# A NUMERICAL STUDY FOR SOLID AND SERRATED ANNULAR FINNED TUBE BUNDLES

by

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*Annular finned tube bundles are commonly used for heat recovery systems. Nowadays, heat recovery systems are important in the energy economy. Cross-flow heat exchangers, one of the heat exchanger types are suitable for waste heat recovery systems. Annular fins are utilized in cross-flow heat exchangers for a long time. In this study, two types of annular fin geometry, namely solid and serrated fins, were studied numerically in the cross-flow heat exchangers. All numerical analyses are performed in ANSYA-FLUENT program and the fin geometries are designed in 3-D geometry. Numerical results obtained for two different geometry fins are validated separately with the literature. It is seen that analysis results are found to be compatible with the literature. In numerical analyses, five different Reynolds Numbers and six different geometric parameters are studied. Effects of these parameters are investigated to determine the flow and thermal performance. According to analysis results, the thermal performance of the serrated annular fin geometry is about 8.2% higher than the solid fin geometry, while the flow performance decreases by 7.5%.*

Key words: *annular finned tube bundles, solid fins, serrated fins*

## Introduction

Cross-flow heat exchangers have been used in industrial areas for many years. The development of these heat exchangers is of great importance in terms of the energy economy. Fins play a major role in improving heat transfer in the cross-flow heat exchangers. The annular solid finned tubes have been widely used in former studies, however, annular serrated finned tubes have been used frequently recently.

In the literature, thermal and hydraulic analytical and numerical analyses related to the tube bundles with different fin geometries are studied by many researchers. Lindqvist *et al*. [1] analyzed the thermal and hydraulic performance of a helical coiled finned tube bundle model numerically using the CFD method. The appropriate verification model for the numerical study was made with four experimental studies. In these four experimental studies, two of them are solid finned tubes and the other two consists of serrated finned tubes. As a result of the studies, it has been observed that the designed model reduces thermal efficiency. It is understood that the geometric arrangement in the heat exchanger is not important in reducing the cost. Kumar *et al*. [2] numerically studied the thermal and hydraulic characteristics of segmented, straight, and wavy geometries of various circular and flat blades in the range of Re = 2500-4000. As a result of the studies, they have revealed that more compact

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heat exchangers can be designed under high pump power costs. Anoop *et al*. [3] studied a sodium-air type part fin heat exchanger experimentally and numerically. In the studies, the RNG $k$-$\varepsilon$ ep turbulence model was found to be compatible with the experimental results in a wide Reynolds number range. As a result, Nusselt equations are derived for the segmented finned tube, which is thought to be useful in such heat exchangers. Kiatpachai *et al*. [4] investigated experimentally the thermal and flow performance of the airside in the range of Re = 4000-5000 for a spirally welded segment finned tube heat exchanger. As a result, they found correlations that can be used in industrial applications. Cathal *et al*. [5] studied how a CFD model of a cross-flow finned tube bundle can predict heat transfer and pressure drop performance. For three helical welded blades, they continued their numerical studies in the range of Reynolds number between 5000 and 30000, including solid and fully segmented geometries. Numerical studies have yielded results consistent with empirical correlations. Furthermore, studies performed that the Nusselt number increases about 23% among partially and fully segmented. Weierman *et al*. [6], Kawaguchi *et al*. [7, 8], and Hofmann [9] compared solid and serrated fins in their works. In these four separate studies in the literature, it has been found that the heat transfer coefficient and Euler number of the serrated finned tube are higher than the solid finned tube. In addition, Kawaguchi *et al*. [8] demonstrated the advantages of the heat transfer coefficient in serrated fins for larger fins pitches. Lemouedda *et al.* [10] studied helically wound serrated fins. This study, which is carried out Reynolds number between 600 and 2600, consists of three main parts. In the first part of the study, the performance of the serrated fin compared to the normal solid fin is compared. In the second part of the study, the effect of bending teeth on the fin in serrated fins is examined. Pongsoi *et al*. [11] experimentally studied the flow and thermal performance of the air side of a finned tube heat exchanger at high Reynolds numbers (3500-13000). As a result of the studies, it has been seen that the number of pipe rows does not have a significant effect on the heat transfer and friction characteristics. However, they noticed that the pipe outer diameter had a great influence on the pressure drop. This study is accepted as a reference for the verification of many numerical studies in the literature. Zhou *et al*. [12] studied the gas side of a fin-tube heat exchanger with a Re = 6000-12000. They focused on the number of serrated fin and twisted serrated fin in their work. As a result of the study, they have seen that the twisted serrated fin has a significant effect on the Nusselt number and Euler number. They observed that the twisted serrated fin performed better than the standard fins. Mon [13] studied a circular finned heat exchanger numerically using the Fluent program. In numerical studies, the constant surface temperature is assumed for the fin and tube surfaces. In the studies, the RNG $k$-$\varepsilon$ turbulence model was used. The optimum fin geometry was tried to be found by examining the thermal and flow performance of the heat exchanger in three dimensions and unsteady situations. Eide [14] modeled the fin-tube heat exchanger in three different geometries. In the studies, it was investigated how close the results of CFD to the literature would be. Numerical analyzes were made time-independent using the SST $k$-$\omega$ turbulence model. As a result, CFD studies were found to be in accordance with the literature. Holfeld [15] experimentally investigated the heat transfer and pressure drop for a compact heat recovery unit. Serrated and solid finned tube bundles were used in the experiments. The experiments were verified by comparing with the literature. Many geometric parameters have been determined on the fins. The effects of these parameters on heat transfer and pressure drop were examined and some correlations were obtained. Bošnjakovic *et al*. [16,17] designed a new star-shaped fin geometry for a heat exchanger. As a result of experimental studies, this star-shaped fin gave 39.3% better high results in terms of heat flux compared to normal circular fins. In addition, this fin geometry is 23.8% lighter than normal annular fins. Thus, they obtained a more useful heat exchanger in terms of weight. It is 23.8% lighter. Thus, a more useful heat exchanger in terms of weight has been obtained. Morales-Fuentes *et al*. [18] numerically studied plain, annular, and pin-shaped wings. For numerical analysis, the geometries are modeled in three dimensions, and wall and fin surface temperatures are constant. In this study, the inlet velocity is in the range of 1 to 4 m/s. As a result of the studies, fins with a larger surface compared to the volume provided better heat transfer. Sahel *et al*. [19] studied hydro-thermal characteristics of fin-and-tube heat exchangers by vortex generators. In this study they focused on inclination angle of the oval tube and they obtained optimal thermal performance factor of 3.58 with the new suggested design. Sahel *et al*. [20] studied the heat transfer characteristics in a smooth tube. They proposed new correlation and this correlation was agreement with literature. Sahel *et al*. [21] investigated numerically effect of tube shape in a finned tube heat exchanger. Simulations were 2-D and Reynolds number varied between 3000-20000 in analyses. They found that circular tube ensured a higher heat transfer coefficient. Djeffal *et al*. [22] examined and compared oval and flat tube in a annular finned tube heat exchangers with conventional circular tube. Sahel *et al*. [23] found a new correlation of hydrothermal characteristic over flat tube banks. They studied numerically and simulations were 2-D in laminar flow condition. Ameur *et al*. [24] studied in a horizontal and baffled tubes. They investigated effect of perforation in baffles. Sahel *et al*. [25] worked fluid flow and heat transfer over a fin and flat tube heat exchangers with complex vortex generators. They studied numerically a lot of angles of vortex generator. Mellal *et al*. [26] studied numerically hydro-thermal performance of a shell and tube heat exchanger under different baffle arrangement and orientation. They showed baffles thermal performances enhancement in shell side. Chamkha *et al*. [27] evaluated numerically *S*-shaped fins in shell-and-tube heat exchangers. They analyzed using the standard $k$-$\varepsilon$ turbulence model. Menni *et al.* [28] studied numerically hydrodynamics performance in shell and tube heat exchangers with using W-baffle vortex generator. They showed thermal and flow performance of the baffle.

In this study, thermal and flow analyses of the solid and serrated annular geometry fin are performed numerically by using ANSYS-FLUENT software. In numerical analyses, five different Reynolds numbers and six different geometric parameters are studied. These parameters are segment height ratio tube diameter, fin height, fin pitch, fin thickness and segment width, respectively. Effects of these parameters are investigated to determine the flow and thermal performance. According to analysis results, the thermal performance of the serrated annular fin geometry is about 8.2% higher than the solid fin geometry, while the flow performance decreases by 7.5%. Obtained results contributed to the literature in terms of detailed examination of the effect of different parameters.

## Three-dimensional solid and serrated fin tube geometries

In this study, two different fin geometries are basically considered. The first geometry annular solid fin and second geometry serrated fin are given in fig. 1. The annular serrated finned tube bundle geometry is shown in fig. 2. The tube bundle consists of eight tubes along with the air-flow. Air side of the tube bundle is taken into consideration in all studies. Then, the air-flow flowing outside of the finned tube is modeled numerically. In numerical analyses, six different main geometric parameters are considered. These parameters are segment height ratio, tube diameter, fin height, fin pitch, fin thickness and segment width, respectively.

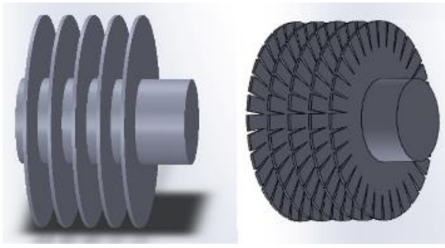

Figure 1. Solid and serrated fin


Figure 2. Annular serrated finned tube bundle

## Numerical model

For numerical studies, 3-D analyses were performed in ANSYS-FLUENT program. The 3-D solid models were prepared according to the parameters for numerical studies. The air side of the flow was taken into account in all numerical studies. Constant surface temperature is also defined for the tube surface and fin surfaces. In most studies in the literature, the definition of constant surface temperature is used [13]. Air inlet is defined from a distance of $3d_o$ and its outlet is defined from a distance of $10d_o$ as shown fig. 3 [14, 15].

Details of numerical scheme: It is used the s*econd-order upwind scheme* which it gives more precise results. For convergence criteria scaled residuals are $10^{-3}$ and energy residual is $10^{-6}$. Moreover, calculation time of the analysis took about 5-6 hours and simulations were done a computer which have i7-5700HQ CPU at 2.70 GHz with 2.70 GHz processor and 16 GB of RAM.

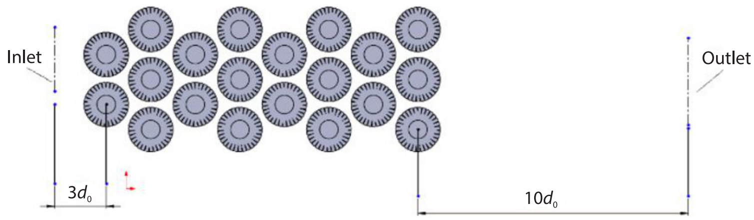


Figure 3. Air inlet and outlet in the numerical model

### *Three-dimensional heat and flow equations*

The continuity and Navier-Stokes equations used in air-flow modeling for the 3-D model can be given:

$$\frac{\partial \rho}{\partial t}+\frac{\partial\left(\rho u_i\right)}{\partial x_i}=0 \tag{1}$$

$$\frac{\partial\left(\rho u_i\right)}{\partial t}+\frac{\partial\left(\rho u_i u_j\right)}{\partial x_j}=-\frac{\partial P}{\partial x_i}+\frac{\partial}{\partial x_j}\left(\tau_{ij}-\rho \overline{u_i' u_j'}\right) \tag{2}$$

The viscous shear stress is:

$$\tau_{ij}=\mu\left(\frac{\partial u_i}{\partial x_j}+\frac{\partial u_j}{\partial x_i}-\frac{2}{3}\delta_{ij}\frac{\partial u_k}{\partial x_k}\right) \tag{3}$$

The Reynolds averaged energy equation used in the calculation of the enthalpy throughout the flow and kinetic energy is given in eq. (4) as:

$$\frac{\partial(\rho h_{\text{total}})}{\partial t}-\frac{\partial P}{\partial t}+\frac{\partial(\rho u_j h_{\text{total}})}{\partial x_j}=\frac{\partial}{\partial x_j}\left(\lambda\frac{\partial T}{\partial x_j}-\rho\overline{u'_j h}\right)+\frac{\partial}{\partial x_j}\left[u_i\left(\tau_{ij}-\rho\overline{u'_i u'_j}\right)\right] \tag{4}$$

where $h_{\text{total}}$ and $k$ are defined as:

$$h_{\text{total}}=h+\frac{1}{2}u^2+k,\quad k=\frac{1}{2}\overline{u_i^2}$$

### *Turbulence model*

Two different turbulence models are generally used in the literature for the calculation of eddy viscosity, $\mu_t$, in this type of tube bundle problem. The Re-normalisation group (RNG $k$-$\varepsilon$) model and shear stress transport (SST $k$-$\omega$) models are turbulence models frequently used in this kind of computational studies [2, 14, 17, 29]. In this study, both turbulence models were tested and the SST $k$-$\omega$ turbulence model which gave the closest results with the literature was used.

### *Thermo-physical properties*

Air was used as the fluid in all numerical studies. In order to determine the thermo-physical properties of the air, a polynomial equation, eq. (5) in accordance with the reference air table [30] in a certain temperature range ($280 \leq T$ [K] $\leq 550$) was interpolated. The coefficients of the polynomial equation were given in tab. 1.

$$\rho/\mu/c_p/k=\text{A}+\text{B}T+\text{C}T^2+\text{D}T^3+\text{E}T^4+\text{F}T^5,\quad 280\leq T\leq 550 \tag{5}$$

**Table 1. Coefficient of the polynomial equation for air properties**

| | $\rho$ [kgm$^{-3}$] | $\mu$ [kgm$^{-1}$s$^{-1}$] | $c_p$ [kJkg$^{-1}$K$^{-1}$] | $k$ [Wm$^{-1}$K$^{-1}$] |
|---|---|---|---|---|
| A | 5.4894 | $-2.2401\cdot10^{-06}$ | $9.4431\cdot10^{-01}$ | $9.2876\cdot10^{-05}$ |
| B | $-3.5035\cdot10^{-02}$ | $1.1108\cdot10^{-07}$ | $8.8556\cdot10^{-04}$ | $9.3537\cdot10^{-05}$ |
| C | $1.1762\cdot10^{-04}$ | $-2.4994\cdot10^{-10}$ | $-4.9308\cdot10^{-06}$ | $-2.5632\cdot10^{-09}$ |
| D | $-2.1926\cdot10^{-07}$ | $5.3773\cdot10^{-13}$ | $1.2709\cdot10^{-08}$ | $-9.4567\cdot10^{-11}$ |
| E | $2.1534\cdot10^{-10}$ | $-6.8193\cdot10^{-16}$ | $-1.4674\cdot10^{-11}$ | $1.2464\cdot10^{-13}$ |
| F | $-8.7095\cdot10^{-14}$ | $3.6122\text{E}\cdot10^{-19}$ | $6.5316\cdot10^{-15}$ | $-5.6940\cdot10^{-17}$ |

### *Boundary conditions*

Velocity inlet and pressure outlet were defined as boundary conditions for numerical studies. In addition, non-slip condition is applied on all walls for numerical solution. Velocity inlet was used at the fluid enters and pressure outlet boundary condition was used for the outlet. Atmospheric pressure is accepted as 0 Pa for the outlet. The wall boundary condition is applied to all fin and tube surfaces that the air comes into contact with these surfaces. Wall temperature is considered to be constant, 283 K. Constant wall temperature is also commonly used in similar studies [5, 18]. In addition, the symmetry boundary condition is also defined in order to shorten the duration of the numerical analysis. The symmetry planes and CFD model of the annular serrated finned tube model for the numerical analysis are shown in fig. 4. Moreover, the symmetry boundary condition due to the repetition of the fins was applied.

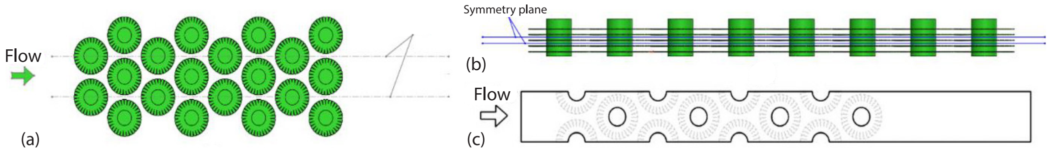


**Figure 4. Symmetry condition of CFD model; (a) top view, (b) side view, and (c) CFD model**

### *Mesh structure and mesh independence*

In the numerical study, the mesh structure was formed separately for solid and serrated fin geometry. For both geometries, smaller mesh element size is used near the fin and tube surface. Thus, more precise results were obtained. In the regions of the flow away from the fin and the pipe wall, a relatively larger mesh element size is used. As a result, numerical analysis calculation time was decreased. The mesh structure developed for the solid and serrated finned tube bundles are shown respectively in figs. 5 and 6. In both geometries, boundary-layers are formed on the tube and fin surfaces where heat transfer and flow friction are important. Boundary-layer thicknesses were calculated for the case $y+ <1$ and the thicknesses of the next layers were determined to increase by 20% [5].

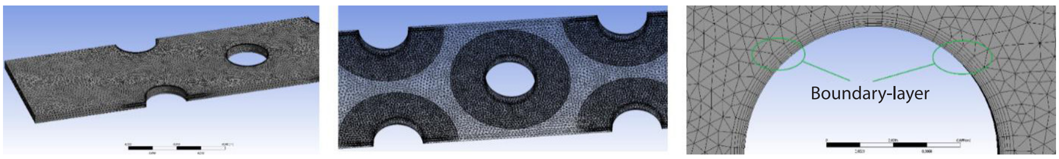


**Figure 5. Solid fin tube bundle mesh and boundary-layer details**

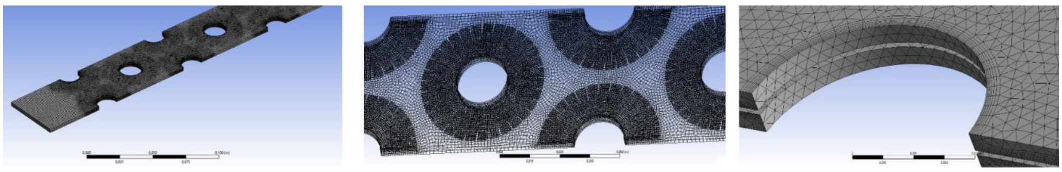

**Figure 6. Serrated fin tube bundle mesh and boundary-layer details**

Reducing the element size and accordingly increasing the number of elements provides more precise results. These situations affect the increase of analysis calculation time. Therefore, separate mesh independence studies were carried out for two different geometries. Increasing the number of mesh elements for the solid and serrated finned tube bundles about over the 450000 and 486000 mesh elements respectively did not change the numerical result. All numerical studies were performed based on this mesh independence status.

### *Pressure drop and Nusselt number equations*

The Reynolds number is calculated for maximum velocity. Maximum speed occurs in the minimum area between the fins. The Euler number is the friction factor due to the pressure drop. Reynolds and Euler numbers is calculated:

$$\mathrm{Re} = \frac{\rho V_{\max} d_o}{\mu} \quad \text{and} \quad \mathrm{Eu} = \frac{2\Delta P}{\rho V_{\max}^2} \tag{6}$$

Logarithmic mean temperature difference and average heat transfer coefficient can be defined respectively as:

$$\Delta T_{\text{LMTD}} = \frac{T_{\text{row,in}} - T_{\text{row,out}}}{\ln\left(\dfrac{T_{\text{row,in}} - T_{\text{w}}}{T_{\text{row,out}} - T_{\text{w}}}\right)} \quad \text{and} \quad U = \frac{\dot{Q}_{\text{total}}}{A_{\text{total}}\Delta T_{\text{LMTD}}} \tag{7}$$

The effective heat transfer coefficient, $\alpha_{\text{e}}$, and actual average heat transfer coefficient, $\alpha_{\text{o}}$, at the outer surface of the finned tube is given in eq. (8). If conduction resistance is neglected, the average heat transfer coefficient can be approximately written as equal to the overall heat transfer coefficient.

$$\alpha_{\text{e}} = \frac{1}{\dfrac{1}{U} - \dfrac{A_{\text{total}}\, ln\left(\dfrac{d_{\text{o}}}{d_{\text{i}}}\right)}{2\pi L k_f}} \cong U \quad \text{and} \quad \alpha_{\text{o}} = \frac{\alpha_{\text{e}} A_{\text{total}}}{A_{\text{bare}} + \eta_{\text{fin}} A_f} \tag{8}$$

where $\eta_{\text{fin}}$ is the fin efficiency, $A_{\text{bare}}$ – the unfinned surface area, and $A_{\text{f}}$ – the fin area. Finally, Nusselt number is expressed in eq. (9):

$$\text{Nu} = \frac{\alpha_{\text{o}} d_{\text{o}}}{k_{\text{air}}} \tag{9}$$

where $k_{\text{air}}$ is the conduction heat transfer coefficient of air.

## Results and discussion

In order to be validated the obtained results from the numerical study is need to be compared with the literature. For this purpose, analytical or experimental studies in the literature were compared.

### *Validation studies for solid finned tube bundle*

For solid finned tube bundles, separate validation studies have been carried out in terms of both heat transfer and pressure drop with several different studies in the literature. In fig. 7(a), two different turbulence models were compared with three different studies in the literature, and a validation study was carried out in terms of pressure drop. The SST *k-ω* and RNG *k-ε* turbulence models are widely used models for flow over cross-flow tube bundles numerical studies [14, 17]. In fig. 7(b), the comparison of solid finned tube bundles in the literature in terms of heat transfer is given. According to these results, a good agreement is observed with

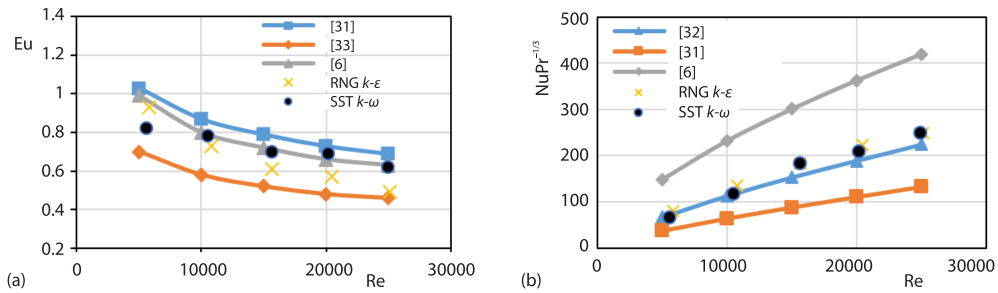


**Figure 7. Validation studies for the solid finned tube by friction factor and Nusselt number**

the literature. In our study, the SST $k$-$\omega$ turbulence model approached the literature with an average error of 7.5 % and RNG $k$-$\varepsilon$ with an error of 10.8 %. Therefore, the SST $k$-$\omega$ turbulence model was used in numerical analysis.

### *Validation studies for serrated finned tube bundle*

For serrated finned tube bundles, separate validation studies have been carried out in terms of both heat transfer and pressure drop with several different studies in the literature. These validation studies are shown in figs. 8(a) and 8(b). The SST $k$-$\omega$ turbulence model gave results with an error of 6.8%, RNG $k$-$\varepsilon$, with an error of 8.3%, consistent with the literature. Thus, SST $k$-$\omega$ was used for numerical analysis.

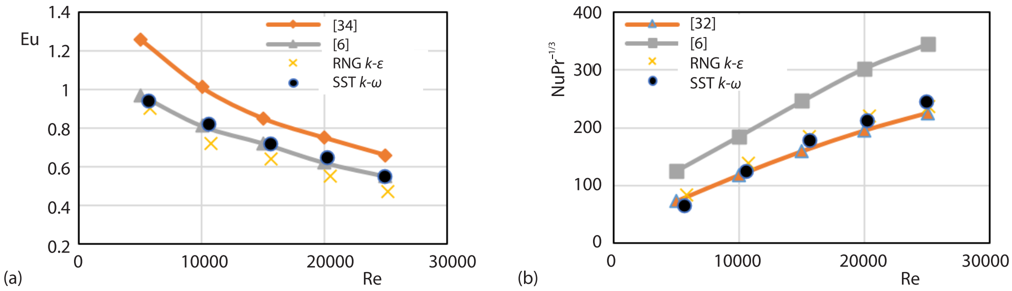


**Figure 8. Validation studies for the serrated finned tube by friction factor and Nusselt number**

### *Variable parameters*

Variable geometric parameters considered in numerical analyses are shown in fig. 9. These parameters are segment height ratio, $h_s/h_f$, tube diameter, $d_o$, fin height, $h_f$, fin pitch $p_f$, fin thickness, $t_f$, segment width, $s_w$, respectively. Variable geometric parameter values taken in numerical analysis are given in the tab. 2. All parameters are given in mm units except $h_s/h_f$ which is a dimensionless parameter.

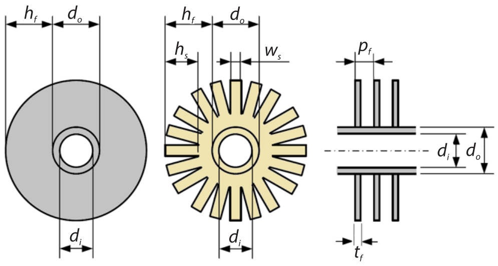


**Figure 9. Solid and serrated fin parameters**

**Table 2. Variable geometric parameters**

| $h_s/h_f$ | $d_o$ | $h_f$ | $p_f$ | $t_f$ | $w_s$ |
|---|---|---|---|---|---|
| 0.0 | 17.2 | 6.0 | 2.5 | 0.4 | 3.0 |
| 0.5 | 24.0 | 8.0 | 3.5 | 1.0 | 4.5 |
| 1.0 | 32.0 | 12.0 | 5.5 | 1.2 | 6.0 |

Note: All parameters except $h_s/h_f$ are in mm.

#### *Effect of tube outer diameter*

Nusselt number and friction Euler number with respect to Reynolds number due to three different tube diameters for solid and serrated finned tubes are shown in figs. 10 and 11. For the heat transfer performance, Nusselt number was increasing with the increasing of tube diameter for both fin geometries. Euler number which is the friction factor is decreasing with the increasing the tube diameter.

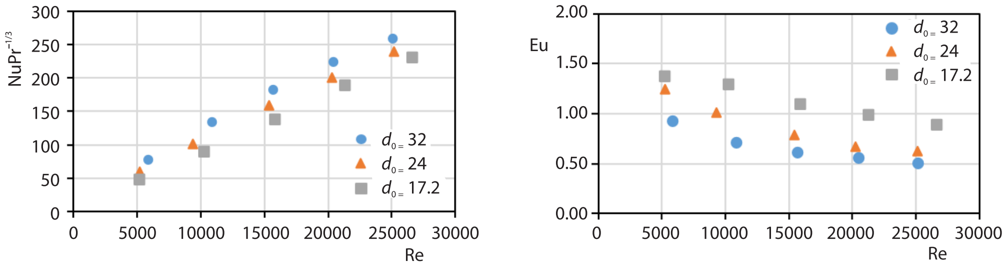


**Figure 10. Effect of tube outer diameter for solid fin**

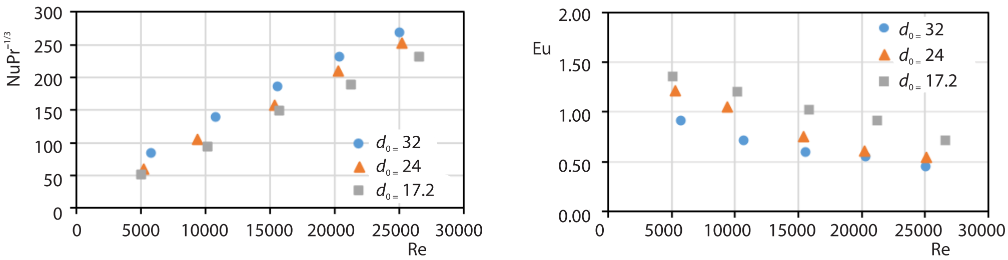


**Figure 11. Effect of tube outer diameter for serrated fin**

### *Effect of the fin height*

The effect on heat transfer and friction factor for three different fin heights is shown for solid and serrated fins in figs. 12 and 13, respectively. For both solid and serrated fins, the Nu number decreases as the fin height decreases. However, the Euler number increases as the fin height increased as well.

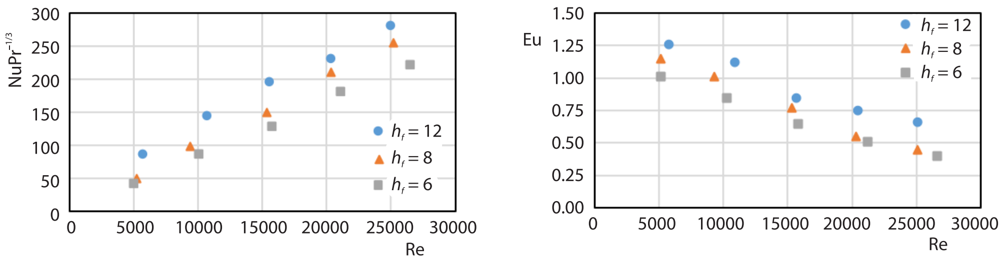


**Figure 12. Effect of fin height for solid fin**

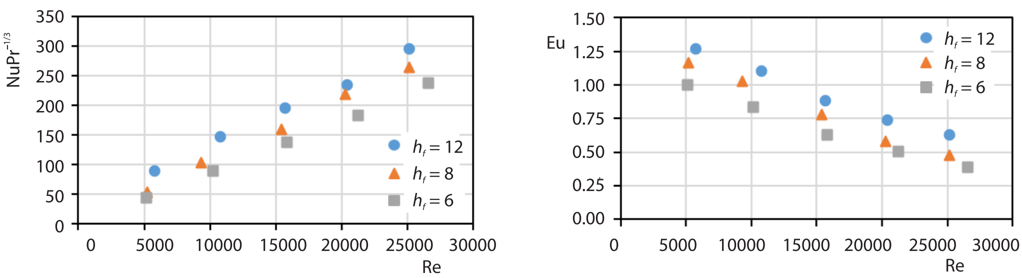


**Figure 13. Effect of fin height for serrated fin**

### *Effect of the fin pitch*

The effect on heat transfer and friction factor for three different fin pitches is shown in figs. 14 and 15. The change in the fin pitch parameter did not significantly change the Nusselt number at low Reynolds numbers, but Nusselt number increases with an increase of the fin pitch at large Reynolds numbers for solid or serrated fins. On the other hand, the friction factor decreases as the fin pitch increases for solid or serrated fins. Moreover, Nusselt number value for serrated fined tubes is higher than solid finned tubes, but Euler number values is nearly close for both finned.

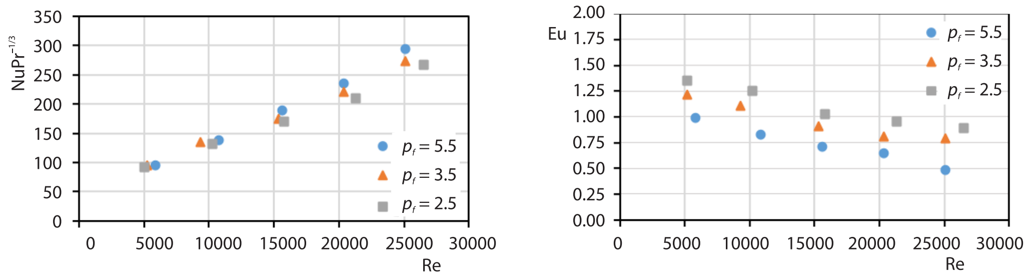


**Figure 14. Effect of fin pitch for solid fin**

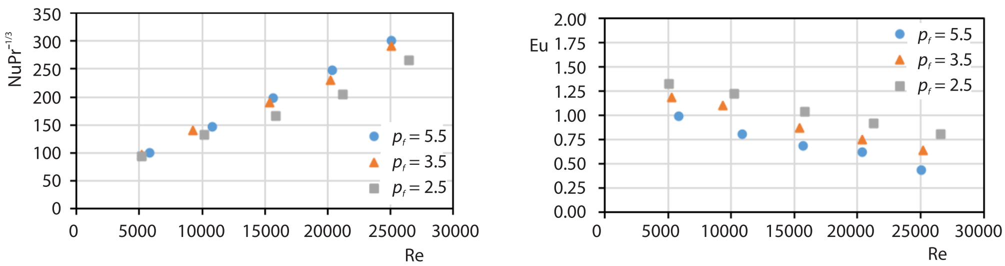


**Figure 15. Effect of fin pitch for serrated fin**

### *Effect of the fin thickness*

The effect of fin thickness on heat transfer and friction factor for the solid and serrated fin is shown in figs. 16 and 17. The change of fin thickness at low Reynolds numbers did not affect the Nusselt number so much for solid and serrated fins. Nusselt number value for serrated fined tubes is a little higher than solid finned tubes. Also, Euler number is increasing with the increasing the fin thickness for solid and serrated fin. The friction factor values for serrated fined tubes are a little higher than the solid finned tubes.

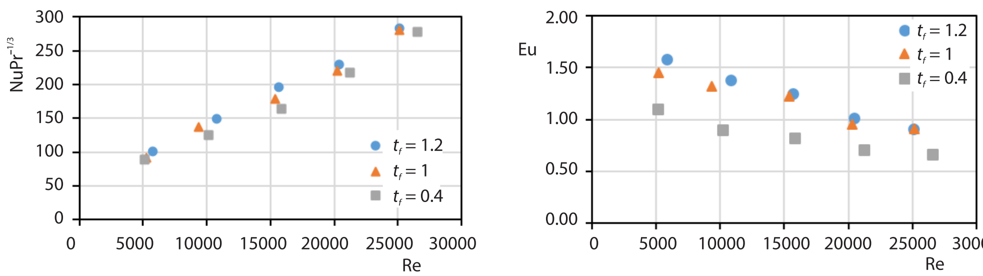


**Figure 16. Effect of fin thickness for solid fin**

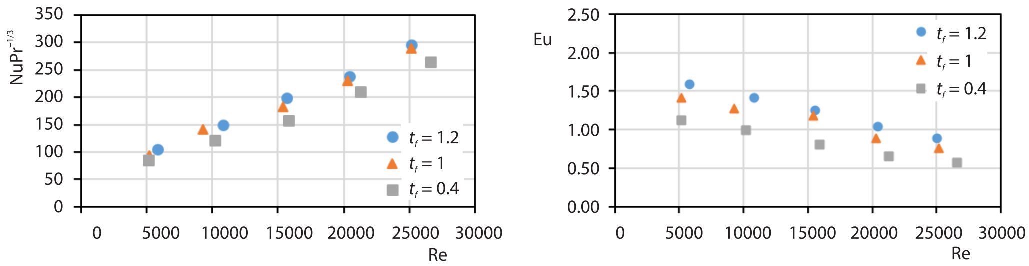


**Figure 17. Effect of fin thickness for serrated fin**

### *Effect of the segment height ratio*

The effect of segment height ratio on heat transfer and friction factor for finned tubes is shown in fig. 18. Segment height ratio is an important parameter for the serrated fin geometry. The segment height ratio equals to zero ($h_s/h_f = 0$) means that the fin is completely solid fin, and this ratio equals to one ($h_s/h_f = 1$) means that fin is serrated fin. The segment height ratio of the serrated fin varies between 0 and 1. Nusselt number increases with increasing the fin segment height ratio. This means that serrated fin shows a better thermal performance than solid fin. Also, the friction factor decreases with the decreasing segment height ratio. Thereby, the flow performance of the solid fin is better than serrated fin.

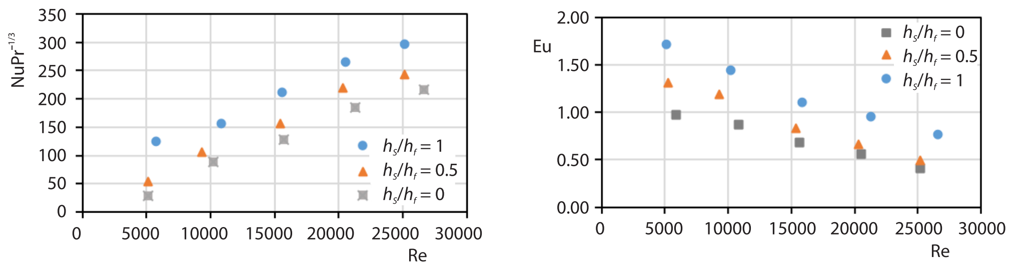


**Figure 18. Effect of segment height ratio for serrated fin**

### *Effect of the serrated fin width*

The effect of serrated fin width parameter which is considering only at the serrated finned tube, on the heat transfer and friction factor is shown in fig. 19. Nusselt number is increasing with decreasing the serrated fin width. At high Reynolds numbers, the fin width effects on the Nusselt number are more remarkable. In addition, the friction factor is increasing with the decreasing fin width.

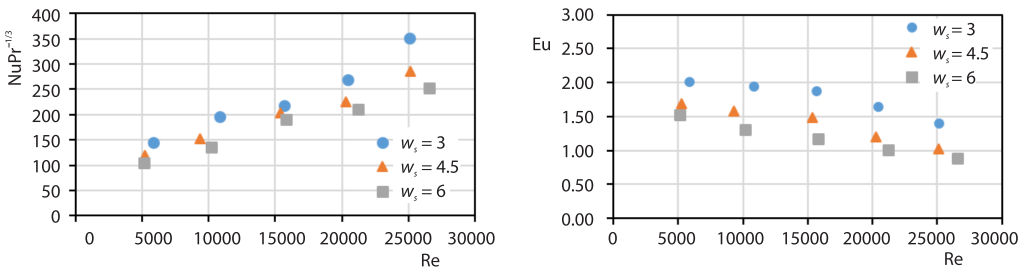


**Figure 19. Effect of serrated fin width**

## Conclusions

In the development of finned tube cross-flow heat exchangers, the change in fin geometries on the gas side significantly effects the heat transfer and pressure drop. In this study, solid and serrated finned tube bundles used in heat exchangers were numerically examined. In the analyses, the effects of some geometric parameters related with solid and serrated fins are investigated. It has been noted that serrated finned tube bundles gave better results in terms of heat transfer in many parameters than solid finned tube bundles, but in respect of pressure drop, solid finned tube bundles gave more affirmative results. Obtained numerical results for solid and serrated finned tube bundles show a good agreement with the literature. The following outcomes of this study can be written are as follows.

- The increases of tube diameter have an effect that increases heat transfer and decreases friction factor for both fin geometries. For thermal performance, serrated finned tubes results is better than solid finned tubes about 10.3%, however solid fin shows a better flow performance by 10.5% in respect to serrated finned tubes.
- The increase of the fin height causes enhancement heat transfer. But friction factor is affected negatively.
- The increase of the fin pitch and fin thickness parameter don't significantly change the Nusselt number sselt number at low Reynolds numbers, but Nusselt number increases with an increase of the fin pitch and thickness at large Reynolds numbers for solid or serrated fins. In addition, the friction factor decreases as the fin pitch and thickness increases for solid or serrated fins.
- The increase of the segment height ratio causes the Nusselt number increases. This means that serrated fin shows a better thermal performance by 8.2% than solid fin. Also, friction factor decreases with the decreasing segment height ratio. Hence, flow performance of the solid fin is better about 7.45% than serrated fin.
- The increase in the serrated fin width affects the heat transfer negatively and flow performance positively. Increasing the fin width causes the decreasing the Nusselt number and the friction factor.
- All these results are compatible with the literature studies on finned-tube heat exchangers. These studies contributed to the literature in terms of detailed examination of the effect of different parameters.

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