of temperature on the fin and also passing gas is in good agreement with Ref.[42]. The maximum temperature difference is less than 1K.

### 4.2. Heat transfer and temperature distribution

We study four types of fins, serrated, semi-serrated, circular and semi-circular fins from the heat transfer point of view at different Reynolds numbers. At first, the amount of heat transfer in each type of fins is investigated. The following relationships are used for this purpose:

$$Q = hA\Delta T_{LMTD} \tag{7}$$

$$\Delta T_{LMTD} = \frac{(T_s - T_{out}) - (T_s - T_{in})}{\log\frac{(T_s - T_{out})}{(T_s - T_{in})}} \tag{8}$$

where $\Delta T_{LMTD}$ is the logarithmic temperature difference, $T_s$ is the pipe surface temperature, $T_{in}$ is the gas inlet temperature and $T_{out}$ is the gas outlet temperature.

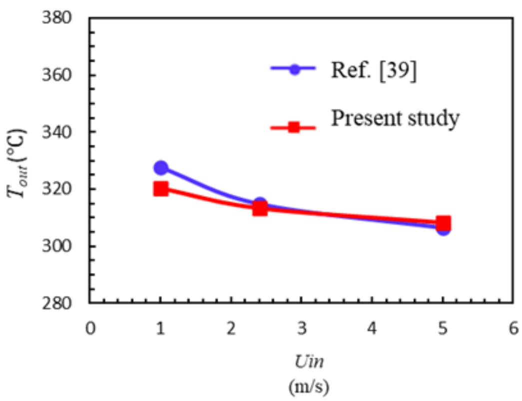


**Figure 5.** Comparison and validation of the results of the present study with reference [39] by calculating the outlet temperature for different inlet velocities.

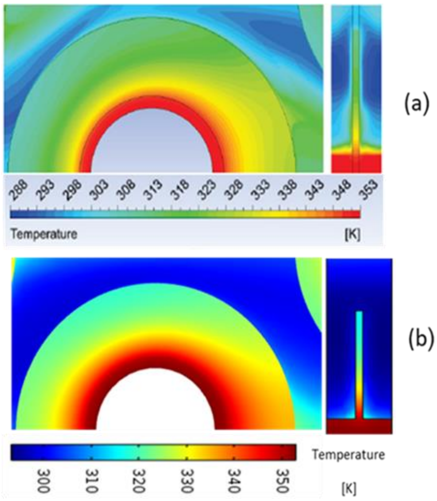


**Figure 6.** Comparison of the temperature contour results of reference [39] (a) and present study (b).

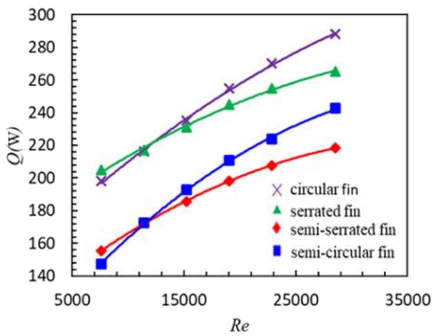


**Figure7.** Heat transfer rate at different Reynolds numbers for serrated, semi-serrated, circular and semi-circular fins.

Figure 7 shows the variations of the heat transfer rate due to the Reynolds number increase. Based on this, it is calculated that the most heat transfer is for the circular fins due to the most amount of surface contact with the fluid. In fact, according to the heat transfer equation (Relation 6), higher contact of the fins surface with the fluid caused the more amount of heat exchange, and consequently the better heat transfer. In Figure 7 the circular fin has the highest heat transfer rate due to the highest contact surface area.

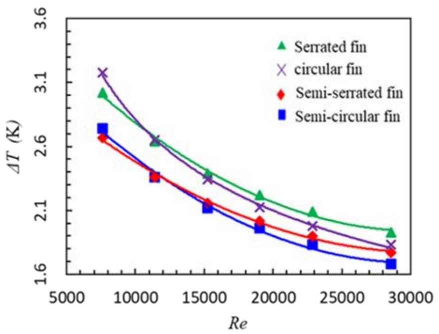


**Figure 8.** Inlet and outlet temperature difference with increasing Reynolds number for serrated, semi-serrated, circular and semi-circular fins.

Figure 8 shows the difference between the inlet and outlet temperatures of the passing gas. It is deduced from Figure 8 that the serrated fin reduces more temperature due to increased local turbulence. Turbulence improves heat exchange and as a result, the serrated fin has the greatest reduction in the gas temperature.

To have a better understanding of the heat transfer performance, the coefficient of convective heat transfer is calculated:

$$h = \frac{\dot{m} c_P \Delta T}{A \Delta T_{LMTD}} \tag{9}$$

where $\dot{m}$ is the mass flow rate, $c_P$ is the specific heat, $A$ is the area of the fin, $\Delta T$ is the gas inlet and outlet temperature difference, and $\Delta T_{LMTD}$ is the logarithmic temperature difference.

The higher surface contact caused the lower heat transfer coefficient. For this reason, the serrated and the semi-serrated fins have a higher heat transfer coefficient than the other two fins. Figures 10 show the variations of the Nusselt number $\left(Nu = \frac{hD}{k}\right)$ versus the Reynolds numbers. It shows that the Nusselt number to the Prandtl number of semi-serrated fins is greater than other fins because of its minimum contact surface area. The Nusselt number is calculated using the heat transfer coefficient, and since this coefficient is inversely related to the contact surface, the Nusselt number inversely varies by the surface area which is the reason behind the fact that Figure 9 and 10 have the same trends.

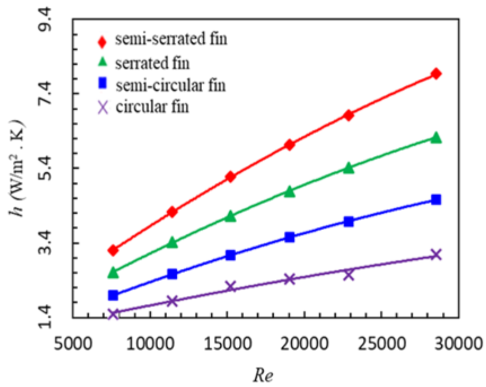


**Figure 9.** Variations of heat transfer coefficient (h) by the Reynolds number for serrated, semi-serrated, circular and semi-circular fins.

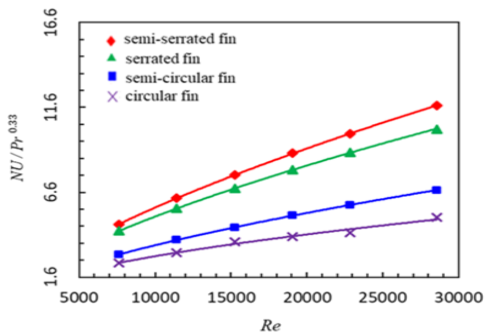


**Figure 10.** Nusselt number versus Reynolds number for serrated, semi-serrated circular, and semi-circular fins.

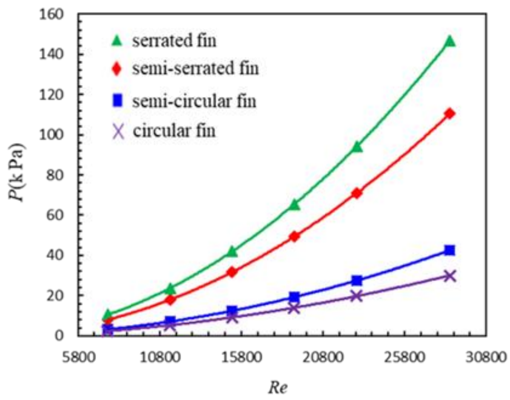


**Figure 11.** Pressure drop in different Reynolds for serrated, semi-serrated, circular, and semi-circular fins.

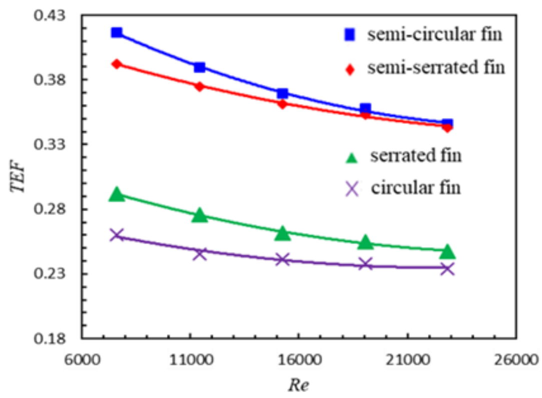


**Figure12.** TEF variations with Reynolds number for serrated, semi-serrated, circular, and semi-circular fins

## 4.3. Pressure drop

Although increasing the fin surface area increases the heat transfer, it increases the pressure drop in a heat exchanger too. So calculating the pressure drop is a crucial factor in the design of a heat exchanger. Figure 11 shows the gas pressure difference between the inlet and the outlet versus Reynolds number. It is predicted that serrated fins cause more pressure drop than circular fins due to their specific geometry and local vorticities. From Figures 10 and 11, It is evident that the semi-serrated fin has the highest h compared to all considered fins and lower pressure drop compared to the serrated fins. Therefore, it is more economical due to less material usage.

## 4.4. Thermal enhancement factor

In this section, we report the calculated thermal enhancement factor (TEF) for each case which is the ratio of the Nusselt number to the friction factor. This coefficient can be used to calculate the ratio of the heat transfer alongside pressure drop simultaneously. The relation of the TEF coefficient is as follows:

$$TEF = \frac{(Nu_E \,/\, Nu)}{(f_E/f)^{\frac{1}{3}}} \tag{10}$$

where $Nu_E$ and $f_E$ are the Nusselt number and friction coefficient for the heat exchanger with fin respectively and $Nu$ and $f$ are the Nusselt number and friction factor for the heat exchanger without any fin respectively. The friction coefficient could be computed as follows:

$$f = \frac{\Delta P}{\frac{L}{D} \times \frac{U_{in}^2}{2g}} \tag{11}$$

where $L$ is the longitudinal length of the domain, $D$ is the hydraulic diameter of the domain, $\Delta P$ is the pressure difference at the inlet and outlet of the flow and $U_{in}$ is the inlet velocity. This coefficient compares the Nusselt number and the frictional drop coefficient of the fin relative to the base state where there are no fins.

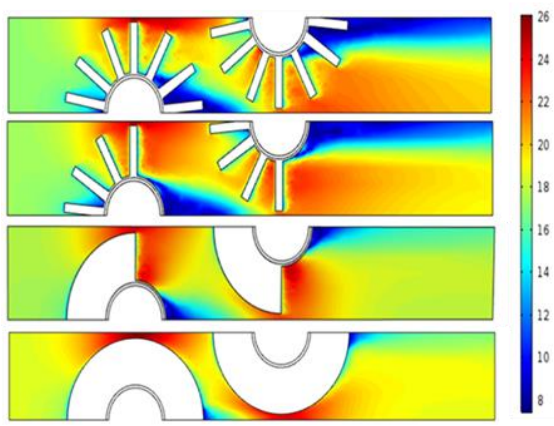


**Figure13.** Contour of gas velocity [m/s] in Reynolds almost 28000 for serrated, semi-serrated, semi-circular, and circular fins.

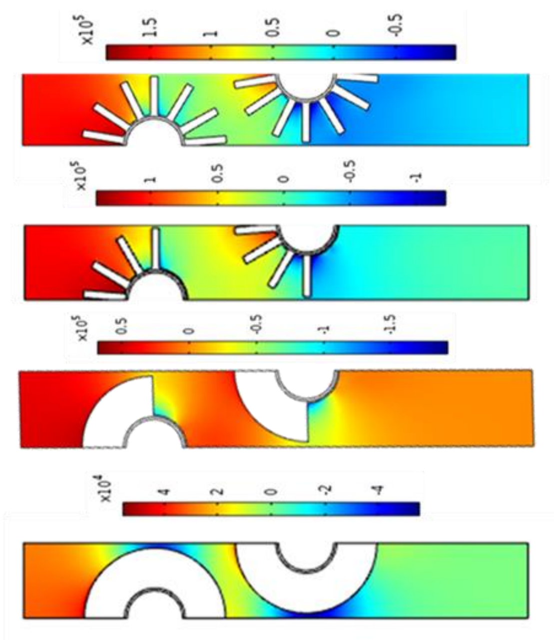

**Figure14:** Pressure contour [Pa] in Reynolds almost 28000 for serrated, semi-serrated, semi-circular and circular fins.

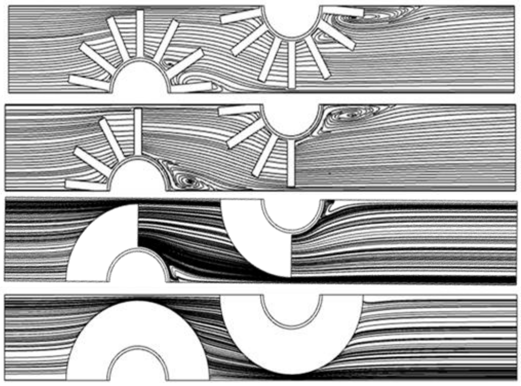

**Figure 15.** Streamlines for serrated, semi-serrated, semi-circular, and circular fins (Re = 28000).

The variation of TEF versus the Reynolds number for all fin arrangements depicted in Figure 12 shows that TEF has the highest value for semi-circular fin followed by semi-serrated. This fact means that both the pressure drop and the Nusselt number must be considered together simultaneously to choose the best fin. The velocity contour for each type of fin is illustrated in Figure 13. As expected, the gas velocity reaches its minimum value behind the fins.

Figure 14 shows the pressure contour of all four types of fins. As expected, the pressure drop for the serrated fin and semi-serrated fins are higher compared to the circular and semi-circular cases. Serrated fins create more pressure drop in addition to causing more turbulence. Figure 15 shows the streamlines inside the section including fins, in which the formation of vortices around different fins can be clearly seen.

## Conclusion

Multiphysics 3D numerical modeling of a finned tube heat exchanger is performed. The model incorporates coupled heat transfer and turbulent flow to analyze the physical phenomenon in the heat exchanger operation.

Four different types of fins were investigated and analyzed, and their strengths and weaknesses were expressed. The results showed that, it is possible to increase the efficiency of heat exchangers by changing of fin type and its geometry. Also discovered that circular fin has highest amount of material and lowest heat transfer coefficient. Serrated fins have highest pressure-drop, so they aren't good choice for heat exchangers. Semi-circular fins have a less heat transfer coefficient than semi-serrated fins but they have lower pressure drop than serrated and semi-serrated fins. In other hand by considering thermal enhancement factor (TEF) parameter which relates the Nusselt number and the friction coefficient, semi-circular fin has highest value of TEF among other fin types. The maximum value of TEF for the semi-circular fin is 0.43, which is reduced by increasing Reynolds number to 0.35.

Finally, it is found that the semi-circular fin is the most efficient and most suitable option compared to the other three types. This type of fin also consumes less material for production, thus reducing production costs. In addition, due to its simple geometry, it has low production costs and its waste is very low.

## References

[1] Shepherd DG. Performance of one-row tube coils with thin-plate fins, low velocity forced convection. Heating, Pip Air Cond 1956;28:137–44.

[2] Rocha LAO, Saboya FEM, Vargas JVC. A comparative study of elliptical and circular sections in one- and two-row tubes and plate fin heat exchangers. Int J Heat Fluid Flow 1997. https://doi.org/10.1016/S0142-727X(96)00063-X.

[3] Kundu B, Das PK. Optimum dimensions of plate fins for fin-tube heat exchangers. Int J Heat Fluid Flow 1997. https://doi.org/10.1016/S0142-727X(97)80011-2.

[4] Abu Madi M, Johns RA, Heikal MR. Performance characteristics correlation for round tube and plate finned heat exchangers: Equations relatives aux performances d'échangeurs de chaleur constitués de tubes ronds et de plaques à ailettes. Int J Refrig 1998. https://doi.org/ http://dx.doi.org/10.1016/S0140-7007(98)00031-0.

[5] Wang CC, Chi KY, Chang CJ. Heat transfer and friction characteristics of plain fin-and-tube heat exchangers, part II: Correlation. Int J Heat Mass Transf 2000. https://doi.org/10.1016/S0017-9310(99)00333-6.

[6] Tutar, Mustafa; Akkoca A. a Computational Study of Effects of Different Geometrical Parameters on Heat Transfer and Fluid Flow in a Wavy and Plain Fin. 6th Bienn Conf Eng Syst Des Anal Instanbul, Turkey 2002:2–7.

[7] Erek A, Özerdem B, Bilir L, Ilken Z. Effect of geometrical parameters on heat transfer and pressure drop characteristics of plate fin and tube heat exchangers. Appl Therm Eng 2005. https://doi.org/10.1016/j.applthermaleng.2004.12.019.

[8] A. Haught MSE. Numerical and experimental Simulation for airflow and heating in a tube fin heat exchanger. Heat Transf Gas Turbines Three-Dimensional Flows 2006:107–13.

[9] Wang CC, Chang YJ, Hsieh YC, Lin YT. Sensible heat and friction characteristics of plate fin-and-tube heat exchangers having plane fins. Int J Refrig 1996. https://doi.org/10.1016/0140-7007(96)00021-7.

[10] Kayansayan N. Heat transfer characterization of plate fin-tube heat exchangers. Int J Refrig 1994;17:49–57. https://doi.org/10.1016/0140-7007(94)90086-8.

[11] Lemouedda A, Schmid A, Franz E, Breuer M, Delgado A. Numerical investigations for the optimization of serrated finned-tube heat exchangers. Appl Therm Eng 2011. https://doi.org/10.1016/j.applthermaleng.2010.12.03.

[12] Tang LH, Zeng M, Wang QW. Experimental and numerical investigation on air-side performance of fin-and-tube heat exchangers with various fin patterns. Exp Therm Fluid Sci 2009. https://doi.org/10.1016/j.expthermflusci.2009.02.008.

[13] Bilir L, Ilken Z, Erek A. Numerical optimization of a fin-tube gas to liquid heat exchanger. Int J Therm Sci 2012. https://doi.org/10.1016/j.ijthermalsci.2011.09.010.

[14] Pongsoi P, Promoppatum P, Pikulkajorn S, Wongwises S. Effect of fin pitches on the air-side performance of L-footed spiral fin-and-tube heat exchangers. Int J Heat Mass Transf 2013. https://doi.org/10.1016/j.ijheatmasstransfer.2012.11.071.

[15] Zeeshan M, Nath S, Bhanja D. Numerical study to predict optimal configuration of fin and tube compact heat exchanger with various tube shapes and spatial arrangements. Energy Convers Manag 2017. https://doi.org/10.1016/j.enconman.2017.06.011.

[16] Senapati JR, Dash SK, Roy S. Numerical investigation of natural convection heat transfer over annular finned horizontal cylinder. Int J Heat Mass Transf 2016. https://doi.org/10.1016/j.ijheatmasstransfer.2016.01.024.

[17] Chen HT, Chiu YJ, Liu CS, Chang JR. Numerical and experimental study of natural convection heat transfer characteristics for vertical annular finned tube heat exchanger. Int J Heat Mass Transf 2017. https://doi.org/10.1016/j.ijheatmasstransfer.2017.01.122.

[18] Zeeshan M, Nath S, Bhanja D. Numerical analysis to predict the optimum configuration of fin and tube heat exchanger with rectangular vortex generators for enhanced thermohydraulic performance. Heat Mass Transf 2020:1–11.

[19] Gupta A, Roy A, Gupta S, Gupta M. Numerical investigation towards implementation of punched winglet as vortex generator for performance improvement of a fin-and-tube heat exchanger. Int J Heat Mass Transf 2020. https://doi.org/10.1016/j.ijheatmasstransfer.2019.119171.

[20] Li D, Yang C, Yang H. Experimental and numerical study of a tube-fin cool storage heat exchanger. Appl Therm Eng 2019. https://doi.org/10.1016/j.applthermaleng.2018.12.024.

[21] Uosofvand H, Abbasian Arani AA, Arefmanesh A. Effect of baffle oientation on shell tube heat exchanger performance. J Heat Mass Transf Res 2017;4:83–90.

[22] Gu L, Min J, Wu X, Yang L. Airside heat transfer and pressure loss characteristics of bare and finned tube heat exchangers used for aero engine cooling considering variable air properties. Int J Heat Mass Transf 2017. https://doi.org/10.1016/j.ijheatmasstransfer.2017.01.047.

[23] Ali Shakir, A. J., Majid, H. M., & Bassam AS. Circular fins with slanted blades attached on the copper pipe: Uniform heat flux and isothermal processes. Int J Mech Eng Technol 2014.

[24] Kundu B, Das PK. Performance analysis and optimization of annular fin with a step change in thickness. J Heat Transfer 2001. https://doi.org/10.1115/1.1351165.

[25] Deka A, Datta D. Geometric size optimization of annular step fin using multi-objective genetic algorithm. J Therm Sci Eng Appl 2017. https://doi.org/10.1115/1.4035838.

[26] Kundu B, Das PK. Performance analysis and optimization of elliptic fins circumscribing a circular tube. Int J Heat Mass Transf 2007. https://doi.org/10.1016/j.ijheatmasstransfer.2006.06.043.

[27] Nemati H, Samivand S. Performance optimization of annular elliptical fin based on thermo-geometric parameters. Alexandria Eng J 2015. https://doi.org/10.1016/j.aej.2015.09.016.

[28] Liu X, Yu J, Yan G. An experimental study on the air side heat transfer performance of the perforated fin-tube heat exchangers under the frosting conditions. Appl Therm Eng 2020;166:114634.

[29] Marković S, Jaćimović B, Genić S, Mihailović M, Milovančević U, Otović M. Air side pressure drop in plate finned tube heat exchangers. Int J Refrig 2019;99:24–9. https://doi.org/10.1016/j.ijrefrig.2018.11.038.

[30] Sadeghianjahromi A, Kheradmand S, Nemati H. Developed correlations for heat transfer and flow friction characteristics of louvered finned tube heat exchangers. Int J Therm Sci 2018;129:135–44. https://doi.org/10.1016/j.ijthermalsci.2018.03.002.

[31] Nemati H, Moghimi MA, Sapin P, Markides CN. Shape optimisation of air-cooled finned-tube heat exchangers. Int J Therm Sci 2020. https://doi.org/10.1016/j.ijthermalsci.2019.106233.

[32] Briggs, D. E., & Young EH. Convection heat transfer and pressure drop of air flowing across triangular pitch banks of finned tubes. 5th AICHE/ASME Natl Heat Transf Conf Houst 1962.

[33] Hewitt, G. F., Shires, G. L., & Bott TR. Process heat transfer. vol. 113. Boca Raton: CRC press.; 1994.

[34] Rabas TJ, Eckels PW, Sabatino RA. The effect of fin density on the heat transfer and pressure drop performance of low-finned tube banks. Chem Eng Commun 1981. https://doi.org/10.1080/00986448108910930.

[35] Gong B, Wang LB, Lin ZM. Heat transfer characteristics of a circular tube bank fin heat exchanger with fins punched curve rectangular vortex generators in the wake regions of the tubes. Appl Therm Eng 2015. https://doi.org/10.1016/j.applthermaleng.2014.09.043.

[36] Robinson, K. K., & Briggs DE. Pressure drop of air flowing across triangular pitch banks of finned tubes. Chem Eng Prog Symp Ser 1966;62:177–84.

[37] Næss E. Experimental investigation of heat transfer and pressure drop in serrated-fin tube bundles with staggered tube layouts. Appl Therm Eng 2010. https://doi.org/10.1016/j.applthermaleng.2010.02.019.

[38] Weirman C. CORRELATIONS EASE THE SELECTION OF FINNED TUBES. Oil Gas J 1976.

[39] Hashizume K, Matsue T, Koyama T. Fin efficiency of serrated fins: Part 1. Analysis of theoretical fin efficiency and experimental results. Heat Transf Res 1999. https://doi.org/10.1002/(sici)1523-149x(1999)28:6<528::aid-htj9>3.0.co;2-b.

[40] Reid DR, Taborek J. Selection criteria for plain and segmented finned tubes for heat recovery systems. J Eng Gas Turbines Power 1994. https://doi.org/10.1115/1.2906835.

[41] Mcilwain SR. A comparison of heat transfer around a single serrated finned tube and a plain finned tube. Proc IJRRAS 2010;2.
[42] Development of a new type of finned heat exchanger. Teh Vjesn - Tech Gaz 2017. https://doi.org/10.17559/tv-20171011071711.
[43] Nada SA, Said MA. Effects of fins geometries, arrangements, dimensions and numbers on natural convection heat transfer characteristics in finned-horizontal annulus. Int J Therm Sci 2019. https://doi.org/10.1016/j.ijthermalsci.2018.11.026.
[44] Hosain ML, Fdhila RB. Literature Review of Accelerated CFD Simulation Methods towards Online Application. Energy Procedia, 2015. https://doi.org/10.1016/j.egypro.2015.07.714.
[45] Qiu G, Sun J, Ma Y, Qu J, Cai W. Theoretical study on the heat transfer characteristics of a plain fin in the finned-tube evaporator assisted by solar energy. Int J Heat Mass Transf 2018. https://doi.org/10.1016/j.ijheatmasstransfer.2018.07.106.
[46] Jones WP, Launder BE. The prediction of laminarization with a two-equation model of turbulence. Int J Heat Mass Transf 1972. https://doi.org/10.1016/0017-9310(72)90076-2.
[47] White FM. Viscous Fluid Flow (MCGRAW HILL SERIES IN MECHANICAL ENGINEERING) 3rd Edition. 3rd ed. New York: McGraw-Hill; 2005.