Journal of Sustainability for Energy
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Enhanced Performance of Structurally Optimized Plate-Fin Heat Exchangers Through Numerical Modeling of Heat Transfer and Pressure Drop

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Received: 04-21-2023 **Revised:** 05-25-2023 **Accepted:** 06-05-2023

Citation: A. H. Hasan, S. M. Banooni, and L. J. Habeeb, "Enhanced performance of structurally optimized Plate-Fin Heat Exchangers through numerical modeling of heat transfer and pressure drop," *J. Sustain. Energy*, vol. 2, no. 2, pp. 39–49, 2023. https://doi.org/10.56578/jse020201.

Abstract: This study aims to optimize the structure of compact Plate-Fin Heat Exchangers (PFHE) by incorporating corrugated fins and validating their improved performance through numerical modeling and simulation. The results provide valuable insights for refining application-specific design guidelines and enhancing the performance of PFHEs. Using Computational Fluid Dynamics (CFD), the PFHE geometry was created in SolidWorks and Ansys Fluent, with fins modeled in three layers inside the heat exchanger both with and without a cover. To investigate the fins' performance, flow field, and heat transfer, fin thickness, entry velocities, and locations of water and air were varied across three wavelengths (10, 20, and 30) during the numerical investigation. The analysis focused on the variations in pressure, temperature, and fluid velocity within the heat exchanger. Key findings include the observation that temperature distribution is influenced by the velocities of both water and air, with the upper layer experiencing a temperature increase due to the warm fluid stream, while the opposite effect is observed near the bottom layer. Furthermore, fluid temperature variation in the depth direction is attributed to conductive heat transfer through side plates and convective heat transfer to the surroundings. The outcomes of this study have the potential to reduce the pressure difference generated during heat exchange and increase the thermal efficiency of PFHEs.

Keywords: Heat transfer performance; Pressure drop; Plate-Fin Heat Exchanger (PFHE); Geometry; Computational Fluid Dynamics (CFD)

1 Introduction

Heat exchangers are a type of device used to transfer thermal energy across the impermeable wall between two or more kinds of fluids. Compact heat exchangers, such as Plate-Fin Heat Exchangers (PFHEs), are highly efficient due to their many merits, including the large heat transfer surface area per unit volume, high heat transfer coefficient, low weight, and reasonable cost, thus they are widely applied in air separation, syngas and petrochemical production, hydrogen and helium liquefiers, gas and oil processing, air conditioners, automobile radiators, aircraft secondary power systems, and other scenarios. Figuring out the performance characteristics of enhanced surfaces in compact PFHE, as well as the heat transfer, fluid flow behavior, and temperature distribution in the heat exchanger is conductive to assessing the overall efficiency and effectiveness of compact PFHEs with different fin geometries, understanding the mechanism of heat transfer, thereby giving more efficient and effective designs for various applications.

2 Literature Review

World field scholars have made numerous efforts to optimize the structure of PFHEs for applications in cryogenic fields, heavy-duty diesel engines, and high-temperature ceramics by employing algorithms and techniques such as $\mathcal{E}$-NTU, NSGA-II, PSO, CFD, and ANN under various fin configurations. For instance, Sanaye and Hajabdollahi [1] applied the $\mathcal{E}$-NTU method to the thermal modeling of tight heat exchangers and used the NSGA-II algorithm to optimize cost and efficiency. Rao and Patel [2] used particle swarm optimization to optimize cross-flow PFHE, whereas Hajabdollahi et al. [3] used CFD analysis and ANN to create a thermal model for compact heat exchangers.

The outlet temperature of the hot fluid of exchanger under wet conditions is lower than the value attained under dry conditions, according to the 3D distributed parameter model developed by Zhang et al. [4] for evaluating and predicting the behavior of PFHE in both wet and dry environments. Yousfei et al. [5] optimized a cross-flow PFHE using an imperialist competitive method, and Goyal et al. [6] modeled a multi-stream PFHE for cryogenic applications.

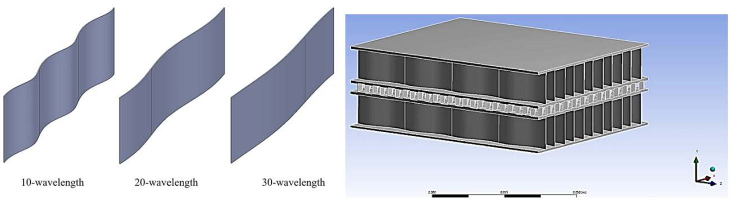
Some studies investigated the fluid flow and heat transfer characteristics in fin and tube heat exchangers by comparing plain and louvred fins; some studies explored novel fin designs and shapes for use in challenging environments; and some studies examined the correlation between friction factor, convective heat transfer coefficient, and j and f factors based on CFD simulations and experimental data. For instance, Feru et al. [7] built a two-phase, modular heat exchanger model for energy recovery in heavy-duty diesel engines. For high-temperature ceramic PFHEs, Nagarajan et al. [8] developed a unique fin arrangement using CFD analysis. Jeong et al. [9] designed a new PFHE for construction equipment in very dusty environments. Taler and Oclon [10] introduced a method for calculating thermal contact resistance in plain and tube fin exchangers, whereas Saad et al. [11] developed a friction factor correlation for microchannels with offset-strip fin geometry. Fernández-Seara et al. [12] experimentally analyzed a titanium-brazed PFHE with serrated fins, and established a correlation for the heat transfer coefficient. Yang and Li [13] investigated heat transfer coefficient and flow friction coefficient numerically for offset strip fins and established correlations for their prediction.

Other mentionable studies include: Carija et al. [14] compared fin and tube heat exchangers with flat and louvred fin exchangers and discovered that louvred heat exchangers have superior heat transfer characteristics and lower pressure drop. By analyzing fluid flow and heat transfer in finned tube heat exchangers with different tube shapes and fin types, Han et al. [15] pointed out that oval fin tubes not only reduce flow resistance and boost heat transfer capacity, but also improve the performance of fins; and it was discovered that for louvered fins, the heat transfer coefficient is more significant than the heat transfer area with regard to the heat transfer rate. Yeom et al. [16] investigated micro pin fin arrays in a constrained rectangular channel and concluded that the fluid dynamic effects around micro pin fins are more crucial for improving heat transfer than area growth. In their work, the micro pin-fin surface with a diameter of 400 mm and a height of 250 mm improved the heat transfer of plain surfaces by a maximum of 79 percent, authors focused on simulating heat exchanger with a specific size to attain the best algorithm to solve it without complicating the simulation process.

After carefully reviewing existing literatures, it's found that they generally tend to focus on designing heat exchangers experimentally without design originality in terms of dimensions or boundaries, therefore, despite all of these research progresses, this study employed FLUENT 2021 R1 for CFD analysis to figure out the effects of different fin boundary profiles and configurations, including fin height, fin spacing, and fin thickness, on the heat transfer performance. In this work, corrugated fins were innovatively added to PFHE and the impact of wave on the fluid bumping process was investigated based on fluid pressure and heat transfer pattern; the pressure, temperature, and velocity distributions within the heat exchanger under various operating conditions were attained with a special attention on how these parameters influence the overall heat transfer performance of PFHE. Specific research steps include the selection of appropriate numerical schemes, construction of turbulence models of PFHE, proposal of convergence criteria, simulation runs, post-processing, statistical analysis, as well as validation and verification using data from previous studies. In addition, several useful recommendations were proposed for research and practical applications of this new-type PFHE in the future. The newly developed PFHE can be used in sensitive and accurate electronic devices due to its small size and ability to transfer heat significantly. Research findings attained in this work can be used for design guideline improvements, heat exchanger design customization, surface performance and energy efficiency enhancement, cost savings, and boosting performance of compact PFHE in special application scenarios.

3 Modelling and Numerical Investigation

3.1 Geometry



Isometric view of heat exchanger fins — Geometry of heat exchanger

Figure 1. Heat exchanger geometry and fin isometry

PFHE with variable wavelengths (10, 20, and 30) was subjected to numerical investigation under conditions of different fin thickness, water and air entry velocities, and entry locations. Geometry of the target PFHE was created in SolidWorks and Ansys Fluent, with fins modeled in three layers inside the heat exchanger with and without a cover. Figure 1 depicts the heat exchanger geometry and fin isometry created between three layers within the heat exchanger; each layer is 19.6 mm in height and 53.64 mm in width, and the thickness of fins ranges from 0.2 to 0.4 mm.

3.2 Model Assumptions

Assumptions of the model include: three-dimensional steady laminar and turbulent flow, incompressible Newtonian fluid, constant inlet temperature, no radiation heat transfer, negligible buoyancy effect, and no heat generation within the heat exchanger. Table 1 shows part of the model specifications adopted in the simulation. The design process is a realistic application of different dimensions to obtain the best results through such parameters as wavelength, for example.

Table 1. Model specifications

Fin type	Wavy	Offset strip
Material	Aluminum	Aluminum
$\delta(\mathrm{mm})$	0.2	0.2
$\delta_{\mathrm{w}}(\mathrm{m})$	0.7	0.7
$\mathrm{Nf}(\mathrm{fpm})$	400	455
$\mathrm{b}(\mathrm{mm})$	11.3	3
landa (m)	10	2.5
Wavy amplitude (mm)	0.8	-

3.3 Meshing

The model domain was meshed into numerous small cells, and quality checks were performed to ensure solution accuracy. The minimum orthogonal quality should be greater than 0.01, with aspect ratio and skewness taken into account. The validation of meshing is critical in finite element analysis. In this study, four meshing cases were investigated, and the outlet temperature was calculated for each of them. Figure 2 shows a screenshot of meshed geometry of the target PFHE. Simulation was carried out in Ansys Fluent with two fluid types representing the heat exchanger core. Besides, model setting entailed defining auxiliary fluid paths, macros, core properties, and conditions. The model was divided into three cell zones: solid for fins and fluid for water and air. The air-to-water velocity ratio ranged between 0.1 and 0.3.

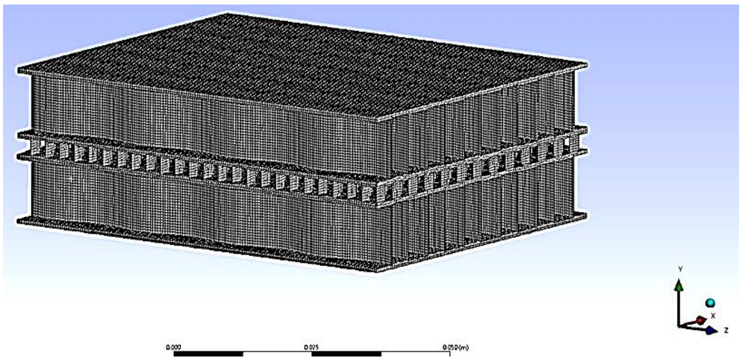


Figure 2. Meshed geometry

3.4 Model Governing Equations

Turbulence model equations, as well as continuity, momentum, and energy equations, were solved by Ansys Fluent; details of mass-averaged velocity, momentum equations, and energy equations are given below. For two- and three-phase fluidized beds, the Turbulence RNG mixture model was adopted, with constants such as turbulent dissipation rate, energy, surface tension, density, velocity, viscosity, and kinetic energy detailed below.

The continuity equation of the mixture is:

$$\frac{\partial}{\partial t}(\rho_m) + \nabla \cdot (\rho_m \vec{v}_m) = 0 \tag{1}$$

where, $\vec{v}_m$ is the mass-averaged velocity:

$$\vec{v}_m = \frac{\sum_{k=1}^{n} \alpha_k \rho_k \vec{v}_k}{\rho_m} \tag{2}$$

and ρ_m is the mixture density:

$$\rho_m = \sum_{k=1}^{n} \alpha_k \rho_k \tag{3}$$

where, α_k represents the volume fraction of phase.

By adding together the separate momentum equations for each phase, the equation of mixture could be created as:

$$\frac{\partial}{\partial t}\left(\rho_m \vec{v}_m\right) + \nabla \cdot \left(\rho_m \vec{v}_m \vec{v}_m\right) = -\nabla p + \nabla \cdot \left[\mu_m \left(\nabla \vec{v}_m + \nabla \vec{v}_m^T\right)\right] + \rho_m \vec{g} + \vec{F} - \nabla \cdot \left(\sum_{k=1}^{n} \alpha_k \rho_k \vec{v}_{dr,k} \vec{v}_{dr,k}\right) \tag{4}$$

where, n is the number of phases, $\vec{F}$ is the body force, and μ_m is the viscosity of mixture:

$$\mu_m = \sum_{k=1}^{n} \alpha_k \mu_k \tag{5}$$

$\vec{v}_{dr,k}$ is the drift velocity of secondary phase:

$$\vec{v}_{dr,k} = \vec{v}_k - \vec{v}_m \tag{6}$$

The energy equation of mixture can be written in the following form:

$$\frac{\partial}{\partial t} \sum_k \left(\alpha_k \rho_k E_k\right) + \nabla \cdot \sum_k \left(\alpha_k \vec{v}_k \left(\rho_k E_k + p\right)\right) = \nabla \cdot \left(k_{eff} \nabla T - \sum_k \sum_j h_{j,k} \vec{J}_{j,k} + \left(\bar{\tau}_{eff} \cdot \vec{v}\right)\right) + S_h \tag{7}$$

where, $h_{j,k}$ is the enthalpy of species j in phase k, $\vec{J}_{j,k}$ is the diffusive flux of species j in phase k, and k_{eff} is the effective conductivity, which can be calculated as:

$$k_{eff} = \sum \alpha_k \left(k_k + k_t\right) \tag{8}$$

where, k_t is the turbulent thermal conductivity defined according to the adopted turbulence model. Divergence solutions were avoided using the under-relaxation method given below:

$$\varphi_{\text{new}} = (1-\alpha)\varphi_{\text{previous}} + \alpha \varphi_{\text{calculated}} \tag{9}$$

where,

φ_{new} = The new under-relaxed value of φ.

$\varphi_{\text{previous}}$ = The value of φ attained from the previous iteration.

$\varphi_{\text{previous}}$ = The calculated value.

SolidWorks, Ansys Workbench, and Fluent Package were taken as processors, and the governing conservation equations were solved by the Finite Volume Method and the SIMPLE algorithm. The first fin thickness and wavelength were numerically analyzed under three air-to-water velocity ratios, and the Fluent Code, a commercial CFD program, was employed to predict the flow and heat transfer performance in the heat exchanger.

Validating and verifying simulation is necessary for ensuring accuracy and reliability and it contains several processes such as comparing simulation results with experimental data based on certain criteria, conducting grid independence studies, code verification, sensitivity analysis, as well as comparing with other numerical models, and literature review. These processes are critical for ensuring accurate and reliable simulation results.

The most important details are the quantification and comparison of effects of fin height, fin spacing, and fin thickness on heat exchanger performance, pressure drop across the heat exchanger, and heat exchanger effectiveness. These metrics provide insights into the efficiency of heat transfer under different configurations. The evaluation of pressure, temperature, and velocity distributions in the heat exchanger can be shown in contour plots or streamlines. Pressure evaluation helps to understand the pressure drop across the heat exchanger and the pressure losses at specific locations; temperature evaluation helps to figure out temperature distribution and heat transfer pattern; while velocity evaluation facilitates the visualization of velocity distribution, the identification of flow characteristics, and the impact of fin parameters.

4 Results and Discussion

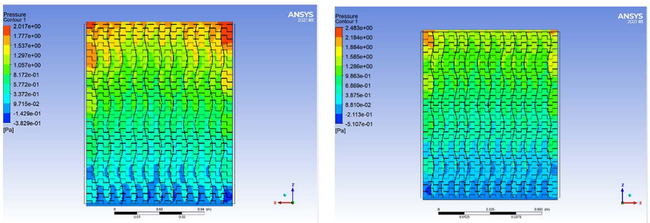
4.1 The Pressure Variation Inside the Heat Exchanger

The pressure contours inside the heat exchanger, as depicted in Figures 3 to 8, vary with air and water velocity and temperature. As the velocity of water increases from 0.1 m/s to 0.2 m/s, the pressure within the heat exchanger rises accordingly, and pressure drop is the same for all layers. As the temperature rises, the mass flow rate decreases; while as the velocity rises, both pressure drop and flow resistance increase, resulting in poor flow performance of thick fins.

In the pressure contours, high-pressure regions can be seen in front of the fins, low-pressure regions can be spotted on either side of the fins' foreparts because of the corrugated configuration of fins. In contrast to linear pressure curves of plain fins, the pressure curves of three corrugated fin configurations present wavy shapes. The pressure drop of cold fluid is greater than that of hot fluid, and both the overall pressure drop and the rate of pressure drop increase with load. The overall pressure drop value is a crucial factor for the design of PFHE.

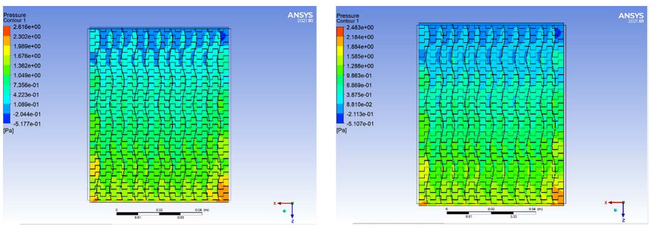
According to the results, increasing fin thickness can intensify disturbance effect and improve heat transfer performance, but it also has a quick negative impact on flow performance and a quick positive impact on flow resistance. Thus, as the fin thickens, the overall performance declines, and this effect exhibits an increasing trend.

These findings suggest that optimizing fin thickness and fluid velocity can significantly affect the design and performance of heat exchangers. Designers must find a balance between heat transfer efficiency and flow resistance to achieve the optimal performance of PFHE, and comparing these results to those attained in previous research is conductive to deepening understandings of heat exchanger behavior and informing future design optimizations.



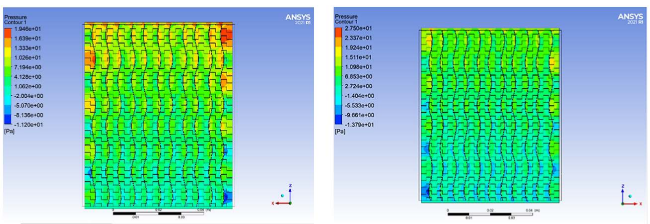
0.1 m/s water velocity 0.2 m/s water velocity

Figure 3. Heat exchanger pressure contours at 25°C water temperature, 1 m/s air velocity, 50°C air temperature, and different water velocities



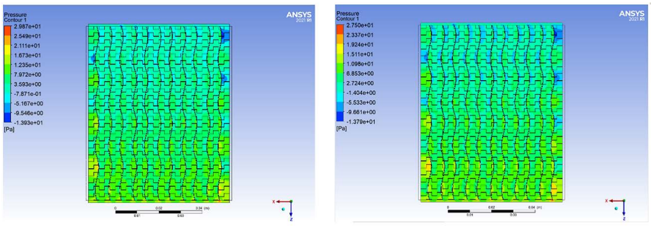
0.1 m/s water velocity 0.2 m/s water velocity

Figure 4. Heat exchanger pressure contours at 50°C water temperature, 1 m/s air velocity, 25°C air temperature, and various water velocities



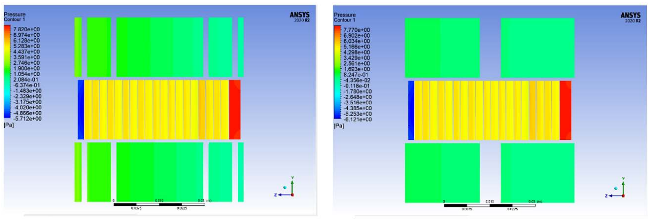
0.1 m/s water velocity 0.2 m/s water velocity

Figure 5. Heat exchanger pressure contours at 25°C water temperature, 5 m/s air velocity, 50°C air temperature, and various water velocities



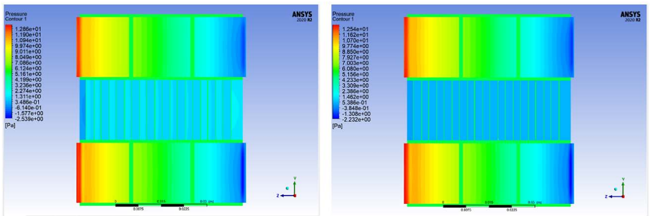
0.1 m/s water velocity 0.2 m/s water velocity

Figure 6. Heat exchanger pressure contours at 50°C water temperature, 5 m/s air velocity, 25°C air temperature, and various water velocities



0.1 air/water velocity 0.3 air/water velocity

Figure 7. Pressure contours of heat exchanger with external side air and water at the middle taken under the condition of 20 fins wavelength and 0.2 mm fin thickness



0.1 air/water velocity 0.3 air/water velocity

Figure 8. Pressure contours of heat exchanger with external side water and air at the middle taken under the condition of 20 fins wavelength and 0.2 mm fin thickness

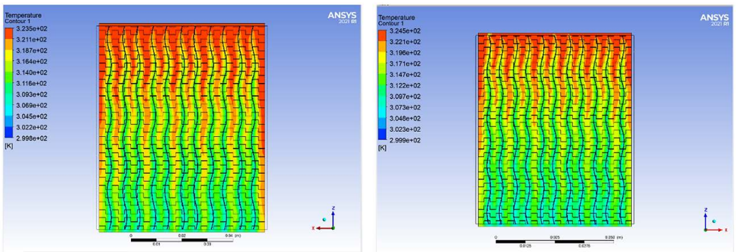
Results of pressure variation in heat exchanger are essential to understanding the flow performance and overall effectiveness of the system as they provide insights into the flow resistance experienced by the fluid as it passes through the heat exchanger, based on these results, regions with high-pressure gradients or local pressure losses could be identified. These information can instruct design and provide evidences for fin parameter selection, thereby optimizing flow performance while maintaining efficient heat transfer. Also, the study helps to identify specific operating conditions and fin configurations that can minimize pressure drop and improve flow performance, assisting in heat exchanger design optimization, energy efficiency enhancement, operational cost reduction, and overall system performance improvement.

4.2 The Temperature Variation Inside the Heat Exchanger

Figures 9 to 14 give the temperature contours of heat exchanger under different water velocities. The conductive heat transmission via side plates and the convective heat transfer to the surroundings caused temperature differences in the depth direction. Also, the convective transfer heat of solid matrix and fluid in channels caused a significant temperature difference in the Y-Z vertical cross-section. The heat exchanger efficiency enhances as the number of layers increases, while the amount of entropy created by each unit cell falls. In terms of thermal efficiency and entropy production, heat exchangers with heat leakage are more sensitive to the quantity of unit cells than adiabatic heat exchangers, and channel walls act as a heat exchange medium that transmits heat between channels.

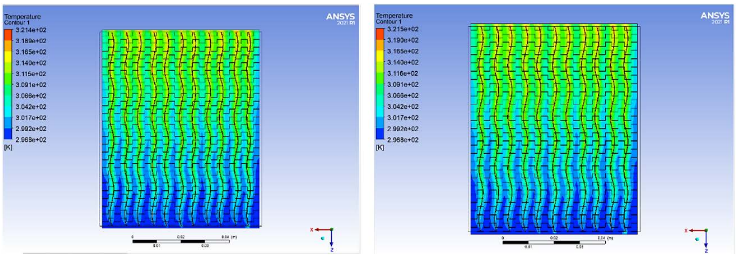
The downstream temperature distributions of fins with corrugation are more uniform and temperature change rates are faster compared to those of fins without corrugation. Under lighter loads, temperature change rate and temperature difference between the inlet and outlet are greater. Downstream regions exhibit a more uniform temperature distribution, and tail-like high-temperature regions between adjacent fins are shorter under conditions of decreased load.

These findings suggest that the corrugated arrangement of fins and adjusting the number of layers can enhance the performance of heat exchanger by increasing temperature uniformity and heat transfer, and the properly set fin arrangement, channel length, and load conditions can optimize the performance of heat exchangers.



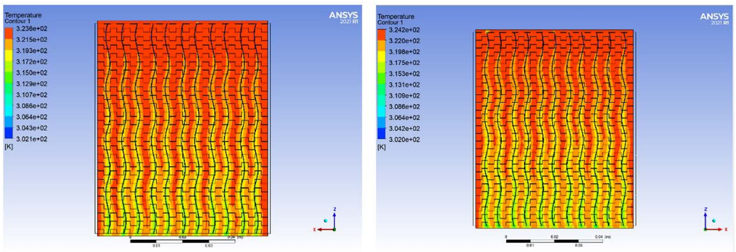
0.1 water velocity 0.2 water velocity

Figure 9. Heat exchanger temperature contours at 25°C water temperature, 1 m/s air velocity, 50°C air temperature, and different water velocities



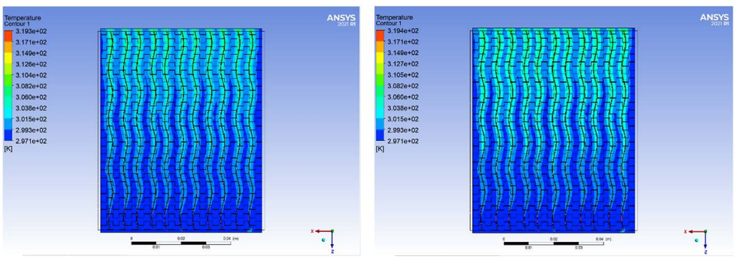
0.1 m/s water velocity 0.2 m/s water velocity

Figure 10. Heat exchanger temperature contours at 50°C water, 1 m/s air, 25°C air, and different water velocities



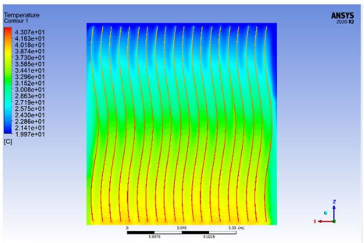
0.1 m/s water velocity 0.2 m/s water velocity

Figure 11. Heat exchanger temperature contours at 25°C water temperature, 5 m/s air velocity, 50°C air temperature, and different water velocities

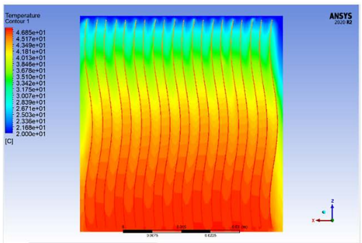


0.1 m/s water velocity 0.2 m/s water velocity

Figure 12. Heat exchanger temperature contours at 50°C water temperature, 5 m/s air velocity, 25°C air temperature, and different water velocities

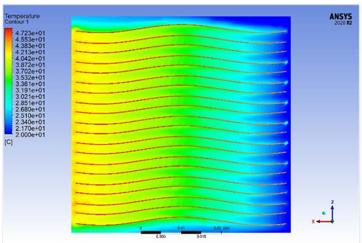


0.1 air/water velocity

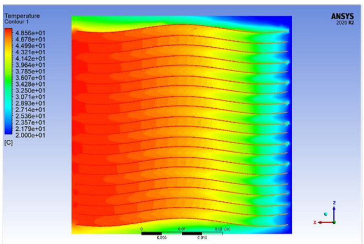


0.3 air/water velocity

Figure 13. Temperature contour of heat exchanger with external side air and water at the middle taken under the condition of 20 fins wavelength and 0.2 mm fin thickness



0.1 air/water velocity



0.3 air/water velocity

Figure 14. Temperature contour of heat exchanger with external side water and air at the middle taken under the condition of 20 fins wavelength and 0.2 mm fin thickness

The results of temperature variation in heat exchanger provide valuable insights into the heat transfer efficiency and overall performance of the system. They can be used to optimize heat exchanger design for enhanced thermal performance, improved energy efficiency, and better temperature control.

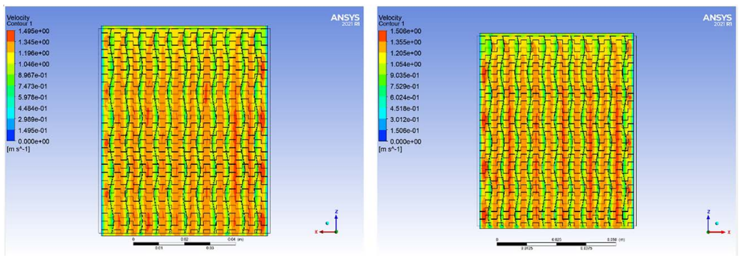
4.3 The Velocity Variation Inside the Heat Exchanger

Figures 15 to 20 give velocity contours of heat exchanger at different water temperatures. These results are consistent with the analysis of Han et al. [15] about fluid flow in finned tube heat exchangers, in which the authors found that fluid velocity significantly speeds up at the intersection of two fins while a low-speed zone forms between two adjacent fins in the flow direction due to the obstruction of fins. Additionally, a change in the direction of fluid flow could be found close to the fins as a result of the back pressure of backflow, which is comparable to the fluid dynamic effects surrounding micro pin fins reported by Yeom et al. [16]. Due to the fully established velocity condition at the intake, the velocity distribution won't change with flow direction, but for corrugated fins, the double-peak characteristic is clearly seen in the velocity distribution. The cross-section shape of fluids changes over the z-coordinate, the fluid between two adjacent parallel fins would be shunted into two parts by the downstream fin, and the fluid in upstream zone would be divided into two parts with the majority of fluid flowing besides downstream fins. This observation relates to the work of Saad et al. [11], who discovered a correlation with the friction factor of micro-channels with flow characteristics taken into account.

The velocity distribution data in the figures were taken from the same coordinate of several models. Due to the longer flow distance from the entry, the velocity distribution is almost fully developed, and the velocity at channel center is higher. A shorter fin structure makes the concentration behavior of fluid traveling closer to fins at a high speed more visible, this is similar with the findings of Carija et al. [14], that the distinctive louvred fin structure of heat exchangers has a higher heat transfer performance and the structure with shorter fins has a significant and direct influence on fluid flow. According to the velocity curves of three models with corrugated fins, clear periodic oscillations could be observed, and this is in line with the periodic disturbance of boundary layer found by Yang and Li [13] in their work on offset strip fins. In the first segment of the three curves, fluid acceleration and fin thickness-induced channel constriction caused a rapid growth in velocity, whereas the channel velocity development in the final segment caused a fast drop in velocity.

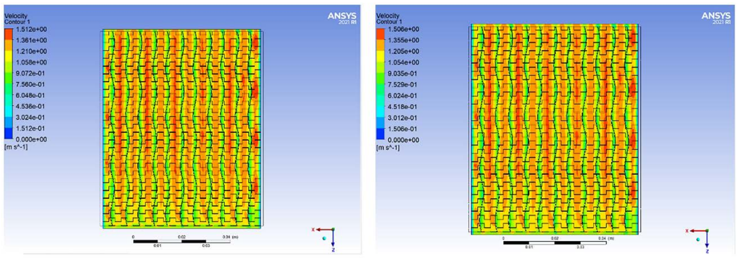
Similar to other conditions, the boundary layer was uniformly destroyed because the velocity decreased first and increased later, which is the overall effect of the corrugated design of fins. The velocity in the boundary layer of corrugated fins is higher than that of plain fins, indicating that the boundary layer of corrugated fins is thinner, which is advantageous in heat transfer. This result is similar to the discovery of Fernández-Seara et al. [12] that corrugated fins improved the heat transfer of the titanium-brazed PFHE they designed.

These findings imply that adjusting the structure, length, and arrangement of fins can impact velocity distribution and heat exchanger performance. By considering these factors, the design of heat exchanger can be optimized for more effective fluid flow control and improved heat transfer performance.



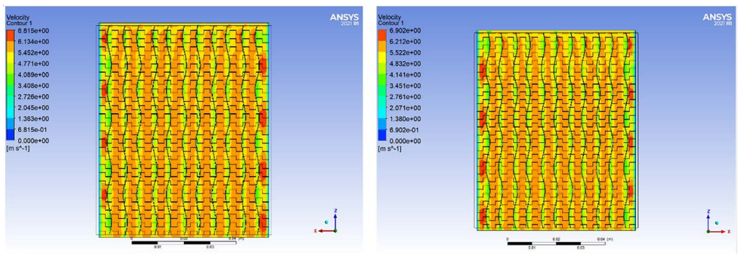
0.1 m/s water velocity 0.2 m/s water velocity

Figure 15. Water velocity contours of heat exchanger under 25℃ water temperature, 1 m/s air velocity, 50℃ air temperature



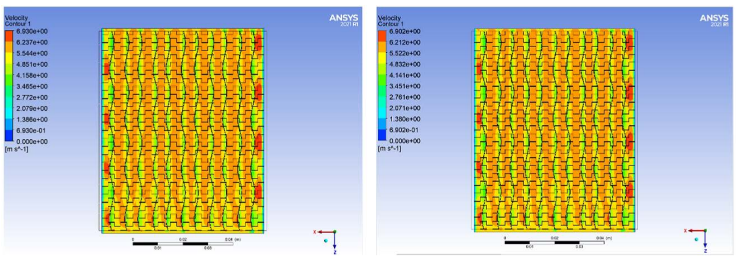
0.1 m/s water velocity 0.2 m/s water velocity

Figure 16. Water velocity contours of heat exchanger under 50℃ water temperature, 1 m/s air velocity, 25℃ air temperature



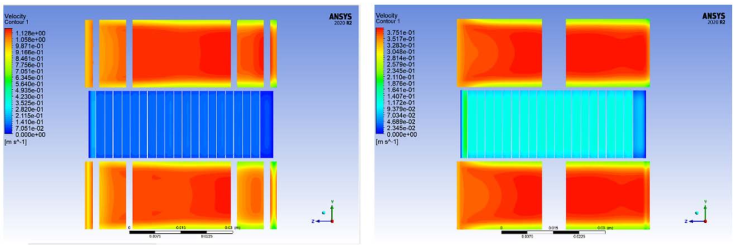
0.1 m/s water velocity 0.2 m/s water velocity

Figure 17. Water velocity contours of heat exchanger under 25℃ water temperature, 5 m/s air velocity, 50℃ air temperature



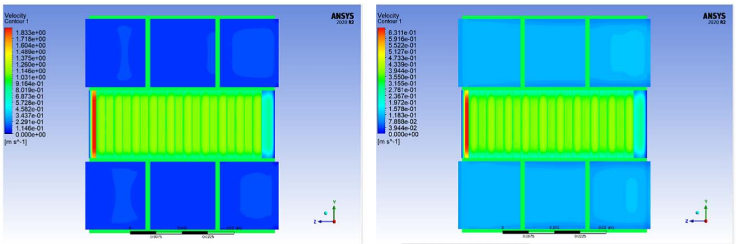
0.1 m/s water velocity 0.3 m/s water velocity

Figure 18. Water velocity contours of heat exchanger under 50℃ water temperature, 5 m/s air velocity, 25℃ air temperature



0.1 air/water velocity 0.3 air/water velocity

Figure 19. Velocity contour of heat exchanger with external side air and water at the middle taken under the condition of 20 fins wavelength and 0.2 mm fin thickness



0.1 air/water velocity 0.3 air/water velocity

Figure 20. Velocity contours of heat exchanger with external side water and air at the middle taken under the condition of 20 fins wavelength and 0.2 mm fin thickness

Results of velocity variation in the heat exchanger can be used as references for the design of heat exchangers to maintain a desired fluid flow rate while minimizing flow resistance, thereby improving system performance, energy efficiency, and operational stability.

Findings of this study provide insights to optimizing design and performance of heat exchangers and increasing potential benefits of putting such design in practical applications.

Fin obstruction of PFHE can cause velocity variation in the heat exchanger. Backflow and back pressure can occur due to flow restrictions caused by fin geometry and channel dimensions. Backflow disrupts the desired flow direction and causes velocity variation, while back pressure can cause non-uniform velocity distribution, flow maldistribution, increased pressure losses, and reduced heat transfer efficiency. Minimizing obstruction by optimizing fin configurations can smoothen flow and improve heat transfer efficiency.

5 Conclusions and Recommendations

This study investigated the performance characteristics of PFHEs with enhanced surfaces using CFD analysis and Fluent 2020 R1.

Adopting higher gas velocity and higher pumping power are two important recommendations for future research. Adopting higher gas velocity involves using different flow control methods, while adopting higher pumping power means using more powerful pumps or optimizing the pump configurations. Benefits of these measures include increased fluid flow rate, improved heat transfer coefficient, and reduced fouling. Challenges include increased energy consumption, potential of increased pressure loss, and limitations in system components and materials. The main findings of this research are summarized below:

1. Pressure inside the heat exchanger increases as water and air velocities increase.
2. Temperature distribution inside the heat exchanger changes with inlet temperatures of water and air.
3. Temperature increases at the top layer due to warm stream passage and decreases near the bottom layer due to cold stream passage.
4. Increasing fin thickness results in intensified fluid turbulence and increased local resistance loss at staggered positions.
5. Variation of fluid temperature in the depth direction is caused by conductive and convective heat transfer.
6. Because of the fully developed velocity at the inlet, the velocity distribution along the flow direction remains unchanged.

Recommendations for future research are:

1. Adopt a higher gas velocity in the design for better flow uniformity.
2. Use different fluids in heat exchangers and consider fluid properties for more reliable simulations.

3. Explore the effects of different fin materials on heat exchanger performance.
4. Validate numerical results and experimental data by creating a rig with the same boundary conditions.
5. Install and maintain suitable filters to prevent plugging of particulates.
6. Conduct pressure tests without water as the medium to meet relevant criteria.
7. Avoid excessive pressure drop to prevent potential mechanical problems.
8. Avoid using vortex generator geometry for very high Reynolds numbers to minimize entropy generation rate.
9. Adopt higher pumping power to address boundary layer separation.

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