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STUDY OF ADDITIONAL FIN TO INCREASE EFFICIENCY OF SUPERHEATER AT HEAT RECOVERY STEAM GENERATOR

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Abstract

Power plants are part of industrial facilities used to produce and generate electricity from various power sources; one of those is PLTGU (Pembangkit Listrik Tenaga Gas dan Uap or Gas and Steam Power Plant). PLTGU is a combined cycle between PLTG and PLTU. It is necessary to achieve a high-capacity target for the PLTGU to increase the generator's efficiency. One way to increase the efficiency of gas and steam power plants is by optimizing heat transfer in the Heat Recovery Steam Generator (HRSG). HRSG has several modules such as superheater, evaporator, economizer, and preheater. One that plays an essential role in absorbed high-temperature steam from the gas turbine is the superheater. The function of the superheater is to utilize the heat from the flue gas to reheat the fluid needed for the steam turbine. In this research, improvements of superheater were made with several fin variations at HRSG PLTGU. Variable of superheater refer to data on the layout of the HRSG PLTGU. Autodesk Inventor is used to modeling flue gas domain, tube, and fin. Additional of the fin has the purpose of optimizing heat transfer distribution in steam through a tube, such as an outlet temperature and efficiency of steam in tube superheater. The use of CFD (Computational Fluid Dynamic) with ANSYS Fluent could use to determine the temperature distribution of the superheater. The most optimal efficiency and outlet temperature of variation fin is the annular fin variation compared to the rectangular and straight fin variations.

Keywords: *Modelling, Simulating, Fin Efficiency, Staggered*

1. INTRODUCTION

The need for electricity in the country will always peak with economic growth and community needs in all sectors. In Indonesia, with the increase in electricity demand (KWh/capita) that occurs every year and starts at the beginning of 2020, the need will increase to 1,142 KWh/capita for national electricity consumption [1]. By optimizing the electricity in the country, economic development can become more stable, as well as being able to improve the quality of people's lives. This reason shows the importance of increasing the efficiency of each power plant, such as optimizing each component of the plant and utilizing high temperatures to produce more energy to reach the country's electricity needs. Using an efficient power plant is also expected to reduce pollution in various media by generating waste flue gas from a gas turbine.

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Chen et al. [2] explained that in increasing the production of electrical energy, the main components of the Gas and Steam Power Plant (Pembangkit Listrik Tenaga Gas dan Uap / PLTGU) must operate optimally. One of the main components of the PLTGU is the Heat Recovery Steam Generator (HRSG). Taler et al. [3] explained that HRSG is a boiler that utilizes residual heat energy from a gas turbine to heat air and water in each module and convert it into hot steam. Using HRSG increases efficiency because it utilizes exhaust gas from gas turbines as a heat source not to require fuel and air as heating [4]. Hennessey [5] studied several factors that affect the HRSG performance: the mass rate of steam and exhaust gases, the temperature of the exhaust gases and steam leaving the turbine, and the pressure and composition of the exhaust gases and water used in the HRSG. HRSG has several main components, superheaters, an evaporator, and an economizer. Superheater, evaporator, and economizer are tubes with a specific structure and number. One of the HRSG components that can significantly increase heat transfer efficiency is the superheater, where this component is vital and impacts the heat transfer process in HRSG. This explanation is based on the idea by Thulukkanam [6].

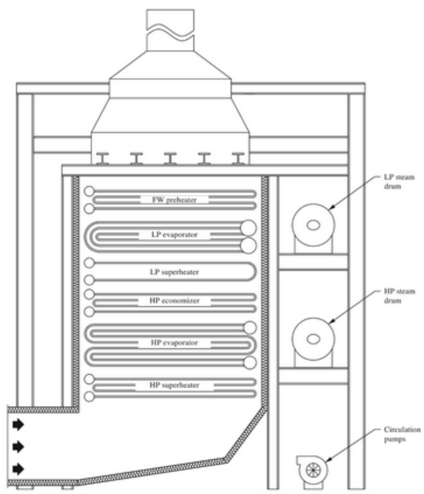


Figure 1. HRSG schematic vertical gas flow, horizontal tube, forced circulation [6]

Methods to increase heat transfer are divided into two types, active and passive. The active method requires additional equipment to improve the system's performance by additional fins, such as spiral fins, annular fins, flat fins, or wavy fins, to increase the overall heat transfer percentage. In comparison, the passive method is the opposite of the active method. This method does not require additional equipment to improve the system's heat transfer Khan et al. [7]. One of the passive methods is staggered arrangement to utilize tubes to produce a flow that can optimize heat transfer between two different fluids. Wei et al. [8] conducted experimental research on water flow in a tube with various turns and under turbulent flow conditions. The heat transfer coefficient and friction factor increase with a constant Reynolds number. Khan et al. and Chen et al. [7, 9] also researched 3D tube geometries with staggered arrangements. This research shows that when the air passes through the tube, the rotating flow begins to occur rapidly, leading to a decrease in velocity suddenly and increase energy. The airstream flows perpendicular to the central axis of the tube and continues to flow along the tube and adds a heat transfer effect.

Sertkaya and Sari [10] explained that the superheater geometry is a critical factor in utilizing flue gas optimally. One of the basic geometries in optimizing heat transfer between flue gas and tube is fins. The benefits of using fins cause heat transfer greater than without using fins. The heat exchanger of the fin depends on the number, the height, the tube length, and the number row of the fin. Therefore, HRSG can be utilized flue gas with high temperatures optimally. In this research, the author will analyze and model a superheater with various fins that a PLTGU will use to increase the Heat Recovery Steam Generator (HRSG).

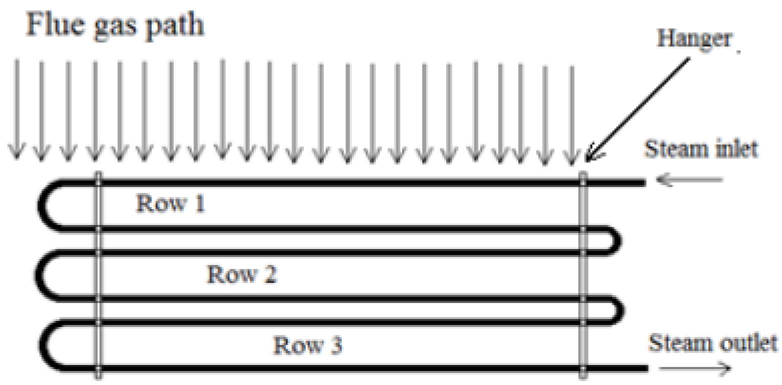


Figure 2. Schematic of using a convection superheater [11]

2. METHODOLOGY

Research for superheater modeling and simulation in this study carried out two software. Modelling the superheater, tube arrangement, and fin geometry using the Autodesk Inventor Professional 2019 software. The modeling results with predetermined dimensions are then converted to a specific format imported into the ANSYS Fluent 20 R2 software. The fluids used are steam and flue gas. Flue gas has a velocity of 31.6718 m/s and a temperature of 533 ^oC. Steam superheater has a velocity of 0.87 m/s and a temperature of 466 ^oC. The geometric configuration of the superheater tube is presented in Table 1.

Table 1. Superheater tube datasheet

DATA TUBE SUPERHEATER		
PROPERTIES	**VALUE**	**UNIT**
Outer diatemer tube	31,8	mm
Tube thickness	3	mm
Tube length	16386	mm
Density	7675	Kg/m^3
Specific heat	720	J/kg.K
Thermal conductivity	33,9	w/m.K
Tube arrangement	Staggered	
Material	10CrMo910 (alloy steel)	

Table 2. Flue gas specification data

FLUE GAS					
SUHU (^OC)	**SPESIFIC HEAT (J/Kg.K)**	**VISCOSITY (Kg/m-s)**	**THERMAL CONDUCTIVITY (w/mc)**	**DENSITY (Kg/m^3)**	**MASS FLOW INLET (Kg/s)**
533 (Inlet)	1100.32	3.7154.10-5	0.05757	0.4323	416,67
507	1094.2	3.6372.10-5	0.05634	0.4469	
500	1092.5	3.6159.10-5	0.056	0.451	
450	1080	3.4606.10-5	0.05355	0.482	

400	1068	3.2999.10-5	0.05094	0.518

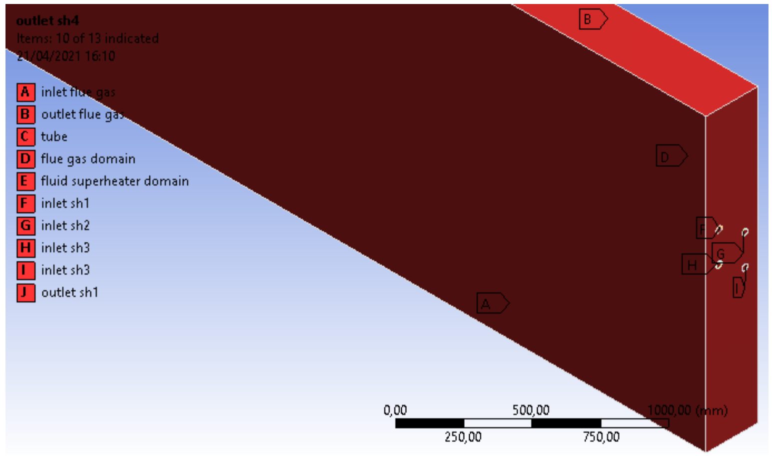


Figure 3. Name selection for the fluid domain

In this research, some data and references are used to build the geometry and be data in the simulation process. The superheater uses a steam fluid with a temperature of 466 ^{o}C, pressure of 79 kg/cm^{2}abs, specific heat 2522.6 J/kg.K, viscosity 27327.7 kg/ms, thermal conductivity 0.0696 w/mc, density 24,524 kg/m^3, and velocity 0.87 m/s. This research focuses on fin geometry variations and the temperature distribution along the fin and tube to affect the efficiency of the superheater. The modeling at this stage simplifies the tube dimensions to 100 cm. The tube material used is 10CrMo910 alloy steel with a density of 7675 kg/m^3, specific heat J/kg.K, and thermal conductivity of 33.9 w/mc.

Figure 4. Geometry straight fin, rectangular fin, and annular fin tube superheater

The modeling of the three fin variations is based on the research by Sertkaya and Sari [10]. According to Fuentes and Saenz [12], in designing dimensions of fins, it is recommended that the ratio of the distance between fins and the height should be less than 1.5. The width dimensions are 2 mm, and the distance between the fins is 15 mm. To ensure temperature distribution on the fin surface is the same and the results are valid, the area of all fin variations has approximately the same. The straight fin tube has 47886,69 mm^2 areas, the rectangular fin tube has 48927,89 mm^2 areas, and the annular fin tube has 48429,41 mm^2 areas.

The turbulent model used in this simulation is reliable K-epsilon and enhances wall treatment because the flue gas passing through the tube is turbulent flow. The solution for pressure and velocity uses a semi-implicit (SIMPLE) algorithm. To the convection estimation, a second-order upwind differential scheme is applied. The boundary condition is the velocity for each inlet and outlet tube. Then the solution method is SIMPLE (Semi-implicit method pressure-linked equations). The tube wall boundary is a thermal convection condition due to heat transfer between the flue gas and steam in the tube [13].

2.1. Mathematic Model

The velocity of the flue gas entering the HRSG inlet duct is uniform throughout the tube can be written as:

$$\overset{o}{m} = \rho \,.\, \mathrm{A} \,.\, \mathrm{V} \tag{1}$$

where ρ= Density of fluid (kg/m^3), A= Cross-sectional area (m^2), m= Mass flow rate (kg/s). Reynolds number and heat transfer coefficient are obtained from Eq. (2) and (3). So, the heat transfer coefficient will be the convection condition input to the simulation setup. Meanwhile, the efficiency of the superheater is obtained from Eq. (4) [14].

$$\mathrm{Re}_{\mathrm{D}} = \frac{\rho.\mathrm{V.D}}{\mu} = \frac{\mathrm{V.D}}{\upsilon} \tag{2}$$

$$\overline{\mathrm{Nu}}_{\mathrm{D}} = \frac{\bar{\mathrm{h}}.D}{k} = \mathrm{C}\,\mathrm{Re}_{\mathrm{D}}^{\mathrm{m}}\,\mathrm{Pr}^{1/3} \tag{3}$$

$$\eta = \frac{T_{c,o} - T_{c,i}}{T_{h,i} - T_{c,i}} \tag{4}$$

With Re= Reynold number, u= Kinematic viscosity (m^2/s), D= Tube diameter (mm), υ= Fluid dynamic viscosity (Ns-s/m^2), V= Velocity of flue gas (m/s), h = Heat transfer coefficient (w/m^2.K), Nu= Nusselt number, C= Parameter constant C, m= Parameter constant m, Pr= Prantl number, η= Efficiency, T= Operating temperature (^{o}C), $T_{c,o}$ = steam superheater outlet temperature (^{o}C), $T_{c,i}$ = steam superheater inlet temperature (^{o}C), and $T_{h,i}$ = flue gas inlet temperature (^{o}C).

The meshing process is carried out using a mesh application in the ANSYS software. Figure 5 shows that the meshing method for steam is hex-dominant, then the tetrahedron for tube and flue gas domain. The meshing process greatly affects the final result of a simulation. A grid independence test is carried out to ensure that the simulation results do not depend on the amount of meshing process. The results of the grid independence test show that for mesh sizes above 1617583, there is no significant change in temperature, so to get good quality and quiet time, a mesh with a total of a bit more 1617583 elements is selected. The mesh quality results show an average value of 0.23572 (excellent) for skewness and 0.7688 (very good) for orthogonal quality. Variations in the type of meshing and sizing used in the meshing process for each variation are the same to ensure that the results obtained are better and valid.

The mesh type is based on the fin part's shape, which is quite complex, so that the tube, fin, and flue gas domains use the tetrahedron type. By choosing the hex-dominant type on the fluid tube, the tube temperature could get more accurate results. The meshing performed on each variation is the same, namely face sizing, method meshing, and edge sizing. The input mesh size also has the same value to ensure that all interpretations have approximately the same quality.

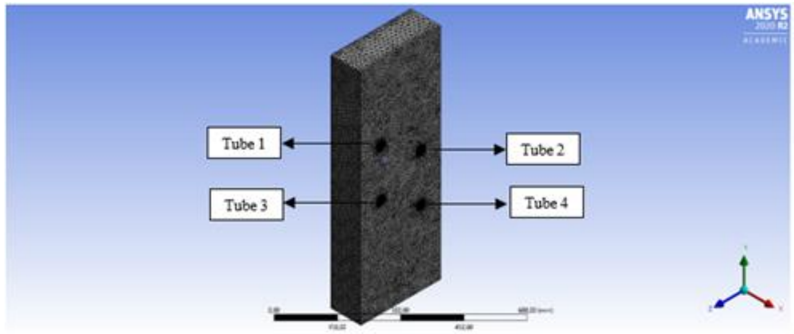


Figure 5. The meshing of geometry flue gas tube without fin

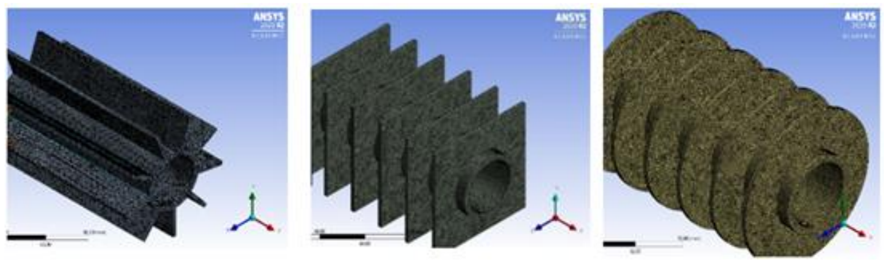


Figure 6. Meshing geometry of straight finned, rectangular fin, and annular fin

3. RESULT

The tube has a diameter of 31.8 mm, a thickness of 3 mm with a staggered tube arrangement, as shown in Figure 5. The distance between the pipes is 134.25 mm in the horizontal direction and 79 mm in the vertical direction. The heat transfer coefficient value of 76.83 w/m^{2}K is obtained from calculations with Eq. (2) and (3), where the constant C and m parameters value is obtained from the Reynold numbers. Prandtl number can be found in Table A.15 [15].

In the tube without fins, the temperature below the tube wall is higher than the upper wall because the flue gas flows to the bottom tube first, then the temperature along the tube slowly increases from the end of the inlet to the outlet. Figure 7 also shows that steam in the tube absorbed heat from the flue gas by convection through the tube so that the steam temperature increases slowly along with the flow. Flue gas has a velocity of 31.6718 m/s along the tube, which only flows in the y-axis direction. Flue gas density will decrease slowly with the relative change in pressure and rate along the tube. Because the fin's height is the same as tube diameter, the heat effect can be distributed along the fin to the pipe. Each tube has increased the average temperature from 466 ^{o}C to 468.250 ^{o}C.

The velocity contour shows that the maximum velocity occurs in the tube wall, especially tubes 3 and 4. This occasion happened because the flue gas flow in direct contact with the bottom tube. Then, because the tube is cylindrical with a circular cross-section, the flow follows the curvature shape of the tube, increasing velocity gradually. The tube arrangement without fins and other variations is staggered. This tube arrangement aims to produce a flow that can optimize heat transfer between two separate tubes. From Figure 8, Bula et al. [16] experimented with air passes through the tube, and the rotating flow occurred rapidly. Turbulent flow occurred, which then leads to a sudden decrease in velocity and an increase in energy. The airstream flows perpendicular to the central axis of the tube and continues to flow along the tube, then adds the heat transfer effect.

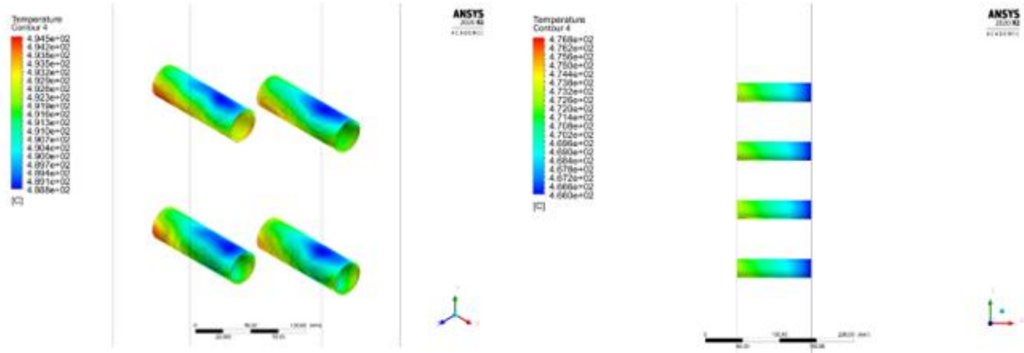


Figure 7. Temperature distribution along the tube without fins and steam superheater

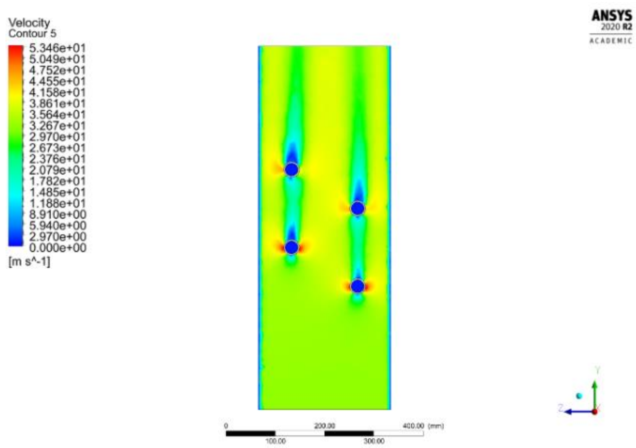


Figure 8: Distribution of flue gas velocity along the tube without fin

3.1. Straight Fin

The temperature distribution of the straight fin variation is shown in Figure 9. It can be seen that the flue gas flow has a maximum temperature at the edge of the fin. This process occurred because the flue gas flows through the straight fins tube, and steam in the tube absorbs heat from the flue gas by convection process. Therefore, the steam temperature in the tube slowly increases. In Figure 9, the maximum temperature distribution is at the bottom fin of the tube. At the same time, the upper fin tube temperature distribution does not receive heat optimally. Then, the lowest temperature is around the wall and tube area. Because the flue gas flow cannot reach the entire fin surface and the flue gas has a high velocity, so the flow cannot pass through the lower and upper fins due to a significant pressure drop, and the velocity drops flow moves away from the fin.

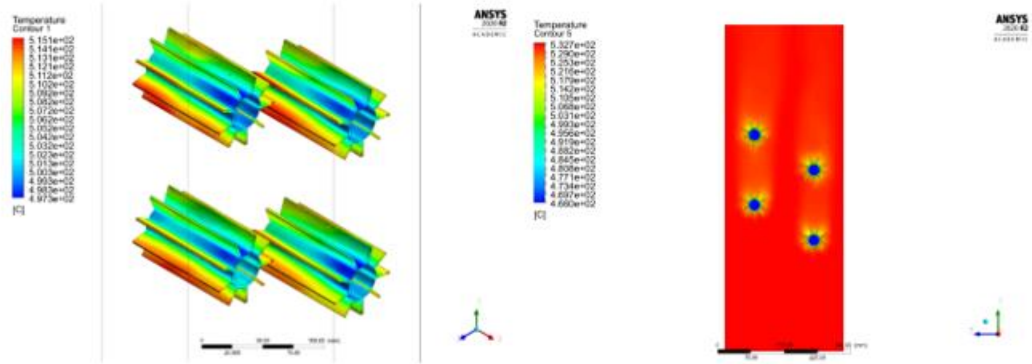


Figure 9. Temperature distribution along straight fin-tube and steam

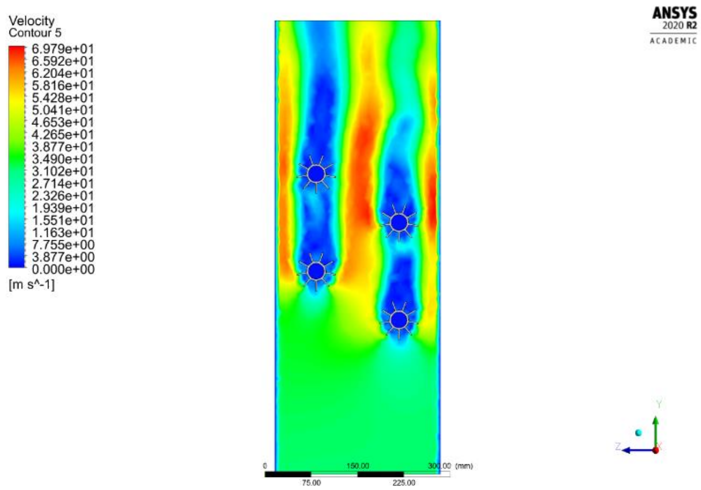


Figure 10. Flue gas velocity distribution along with a straight fin tube

The maximum velocity contour is near the tube wall, as same as with the tube without fin. The curvature causes maximum velocity on the cylindrical tube. In straight fin, flue gas flow distribution is not optimal compared to other variations because the flue gas heat cannot reach every corner of the fin, especially the fin surface close to the tube. Each tube increases the average temperature from 466 ºC to 469 ºC.

3.2. Rectangular Fin

In Figure 11, it can be seen that the flue gas flows to the bottom tube first and causes optimal use of flue gas heat in the two bottom tubes. Moreover, in Figure 12, the maximum velocity contour is near the tube wall the same as other fin tube variations. The smaller the distance between the fins can increase turbulent flow effectively around the tube wall and enhance heat transfer performance.

Figure 11 shows that the maximum temperature is at the end of the fin, especially at the corners of the fin. The heat from the flue gas stream is in direct contact with the fin to heat transfer by the convection process. However, the temperature distribution of the rectangular variation has a smaller value rather than the annular variation. In that case, the temperature distribution is lower than the annular fin because when flue gas flows along the fin, there are angles of 90^o on each side, or there is a sudden change in the area and formed a region called vortex region. It has meant that the gas cannot be utilized by the fin optimally because a high-pressure drop occurred around the corners of the fin. At the same time, the lowest temperature is around the wall and tube area. Each tube increases the average temperature from 466 ^{o}C to 469 ^{o}C.

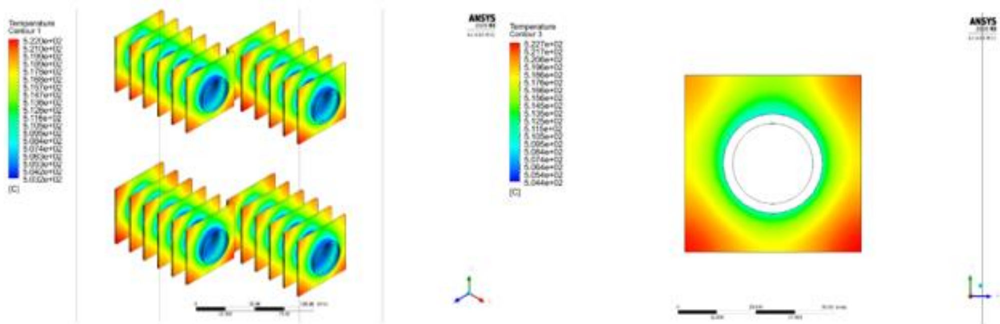


Figure 11. Temperature distribution along rectangular fin tube

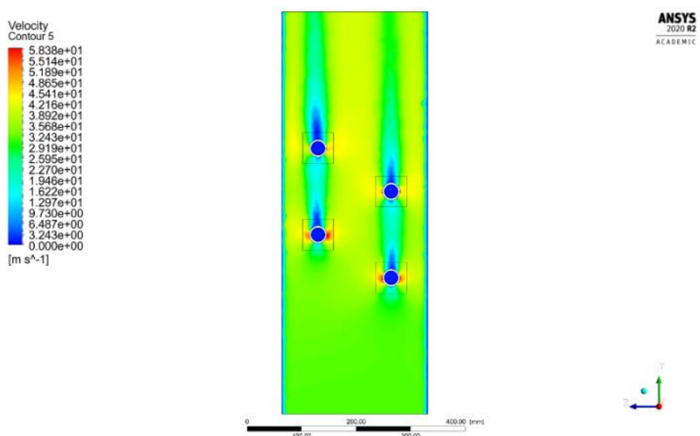


Figure 12. Flue gas velocity distribution along with a rectangular fin tube

3.3. Annular Fin

The flow characteristics in the tube are strongly influenced by the tube and fin geometry. The flow accelerates around the heated annular fin-tube and forms a wake region behind the tube with low velocity. The boundary layer above the hot tube increases heat around the tube. Then, convection heat transfer has the highest value in the edge part of the fin and the lowest in the wake region of the fin behind the tube from the flow direction. Because of this, the heat transfer coefficient along the fin is even. The streamlined annular tube flow pattern is shown in Figure 13. Similar to the temperature distribution in other variations, the maximum temperature is near the walls of tubes 3 and 4 because the flue gas flows to the bottom tube first. This maximum velocity is caused by the curvature of a tube, which is circular so that the flue gas flow has a smooth flowed type and increased velocity. Velocity near the tube wall has a higher value than in other areas.

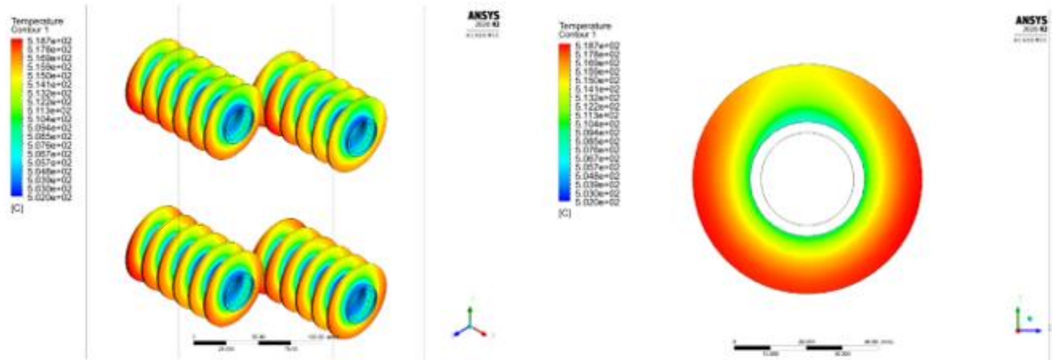


Figure 13. Temperature distribution along the annular fin tube

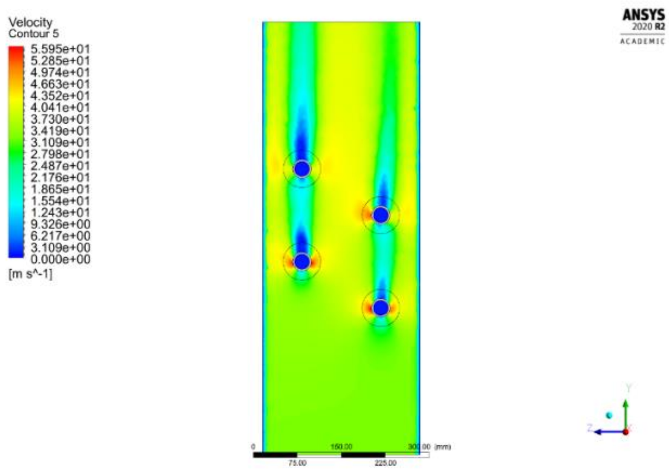
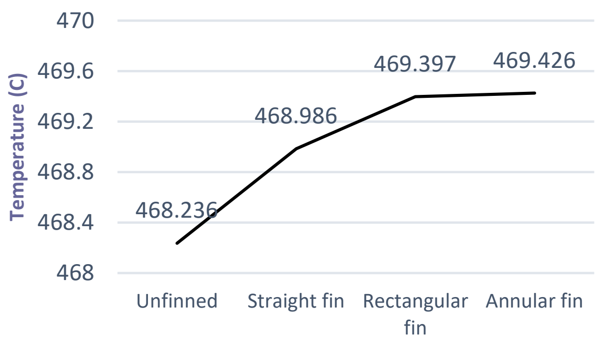


Figure 14. Distribution of flue gas velocity along with the annular fin tube

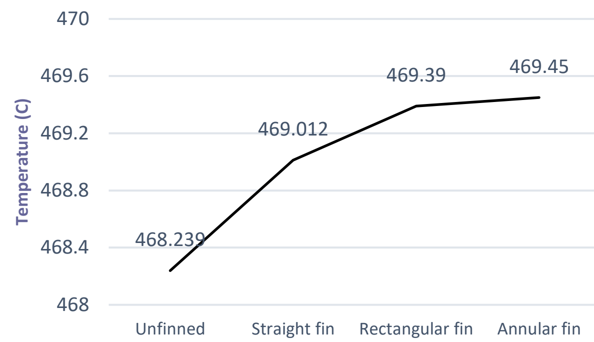
Figure 13 shows the temperature distribution on tube 1. The maximum temperature is at the bottom end of the fin. The heat from the flue gas flow is in direct contact with the fin to heat transfer by convection with the tube. At the same time, the lowest temperature is around the wall and tube area. The temperature distribution of the annular fin variation has a better value than the other variations. The annular fin geometry gives the advantage by distributing heat on the surface with a high-efficiency value. The fillet geometry of the annular fin causes the flue gas heat to have a high-velocity value on the sidewall of the tube, thus enhancing the performance of heat transfer. In addition, the annular fin area has a smaller area than the rectangular fin geometry, which is 48429.41 compared to 48927.89. However, with a reduction area and weight, the efficiency of the annular fin still has a good performance in the distribution heat along the tube and fin surface.

The temperature contour in Figure 13 shows the effect of the distance between two edges of fins on each tube. According to Bhale et al. [17], if the distance between the two tubes increases, the heat transfer performance also increases slightly by around 1-2%. There is a more extended contrast color in the area behind the annular fin tube from Figure 14. The existence of this gap can offset the effects of secondary flow in the area behind the tube. Each tube increases the average temperature from 466 ^oC to 469.5 ^oC.

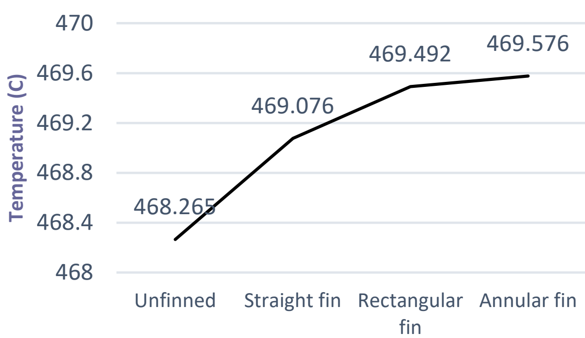
3.4. Graphic of Outlet Tube Temperature



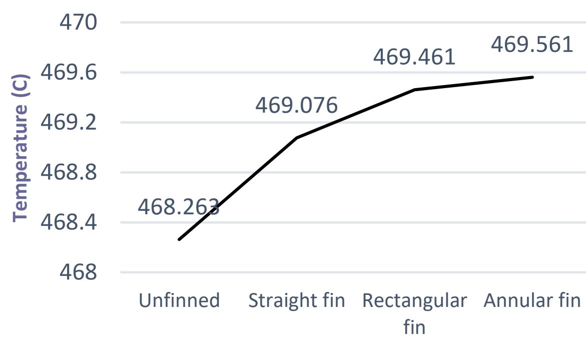
a. Graph of fin variations with the average temperature on tube 1



b. Graph of fin variations with the average temperature on tube 2



c. Graph of fin variations with the average temperature on tube 3



d. Graph of fin variations with the average temperature on tube 4

Figure 15. Graph of the average superheater tube temperature

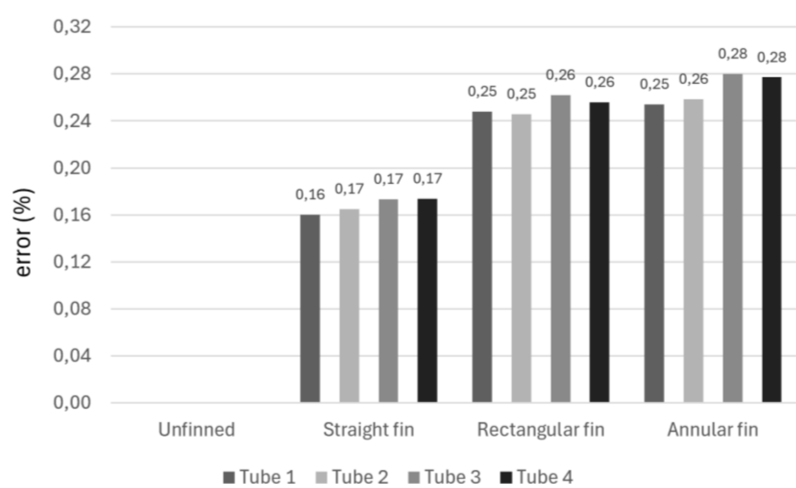


Figure 16. Graph of the average error superheater tube temperature compared to Unfinned

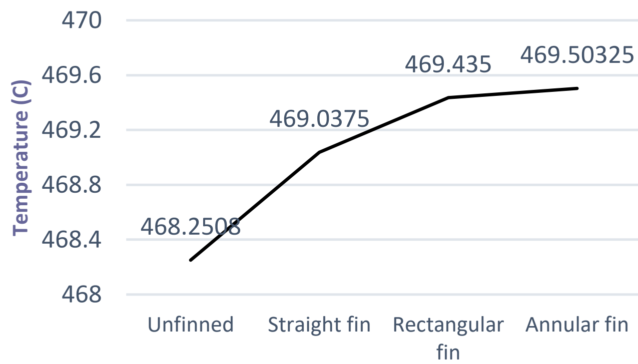


Figure 17. Graph of the average temperature fin variation

Figure 15, 16, and 17 explains that the steam outlet temperature slowly increases with more efficient fin variations, which means annular fin. The annular fin variation has a higher average outlet temperature than the other variations as seen in Figure 15. Figure 16 shows errors for each fin variations compared to unfinned model. The steam outlet temperature on tubes 1 and 2 has a lower value than the tube below it, namely tubes 3 and tube 4, because the flue gas heat is utilized by the tube below first before contact with tube upper.

3.5. Graph and Value of Fin Variation Efficiency

The overall efficiency calculation will use the average steam outlet temperature for each tube. By using Eq. (4), the efficiency of each tube geometry can be calculated. The efficiency of tubes without fin is 3.35% which will be a reference to see the difference in efficiency increase in each geometry of fin tube. The straight fin efficiency value is 4.48%, the rectangular efficiency value is 5.12%, and the annular fin efficiency value is 5.22 %.

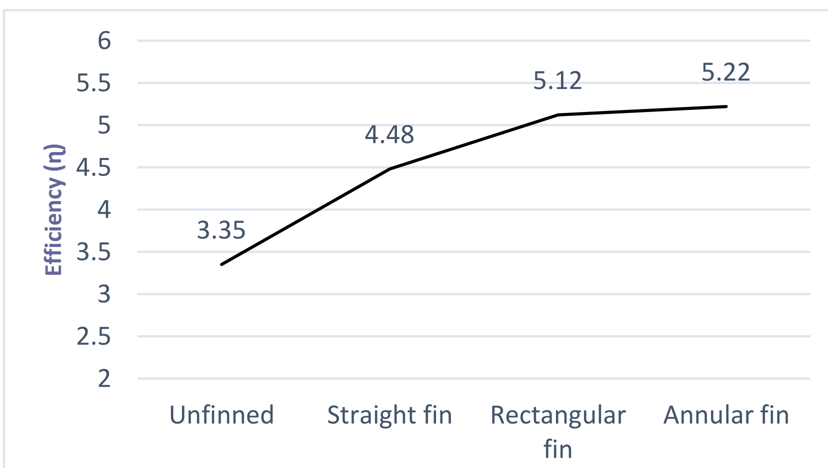


Figure 18. Efficiency graph of fin variation

In calculating the efficiency value above, the annular fin has a better efficiency value than other variations in the optimal utilization of flue gas heat around the tube. In Figure 18, The efficiency value of the fin variation is directly proportional to the value of the steam outlet temperature for each fin variation. This means that the higher the outlet temperature of steam, the higher the efficiency in utilizing flue gas heat by convection. From the results of the efficiency calculation above, Zhang et al. [18] concluded that the average heat transfer from tubes using fins can reach 3.1 greater than tubes without using fins.

4. CONCLUSION

From the fin variations results, with the area value of each fin approximately the same, the most efficient variation is an annular fin with an efficiency of 5.22% compared to a tube without a fin, which has an efficiency of efficiency 3.35%. The use of flue gas heat in steam through convection tubes depends on the fins' geometry. The heat optimization is indicated by the better value of the steam outlet temperature. The annular fin variation has a higher steam temperature than other variations, 469.503 ^{o}C. The fin geometry of the tube affects the heat transfer performance that occurs on the flue gas flow when passing through it, such as the size of the vortex area, the high intensity of turbulence, the magnitude of the pressure drop, and the temperature distribution on its surface. The annular fin has the highest performance in heat transfer. It has a small vortex area formed around the fin, turbulent flow occurs.

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