Tube heat exchanger with new star shaped fins

M Bošnjaković, A Čikić, S Muhič and M Stojkov

1 College of Slavonski Brod, Dr Mile Budaka 1, 35000 Slavonski Brod, Croatia
2 Technical College in Bjelovar, Trg Eugena Kvaternika 4, 43000 Bjelovar, Croatia
3 Faculty of Technologies and Systems, Na Loko 2, 8000 Novo Mesto, Slovenia
4 Mechanical Engineering Faculty in Slavonski Brod, Trg Ivane Brlić Mažuranić 2, 35000 Slavonski Brod, Croatia

E-mail: mladen.bosnjakovic@vusb.hr

Abstract. The article compares the tube heat exchanger with annular fins with innovative star shaped fins. For the analysis of the heat transfer and pressure drop, the computational fluid dynamics analysis with ANSYS Fluent software was applied. Through the tubes flow the hot water and heat the cold air flowing around the fins and tubes. The validation of selected turbulence models (k-ω SST and k-ε Realizable) was performed by annular fins comparing with the results in the literature. The results of the numerical analysis for star shaped fins for 2200 < Re < 12000 show an average of 15 % higher fins efficiency, 13 % higher Nu number on the air side with reduction of the exchange surfaces weight of 23.8 % compared to annular fins.

1. Introduction

Efficiency of heat exchanger and its dimensions are ones of the most important parameters to consider in engineering design activities. Process heat transfer intensification is a design philosophy that aims at achieving reduced size or mass of heat transfer equipment and the associated benefits. Intensified heat transfer will increase efficiency, which will lead to energy conservation and reduced costs.

Heat transfer intensification can be achieved by applying different fins shapes that increase the turbulence of the flow, so the thermal boundary layer is periodically interrupted which results in more efficient heat exchange.

1.1. Literature review

The analytical approach to solving heat transfer for simple geometry of finned surfaces dates from the last century when basic functions were defined for temperature profiles and fin efficiency for different fin shapes. Hashizume et al. [1] analysed the thermal efficiency of rectangular cross sectional circular fins and performed an analytical solution in the form of a modified Bessel function. Kays and London [2] provided experimental correlations for a wide range of configurations adopted in compact extended surface heat exchangers and presented test data on finned circular tubes.

Ward and Young [3] developed heat transfer and pressure drop correlations for plain finned tubes with triangular pitch. They compared their pressure drop data with a correlation from literature.

Stasiulevicius et al. [4] developed correlations of the convective heat transfer coefficient and resistance of finned-tube bundles in a cross flow including the effects of geometric parameters of fins and tube arrangement within the bundle. Anoop, Balaji, and Velusamy [5] investigated the heat transfer on one tube with serrated fins combining experimental approach and numerical calculation.
The depth of the segment, the pitch of the fins, fin height and thickness were investigated. Widely cited correlations for pressure drop are authorized by Briggs and Young [6], Schmidt [7], Ward and Young [8] and Nir [9].

Numerous methods used in computational fluid dynamics have been developed today. A review of all the methods applied today in computer fluid dynamics was given by Lokman and Fdhil [10]. Detailed analysis shows that computational fluid dynamics can be used as a flow visualization tool. Due to certain deviations in the results between certain models it is necessary to compare the results with experimental research.

2. Physical model of heat exchanger with star shaped fins
The geometric model of the tube heat exchanger on which its thermal efficiency and thermal effectiveness will be estimated in relation to the reference model, consists of a tube bundle with star shaped fins. Water as a heating medium passes through the tubes. Across the finned tubes flows colder air. The new and innovative star shaped fins enable achievement of higher turbulence levels of the fluid flowing around the fins, and thus higher heat transfer. For research implementation, the tube Ø20×1.5 is selected. As a reference geometry, annular fin Ø40/Ø20 with thickness 0.5 mm was selected (Figure 1a). The new star shaped fin is shown in Figure 1b and has eight vertices spaced uniformly along the circumference. For the implementation of comprehensive research, the selected material of the tubes and fins is stainless steel.

![Figure 1. Geometry of annular fin (a) and star shaped fin (b).](image)

3. Numerical analysis
Certain assumptions and simplifications were adopted for the mathematical description of the thermal phenomena within the heat exchanger. The most common assumption that is applied when defining fluid flow is fluid continuity. The assumptions of homogeneity and isotropy were also introduced. It is assumed that the thermal conductivity of the fin is equal in all directions. Air is considered as homogeneous mixture of gases and is treated as a one-component fluid (dry air). There is no fluid leakage through the walls of the finned exchanger and that the heat exchange with the environment is negligible. The fins are completely and tightly attached to the tube and there is no contact resistance.

![Figure 2. Mathematical model of computational domains.](image)
between the tube and the fin surface. Steady state analysis is performed.

The numerical domain was chosen to perform the numerical analysis of the tube heat exchanger. For a heat exchanger with a uniform fluid flow field at the inlet, a typical repeating section in the heat exchanger is selected for the domain (Figure 2), and the solutions obtained are assumed valid for the entire heat exchanger.

Boundary conditions are determined when defining the computational model and the physical properties of fluids and materials of the exchange surfaces. Boundary conditions include defining:

- The air entering the computational domain is assumed to have uniform velocity over the cross section and ranges from 1 m/s to 5 m/s, and turbulence intensity 5 %.
- The air temperature at the inlet to the heat exchanger is 288 K.
- Hot water at the inlet to tubes has temperature 353 K. Because water has a high thermal capacity, it is assumed that the temperature of the tube inner wall is constant and equal to the water temperature.
- At the sides of the computational domain it is set symmetry boundary condition.
- The gauge pressure $p_{out,c}$ at the outlet of channel is set to zero. This corresponds to atmospheric conditions $p_a$.
- Hydraulically smooth walls was defined for outer tube and the fin walls.
- Symmetry condition is set for top, bottom, left and right side of computational domain. Symmetry condition is applied due to simplify the calculation as reduce computational domain where it is possible. At the symmetry planes a zero heat flux is assumed. The normal velocity component at the symmetry plane equals to zero, i.e. no convective flux across that symmetry plane exist. Thus, the temperature gradients and tangential components of the velocity gradients in normal direction are set to zero.
- Physical properties of air are defined as polynomial function of temperature (and pressure) and thus set in Ansys Fluent software. Density of the air is based on incompressible ideal gas law.
- The physical properties of the fin and tube material are set constant ($\lambda_t=16.2$ W/(m·K), $\rho_t=7860$ kg/m$^3$).

The mathematical model used to describe a physical problem is a set of differential equations and constitutive relations and initial and boundary conditions. The basic equations of fluid dynamics are derived from conservation of mass, conservation of momentum and conservation of energy. The established mathematical model is solved by using the finite volume method. Computational domain is meshed by ANSYS software.

Meshing was performed by using a hybrid mesh where most of the volume is structured mesh, and the smaller part around the fins is unstructured mesh. The area along the fins and tubes where the convection heat transfer is performed is covered with eight boundary layers and the thickness of the first layer is 0.025 mm (Figure 3). By successive refining of the mesh, we found the solution, independent of the computational mesh. In the final analysis we used the mesh with 10.9 million of the finite volumes (elements).

Numerical calculation was performed with ANSYS Fluent software. A steady state model of heat exchange is assumed. Two models of turbulence were used: $k$-$\varepsilon$ Realizable with Enhanced Wall Treatment (EWT) method and $k$-$\omega$ SST. The calculation results of both models were used for the computation of $Nu$ and $Eu$, which were compared with the results of the literature with the aim of selecting a turbulence model whose results more coincide with the results of the literature.

**Figure 3.** Part of mesh with annular fins.
4. The results of numerical calculation
The input and output data of the heat calculations are summarized in the tables below.

**Table 1.** Characteristic dimensions of tube and fins.

| Tube and fin material | - | - | Stainless steel |
|-----------------------|---|---|------------------|
| Tube outside diameter | \( d_0 \) mm | 20 |
| Tube inside diameter | \( d_i \) mm | 17 |
| Tube rows configuration | - | - | staggered |
| Transverse tube pitch | \( s_t \) mm | 50 |
| Longitudinal tube pitch | \( s_l \) mm | 40 |
| Fin height | \( h_f \) mm | 10 |
| Fin thickness | \( t_f \) mm | 0.5 |
| Fin pitch | \( s_f \) mm | 4.5 |
| Number of rows | \( N_l \) | 5 |

**Table 2.** Output data from Ansys Fluent for \( k-\varepsilon \) Realizable with EWT turbulence model.

| Item                                      | \( u_{in}=1.0 \) m/s | \( u_{in}=2.4 \) m/s | \( u_{in}=5.0 \) m/s |
|-------------------------------------------|------------------------|------------------------|------------------------|
| Outlet air temp. from tube bundle         | \( T_{out} \) K        | 327.3                  | 315.4                  | 307.1                  |
| Pressure at the tube bundle inlet         | \( p_{in} \) Pa        | 9.30                   | 36.98                  | 115.1                  |
| Pressure at the tube bundle outlet        | \( p_{out} \) Pa       | -1.35                  | -8.11                  | -31.13                 |

**Table 3.** Output data from Ansys Fluent for \( k-\omega \) SST turbulence model.

| Item                                      | Annular fins | Star shaped fins |
|-------------------------------------------|--------------|------------------|
|                                           | \( u_{in}=1.0 \) m/s | \( u_{in}=2.4 \) m/s | \( u_{in}=5.0 \) m/s | \( u_{in}=1.0 \) m/s | \( u_{in}=2.4 \) m/s | \( u_{in}=5.0 \) m/s |
| Air temp. at tube bundle outlet           | \( T_{out} \) K | 327.6 | 314.8 | 306.5 | 321.8 | 311.1 | 304.3 |
| Pressure at the tube bundle inlet         | \( p_{in} \) Pa | 9.30 | 40.55 | 149.2 | 8.10 | 37.48 | 142.84 |
| Pressure at the tube bundle outlet        | \( p_{out} \) Pa | -1.85 | -11.28 | -49.85 | -2.10 | -12.28 | -52.52 |

5. Data reduction and interpretation
In order to account for the variation of the properties with temperature, the fluid properties are usually evaluated at the boundary layer temperature, defined as

\[
T_{bl} = \left( T_{tot, av} + T_{av} \right) / 2
\]

that is the arithmetic average of the surface and the average free-stream temperatures. Average free-stream temperature is defined as

\[
T_{av} = (T_{in} + T_{out}) / 2
\]

The surface temperature of the fins by height is not constant and it is different in each row of tubes. Average fin temperature \( T_{f, av} \) is calculated based on the fin base surface temperature \( T_{c, v} \) and fin efficiency \( \eta_f \):

\[
T_{f, av} = T_{c, v} + \frac{U}{\alpha_c} (1 - \eta_f) \cdot (T_{av} - T_{c, v})
\]

Average temperature of the entire surface of tubes and fins \( A_{tot} \) is calculated according:

\[
T_{tot, av} = \frac{T_{f, av} \cdot A_f + T_{c, v} \cdot A_c}{A_{tot}}
\]
where the surface area of the fins is \( A_f \) and \( A_c \) the surface of the tubular part without fins. For calculating the Reynolds number, the physical properties (density, dynamic viscosity) are taken at boundary layer temperature \( T_{bl} \). Characteristic dimension is based on the outer diameter of the tube \( (d_o) \), and the mass flux, i.e. velocity through the narrowest free flow area within the tube bundle. This is consistent with the relevant literature

\[
Re = \frac{\rho_{bl} \cdot u_{\text{max}} \cdot d_o}{\mu_{bl}} = \frac{m \cdot d_o}{\mu_{bl} \cdot A_{\text{min}}}
\]

For the geometries considered in the present study, the narrowest free flow area is the area between two tubes in the transversal direction, and can be defined as

\[
A_{\text{min}} = s_{t} \cdot (s_{t} - d_{o}) - 2 \cdot h_{t} \cdot t_{t}
\]

For heat transfer correlations, the Nusselt number \((Nu)\) is defined as

\[
Nu = \frac{\alpha_{o} \cdot d_{o}}{\lambda_{bl}}
\]

where \( \alpha_{o} \) is actual average gas-side heat transfer coefficient. For pressure drop correlations, the Euler number \((Eu)\) is defined as

\[
Eu = \frac{\Delta p}{N_{1} \cdot \rho_{bl} \cdot u^{2}_{\text{max}}}
\]

The heat transfer rate is calculated from the calculated mass flow rate and overall temperature changes.

\[
m = \rho_{in} \cdot A_{in} \cdot u_{in}
\]

\[
\dot{Q} = m \cdot c_{p,\text{av}} \cdot (T_{\text{out}} - T_{\text{in}})
\]

To calculate the overall heat transfer coefficient \((U)\) the logarithmic mean temperature difference for counter-current flow is chosen. This method has proven to be accurate to within 0.1 % of the actual cross-counter-current arrangement mean temperature difference.

\[
\Delta T_{\text{In}} = \frac{T_{\text{in}} - T_{\text{out}}}{\ln \left( \frac{T_{\text{in}} - T_{c,v}}{T_{\text{out}} - T_{c,v}} \right)}
\]

The overall heat transfer coefficient is then calculated

\[
U = \frac{\dot{Q}}{A_{\text{tot}} \cdot \Delta T_{\text{In}}}
\]

The effective/apparent air-side heat transfer coefficient is

\[
\alpha_{e} = \frac{1}{U} \left( \frac{\ln d_{o}}{\ln d_{l}} - A_{\text{tot}} \frac{1}{A_{\text{tot}} \cdot \alpha_{t}} \right)
\]

where \( L_{t} \) is tube length that is equal to model width. In the case of air temperatures lower than 300 °C, the value of radiation heat transfer coefficient could be negligible and then its value may be taken as zero. Outside and inside fouling factors is also taken as zero. The effective heat transfer coefficient in
equation (13) is the apparent heat transfer coefficient including fin efficiency. In order to calculate the actual average gas-side heat transfer coefficient \( \alpha_o \), the fin efficiency \( \eta_f \) is needed.

\[
\alpha_o = \frac{\alpha_{e} \cdot A_{tot}}{(A_{c,v} + \eta_f \cdot A_f)}
\]  \( (14) \)

The annular fins efficiency is calculated according Schmidt

\[
m = \left( \frac{2 \alpha_0}{\lambda_f \cdot t_f} \right)^{1/2}
\]  \( (15) \)

\[
h_e = h_t + \frac{t_e}{2}
\]  \( (16) \)

Corrected fin length \( h_e \) is based on assumption of equivalence between heat transfer from the actual fin with tip convection and heat transfer from a longer, hypothetical fin with an adiabatic tip.

\[
\eta_{th,f} = \frac{\tanh(\psi \cdot m \cdot h_e)}{\psi \cdot m \cdot h_e}
\]  \( (17) \)

\[
\psi = 1.0 + 0.35 \cdot \ln \left( 1.0 + 2.0 \cdot \frac{h_e}{d_o} \right)
\]  \( (18) \)

\[
E = 0.76 + 0.24 \cdot \eta_{th}
\]  \( (19) \)

\[
\eta_t = E \cdot \eta_{th}
\]  \( (20) \)

For new type of fins proposed in the article, there is no expression for efficiency in literature. As serrated fins are similar to new type of fins expression for serrated fins will be used. This means some error in calculation and have to be examined in future. The appropriate theoretical fin efficiency for serrated fins, which compensates for the finite conductance in the fins, under the assumption of a uniform air side heat transfer coefficient, can be expressed as \[1\]

\[
\eta_{th,f} = \frac{\tanh(m_e \cdot h_e)}{m_e \cdot h_e}
\]  \( (21) \)

where the parameter \( m_e \) is given by

\[
m_e = \left( \frac{2 \alpha_0}{\lambda_f \cdot t_e} \right)^{1/2}
\]  \( (22) \)

\[
t_e = \frac{t_f}{1 + \frac{t_f}{w_s}}
\]  \( (23) \)

\[
w_s = \frac{d_{r,f} \cdot \pi}{2 \cdot N_s}
\]  \( (24) \)

where \( w_s \) is average width of star shaped fin and \( t_e \) is equivalence thickness of fin, \( d_{r,f} \) is root diameter of star shaped fin. Due to the formation of vortices and boundary layers, the assumption of uniformity of heat transfer coefficient across the fin does not hold in actual situations. Experiments have shown lower heat transfer coefficients than predicted theoretically. In the present analysis for star type fins an empirical correction factor to the theoretical fin efficiency proposed by Weierman \[11\] for serrated fins is used.

\[
E = 0.9 + 0.1 \cdot \eta_{th}
\]  \( (25) \)
Efficiency of serrated fins is defined acc. equation (21). Numerical pressure drop in tube bundle is calculated from the next equation

$$\Delta p = p_{in} - p_{out}$$

(26)

where $p_{in}$ and $p_{out}$ is the mass weighted average pressure at inlet and outlet of tube bundle (Figure 2).

6. Validation with literature results for annular fins

For the validation of selected turbulence models, a comparison of the $Nu$ and $Eu$ obtained by heat calculation for the described model and calculation by correlation with Briggs [12], VDI-Warmeatlas [13], Ward and Young [3], Schmidt [8] and PFR [14]. In Figure 4 and Figure 5 there is a comparison of the calculation results of the analysed model with annular fins and the results obtained by correlations from the literature.

![Figure 4. Calculated $Nu$ and the results from literature for the annular fins.](image)

The values of $Nu$ numbers obtained on the SST $k-\omega$ turbulence model show max. deviation of 13 % according to the reference correlations which can be evaluated very good (Figure 4). $Eu$ values for the SST $k-\omega$ turbulence model show slightly greater deviation from literature. The deviations are higher for lower $Re$ values and up to 25 % (Figure 5). Also, the values of $Eu$ are less than by correlations. From the previous diagrams, it can be seen that the SST $k-\omega$ model of turbulence in the observed area of $Re$ numbers shows a better match to the results of literature than the $k-\varepsilon$ Realizable model. In accordance with the previously discussed for further comparison of the annular fins and the new type of star shaped fins, the SST $k-\omega$ turbulence model will be selected.

![Figure 5. Calculated $Eu$ and the results from literature for annular fins.](image)
7. Comparison of calculation results for annular fins and star shaped fins

The numerical results of the calculation based on the data given in Table 2 and Table 3 can be graphically displayed. As can be seen from Figure 6 $Nu$ number is greater than 13 % for star shaped fins compared to $Nu$ number for annular fins. The difference is more significant for higher $Re$ numbers. In the analysis of finned heat exchanger with segmented fins Hofmann [15] stated the average increase in the $Nu$ number of 22 % relative to the finned exchanger with circular annular fins. In our case, increase in $Nu$ number is higher and it is around 26 %. Figure 7 show that $Eu$ number is approximately equal for both fin types. Even for the star shaped fin $Eu$ decreases a little with its value. Although the fin shape varies the net free area in a tube row is nearly the same in both cases. Thus, a significant difference among pressure drop is not expected. The efficiency of the star shaped fins is greater than the efficiency of the annular fins (Figure 8), which means that there is a slight difference in surface

![Figure 6. Computational Nu results for annular and star shaped fins.](image6)

![Figure 7. Computational Eu results for annular and star shaped fins.](image7)

![Figure 8. The average efficiency of the fins.](image8)
temperature between the root of the fins and the top of the fins. In addition it should be noted that the mass of the star shaped fins is 43.5 % less than the mass of annular fins or mass reduction of the exchanger bundle is 23.8 %. The heat flux is greater for the star shaped fins for 35 % at $Re = 2255$ and 39.3 % at $Re = 11608$ (Figure 9).

8. Conclusion
Comparison was made between the $Nu$ and $Eu$ numbers for round fins and star shaped fins. The star shaped fins showed better results in terms of increasing the $Nu$ number to 13 % and heat flux up to 39.3 % while simultaneously reducing the weight of the exchange surfaces by 23.8 %.

References
[1] Hashizume K and Matsue T 1999 Fin Efficiency of Serrated Fins: Part 1. Analysis of Theoretical Fin Efficiency and Experimental Results, *Heat Transfer-Asian Research* **28**(6) 528-540
[2] Kays W M and London A L 1964 *Compact Heat Exchangers*, McGraw-Hill
[3] Ward D J and Young E H 1959 Heat Transfer and Pressure Drop of Air in Forced Convection Across Triangular Pitch Banks of Finned Tubes, *Chemical Engineering Progress Symposium Series* **55**(29) 37-44
[4] Stasiulevičius J and Skrinska A 1988 *Heat Transfer of Finned Tube Bundles in Crossflow*, Hemisphere Publishing
[5] Anoop B, Balaji C and Velusamy K 2015 A Characteristic Correlation for Heat Transfer over Serrated Finned Tubes, *Annals of Nuclear Energy* **85** 1052-1065
[6] Briggs D E and Young E H 1963 Convection Heat Transfer and Pressure Drop of Air Flowing Across Triangular Pitch Banks of Finned Tubes, *Chemical Engineering Progress Symposium Series* **59**(41) 1-10
[7] Schmidt T E 1966 Verbesserte Methoden zur Bestimmung des Wärmeaustausches an berippten Flächen, *Kältetechnik-Klimatisierung* **18** 135-138
[8] Ward D J and Young E H 1959 Heat Transfer and Pressure Drop of Air in Forced Convection Across Triangular Pitch Banks of Finned Tubes, *Chemical Engineering Progress Symposium Series* **55**(29) 37-44
[9] Nir A 1991 Heat Transfer and Friction Factor Correlations for Cross Flow Over Staggered Finned Tube Banks, *Heat Transfer Engineering* **12**(1) 43-58
[10] Lokman H M and Fdhila R B 2015 Literature Review of accelerated CFD Simulation Methods towards Online Application, *Energy Procedia* **75** 3307-3314
[11] Weierman C 1976 Correlations Aase the Selection of Finned Tubes, *The Oil and Gas Journal* **74**(36) 94-100
[12] Briggs D E and Young E H 1963 Convection Heat Transfer and Pressure Drop of Air Flowing Across Triangular Pitch Banks of Finned Tubes, *Chemie Ingenieur Technik* **33**(6) 431-438
[13] VDI-Wärmeatlas 2000, Berechnungsläfter feur den Wärmeübergang, Springer
[14] Dry Cooling Tower Program 1976 *Heat Transfer and Pressure Drop Characteristics of Dry Tower Extended Surfaces*, PFR Engineering Systems Inc
[15] Hofmann R, Frasz F and Ponweiser K, 2007 Heat Transfer and Pressure Drop Performance Comparison of Finned-Tube Bundles in Forced Convection, *WSEAS Transactions on Heat and Mass Transfer* **4**(2) 72-88