FLYWHEEL: JURNAL TEKNIK MESIN UNTIRTA

Homepagejurnal: http://jurnal.untirta.ac.id/index.php/jwl

Numerical Analysis of the Effect of Serrated Fin to the Heat Transfer in the Condenser

Lohdy Diana[1*], Febrian Kusumawardani[1], Wahyu Nur Fadilah[1], Muhammad Nabiel Ariansyah[1], Muhammad Farid Irfianto[1]

[1]*Program Studi Sistem Pembangkit Energi, Politeknik Elektronika Negeri Surabaya, Surabaya 60111, Indonesia.*

*Corresponding author: *lohdydiana@pens.ac.id*

ARTICLE INFO

Received 30/07/2021
revision 25/10/2021
accepted 20/11/2021
Available online 26/10/2021

ABSTRACT

Heat transfer efficiency improvements have been conducted in various methods. Increasing surface area by installing serrated fin will increase exchanger efficiency. This research uses the numerical method of CFD 2D to compare tube heat transfer without and with a serrated fin on 7 m/s and 9 m/s Velocity. A serrated fin with an outer diameter of 0,03175 m has 0° and 30 segments per period. Give inlet parameter 350.15 °K, resulting in outlet serrated fin tube temperature lower than the annular tube (without serrated fin). Simulation Serrated fin tube an inlet velocity of 7 m/s resulting outlet temperature of 343.2 °K, lower than tube simulation which produces outlet temperature of 344.53 °K.

Keywords: *Condenser, CFD, Shell and Tube, Fin, Serrated Fin, Annulus Tube*

1. INTRODUCTION

The condenser condenses steam output from a low-pressure turbine into a condensate water steam turbine usually uses a shell and tube condenser, consisting of a collection pipe located inside a shell (Figure 1).

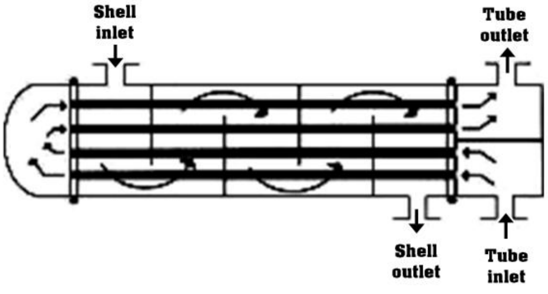

Figure 1. Shell and Tube Heat Exchanger [1].

Condenser performance degradation potential is caused by fouling, tube design, and tube arrangement, which is less precise [1].

One solution to improve condenser performance is by adding fin at the outer tube that aims to enlarge the heat transfer surface, increase fluid turbulence, and improve heat transfer rate [2]. An addition of a fin will make a flowing contour that occurs in the shell side randomly [3]. Fin tube consists of several kinds: annular, studded, plain, serrated, and others. Various types of the fin (Figure 2) can be affixed on the outer side of the tube with some methods, such as tension winding using adhesive bonding, soldered, brazing, welding, or extrusion.

Previous research has compared characteristics of serrated fin-tube and plain tube (Figure 3) using numerical methods of *Computational Fluid Dynamics* (CFD).

The result is known that serrated fin has better heat transfer characteristics due to turbulence of flow around tube increased compared with plain tube [4]. Researched fin tube with the annular tube to determine how much fin addition influenced heat transfer performance. This study contributes to the investigated flow phenomenon across tubes which is not possible experimentally.

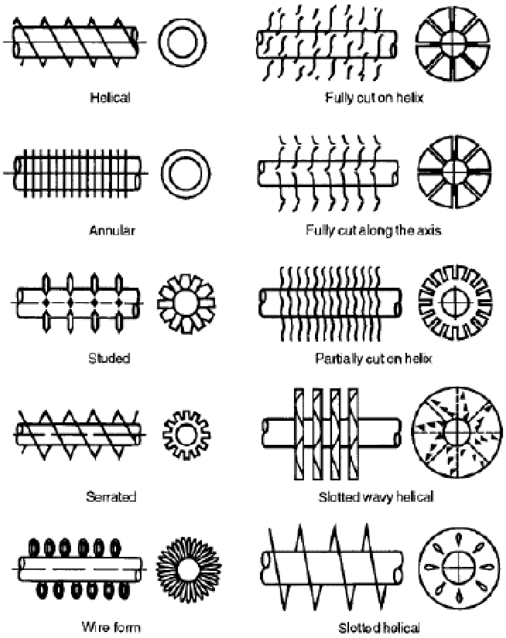

Figure 2. Type Fin Tube

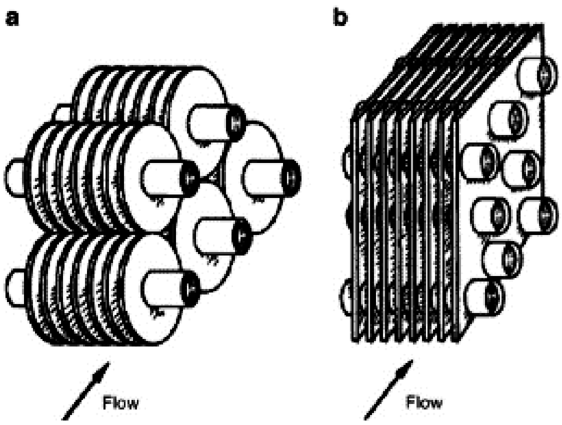

Figure 3. Fin Tube Exchanger Configuration. (a) Individually Fnned Tubes. (b) Flat Fin with Continuous Form On an Array of Tubes [5]

Experiments were carried out at Reynolds numbers of 3000–13,500 at turbulent flow regime [6]. Higher heat flux is observed at the shell's inlet due to two reasons. Firstly due to cross-flow and secondly, to a higher temperature difference between tubes and shell-side fluid [7]. Segmental baffle forces liquid in a Zigzag flow, improves heat transfer and high-pressure drop and increases fouling resistance. Helical Baffle effectively increases heat transfer performance [8]. Heat flux variation in condenser tubes is more complex than a simple polynomial curve fit [9]. The natural convection air stream mixture significantly influenced Condensation heat transfer coefficients on a vertical tube [10]. An increase in the difference between steam and tube wall temperature hinders the mean within-tube condensation heat transfer coefficient [11].

Experiment results show that smaller baffle spacing results in higher thermal effectiveness [12]. NTU and effectiveness are influenced significantly by injected airflow, shell-side flow, and coil-side flow rates without a noticeable temperature difference [13]. Total condenser cost is nearly independent of the tube diameter. The total cost depends weakly on brine velocity and more strongly on brine flow rate [14]. Water mass flow rate will come from a compromise between complete wettability of heat transfer geometry and limited water pumping consumptions [15]. A numerical simulation was presented to investigate and visualize flow characteristics of reflux condensation in core tubes related to the air-cooling condenser. The simulation had been performed in a flat tube under a saturation pressure of 13 kPa [17]. Flow patterns of condensation were identified to use relevant correlations for heat transfer and pressure drop [16]. Uneven distribution of internal temperature and airflow field is considered essential defects of *Λ-frame air-cooled condenser* (ΛACC) in a power plant [18].

2. METHODOLOGY

The author researched by using the numerical methods of CFD to determine the effect of the serrated fin on the heat transfer.

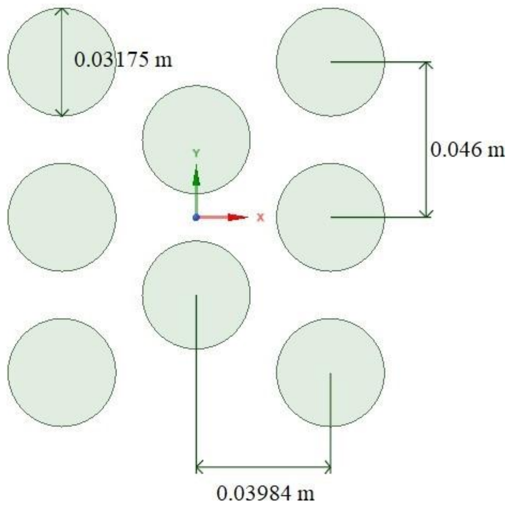

Figure 4. Design of Annulus Tube

The simulation is performed in dimensionless 2D by comparing the heat transfer between the annular tube (Figure 4) and serrated fin tube (Figure 5) at a different velocity, 7 m/s, and 9 m/s. From this research, qualitative and quantitative data are obtained.

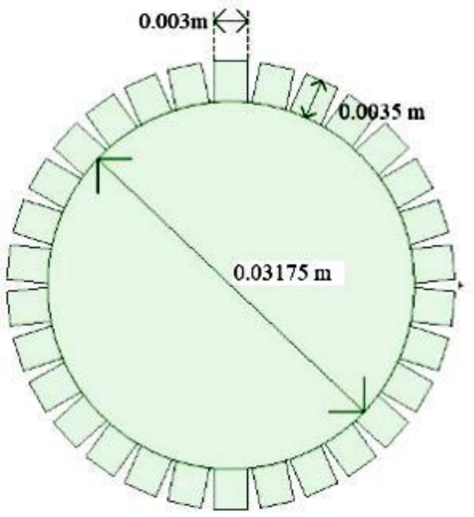

Figure 5. Design of Serrated Tube

2.1. Tube Design

Making design was done in SpaceClaim with parameters as in Table 1, using fins with 0° angle [19].

Table 1. Parameter Design of Geometry Serrated Fin Tube

Design	Design Parameter	
	Value	Unit
Tube OD	0,03175	m
Fin High	0,0035	m
Fin Angle	0	°
Fin Width	0,003	
Tube Arrangement	Staggered	
Tube Bank Angle	60	°
Transversal Pitch	0,046	m
Longitudinal Picth	0,03984	m

2.2. Meshing

Meshing is the numeration of geometry into small parts that will be done calculations for computing. Tightly meshing will produce a value close to accuracy.

Since there are significant conditions gradation around the tube, such as decreased temperature, increased velocity, meshing around the tube area is done very tightly. Meshing type simulation used *Quadrilateral-Map*.

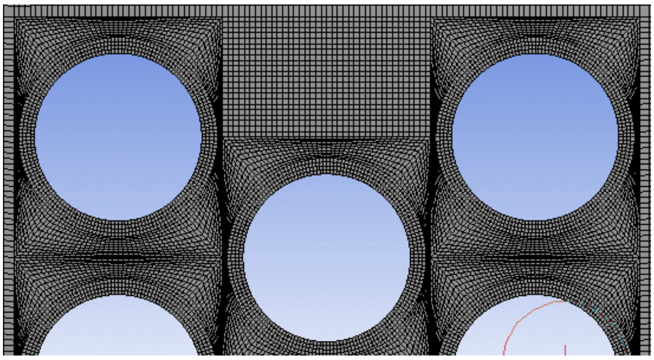

Figure 6. Details of Meshing Annulus Tube

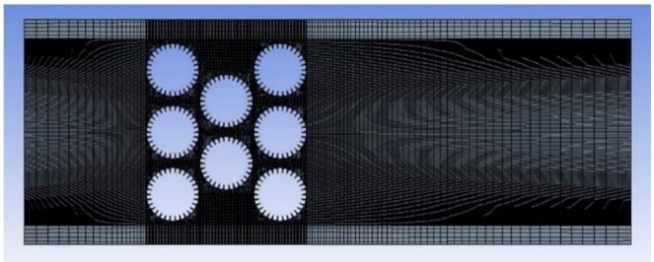

Figure 7. Meshing Serrated Fin Tube

Results of meshing on simulation can be seen in Fig. 6 and Fig. 7, While parameters and quality of meshing can be seen in Table 2.

Table 2. Meshing Parameter

Tube Type	Meshing Parameter			
	Element Mesh	Nodes	Orthogonal Quality	Skewness Quality
Annulus Tube	50704	51960	0,92225	0,17515
Serrated Tube	40043	42527	0,9859	6,62E-02

2.3 Processing

In this stage, the processing is done by settings solver model, viscous models, determination type of material, determination of boundary conditions, control & monitoring conditions, and initial conditions. The solver model enables energy equation to support completion of heat transfer, which uses the equation of energy transport (Eq.1).

$$\frac{\partial(\rho E)}{\partial t} + \nabla(E+\rho) = \nabla \cdot \left[k_{eff}\nabla T - \sum_j h_j \boldsymbol{J}_j + \tau_{eff} \cdot \boldsymbol{V} \right] + S_h \qquad (1)$$

While for turbulence modeling of selected k-RNG (Renormalization Group) for support of turbulence in the outer side of the tube is using transport equation 3.

$$\frac{\partial}{\partial t}(\rho k) + \frac{\partial}{\partial xi}(\rho k u_i) = \frac{\partial}{\partial xj}\left[\left(\mu + \frac{\mu_t}{\sigma_k}\right)\frac{\partial k}{\partial xj}\right] + P_k - \rho\varepsilon \qquad (2)$$

$$\frac{\partial}{\partial t}(\rho\epsilon)+\frac{\partial}{\partial x_i}(\rho\epsilon u_i)=\frac{\partial}{\partial x_j}\left[\left(\mu+\frac{\mu_t}{\sigma_\epsilon}\right)\frac{\partial\epsilon}{\partial x_j}\right]+C_{1\epsilon}\frac{\epsilon}{k}P_k-C_{2\epsilon}^{*}\rho\frac{\epsilon^2}{k} \quad (3)$$

This simulation uses two materials; one is water vapor, which flows on the shell side, other is titanium which is used as tube and fin material. Properties of water vapor and titanium are analyzed based on the average temperature of the fluid within the shell. Next, the boundary condition is determined in this simulation are shown in Table 3.

Table 3. Parameter Boundary Conditions

Boundary Condition	Boundary Conditions Parameter	
	Type	Parameter
Inlet	Velocity	v = 7 & 9 m/s T = 350,15° K
Outlet	Outflow	
Tube	Wall	T = 315,13° K
Fin	Wall	T = 315,9° K
Wall type	Symmetry	

Equation 4 is used to get the value of the temperature of the surface of the tube.

$$T_w=\frac{h_i t_{ave}+h_o\left({}^{D_o}/_{D_i}\right)T_{ave}}{h_i+h_o\left({}^{D_o}/_{D_i}\right)} \quad (4)$$

Equation 5 is used to get the value of the temperature of the surface of the fin.

$$T_{wtd}=\frac{h_i\eta_w t_{ave}+\left[h_i(1-\eta_w)+h_o\eta_w\left({}^{A_{Tot}}/_{A_i}\right)\right]T_{ave}}{h_i+h_o\eta_w\left({}^{A_{Tot}}/_{A_i}\right)} \quad (5)$$

$$\eta_w=\frac{A+\eta_f A_{fins}}{mb_c} \quad (6)$$

$$\eta_f=\frac{\tanh(mb_c)}{mb_c} \quad (7)$$

Furthermore, the solution method used is SIMPLEC and spatial discretization on second-order upwind for all parameters. Value 10 – 4 is set in the residual setting element to improve the accuracy of the simulation. Last initialization is done using the hybrid method before finally starting iteration to get a convergent result in controlled conditions.

2.4 Post-Processing

Results of numerical simulations of velocity and temperature distribution will be analyzed qualitatively in image form. Moreover, to determine the effect of serrated fin against heat transfer characteristics, such as *Re* and *Nu* quantitatively presented in bar chart form.

3. RESULTS AND DISCUSSION

The research analysis results are completed using numerical method computationally with fluid mechanics theory and heat transfer.

3.1. Comparison of Velocity Distribution for Annulus and Serrated Fin

Figure 8 obtained qualitative data in contour image form. The red color and blue color represent the highest velocity and lowest velocity. Flow velocity image contour through tube bank is taken at a velocity of 7 m/s, both for simulation of the annular tube or serrated fin tube.

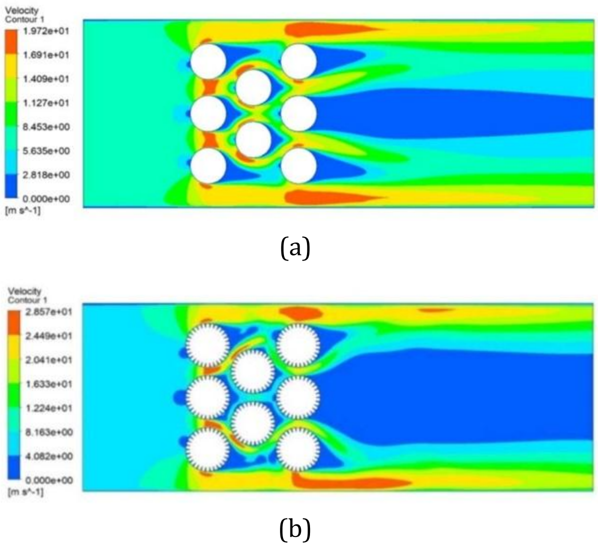

Figure 8. Comparison of Velocity Distribution (a) Annulus and (b) Serrated Fin Tube

On the simulation of an annular or serrated fin tube, the maximum fluid velocity is obtained at the first transverse rows area due to a narrowing area. While on the second and third parts of transverse rows has been a decreased velocity caused by colliding with the tube wall [5]. The blue color behind the first tube line, second or third, shows a shortage momentum area caused by separating fluid flow passed through a pipe and creates a gap.

Figure 9 shows the resulting *Vmax value* between the annular tube and serrated fin tube quantitatively. Serrated fin addition produces a *Vmax* of 13,1 m/s, which is more significant than the annular tube, which only produces a *Vmax* value at 11,97 m/s.

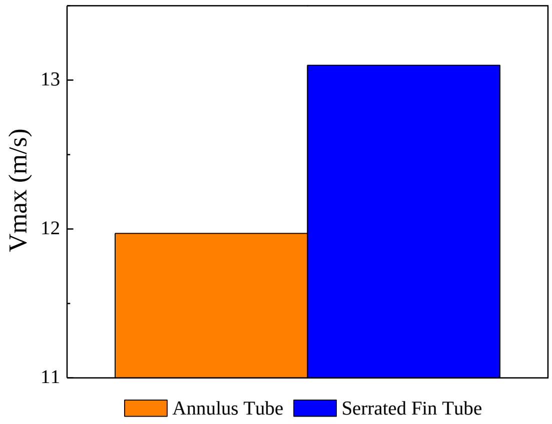

Figure 9. *Vmax* Comparison Chart for Annulus and Serrated Fin Tube

A high value of *Vmax* of serrated fin tube in simulation due to the existence of fin on the outside of the tube will enlarge the heat transfer surface and narrow distance of transverse and longitudinal between tubes, causing heat transfer to be better [2]. The existence of serrated fin will also randomize fluid flow, which causes a more excellent *Re*-value. More random or turbulent the fluid flow, the heat transfer that occurs would be better.

3.2 Comparison of Temperature Distribution for Annulus and Serrated Fin Tube

Figure 10 obtained qualitative temperature data in image contour form with indicator red representing the highest temperature and blue color showing the lowest temperature.

Image of temperature contours that pass through tube bank is taken at a velocity of 7 m/s, both on simulation of the annular tube or serrated fin tube. In a simulation, inlet temperature parameters input is 350,15 °K with uniform temperature conditions for either the annular tube or the serrated fin tube simulation. While water cooling, the temperature is low within the tube, enabling a decrease of temperature at the outlet side because there was a heat transfer around the tube bank. Heat will flow from high temperatures heading to low temperatures [2].

An annular tube is dominated by yellow color on the outlet, indicating that wasted fluid temperature is still high. Different from qualitative data serrated fin tube which is dominated by light green color on the outlet side, which fluid temperature lower indicates that heat more transfer around tube bank [3].

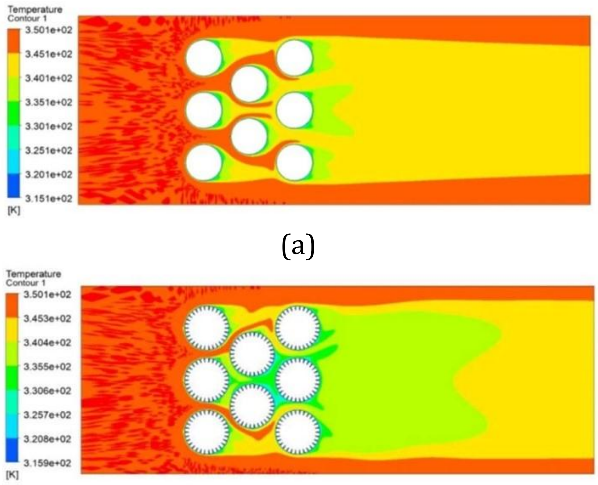

Figure 10. Comparison of Temperature Distribution (a) Annulus and (b) Serrated Fin Tube.

Figure 11 obtained quantitative data outlet temperature of the annular tube is 344,53 °K, with 5,62 °K differences from inlet temperature. While outlet temperature serrated fin tube is 343,20 °K, difference 6,95 °K from inlet temperature. It means that the installation of a serrated fin tube makes heat around the tube bank better transferred. The surface of the serrated fin will make heat transfer more extensive compared to the annular tube. The greater area of the heat transfer surface, the better heat transfer occurs [2].

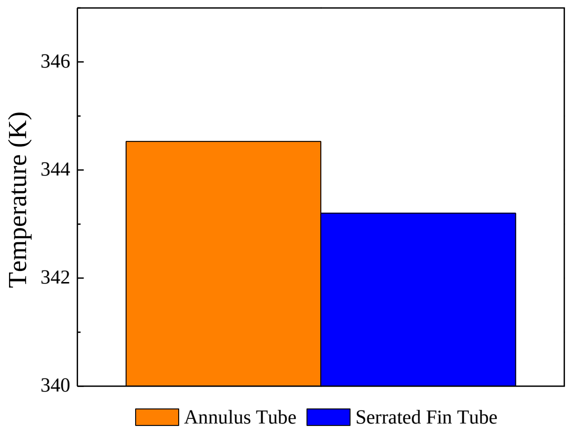

Figure 11. Temperature Comparison Chart for Annulus and Serrated Fin Tube.

The outlet temperature of the annular tube is 344,53 °K, which means it has a 5,62 °K difference from inlet temperature. While outlet temperature serrated fin tube is 343,20 °K, difference 6,95 °K from inlet temperature. It means serrated fin tube installation makes heat transfer around the tube bank better because the serrated fin will make the surface of heat transfer more extensive than the annular tube. With a greater area of the heat transfer surface, better heat transfer occurs [2].

3.3 Analysis of Effect of Velocity on Reynold Numbers for Annulus and Serrated Fin Tube

Figure 11 obtained Reynold Number value (*Re*-value) of the annular tube or serrated fin tube at velocity 7 m/s and 9 m/s to determine the effect of velocity changes against *Re* to determine the type of fluid flow that occurs.

Re-value is also defined as the ratio between inertia force with viscosity force in boundary layer velocity (Eq. 8). Low *Re*-value will not cause significant inertia force compared to viscosity force. At the same time, the high *Re*-value has high inertial force, which has more significance than viscosity force. The high *Re*-value will make the fluid boundary layer thinner because depressed by inertia force.

The thinness boundary layer will increase the fluid velocity (Eq.9), making the heat transfer process faster. The Re value is also dependent on the *Vmax* value. More excellent maximum velocity value then *Re*-value will be even greater.

$$ReD = \frac{\rho \cdot V_{max} \cdot d}{\mu} \tag{8}$$

$$V_{max} = \frac{ST}{2(SD - D)} \tag{9}$$

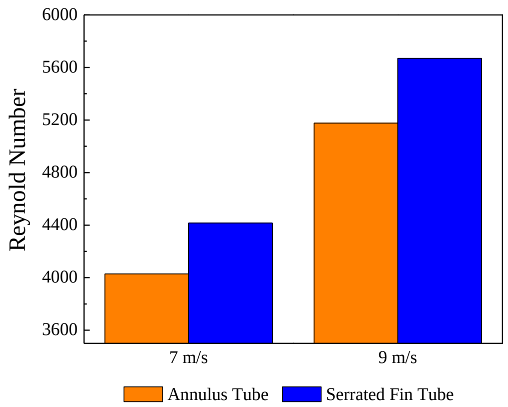

Figure 12. Reynold Number Comparison Chart for Annulus and Serrated Fin Tube at 7 m/s and 9 m/s Velocity.

Figure 12 obtained *Re*-value in the simulation of annular and serrated fin tube. Serrated fin tube generates large *Re*-value, each worth 4415,97 at a velocity of 7 m/s and 5669,34 at 9 m/s. In comparison, the annular tube produces *Re*-value of only 4029,3 at 7 m/s and 5176,23 at a velocity of 9 m/s.

The high *Re*-value of the serrated fin tube shows that adding a serrated fin on the outer tube will make fluid flow more random. When inlet velocity increases, the value of *Vmax* will be more excellent, also with *Re*-value.

3.4 Analysis of Effect of Velocity Changes on Nusselt Numbers for Annulus and Serrated Fin Tube

Figure 13 shows the correlation between *Nu*-value and velocity value. A *Nu*-value is a number ratio of heat transfer by convection and conduction on the fluid boundary condition.

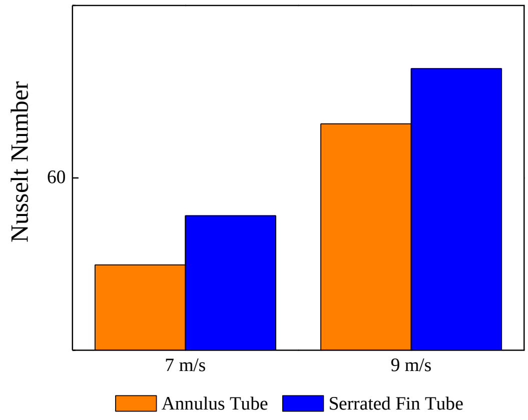

Figure 13. *Nu* Comparison Chart for Annulus and Serrated Fin Tube at 7 m/s and 9 m/s Velocity

Nu-value is also influenced by *Prandtl Number* (Eq. 10). *Prandtl Number* is defined as the ratio of diffusivity of momentum to diffusivity thermal. In contrast, thermal conductivity is defined as heat transfer rate by conduction through the cross-section area of the material. The value of C_1 is shown in Figure 14.

$$N_L \geq 10$$

$$\overline{Nu_D} = 1{,}13 C_1 Re_{D,max}^{m} Pr^{1/3} \tag{10}$$

$$Pr \geq 10$$

Nu-value of the annular tube only of 54,95 at velocity 7 m/s and 63,13. At a velocity of 9 m/s. In comparison, *the* *Nu*-value of serrated fin tube is 57,8 at velocity 7 m/s and increased to 66,34 at velocity 9 m/s. Increasing fluid velocity will increase the value of *Re* because the thickness of the boundary layer is getting thinner; the Fluid boundary depletion layer will cause *Nu*-value to increase, and vice versa [20]. Therefore, increasing *Nu*-value will also make better heat transfer. The serrated fin outside the tube will produce a large *Nu*-value, making heat transfer in the tube bank better.

	S_T/D							
	1.25		1.5		2.0		3.0	
S_L/D	C_1	m	C_1	m	C_1	m	C_1	m
Aligned								
1.25	0.348	0.592	0.275	0.608	0.100	0.704	0.0633	0.752
1.50	0.367	0.586	0.250	0.620	0.101	0.702	0.0678	0.744
2.00	0.418	0.570	0.299	0.602	0.229	0.632	0.198	0.648
3.00	0.290	0.601	0.357	0.584	0.374	0.581	0.286	0.608
Staggered								
0.600	—	—	—	—	—	—	0.213	0.636
0.900	—	—	—	—	0.446	0.571	0.401	0.581
1.000	—	—	0.497	0.558	—	—	—	—
1.125	—	—	—	—	0.478	0.565	0.518	0.560
1.250	0.518	0.556	0.505	0.554	0.519	0.556	0.522	0.562
1.500	0.451	0.568	0.460	0.562	0.452	0.568	0.488	0.568
2.000	0.404	0.572	0.416	0.568	0.482	0.556	0.449	0.570
3.000	0.310	0.592	0.356	0.580	0.440	0.562	0.428	0.574

Figure 14. Constant Value C_1 and m [1]

3.5 Analysis of Effect of Velocity Changes on Convection Coefficient for Annulus and Serrated Fin Tube

Theoretically, a more significant value convection coefficient means better heat transfer. Two types of convection occur, natural convection and forced convection. Natural convection occurs if there is no fluid velocity affects the heat transfer process. Forced convection occurs when the heat transfer is influenced by fluid velocity occurs in the vicinity.

Heat transfer by convection occurs because water vapor with a higher temperature releases heat to the fin walls tube. The convection coefficient value is also affected by *Re*-value. The greater *Re*-value, the more excellent process of convection occurs. The convection coefficient is also affected by the *Prandtl Number* and thermal conductivity of water vapor.

$$H_{kforced} = \frac{(0{,}4 \cdot Re^{0{,}5}) + (0{,}06 \cdot Re^{0{,}5}) \cdot Pr^{0{,}4} K^{0{,}25}}{D} \qquad (11)$$

Figure 15 shows the convection coefficient value of serrated fin-tube and annular tube. Compared to the annular tube, the addition of serrated fin resulting a better convection coefficient value of heat transfer. At a velocity of 7 m/s and 9 m/s. At velocity 7 m/s, convection coefficient Heat transfer of serrated fin tube is 883,41 and increased to 1000,96 at velocity 9 m/s. While in simulation, the annular tube produces a convection heat transfer coefficient is 843,8 at velocity 7 m/s and increased to 956,44 at velocity 9 m/s. A serrated fin on the outer side of the tube can increase heat transfer occurrence, which is proven by increasing the heat transfer convection coefficient.

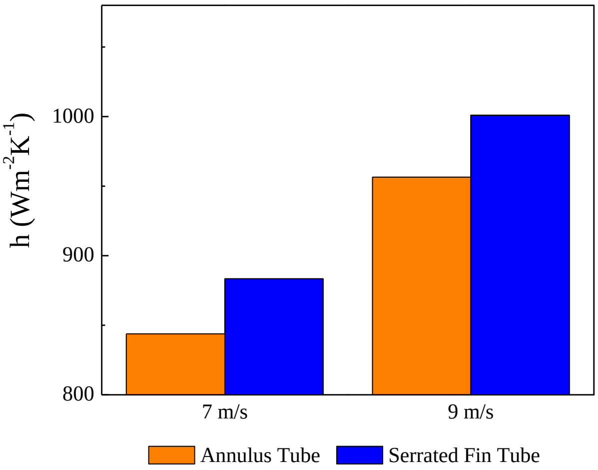

Figure 15. Comparison of Convection Heat Transfer Coefficients for Annulus and Serrated Fin Tube at Velocity 7 m/s and 9 m/s.

4. CONCLUSION

Adding a serrated fin on the outer tube can improve the heat transfer around tube banks, which causes a lower outlet temperature. Inlet temperature 350,15 °K, at a velocity of 7 m/s of an annular tube in simulations, produces an outlet temperature of 344,53 °K, while a serrated fin tube simulation produces an outlet temperature of 343,2 °K.

A serrated fin tube can also increase *Re*-value, *Nu*-value, and convection coefficient heat transfer around the tube bank. The simulation of serrated fin tube at a velocity of 7 m/s produced a *Re*-value of 4415,97, *Nu*-value of 57,8, and convection heat transfer of 883,41. While the annular tube at a velocity of 7 m/s only produces a *Re*-value of 4029,3 *Nu*-value of 54,95 and coefficient of convection heat transfer of 843,84. To further improve the heat transfer performance of serrated fins, further research could expand the heat transfer surface to increase the number of segments in each period.

ACKNOWLEDGMENTS

Authors would like to convey their appreciation to Politeknik Elektronika Negeri Surabaya.

REFERENCES

1. R.W. Serth. Process Heat Transfer Principles and Applications. Elsevier; 2017.

2. Incropera FP, Dewitt DP. Fundamental of Heat and Mass Transfer seventh edition. John Wiley & Sons; 2011.
3. Nilasari D. Analysis of Pitch Tube Angle Arrangement Effect in Fluid Flow Characteristics and Heat Transfer. Institut Teknologi Sepuluh Nopember; 2017.
4. Putra RA. Fin Shape Analysis For Thermal Efficiency On High Pressure Economizer Heat Recovery Steam Generator PLTGU PT. PJB-UP Gresik. Institut Teknologi Sepuluh Nopember; 2017.
5. Shah RK, Sekuli DP. Selection of Heat Exchangers and Their Components. Fundamentals of Heat Exchanger Design. John Wiley & Sons; 2003. 673-734 p.
6. Pourahmad S, Pesteei SM. Effectiveness-NTU analyses in a double tube heat exchanger equipped with wavy strip considering various angles. Energy Convers Manag. 2016;123:462–9.
7. Rehman UU. Heat transfer optimization of shell-and-tube heat exchanger through CFD studies. 2011.
8. Arjun KS, Gopu KB. Design of Shell and Tube Heat Exchanger Using Computational Fluid Dynamics Tools. 2014;3(7):8–16.
9. Li J De. CFD simulation of water vapour condensation in the presence of non-condensable gas in vertical cylindrical condensers. Int J Heat Mass Transf. 2013;57(2):708–21.
10. Lee YG, Jang YJ, Choi DJ. An experimental study of air–steam condensation on the exterior surface of a vertical tube under natural convection conditions. Int J Heat Mass Transf. 2017;104:1034–47.
11. Zhu X, Chen S, Shen S, Ni S, Shi X, Qiu Q. Experimental study on the heat and mass transfer characteristics of air-water two-phase flow in an evaporative condenser with a horizontal elliptical tube bundle. Appl Therm Eng. 2020;168(September 2019):114825.
12. Zebua MA, Ambarita H. Numerical simulation of the effect of baffle spacing to the effectiveness of a shell and tube heat exchanger. IOP Conf Ser Mater Sci Eng. 2018;420(1).
13. Baqir AS, Mahood HB, Kareem AR. Optimisation and evaluation of NTU and effectiveness of a helical coil tube heat exchanger with air injection. Therm Sci Eng Prog. 2019;14:100420.
14. Karameldin A, Atomic E, Authority E. Optimum Design of a Horizontal Condenser. 2015;(February).
15. Fiorentino M, Starace G. Numerical and Experimental Performance Analysis of Evaporative Condensers. Energy Procedia. 2016;101(September):26–33.
16. Hua N, Xi H, Xu RJ, Chen Y, Wang HS. Numerical simulation of multi-pass parallel flow condensers with liquid-vapor separation. Int J Heat Mass Transf. 2019;142:118469.
17. Deng H, Liu J, Yang T, Wu S. Numerical study and visualization on flow characteristics of reflux condensation in air-cooled condenser. Appl Therm Eng. 2019;148:1310–23.
18. Zhou Y, Cheng YL, Zhang N, Shi HB. Numerical simulation study of novel air-cooled condenser with lateral air supply. Case Stud Therm Eng. 2019;13(September 2018):100354.
19. 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;31(8-9):1393–401.
20. Kumar A, Layek A. Numerical Analysis for Nusselt Number and Heat Transfer Augmentation on Solar Air Heater Roughened With Square Rib Roughness on the Absorber Plate. Agribus J. 2019;13(1):31–6.