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Review

Effect of Fin Type and Geometry on Thermal and Hydraulic Performance in Conditions of Combined-Cycle Nuclear Power Plant with High-Temperature Gas-Cooled Reactors

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Abstract: One method of nuclear energy development involves using helium. Its properties make using extended surfaces obligatory. However, currently nuclear technology does not typically use finned tubes. This study explores ways of enhancing heat transfer efficiency in a high-temperature gas-cooled reactor system by using novel fin designs in the heat exchanger for residual heat removal. Four different types of fins were studied: annular, serrated, square, and helical. The effect of fin height, thickness, and number was evaluated. Serrated and helical fins demonstrated superior performance compared to conventional annular fin designs, which was expressed in enhanced efficiency. The thickness of fins was found to have the strongest influence on the efficiency, while the height and number of fins per meter had weaker effects. In addition, the study emphasized the significance of considering complex effects when optimizing fin design, like the effect of fin geometry on the velocity of helium. The findings highlight the potential of creative fin designs to greatly enhance the efficiency and dependability of gas-cooled reactor systems, opening up possibilities for advancements in nuclear power plant technology.

Keywords: high-temperature gas-cooled reactor; combined-cycle nuclear power plants; finned tube; heat transfer

Citation: Ramadan, K.A.A.; Slyusarskiy, K.V. Effect of Fin Type and Geometry on Thermal and Hydraulic Performance in Conditions of Combined-Cycle Nuclear Power Plant with High-Temperature Gas-Cooled Reactors. *Thermo* **2024**, *4*, 382–393. https://doi.org/10.3390/thermo4030020

Academic Editor: Kian Jon Chua

Received: 24 June 2024
Revised: 2 August 2024
Accepted: 4 August 2024
Published: 9 August 2024

1. Introduction

Properly managing heat removal is crucial for the safe and effective operation of high-temperature gas-cooled reactors (HTGRs). HTGRs are recognized for their excellent safety characteristics and ability to operate at high temperatures, allowing them to be used in a wide range of applications, such as power generation or chemical production. HTGRs use a mix of passive and active cooling systems to effectively remove heat from the reactor core. Passive systems utilize the phenomena of natural convection and radiation to offer safety through the principles of physics. Active systems, such as gas circulation and dedicated cooling loops, allow for better control over heat removal rates [1].

The studies emphasized by [2] provide insights into experimental thermal fluid behavior and safety studies for HTGR designs. These studies explore core heat transfer, plenum behavior, and safety aspects, enhancing our understanding of HTGR thermal behavior. The plenum is a vital element in the HTGR design, providing a key function of enabling effective energy transfer and ensuring reactor stability. Huning et al.'s study [2] explores the plenum's behavior, including the intricacies of gas flow, pressure distribution, and temperature gradients. This information is crucial for enhancing plenum design and guaranteeing the best thermal performance. A properly constructed plenum improves the efficiency of heat transfer and enhances the safety and reliability of HTGRs.

Efficient heat removal from the reactor should result in its smooth transition to the turbine for power generation. Utilizing advanced steam reheating technologies can enhance efficiency and power output by incorporating reheated steam in the turbine cycle [3]. It is essential to have a strong control strategy in place to safely and efficiently operate the combined cycle at different loads, maintaining peak performance throughout the HTGR's operational range [4]. A study by Florido et al. [5] evaluated the economic viability of power plants that combine nuclear and gas turbine technologies. The combined thermal cycle is seen as a beneficial connection between nuclear and natural gas fuels, providing a cost-effective alternative for generating electricity.

A combined-cycle gas turbine power plant is essentially an electrical power plant in which a gas turbine and steam turbine are used in combination to achieve greater efficiency. The gas turbine drives an electrical generator while the gas turbine exhaust is used to produce steam for the steam turbine, whose output provides the means to generate more electricity [6,7]. This process of steam production is deeply rooted in the principles of thermodynamics and heat transfer, as expounded by Kays and London [8].

The emergency cooling systems in the core of HTGRs are based on gas–water heat exchangers, which include the combination of conduction through tubes and convection from tubes to cooling water or steam and from helium to tubes. However, due to the much lower rate of the latter, studies in this area usually deal more thoroughly with convection in order to provide tools for the critical assessment of correlations and data and to provide pointers for the development of surfaces [9]. There are numerous researchers who have modeled and/or optimized heat exchangers of HTGR systems [10–12], highlighting their importance. Using finned tubes seems to be a viable solution for such systems because it may result in a reduction in capital costs, as well as hydraulic resistance, which influences the overall efficiency of corresponding systems [10,13].

Computational studies, such as computational fluid dynamics (CFD) simulations, have been conducted to compare different types of fins, like segmented and solid fins, especially for the economizer section of the steam generator of steam–gas power plants [14].

HTGRs have experienced a long and tortuous development since the middle of the previous century. The final technical route focuses on small modular HTGRs with inherent safety. HTGRs use helium as the cooling medium, whose reactor outlet temperature (ROT) can reach 700–950 °C. Compared with HTGRs, the outlet temperature of very-high-temperature gas-cooled reactors (VHTRs) is higher and can exceed 1000 °C [15].

Wang J. meticulously developed and contrasted a simplified and a complex combined-cycle scheme specifically tailored for nuclear power plant applications, taking inspiration from the conventional designs, which are prevalent in thermal power combined-cycle systems [16]. This comparative analysis underscored the inherent complexity in adapting heat recovery steam generators, which are standard in thermal power plants, for use within HTGRs, noting the advanced design and operational complexities that such applications entail [17].

Recognizing the pivotal role of finned tubes due to their superior heat transfer efficiencies [18,19], this study argues for their suitability across various emergency cooling system heat exchangers in HTGR contexts. Despite the apparent benefits, a notable gap in the literature regarding the performance characteristics of finned tubes under forced convection conditions with helium prompts an in-depth investigation.

Thus, this study embarks on a focused endeavor to optimize geometric characteristics of four distinct finned tube designs to enhance the heat transfer dynamics between water and helium, addressing a critical need for extended knowledge in this domain.

2. Materials and Methods

2.1. Thermal Assumption

This research delves into a comprehensive investigation, meticulously accounting for a broad spectrum of inlet and outlet conditions to mirror the complexities and variabilities characteristic of actual HTGR and VHTR operational scenarios. To establish a foundation for this exploration, Table 1 systematically delineates all presumed initial parameters.

Table 1. Initial parameters of helium and tubes for calculation.

Parameter, Unit.	Symbols	Value
Pressure of He, MPa	P_1	2.44
Inlet temperature of He, °C	t_1	527.5
Outlet temperature of He, °C	t_2	241.4
Outer diameter of tubes, m	d_{out}	0.0254
Thickness of tubes, m	δ_{tube}	0.002415
Tube height, m	l_{tube}	8
Material	Stainless steel	

This study positions helium, known for its unique thermophysical properties such as its low density, compressibility, and high thermal conductivity, in the spotlight as the primary coolant, contrasting its utilization against the backdrop of more conventional coolants, like molten metals or pressurized water, employed in the industry.

This study pays particular attention to the critical importance of tube geometry used in heat exchanger systems [12], asserting that the dimensions such as tube diameter, thickness, and height, as well as the allocative density of tubes per row, inherently influence the overall efficacy of heat transfer.

In adhering to established guidelines and leveraging empirical data drawn from [20,21], this investigation ensures a robust and adaptive framework to encapsulate the inherent variabilities integral to nuclear power plant operations, thereby contributing meaningful insights towards systematic optimization within this field.

As the Nusselt number indicates the contribution of the convective heat transfer of the helium motion to the conductive heat transfer, the Nusselt criteria for forced convection were calculated as follows [22]:

$$Nu = C_Z \cdot Nu_0 \begin{cases} Nu_0 = \frac{0.023 \cdot Pr \cdot Re^{0.8} \cdot C_t}{1+2.14 \cdot Re^{-0.1} \cdot \left(Pr^{0.7}-1\right)} \begin{cases} Re = \frac{w \cdot d}{v} \to w = \frac{G}{f \cdot \rho} \to f = l_{tube}(W_d - N_{row} \cdot d) \\ C_t = \left(\frac{Pr_{\text{fluid}}}{Pr_{\text{wall}}}\right)^n \end{cases} \\ C_Z = 1.048 - \frac{0.712}{z} + \frac{0.2837}{z^2} \to z = \mathrm{N_{row}} \end{cases} \quad (1)$$

where Pr is Prandtl number; C_t is the allowance for non-isothermal flow [23]; Nu_0 is Nusselt number of the single tube in center of a bundle (crossflow); C_Z is the correction factor; n is equal to 0.11 [23]; Re is Reynolds number; w is velocity of He in tube pack (m/s); and v is the kinematic viscosity at a given temperature. The calculations were made for the conditions of the crossflow over tube bundle.

Then, the heat transfer coefficient, which describes the intensity of the heat transfer process between the tube surface and helium, was calculated as

$$\alpha = \frac{Nu \cdot \lambda}{d}, \ \mathrm{W/(m^2 \cdot {}^\circ C)} \quad (2)$$

where λ is the thermal conductivity of the tube material. It was assumed to be equal to 15 W/m·°C, which is the typical value for stainless steel.

The higher the heat transfer coefficient, the greater the ability for thermal energy to move from one medium to another. The challenge is to increase this equivalent heat transfer coefficient by increasing the heat transfer area using extended surfaces. Using fins is the most widespread solution for such extended surfaces.

2.2. Fin Types and Characteristics

The strategic integration of fins into the design significantly amplifies the surface area available for heat transfer, thus propelling the efficiency of thermal management in nuclear reactor applications. This methodology not only proves to be cost-effective, bypassing the need for additional pressure piping, but also adapts to gradients of heat transfer coefficients, optimizing areas with inherently lower rates.

Crucially, the assumption of impeccable thermal contact between the fins and the tube walls underpins the reliability of subsequent calculations, setting a stringent manufacturing standard for the production of finned surfaces.

It is, however, essential to acknowledge the limitations inherent in current analytical methods, which fall short of accurately predicting heat transfer coefficients across the spectrum of finned surface typologies. Therefore, this exploration underscores the necessity of employing numerical simulations, corroborated by empirical testing, to bridge the gap in precision, thereby ensuring the fidelity and applicability of findings within the nuanced context of heat exchanger design and optimization of settings.

The overall heat transfer coefficient in the current study was determined as follows:

$$k = \left[\frac{A}{A_0 + A_f \cdot \eta_f} \cdot \frac{1}{\alpha_1} + \frac{\delta_{tube}}{\lambda_{tube}} + \frac{\delta_{fo}}{\lambda_{fo}} + 2\frac{\delta_{ox}}{\lambda_{ox}} + \frac{1}{\alpha_2} \right]^{-1}, \mathrm{W/(m^2 \cdot {}^\circ C)}. \quad (3)$$

where $\frac{\delta_f}{\lambda_f}$ is heat resistance caused by fouling layer (it was assumed to be equal to 0.000081 according to [24]) ($m^2 \cdot K/W$); $\frac{\delta_{tube}}{\lambda_{tube}}$ is heat resistance caused by tube wall ($m^2 \cdot K/W$); α_1 is convective heat transfer coefficient from the He to the tube wall; α_2 is convective heat transfer coefficient from the tube wall to the working fluid; λ_{tube} is the thermal conductivity of tube material (it was assumed to be equal to 15) ($W/m \cdot K$); $\frac{\delta_{ox}}{\lambda_{ox}}$ is heat resistance caused by oxidized layer (it was assumed to be equal to 0.00001 according to RD 24.035.05-89 thermal and hydraulic calculation [25]); A, A_0, A_f are the surface area of non-finned tube, section of tube between the fins, and area of the fins themselves (m^2); and η_f is fin efficiency.

In general form, the equation can be written as $k = \left[\frac{1}{\alpha_1} + \sum \frac{\delta_i}{\lambda_i} + \frac{1}{\alpha_2} \right]^{-1}$ where $\frac{1}{\alpha_1}$ is the convective heat resistance on one side, $\sum \frac{\delta_i}{\lambda_i}$ represents the sum of conductive heat resistances through various layers, and $\frac{1}{\alpha_2}$ is the convective heat resistance on the other side. The term $\frac{A}{A_0 + A_f \cdot \eta_f}$ in Equation (3) enhances the effective heat transfer area due to the presence of fins, thereby improving the overall heat transfer performance [26].

The concept of fin efficiency is crucial for evaluating fin performance. Understanding fin efficiency is critical for optimizing heat exchanger design. A high-efficiency fin maximizes the use of its surface area for heat transfer. A low-efficiency fin may be underutilized, resulting in poor heat transfer performance. However, increasing fin dimensions always results in decreasing efficiency, making optimization necessary. The following is the expression for fin efficiency.

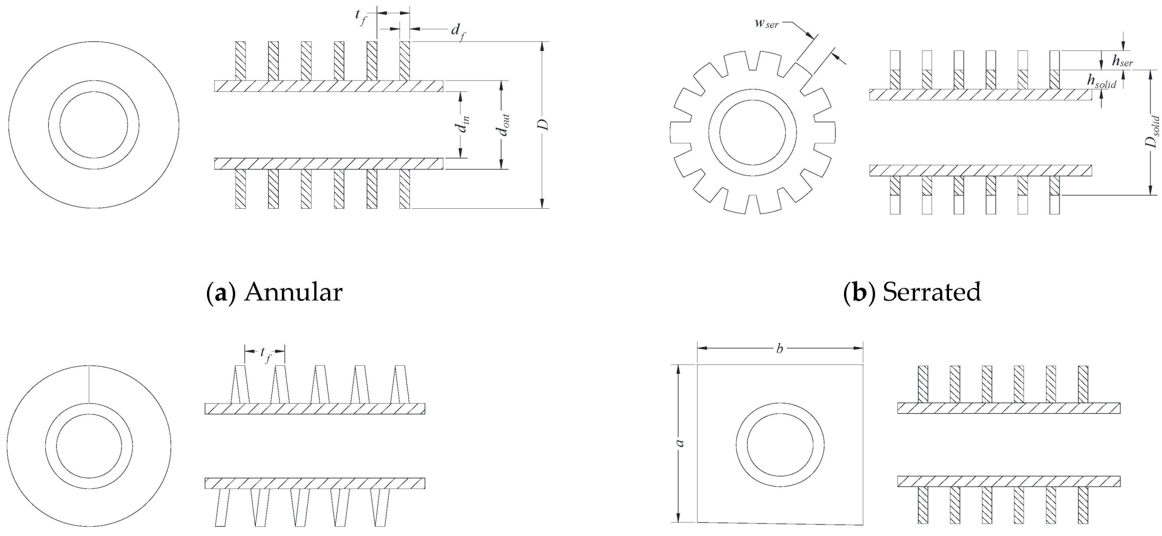
$$\eta_f = \frac{\tanh X}{X} \quad (4)$$

The dimensionless value X is calculated as

$$X = \varphi \cdot \frac{d_{out}}{2} \cdot \sqrt{\frac{2 \cdot \alpha}{\lambda \cdot \delta_f}} \quad (5)$$

where φ is the correction factor in different geometries; δ_f is the thickness of the fin.

Among the numerous variations of fin types, four basic types emerge: annular, helical, serrated, and square, each with unique characteristics, advantages, and disadvantages (Figure 1).



(**a**) Annular (**b**) Serrated

(**c**) Helical (**d**) Square

Figure 1. Image of finned tube types considered in current study.

Annular fins emerge as a paradigm of enhanced structural integrity and thermal dynamics, attributed to their circular form facilitating a broader heat transfer surface. Despite the allure of their efficacy, it is imperative to weigh the economic considerations stemming from their intricate manufacturing processes.

Helical fins have a spiraled configuration, which not only accentuates thermal transfer rates, but also adeptly navigates the challenges posed by fluctuating flow conditions, albeit with elevated production costs and potential constraints in application breadth.

Serrated fins introduce a novel tactic in thermal optimization by leveraging their distinctively notched edges to induce turbulence, thereby magnifying heat exchange effectiveness. Yet, this approach is not without its complexities, presenting potential hurdles in both manufacturing and ongoing maintenance.

Square fins offer a straightforward, cost-efficient solution to heat transfer enhancement, albeit potentially at the expense of reduced efficiency relative to their more complex counterparts. This exploration underlines the intricate balance between geometric design, manufacturing feasibility, and the overarching aim of optimizing thermal performance in heat exchanger applications specific to nuclear reactor technologies.

The choice of fin type can be determined after demonstrating the effect of each of them on the efficiency of heat transfer, taking into account their advantages and limitations.

The surface area A of the non-finned tube can be calculated as

$$A = \pi \cdot d_{out} \cdot l_{tube} \tag{6}$$

The surface area A_0 of the tube between the fins is as follows.

For annular, square, and serrated:

$$A_0 = \pi \cdot d_{out} \cdot l_{tube} \cdot \left(1 - \frac{\delta_f}{t_f}\right)$$

For helical:

$$A_0 = \pi \cdot d_{out} \cdot l_{tube} - \left(\mathrm{L_{turn}} \cdot l_{tube} \cdot \frac{\delta_f}{t_f}\right)$$

Surface area of the fins, A_f:

Annular:

$$A_f = 2 \cdot \frac{\pi}{4} \cdot \left(D^2 - d_{out}^2\right) \cdot \frac{l_{tube}}{t_f}$$

Square:

$$A_f = 2 \cdot \left(\mathrm{a}^2 - \frac{\pi}{4} \cdot d_{out}^2\right) \cdot \frac{l_{tube}}{t_f}$$

Serrated:

$$A_f = 2 \cdot \left(\frac{\pi}{4} \cdot \left(D_{solid}^2 - d_{out}^2\right) + \mathrm{n_{ser}} \cdot \mathrm{h_{ser}} \cdot \mathrm{w_{ser}}\right) \cdot \frac{l_{tube}}{t_f}$$

Helical:

$$A_f = 2 \cdot \left(\frac{\left(\mathrm{L_{turn}^{out}} + \mathrm{L_{turn}^{in}}\right) \cdot \mathrm{h}_f}{2}\right) \cdot \frac{l_{tube}}{t_f}$$

where $\mathrm{L_{turn}^{in}}$ is the length of one complete turn of the helical fin around the tube, and $\mathrm{L_{turn}^{out}}$ is the same parameter with extended fin height.

The correction factor φ is a crucial component in the fin efficiency equation, acting as a dynamic element that encompasses various influences on the heat transfer process. It serves as a multiplier, enhancing the accuracy in representing the unique attributes of various fin types within the system.

Essentially, φ takes into consideration variations in fin geometry, including factors like shape, surface irregularities, and structural intricacies. Flexibility is essential in situations where the fundamental variables in the equation may not fully encompass the intricacies of the heat transfer process. However, φ explores the various heat transfer characteristics displayed by different materials. Considering the crucial role of materials in determining heat exchange efficiency, the correction factor enables the equation to be customized according to the thermal properties of the fin material. This guarantees that it accurately reflects heat transfer in real-world scenarios [26,27].

$$\varphi = \left(\varphi' - 1\right) \cdot \left[1 + 0.35 \cdot \ln\left(\varphi'\right)\right] \tag{7}$$

Annular:

$$\varphi' = \frac{D}{d_{out}}$$

Square:

$$\varphi' = 1.28 \cdot \left(\frac{\mathrm{b}}{d_{out}}\right) \cdot \sqrt{\frac{\mathrm{a}}{\mathrm{b}} - 0.2}$$

Serrated:

$$\varphi' = \frac{D_{solid} + \mathrm{h_{ser}}}{d_{out}}$$

Helical:

$$\varphi' = \left(\frac{\mathrm{D}}{d_{out}}\right) \cdot \sqrt{\frac{\mathrm{D_1}}{\mathrm{D_2}} - 0.1}$$

$$\mathrm{D_1} = \sqrt{D^2 + {t_f}^2};\ \mathrm{D_2} = \sqrt{D^2 + \left(\frac{t_f}{2}\right)^2}$$

In the case of serrated and helical fins, the correction factor is mostly based on φ′, which describes the effective diameter or the ratio of diameter projection from one side to the ratio of diameter projection with 90° rotation as represented before.

While nuclear energy is known for its unique equipment requirements, the constant demand for reducing capital costs creates constant pressure to use more common and standard solutions. The fin height h_f and fin thickness δ_f are the most studied dimensions in the literature, but it was established that the number of fins per tube, which can also be normalized as fins per meter, should also be taken into account. These parameters were set at the following range as the most widespread values, which are currently used in various industrial areas (predominantly in fossil-fired energy production areas) according to the literature [27–29]:

For fin height: from 0.005 m to 0.01 m, with a step of 0.0005 m.

For fin thickness: from 0.0005 m to 0.005 m, with a step of 0.00045 m.

For fin frequency: from 160 to 200 fins per meter, with a step of 10.

3. Results and Discussion

As the principal aim of integrating fins is to enhance the heat transfer from the helium side, which leads to a noticeable reduction in the overall layout of the heat exchanger, it is crucial to emphasize that the optimization of fin dimensions directly affects the efficiency and safety of VHTR systems. These reactors, characterized by their high outlet temperatures and efficiency, significantly benefit from the improved heat transfer capabilities facilitated by optimized fin designs. However, in actual applications, manufacturing, maintenance, and cost constraints are also considered, putting certain limits on the optimization process.

3.1. Annular Type

The relationship between thicker fins and higher fin efficiency can be attributed to their thermal conductivity. Thicker fins provide a greater surface area for heat transfer with more constant temperature over their height, resulting in improved heat dissipation efficiency, as can be seen in Figure 2. The increase in the thickness of the fins from 0.0005 m to 0.005 m resulted in a noticeable increase in fin efficiency of almost 37%. Nevertheless, there is an ideal thickness that, when exceeded, leads to diminishing returns. It is probable that the observed phenomenon is a result of a rise in thermal resistance when the thickness surpasses the threshold for optimal conduction.

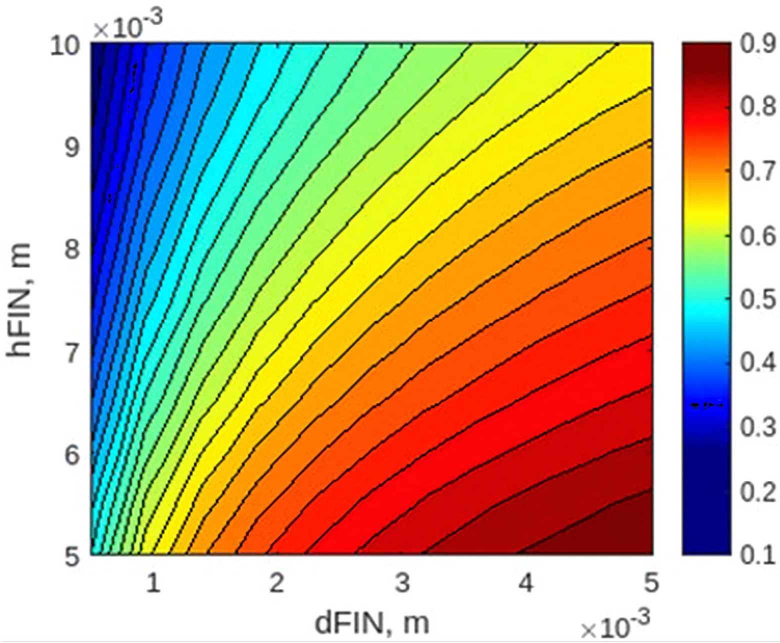


Figure 2. Dependence of annular−type fin efficiency on height and thickness.

On the other hand, an inverse relationship between fin height and fin efficiency was observed. The favorable fluid dynamics around the fins are influenced by the low density and viscosity of helium. Reducing the height of the fins can enhance convective heat exchange, allowing helium to flow more smoothly through the passages, and this can be established as well from Figure 2, which shows that fin efficiency decreases by about 25% when the height increases from 0.005 m to 0.01 m. In addition, shorter fins may undergo less heat loss to the helium environment because they have a smaller surface area exposed. However, smaller fins provide less surface area, meaning larger dimensions of corresponding equipment.

3.2. Square

The consistent factors of fin height, thickness, and number of fins per meter remain unchanged for both annular and square fins. Nevertheless, the different geometric configurations of annular and square fins result in differences in heat transfer properties, including surface area and helium flow patterns. Annular fins, with their circular shape, offer a continuous and smooth surface that may promote more consistent helium flow patterns. The geometry of this structure may play a role in enhancing convective heat exchange, leading to improved fin efficiency. Nevertheless, in the case of square fins, the presence of edges and corners can potentially disturb the flow, resulting in irregular flow patterns and heightened air resistance.

It can be clearly seen in Figure 3 that as the thickness increases from 0.0005 m to 0.005 m and height from 0.005 m to 0.01 m the maximum fin efficiency reaches 88.6%, which is lower than that of the annular shape. When discussing thermal conduction and heat distribution, it is worth noting that annular fins possess a continuous circular shape that enhances the efficiency of temperature distribution over the fin surface. However, when it comes to square fins, thermal gradients may arise due to the presence of distinct corners and edges, which could result in an uneven distribution of heat and lower efficiency.

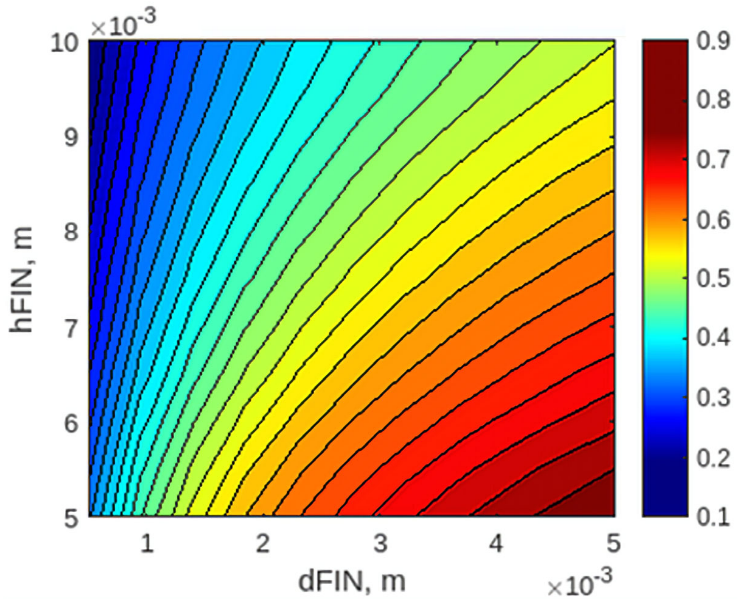


Figure 3. Dependence of square−type fin efficiency on height and thickness.

3.3. Serrated Fins

This study has shown that serrated fins have a higher heat transfer efficiency compared to annular and square fin types. Several factors contribute to this remarkable efficiency, such as the surface area and the serrated edge, which provides additional surface area, promoting enhanced convective heat exchange. As clearly shown in Figure 4, there is an increase in efficiency from 64.5% to 93.6% due to varying the thickness with the same values as in the previous shapes.

The presence of serrated ends enhances convective heat transfer coefficients and facilitates heat distribution. The serrations distribute heat along the edges, creating multiple smaller channels for heat conduction, as well as for reduced flow resistance. It has been observed, that the serrated configuration can potentially decrease flow resistance in comparison to square fins [29]. All these factors contribute to an enhanced fin efficiency.

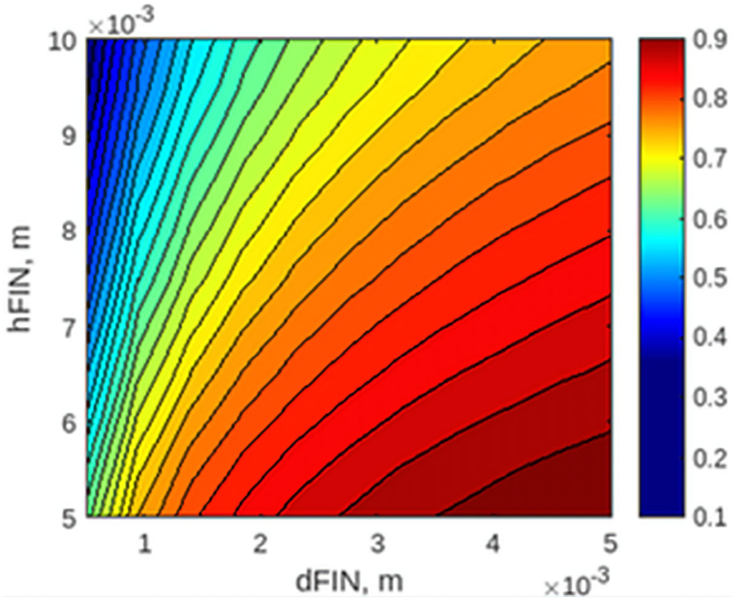


Figure 4. Dependence of serrated—type fin efficiency on height and thickness.

3.4. Helical Fins

Among the various types of fins, helical fins have emerged as the most efficient in terms of performance. Various factors contribute to their exceptional performance, including an extended surface area and an enhanced heat transfer path that offers a longer and uninterrupted surface area for heat exchange.

The helical fin has the highest efficiency compared to all other shapes, which is demonstrated in Figure 5. The highest recorded efficiency is 95.1% when both the fin height and thickness are 0.005 m, which is about 1.5% higher compared to the same-sized serrated fin. Additionally, the increased number of fins per meter increases velocity within the helium flow at a higher rate compared to other types. This is due to the different shape of each fin from side to side. This phenomenon can be attributed to the correlation between the number of fins per meter and the subsequent enhancement in fin efficiency.

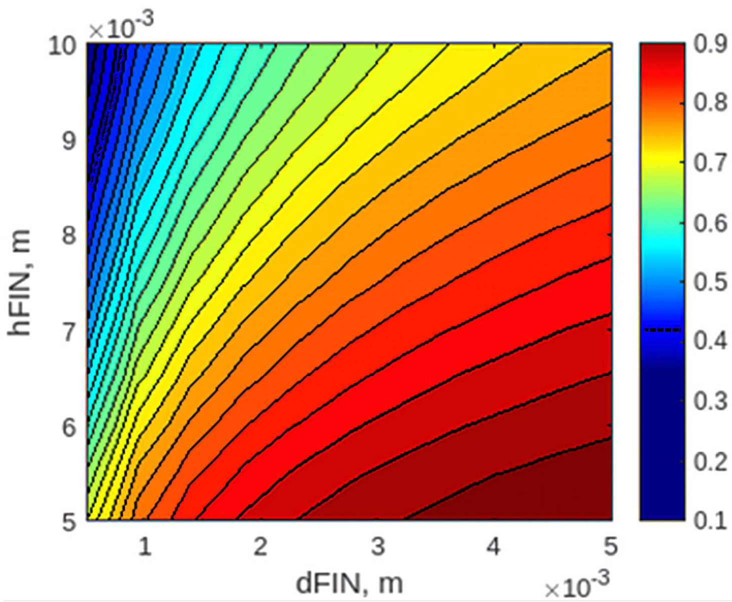


Figure 5. Dependence of helical—type fin efficiency on height and thickness.

Based on the data of Figure 6, it can be observed that there is a correlation between the number of fins per meter and fin efficiency. While this is quite small (less than 0.5% for all samples), it is still worth being considered, especially with the further expansion of the studied range of parameter variation. The effect of the number of fins per meter was stronger for thin fins than for thick ones: variation in efficiency was around 0.5% for 0.5 mm thickness fins, while for 5 mm thickness fins it was 0.1%. It is commonly observed that an increased number of fins tends to result in reduced fin efficiency in many cases.

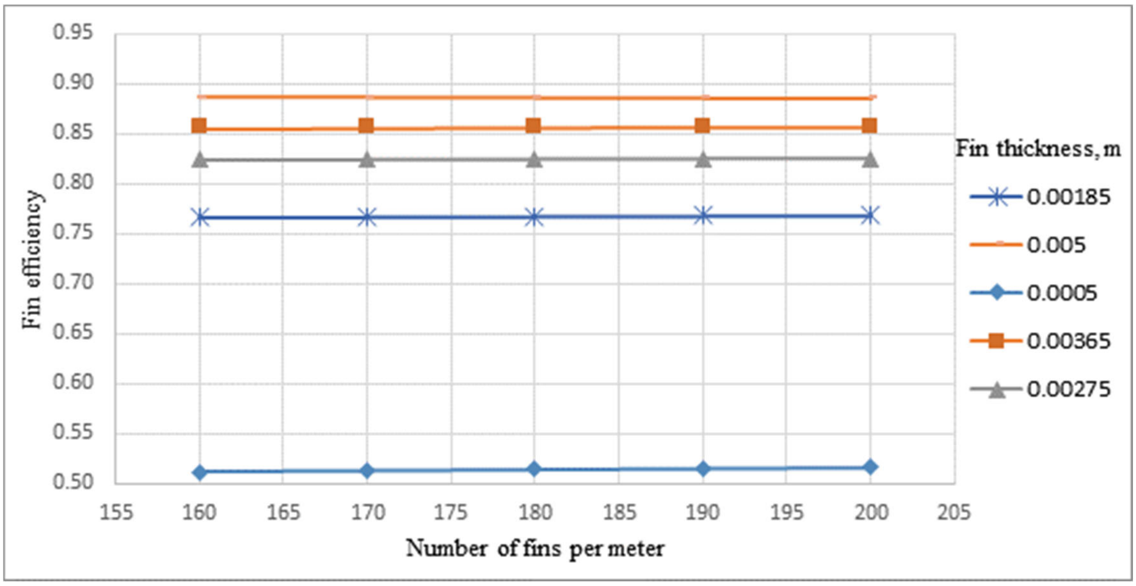


Figure 6. Effect of fin efficiency on amount of fins per meter at varying thicknesses of fins.

4. Conclusions and Prospects

This study's findings on the impact of creative fin designs on heat transfer efficiency have direct implications for improving the performance and reliability of VHTRs and HTGRs. As improving heat transfer efficiency in these reactors progresses, it becomes clear that creative fin designs offer great potential in boosting the performance and dependability of gas-cooled reactor systems. This research explored different fin geometries, providing insights into their unique characteristics and how they affect heat transfer efficiency.

After thorough analysis and experimentation, it was proven that innovative designs like serrated and helical fins surpass traditional fin types in terms of heat transfer efficiency, providing significantly better results. Serrated fins, with their irregular edges, bring about significant enhancements in efficiency, while helical fins, with their extended surface area and unique geometry, showcase exceptional performance in boosting heat transfer, whereas annular fins showed visibly worse performance.

These insights underscore the potential for these designs to contribute to the development of more efficient and safer VHTR and HTGR systems. However, it is important to acknowledge the limitations that arise from making assumptions based on existing research. These inaccuracies highlight the necessity for further research, especially in the realm of simulation models for finned heat exchangers in VHTR and HTGR conditions. This ongoing effort aims to refine thermal management strategies in advanced reactor technologies, ensuring the optimization of fin designs for enhanced performance and reliability.

Funding: This work was supported by the Ministry of Science and Higher Education of the Russian Federation.

Data Availability Statement: Data are available within the article.

Acknowledgments: Authors express gratitude to the TPU development program.

Conflicts of Interest: The authors declare that they have no known conflict or competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

Abbreviations

Nomenclature	
BC	Brayton cycle
CC	Combined cycle
CCPP	Combined-cycle power plant
GCR	Gas-cooled reactors
GT	Gas turbine
He	Helium
HGTS	Helium Gas Turbine System
HTGR	High-temperature gas-cooled reactor
NPP	Nuclear power plant
TPs	Thermodynamic properties
Parameters	
d	Diameter, m
f	Cross-section area, m^2
G	Flow rate of helium coolant, kg/s
H	Heat drop, kJ/kg
h	Specific enthalpy, kJ/kg
k	Overall heat transfer coefficient, W/(m^2·K)
l, L	Length, height, m
Nu	Nusselt number
P	Pressure, MPa
Q	Thermal power, MW
Re	Reynolds criteria
t	Temperature, °C
v	Kinematic viscosity, m^2/s
α	Heat transfer coefficient, W/(m^2·K)
δ	Thickness, m
Δt	Temperature difference, °C
η_f	Fin efficiency, %
λ	Thermal conductivity, W/(m·K)
ρ	Density, kg/m^3
Subscript	
Δt_{pp}	Temperature difference at pinch point
δ_f/λ_f	Thermal resistance of fouling layer
δ_{ox}/λ_{ox}	Thermal resistance of oxidizing layers

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