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Numerical Study of Flow and Heat Transfer on Plate-Fin Heat Exchanger in a High Temperature Gas-Cooled Reactor

Nuraqilah Zahari[1], Bukhari Manshoor[1,*], Ishkrizat Taib[1], Izz Mustaqim Mohd Fauzi[1], Muhammad Izzul Iqmal Shahrizal[1], Nur Maizaratul Hanisha Khairi Azhar[1], Nur Izzati Ruslie[1], Nurnabila Syuhada Azian[1], Azlan Adam[2], Wan Mahafiz Rosni[2], Mithun Mondal[3]

[1] Faculty of Mechanical and Manufacturing Engineering, Universiti Tun Hussein Onn Malaysia, Parit Raja, Batu Pahat, 86400, MALAYSIA
[2] Department of Mechanical Engineering, Polytechnic Kuching, 93050 Kuching Sarawak, MALAYSIA
[3] School of Engineering of Barcelona, Universitat Politècnica de Catalunya | UPC · ETSEIB, SPAIN

*Corresponding Author

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Abstract: Plate heat exchangers, known for their high efficiency and low mass, are a type of compact heat exchanger widely used in aerospace, automotive, and power machinery applications. The objective of this study is to model both plain and staggered fin configurations and to analyze the flow characteristics and heat transfer within a plate-fin heat exchanger. The analysis focuses on examining fluid dynamics and evaluating heat transfer efficiency, contributing to the optimization of plate-fin heat exchanger designs for various industrial applications. The analyses utilize Computational Fluid Dynamics (CFD) approaches to evaluate heat transfer of the heat exchanger. The simulation uses models including the energy equation and the k-epsilon viscous model. The SIMPLE method is used for solving equations to ensure accurate pressure calculations. At various power levels, Model 2 consistently shows better temperature distribution in terms of uniformity and clearer separation between hot and cold zones compared to Model 1, demonstrating superior heat exchanger performance due to its staggered fin design. Moreover, both models have high upper section velocities promoting efficient heat transfer, with Model 1's lower section showing slower velocities and higher thermal exchange, while Model 2's staggered fins enhance overall heat exchange and turbulence. Therefore, the study concludes that staggered fins (Model 2) are more effective for applications that require high heat transfer efficiency compared to plain fins (Model 1) in HGTR.

Keywords: Plate heat exchanger, heat transfer, CFD, k-epsilon, plain, staggered

1. Introduction

Plate heat exchangers, known for their high efficiency and low mass, are a type of compact heat exchanger widely used in aerospace, automotive, and power machinery applications. In plate heat exchangers, fins play a crucial role in enhancing the heat transfer efficiency by increasing the surface area available for heat exchange [1]. Due to the complexity of these geometries, calculating flow friction and overall heat transfer coefficients is difficult. Kays and London [2] conducted systematic experiments and determined the heat transfer factors and friction factors for various fins in plate-fin heat exchangers.

A plate fin heat exchanger is a type of compact exchanger composed of alternating flat plates, known as parting sheets, and corrugated fins brazed together into a block [3]. Heat exchange occurs as streams flow through the passages created by the fins between the parting sheets [4]. The parting sheets serve as the primary heat transfer surfaces, while the fins, intimately bonded to these surfaces, act as secondary heat transfer surfaces [5]. Enhancing the efficiency of heat exchangers holds significant engineering importance [6]. Optimizing the internal flow field of a heat exchanger is crucial for maximizing its performance [7]. Plate-fin heat exchangers are among the most widely used in the industry today.

**Corresponding author: bukhari@uthm.edu.my*

This is due to their numerous advantages over traditional tube heat exchangers [8] primary advantage of plate-fin heat exchangers is their compact design, which provides a large heat exchange surface area while maintaining moderate hydraulic resistance. There are various compact heat exchangers with plate fins, there are many different designs, like plain triangle fins, wavy fins, and perforated fins, among others [9]. However, this study is focusing on just two models: the plain rectangular fin and the serrated fin. These models were picked because they're commonly used and help us understand how heat exchangers work. By studying these designs, we hope to learn more about their performance and features.

A plain rectangular heat exchanger is a compact and efficient device used to transfer heat between two fluids through a series of parallel [10], flat plates made from materials with high thermal conductivity [11]. These plates create narrow channels, which can be configured for counterflow, parallel flow, or crossflow, optimizing the heat transfer process. The large surface area provided by the multiple plates enhances thermal efficiency, making these exchangers ideal for applications in HVAC systems, the process industry, and power plants. Despite their advantages, including a compact design and high efficiency, plain rectangular heat exchangers can experience significant pressure drops and are prone to fouling, necessitating regular maintenance.

A serrated fin heat exchanger employs a series of corrugated fins, often arranged in a staggered pattern, to enhance heat transfer between fluids [12,13]. These fins are typically brazed to flat plates, creating a structure with increased surface area compared to plain rectangular exchangers. The serrated design disrupts boundary layers and promotes turbulence, improving convective heat transfer [14]. Serrated fin exchangers are particularly effective in applications requiring high thermal performance, such as in aerospace, automotive cooling systems, and refrigeration [13]. Their design allows for efficient heat exchange in compact spaces, making them valuable in situations where space is limited.

However, these exchangers may require careful maintenance due to the intricate nature of the fin structure, and they can likely foul. The flow and heat transfer characteristics of a plate-fin heat exchanger can be analyzed and compared using numerical results. Some investigations have been conducted to construct these numerical simulations. Zhang [15] conducted numerical simulations of laminar forced convection in two-dimensional wavy-plate-fin channels while Li [16] investigated laminar forced flow and heat transfer in plate by using isosceles triangular duct. Each study shows the flow was fully developed with hydrodynamically and uniformly wall temperature.

Wang [17] studied a fluid flow and heat transfer for plain fins and serrated fins at low Reynolds number by employing the simplified model using CFD in code FLUENT. The numerical simulation results for heat transfer rate and pressure drop are compared with experimental data obtained from literature.

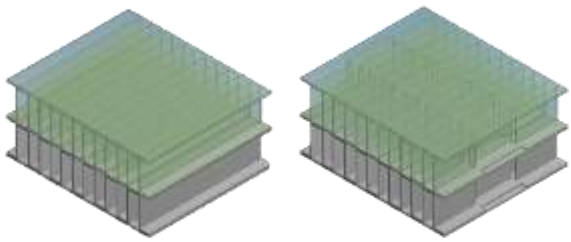
The objective of this study is to model both plain and staggered fin configurations and to analyze the flow characteristics and heat transfer within a plate-fin heat exchanger. By developing these models, the study aims to understand how different fin arrangements impact the performance of the heat exchanger. The analysis focuses on examining fluid dynamics and evaluating heat transfer efficiency, contributing to the optimization of plate-fin heat exchanger designs for various industrial applications.

2. Methodology

This methodology explained in detail the steps and procedures involved in the simulation using CFD via ANSYS Fluent software to analyze the flow and heat transfer of the plate-fin heat exchanger in an HTGR.

2.1 Geometry of the heat exchanger

The plate-fin heat exchanger was modeled using the Design Modeler in the Ansys software. Two different types of fins, which as plain fins and staggered fins, are created and simulated in this study. The heat exchanger is illustrated in three dimensions in Figure 1 and comprises the following parts: hot and cold fluid flow channel, plates, and fins. The dimension of the heat exchanger is obtained from [18] and is shown in Table 1.



(a) (b)

Fig. 1 – Geometry of the heat exchanger (a) Plain fin, model 1; (b) Staggered fin, model 2

Table 1 – Dimension of the model

Parameter	Diameter (*mm*)
Fin spacing	1.7
Fin thickness	0.2
Fin height	5.2
Plate thickness	0.3
Fin length (model 1)	21
Fin length (model 2)	7

2.2 Meshing of the heat exchanger

Mesh size impacts the accuracy of simulation results, with a finer mesh generally yielding more precise outcomes. However, a finer mesh requires more computational resources and processing time. The fluid domain and heat exchanger were created using tetrahedral mesh shapes with an average skewness of 0.89 as illustrated in Figure 2. The full-scale model, consisting of 232,253 and 248, 152 nodes for model 1 and 2 respectively with the same element size of 0.4 mm was utilized in all simulations.

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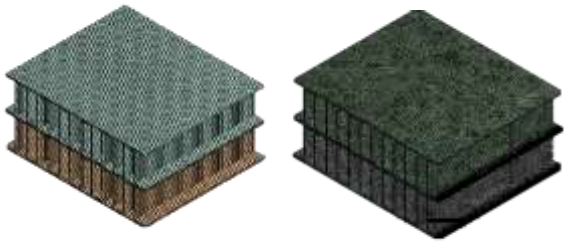


(a) (b)

Fig. 2 – Meshing of the heat exchanger (a) model 1; (b) model 2

2.3 Governing equations

In this study, the governing equations for continuity, momentum, and energy need to be solved simultaneously [18]. The equations are the continuity equation, momentum equation and energy equation as written in the following Eqn. 1 – 6.

Continuity equation;

$$\frac{\partial u}{\partial x}+\frac{\partial v}{\partial y}+\frac{\partial w}{\partial z}=0 \quad (1)$$

Momentum equation:

$$\rho(u\frac{\partial u}{\partial x}+v\frac{\partial u}{\partial y}+w\frac{\partial u}{\partial z})=-\frac{\partial p}{\partial x}+\eta\left(\frac{\partial^2 u}{\partial x^2}+\frac{\partial^2 u}{\partial y^2}+\frac{\partial^2 u}{\partial z^2}\right) \quad (2)$$

$$\rho\left(\frac{\partial v}{\partial x}+v\frac{\partial v}{\partial y}+w\frac{\partial v}{\partial z}\right)=-\frac{\partial p}{\partial y}+\eta\left(\frac{\partial^2 v}{\partial x^2}+\frac{\partial^2 v}{\partial y^2}+\frac{\partial^2 v}{\partial z^2}\right) \quad (3)$$

$$\rho\left(\frac{\partial w}{\partial x}+v\frac{\partial w}{\partial y}+w\frac{\partial w}{\partial z}\right)=-\frac{\partial p}{\partial z}+\eta\left(\frac{\partial^2 w}{\partial x^2}+\frac{\partial^2 w}{\partial y^2}+\frac{\partial^2 w}{\partial z^2}\right) (4)$$

Energy equation for the fluid zone:

$$\left(u\frac{\partial T}{\partial x}+v\frac{\partial T}{\partial y}+w\frac{\partial T}{\partial z}\right)=\frac{\lambda}{\rho x^2}\left(\frac{\partial^2 T}{\partial x^2}+\frac{\partial^2 T}{\partial y^2}+\frac{\partial^2 T}{\partial z^2}\right) \quad (5)$$

Energy equation for the solid zone:

$$\frac{\partial^2 T}{\partial x^2}+\frac{\partial^2 T}{\partial y^2}+\frac{\partial^2 T}{\partial z^2}=0 \quad (6)$$

2.4 Turbulence model

The turbulence model used in this study is the *k*-*ε* turbulence model. Transport equation for *k* and *ε*:

$$\frac{D}{Dt}(\rho k)=\frac{\partial}{\partial x_j}\left[\left(\mu+\frac{\mu_t}{\sigma_k}\right)\frac{\partial k}{\partial x_j}\right]+G_k-\rho\varepsilon \quad (7)$$

$$\frac{D}{Dt}(\rho \varepsilon)=\frac{\partial}{\partial x_j}\left[\left(\mu+\frac{\mu_t}{\sigma_\varepsilon}\right)\frac{\partial \varepsilon}{\partial x_j}\right]+C_{\varepsilon 1}\frac{\varepsilon}{k}G_k-\rho C_{e2}\frac{\varepsilon^2}{k} \quad (8)$$

where,

$C_\mu = 0.09$, $C_{\varepsilon 1} = 1.44$, $C_{\varepsilon 2} = 1.92$, $\sigma_k = 1.0$, and $\sigma_\varepsilon = 1.3$

2.5 Parameter equations

The parameters investigated throughout the simulation involve the temperature distribution, velocity, heat exchange rate, and Reynolds number. All the parameters can be determined using the equation as follows:

Temperature:

$$\rho u\cdot\nabla T=\nabla\cdot(k\nabla T)+Q \quad (9)$$

Velocity:

$$\rho\left(\frac{\partial u}{\partial t}+u\cdot\nabla u\right)=-\nabla p+\mu\nabla^2 u+f \quad (10)$$

Heat exchange rate:

$$\dot{Q}=\dot{m}c_p\Delta T \quad (11)$$

Reynolds number:

$$R_e=\frac{\rho u L}{\mu} \quad (12)$$

Where ρ is the density, u is the velocity, T is the temperature, k is the thermal conductivity, Q is the volumetric heat sources, p is the pressure, μ is the dynamic viscosity, f is the body force, $\dot{Q}$ is the heat exchange rate, $\dot{m}$ is the mass flow rate, c_p is the specific heat capacity, ΔT is the temperature difference, R_e is the Reynolds number, and L is the length.

2.6 Boundary conditions

Two primary domains were identified in this study: hot and cold helium gas zones. The upper part is hot gas while the lower part contains cold helium gas. The simulations included four operating settings, with reactor power levels of 100%FP, 70%FP, 50%FP, and 30%FP. The inlet was set to fully developed inlet velocity boundary conditions, while the outlet was set to pressure-outlet boundaries. The boundary conditions are shown in Table 2.

Table 2 – The boundary conditions

Boundary conditions	Hot helium	Cold Helium
Inlet pressure (MPa)	0.67	1.57
Inlet temperature (K)	650	600
Ave. inlet velocity 100% FP	25	9.9
Ave. inlet velocity 70% FP	17.5	6.9
Ave. inlet velocity 50% FP	12.5	4.9
Ave. inlet velocity 30% FP	7.5	3

3. Results and Discussions

3.1 Model validation

The plate-fin heat exchanger model consistency at low Reynolds number was evaluated by comparing the numerical findings with the experimental results of Chen et al, [18] shown in Figure 3 and 4. The comparison shows that the average of the errors is less than 10%, which means that the model is suitable for this numerical study.

Table 3 – Heat Exchange Rate for model 1

Reynolds Number, Re	Experiment	Simulation	%Error
500	35.12	33.3	5.18
600	36.18	34.73	4.01
800	37.75	35.04	7.18
1000	38.98	36.81	5.57
Average			5.48

Table 4 – Heat Exchanger Rate for model 2

Reynolds Number, Re	Experiment	Simulation	%Error
300	5.93	5.25	11.47
400	6.41	6.71	4.68
500	6.86	7	2.04
800	8.00	8.68	8.50
Average			6.67

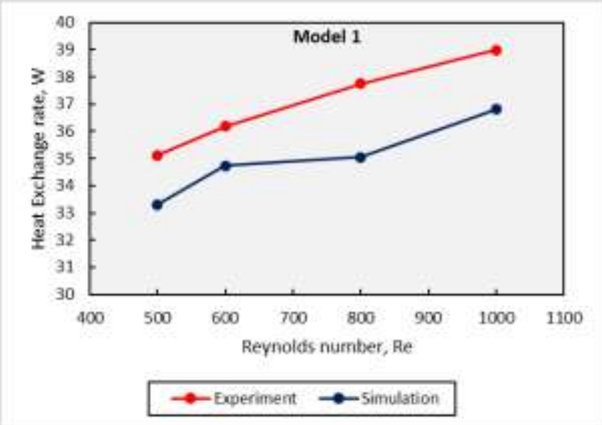


Fig. 3 – Model validation model 1

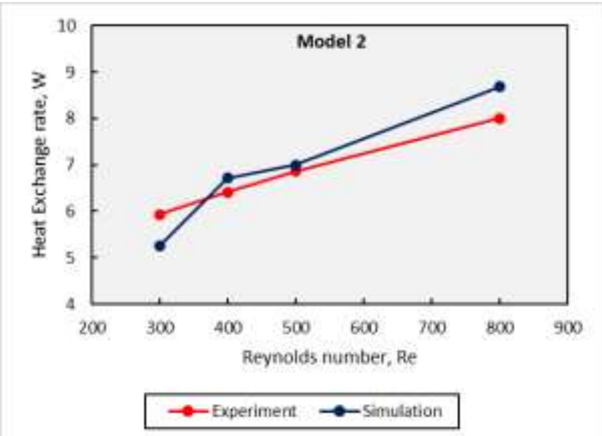


Fig. 4– Model validation model 2

Table 5 – Heat Exchange rate, W

Reynolds Number, Re	model 1	model 2
300	31.24	5.25
400	32.62	6.71
500	33.33	7.00
600	34.73	7.88
800	35.04	8.68
1000	36.81	9.27

Figure 5 demonstrates that Model 1 consistently outperforms Model 2 in terms of heat exchange rate across a range of Reynolds numbers (200 to 1000). For Model 1, the heat exchange rate increases from approximately 30W to about 37W as the Reynolds number rises, indicating a robust performance in enhancing heat transfer. In contrast, Model 2 shows a more modest increase in heat exchange rate from around 5 W to about 10 W over the same Reynolds number range. This significant performance disparity suggests that Model 1's design, materials, or structural features are more effective in facilitating heat transfer in this high-temperature reactor context.

The design of fins in a heat exchanger affects the heat exchange rate primarily through its impact on surface area and fluid flow characteristics. Plain fins, as seen in Model 1, provide a continuous, smooth surface that maximizes the contact area with the fluid, leading to efficient heat transfer. The smooth flow path minimizes resistance and pressure drop, allowing for steady and effective heat exchange [19]. On the other hand, staggered fins, as used in Model 2, introduce more surface area and create turbulence in the fluid flow, which can enhance local heat transfer by constantly mixing the fluid and bringing more of it into contact with the fin surfaces. However, the increased turbulence also results in higher flow resistance and pressure drop, which could reduce overall efficiency if the energy required to move the fluid through the heat exchanger becomes too high.

Additionally, the heat transfer coefficient, which measures the heat transfer capability of the surface, is influenced by fin design. Plain fins generally have a higher heat transfer coefficient under laminar flow conditions due to their uninterrupted surface, leading to better performance in steady-state applications [20]. Staggered fins, while potentially increasing the local heat transfer coefficient due to induced turbulence, may not be as efficient in overall heat exchange if the increased flow resistance and pressure drop outweigh the benefits of turbulence [21]. Overall, the graph underscores Model 1's superior thermal performance, making it a more suitable option for high-efficiency heat exchange tasks in gas-cooled reactors.

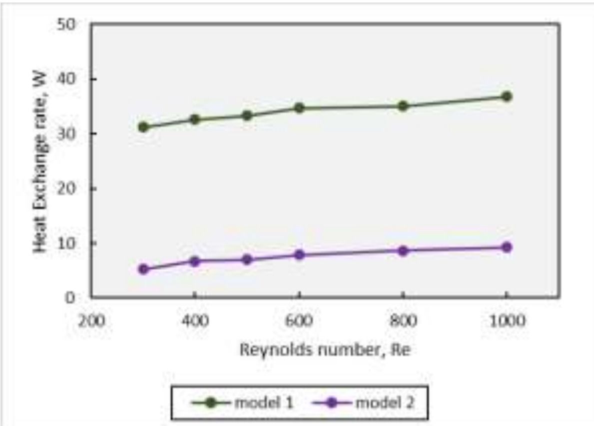


Fig. 5– Comparison between model 1 and model 2

3.2 Temperature distribution

The temperature distribution contours at various levels of FP for each model are presented in Figure 6 and Figure 7. At 100% FP, Model 1 displays significant non-uniformity with intricate patterns throughout, but Model 2 displays better overall uniformity, especially in the upper section. Both models show better uniformity when power drops to

70% FP, but Model 2 has a more obvious transition zone and better consistency in the upper and lower parts. Both models show greater separation between hot and cold zones at 50% FP, with Model 2 showing clearer borders and stable internal patterns. Both models demonstrate the highest level of uniformity at 30% FP, but Model 2 shows nearly entirely uniform upper region and highly uniform lower section patterns.

The staggered fin design of Model 2 allows for more efficient mixing inside flow areas at all power levels, which produces more uniform temperature distributions. Better flow separation and heat transfer efficiency are indicated by Model 2's constantly narrower and more defined transition zones between hot and cool regions. In comparison to Model 1's plain fin design, Model 2 demonstrates higher overall temperature uniformity across all power levels, maintaining more constant patterns and a clearer separation between hot and cold sections, indicating better heat exchanger performance overall.

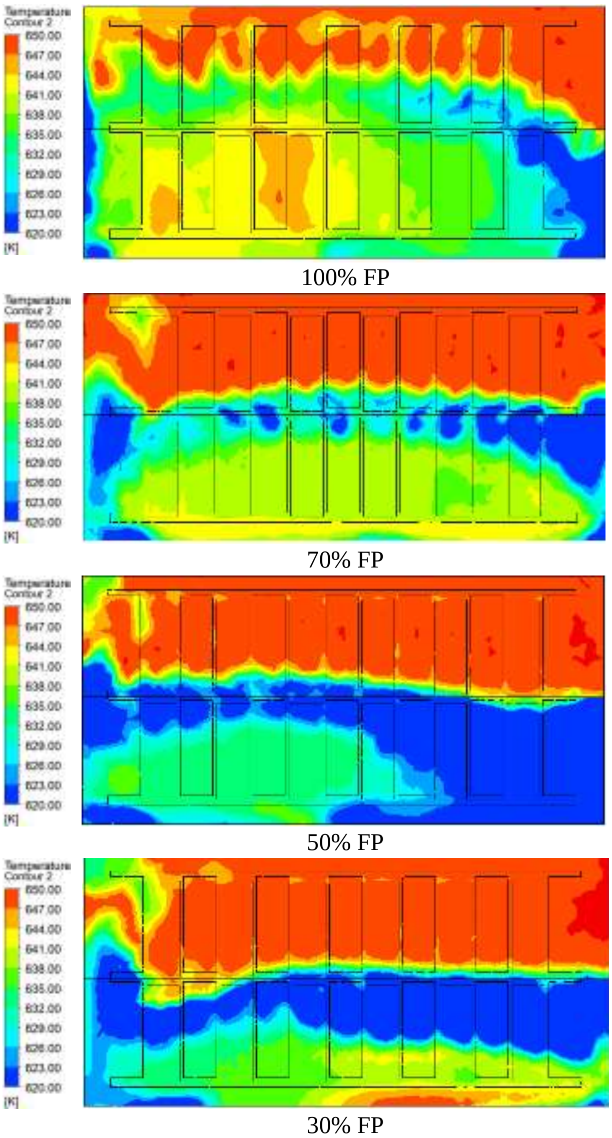


Fig. 6– Temperature distribution for model 1

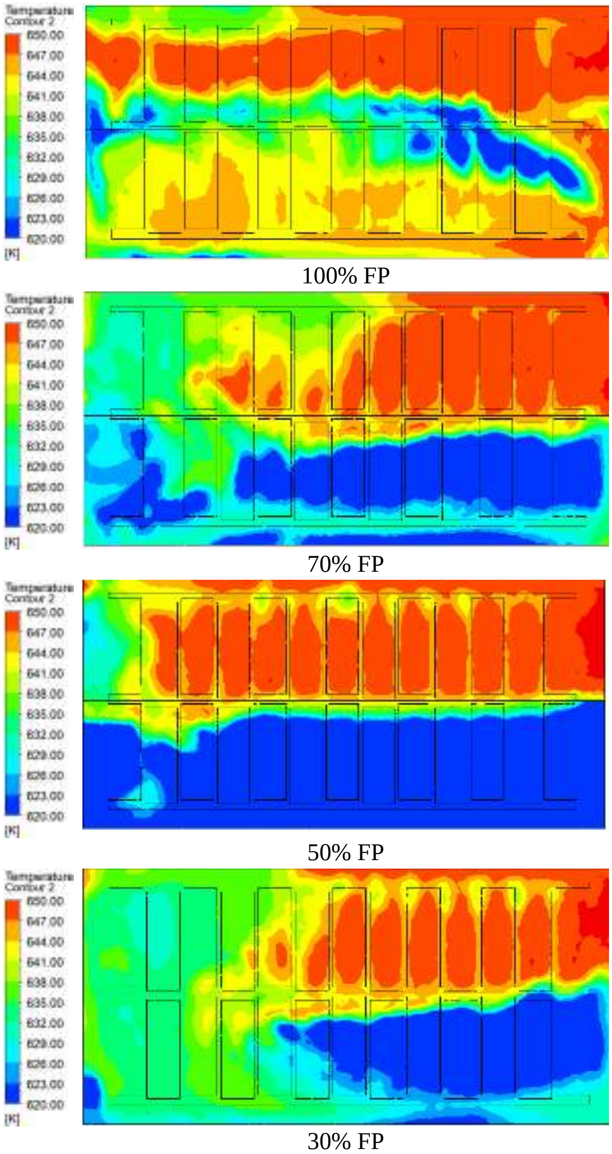


Fig. 7– Temperature distribution for model 2

3.3 Velocity

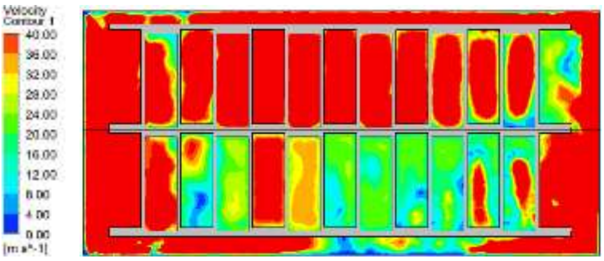
As the red areas, the hot helium in the upper part has high velocities at 100% power in both models as shown in Figure 8 and Figure 9, which promote efficient heat transfer. The cold helium in the lower section of Model 1 travels more slowly, with notable velocity differences increasing total thermal exchange. Similar to this, the lower section of Model 2 likewise has lower but considerable velocities, with the overall heat exchange and turbulence being enhanced by the staggered fin design.

On the other hand, both models' top section velocity marginally drops but stays high at 70% power. The distribution of velocity in Model 1 is more varied, whereas in Model 2, the staggered fins contribute to the maintenance of turbulence and efficient heat exchange. Both models possess a lower section velocity that is still sufficient for heat transfer; but, because of the fin design, Model 2 appears to have a little advantage in terms of flow homogeneity.

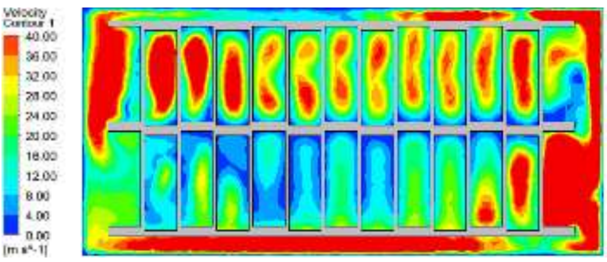
For both models, there is a noticeable decrease in the hot and cold section velocities at 50% power. Model 1's lower section displays a more uniform and lower velocity profile, while the top section has fewer high-velocity zones, which may reduce the convective heat transfer rate. Even if Model 2's velocities decrease as well, the staggered fins manage to create turbulence and keep the heat transfer rate manageable. More so than in Model 1, the fin design in Model 2 assists to reduce the possible performance reduction.

When the reactor runs at 30% power, velocities in both sections drop further in both models. The upper section of Model 1 presents difficulties with regard to effective heat exchange due to its generally lower velocities, while the lowest section displays uniformly low velocities, indicating a possible compromise in performance. Model 2's staggered fins continue to create some turbulence, which helps with heat transfer and somewhat mitigates the performance impact as compared to Model 1. Model 2's velocities also significantly decrease.

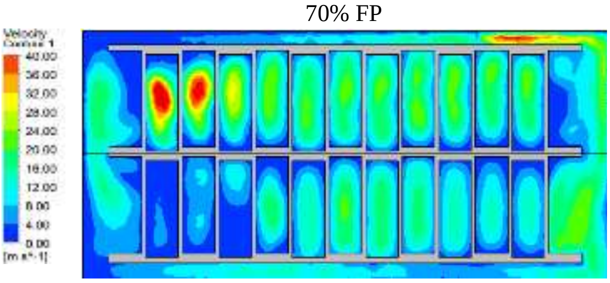
In summary, in both models, the power level of the reactor has a significant impact on the flow dynamics of the plain fin heat exchanger. At different power levels, the Model 2 with staggered fins maintains superior transfer of heat and turbulence. Both models have high velocities at 100% power, but Model 2 has a more useful distribution. Model 2 maintains more efficiency and turbulence at 70% and 50% power. Because of its staggered fin design, Model 2 performs better even at 30% power. Maintaining efficient heat management in HTGRs requires optimizing power levels and design options.



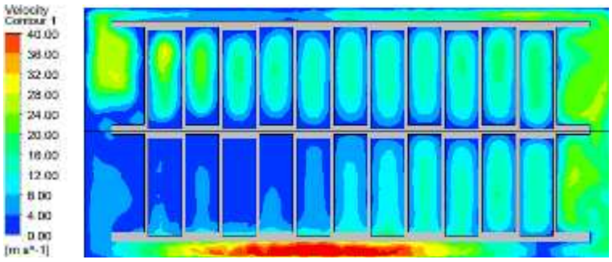
100% FP



70% FP

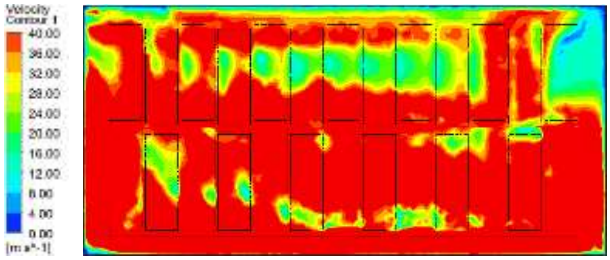


50% FP

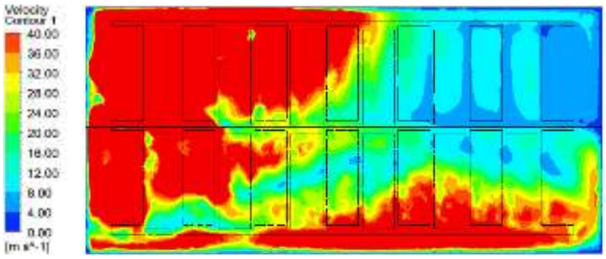


30% FP

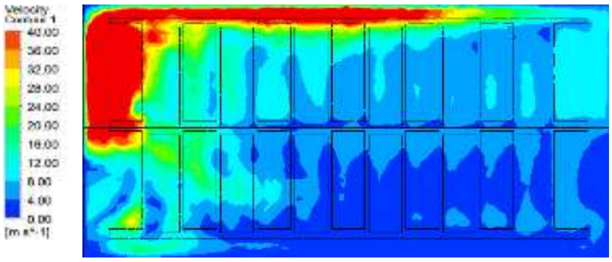
Fig. 8– Velocity for model 1



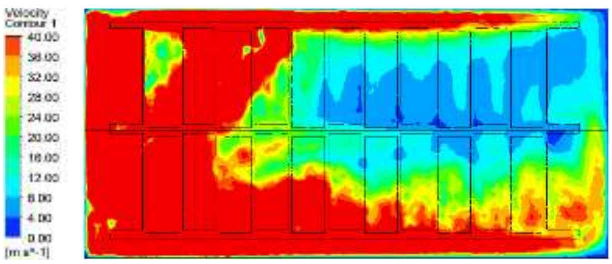
100% FP



70% FP



50% FP



30% FP

Fig. 9– Velocity for model 2

4. Conclusion

In this study, a detailed analysis was conducted on the thermal performance and flow characteristics of plate-fin heat exchangers with different fin configurations within a high-temperature gas-cooled reactor. Model 2, with its staggered fin configuration, was observed to achieve better temperature uniformity and more efficient mixing of the fluid due to the disruptions created by the staggered fins. These disruptions promote turbulence, which can enhance

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heat transfer. Therefore, the study concludes that plain staggered fins (Model 2) are more effective for applications that require high heat transfer efficiency, particularly in high-temperature environments. This conclusion is critical for the design and selection of heat exchangers in industrial applications, as it highlights the importance of fin configuration on thermal performance and operational efficiency.

The study could be expanded to include an analysis of different materials and surface treatments for the heat exchanger fins, as these factors could significantly impact the heat transfer coefficient and overall performance of the heat exchanger. Additionally, further research could explore the optimization of fin geometry, such as the angle and shape of serrations, to find the optimal balance between heat transfer enhancement and pressure drop. The study compared plain and serrated fin designs, but more detailed investigations into these geometrical aspects could yield more efficient designs. Moreover, the study compared numerical findings with experimental results to validate the model. Conducting additional experiments with a wider range of operating conditions could provide more data points for model validation and help refine the CFD model further.

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