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Article

Numerical Investigation of Heat Transfer Intensification Using Lattice Structures in Heat Exchangers

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Abstract: Heat exchangers make it possible to utilize energy efficiently, reducing the cost of energy production or consumption. For example, they can be used to improve the efficiency of gas turbines. Improving the efficiency of a heat exchanger directly affects the efficiency of the device for which it is used. One of the most effective ways to intensify heat exchange in a heat exchanger without a significant increase in mass-dimensional characteristics and changes in the input parameters of the flows is the introduction of turbulators into the heat exchangers. This article investigates the increase in efficiency of heat exchanger apparatuses by introducing turbulent lattice structures manufactured with the use of additive technologies into their design. The study is carried out by numerical modeling of the heat transfer process for two sections of the heat exchanger: with and without the lattice structure inside. It was found that lattice structures intensify the heat exchange by creating vortex flow structures, as well as by increasing the heat exchange area. Thus, the ratio of convection in thermal conductivity increases to 3.03 times. Also in the article, a comparative analysis of the results obtained with the results of heat transfer intensification using classical flow turbulators is carried out. According to the results of the analysis, it was determined that the investigated turbulators are more effective than classical ones, however, the pressure losses in the investigated turbulators are much higher.

Keywords: additive technologies; energy; heat exchange; heat regeneration; intensification of heat transfer; lattice structures; turbulators; energy

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1. Introduction

Power engineering is the backbone of any state. This sector is engaged in the production of equipment for the energy industry, the main objects produced are gas turbine and steam turbine units for CHPPs, HPPs, and NPPs. The ability of the state to realize the production of these turbomachines reflects its technical, economic, and environmental conditions.

The current stage of turbomachinery development is directly related to increasing its economy, environmental friendliness, and efficiency. The ways to achieve these goals are different and can involve both classical and modern approaches. The authors employed a hybrid approach to improve efficiency, environmental friendliness, and economy, which consists of the use of high-efficiency heat exchangers. The presence of a heat-exchange apparatus in the gas turbine unit (GTU) allows for an increase in the efficiency of the power plant by reducing the amount of heat required to supply the thermodynamic cycle, to increase environmental friendliness due to effective utilization of heat energy at the outlet from the turbine part of the GTU, as well as to increase efficiency by reducing the consumption of fuel burned in the GTU.

A heat exchanger is a type of auxiliary heat engineering equipment used to transfer heat energy from a heat carrier to a less heated body for the realization of various thermal processes. One of the parameters characterizing its efficiency is the degree of heat recovery, which is equal to the ratio of the real amount of heat supplied to the body to the maximum theoretically possible. Convective heat exchange, which is seen in surface heat exchangers, is characterized by Nusselt number, which is the ratio between the intensity of heat exchange due to convection and intensity due to heat conduction.

However, the high efficiency of heat exchangers is mainly achieved by realizing multi-pass systems of working body motion, which leads to an increase in mass and dimensional characteristics. Also, factors such as temperature difference between hot and cold medium, flow velocity, heat exchange surface area, and heat exchanger design influence heat exchange efficiency. Thus, increasing heat exchanger efficiency in modern technologies is urgent.

The field of application of heat exchangers is wide and covers technological processes in oil refining, chemical, refrigeration, gas, power, and other industries. The use of heat exchangers allows for increased efficiency of the device for which it is used. For example, the use of heat exchangers in gas-turbine units reduces fuel consumption, which will have a positive impact on economic indicators.

This article is devoted to the study of increasing the efficiency of heat exchangers manufactured by additive technologies using lattice turbulators, which are a single structure with the heat exchanger. In the course of this research, the previously used designs of heat exchanger turbulators [1,2] as well as metal volume printing technologies were analyzed. The aim of the work was to create models of heat exchanger sections with and without integrated turbulent lattice, to perform gas-dynamic calculations, as well as a comparative analysis of the calculation results.

The use of heat exchangers to improve the efficiency of gas turbines has been applied since the middle of the last century, but the heat exchanger as an element of the gas turbine unit has a relatively low efficiency. The increase in efficiency of this equipment is faced with the observance of balance between realized heat exchange processes and optimal hydraulic resistance from the system side.

Various research groups around the world are searching for ways to achieve this balance. Studies on increasing the efficiency of heat exchangers are carried out in the following countries: the USA, China, Russia, and others. Each of the countries uses its own approach to solve this problem, relying on its technological, design, and experimental experience.

Considering the possible ways to improve the efficiency of heat exchangers, including:

1. Increasing the heat exchange area: can be achieved by increasing the number of plates or tubes in the heat exchanger.
2. Optimizing the parameters of the working bodies: adjusting the flow rate of the fluid can improve efficiency. Changing the flow rate, pressure, or temperature of the medium.
3. Improving thermal insulation: minimizing heat loss to the environment can also increase the efficiency of the heat exchanger.
4. Cleaning and maintenance: regular cleaning and maintenance of the heat exchanger helps to keep it running and efficient. Accumulation of deposits and contaminants on the heat exchange surface can significantly reduce the efficiency of the apparatus.
5. Using additional devices to improve heat exchange: for example, using special coatings on the heat exchanger surface or using additional devices such as turbulators.

The use of turbulators as a method of heat exchange intensification is more advantageous; it increases the efficiency of heat exchange at the same boundary conditions obtained at the inlet without significant changes in mass-dimensional characteristics.

A turbulator in a heat exchanger is a structural element that is used to increase the turbulence of the medium flow. It helps to increase the efficiency of heat transfer between the two media by increasing the heat transfer coefficient.

The role of the turbulator is to create a pulsating or swirling motion of the liquid or gas inside the heat exchanger. This, in turn, promotes mixing of thermal boundary layers and increases heat transfer between heat exchange surfaces.

One of the confirmations of the effectiveness of turbulators for heat exchange intensification is the experimental study of Chernyaev L. A. [1] in which an experiment was carried out to study the effect of fluid turbulization on the parameters of the heat exchanger. In this experiment, heat exchangers without turbulators and with turbulators in the form of twisted inserts were alternately placed in a thermal wind tunnel. The average heat transfer coefficient of the heat exchanger surface was given as a parameter for evaluating the efficiency of the heat exchanger:

$$U = 1163 \times Q_W / (\Delta T_{ln} \times A),\ W/\left(m^2 \times K\right), \quad (1)$$

where Q_W—heat transfer from the working body to the heat exchanger surface, ΔT_{ln}—mean logarithmic mean temperature difference between the domains, and A—heat transfer surface area.

Figure 1 shows graphs of changes in hydraulic resistance and heat transfer coefficient depending on the flow rate of the working body for heat exchangers with and without turbulator [1].

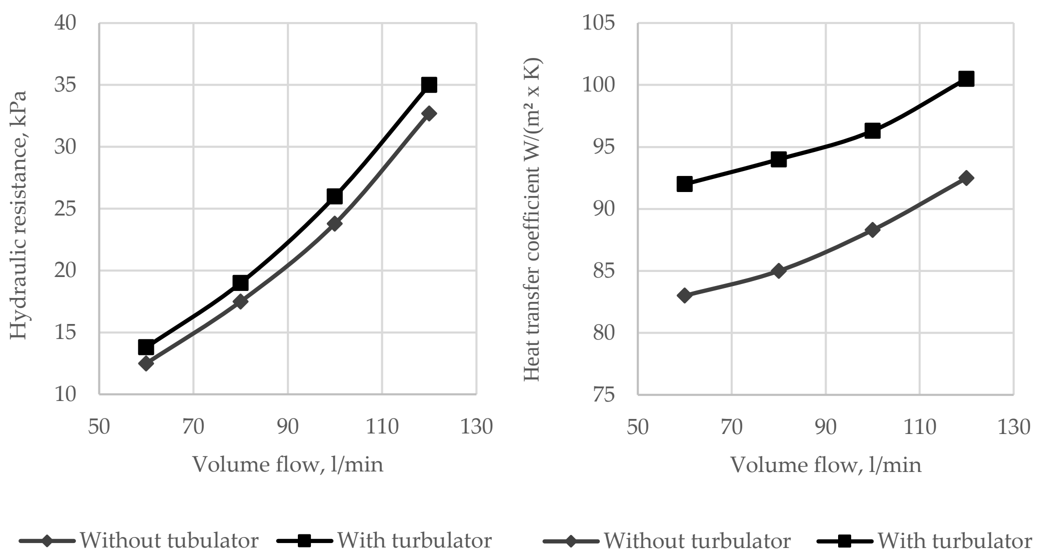

Figure 1. Plots of changes in hydraulic resistance and heat transfer coefficient depending on the flow rate of the working body for heat exchangers with and without turbulators.

Thus, it can be seen from this experiment that the use of turbulators has a positive effect on the heat transfer coefficient, but increases the hydraulic resistance, thereby reducing the pressure.

Another illustrative example is the study of Gorobets V.G. [2]. The comparative analysis of heat transfer and hydraulic resistance of tube bundles with different types of finning is carried out. The investigated surfaces with different types of finning are shown in Figure 2.

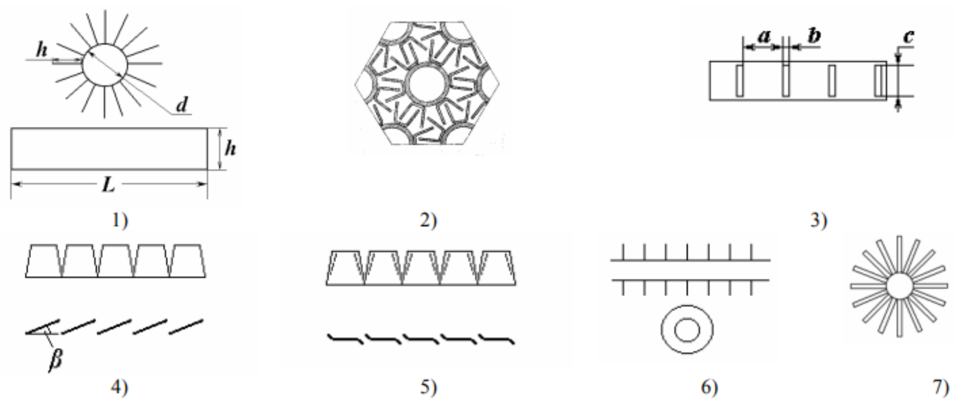

Figure 2. Surfaces with different types of finning: (**1**)—longitudinal continuous finning; (**2**)—longitudinal finning in close bundles; (**3**)—longitudinal perforated finning; (**4**)—longitudinal incised finning with rib rotation; (**5**)—longitudinal incised finning with edge bending; (**6**)—transverse continuous ribbing; (**7**)—transverse petal ribbing.

In this study, as a qualitative criterion reflecting the efficiency of heat transfer intensification with the addition of fin-turbulators, the ratio of Nu/Nu_{smooth}. It shows how many times the Nusselt number changes for a finned tube relative to a smooth tube. The graph of dependence Nu/Nu_{smooth} of Reynolds number for tube bundles with different types of finning is shown in Figure 3.

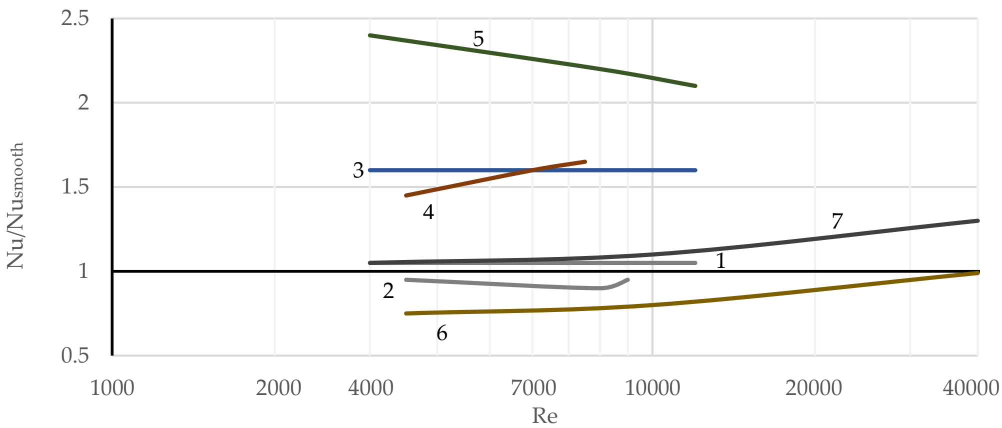

Figure 3. Dependence of Nu/Nu_{smooth} parameter on Re number for tube bundles with different types of finning: 1—longitudinal continuous finning; 2—longitudinal finning in close bundles; 3—longitudinal perforated finning; 4—longitudinal incised finning with rib rotation; 5—longitudinal incised finning with edge bending; 6—transverse continuous ribbing; 7—transverse petal ribbing.

In this study [3], the authors installed limiters in the gaps between the batteries, which can effectively improve the heat dissipation characteristics of the battery. In fact, these are also a kind of turbulator. In another study [4], various technologies were numerically tested to achieve maximum thermal efficiency of the Parabolic trough collector. One of the methods was the use of flow turbulators, mainly inserts and internal ribs or recesses in pipes. According to the final results, the use of internal ribs increases efficiency by 1.10%. Although it seems to be a small value, taking into account the longer operating time, it will give significant cost reductions. Despite the long-standing use of turbulators in heat exchangers, the works of various authors who optimize their designs or propose new designs continue to be relevant [5].

Having considered various types of turbulators, it is possible to pay attention to their simple design, depending on the possibilities of manufacturing technologies by classical methods (casting, stamping, metal cutting and so on). With the development of additive technologies, it has become possible to integrate more complex turbulent surfaces directly into the design of the heat exchanger at the design stage. In this case, the heat exchanger and turbulator represent monolithic construction. In this way, a comprehensive improvement of the heat exchanger is realized in terms of solving the problems of increasing the efficiency of heat exchange between two moving working bodies and improving the strength characteristics caused by the addition of internal ribs that increase the rigidity of the structure.

Additive manufacturing is the process of creating objects by applying material layer by layer based on a three-dimensional model. Unlike traditional manufacturing methods where materials are usually removed or specially deformed to create the shape of an object, in additive manufacturing, materials are added sequentially to create the final product. The principle of metal 3D printing has made it possible to create objects with geometries of high complexity.

The additive manufacturing process typically involves the following steps:

1. Creating a three-dimensional model of the object;
2. Partitioning the model into thin horizontal layers;
3. Sequential printing of each layer using an appropriate manufacturing method (e.g., material surfacing, resin polymerization, powder splicing).

Additive technologies are widely used in a variety of industries. These technologies provide a high degree of flexibility and the ability to customize production. Also, they can be used to create complex geometries and lightweight lattice structures that are often not available using traditional manufacturing methods.

Lattice structures have already found applications in mechanical engineering, prosthetics, sports equipment manufacturing, and many other sectors [6]. Examples of the use of lattice structures are shown in Figure 4.

Figure 4. Examples of the use of lattice structures.

Having considered various types of turbulators, it is possible to pay attention to their simple design.

Lattice infill has unique properties that often cannot be fully achieved using traditional fabrication methods. In certain cases, the use of lattice structures can reduce the amount of material used. Another beneficial property of such structures is their increased strength, as they effectively absorb energy and distribute stresses evenly. Also, they are highly rigid which will have a positive effect on both static and vibration strength. In addition, lattice structures have a relatively high surface area, relative to the counterpart without filling, and this will have a positive effect on the efficiency of heat exchangers.

In the article [7] a review of different types of gratings, their properties, and applications has been carried out. From the point of view of construction, gratings can be divided into several types: gratings based on beams, as well as gratings based on triply periodic minimal surfaces (TPMS). Figure 5 shows some types of grids classified by construction method.

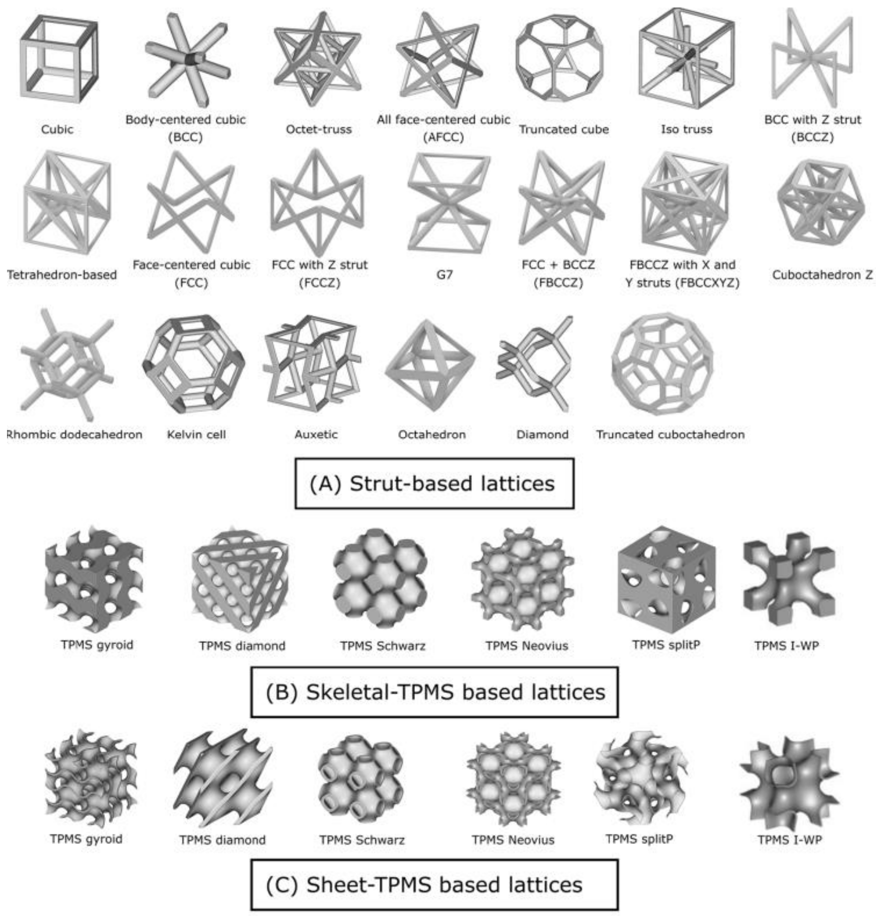

Figure 5. Types of lattices.

A minimal surface is an implicitly equivalent surface with zero mean curvature. If the minimal surface is periodic in three independent directions, it is called a triply periodic minimal surface. This structure can be expressed by a trigonometric function. By varying the parameters of the TPMS lattice, the internal pore structure can be precisely controlled, the pore gradient structure can be optimized, and the specific surface area of the framework can be maximized [8].

In contrast to other methods of grating construction, based on the construction of various beam shapes or on the Voronoi tessellation method, TPMS grids have a larger specific surface area, which can positively affect the intensification of heat transfer. Also, such grids have a smoother transition from the channels and thus, in theory, a comparatively lower hydraulic resistance. Another advantage of TPMS grids is the best mechanical properties among other types of grids [8].

One type of TPMS-type lattice structures is the gyroid. Gyroids were first described in 1970 by the American scientist Alan Schoen. There are examples of gyroid in nature: structures of some materials, viscous liquids, mitochondrial membranes, butterfly wing scales, and many others.

The gyroid is a periodic surface, trigonometrically approximated by the formula:

$$\sin(x) \times \cos(y) + \sin(y) \times \cos(z) + \sin(z) \times \cos(x) = 0. \quad (2)$$

Figure 6 shows a gyroid lattice.

By applying such structures in the design of heat exchangers it is possible to increase the degree of heat recovery. This is achieved by creating vortex structures within the flow and increasing the heat exchange surface.

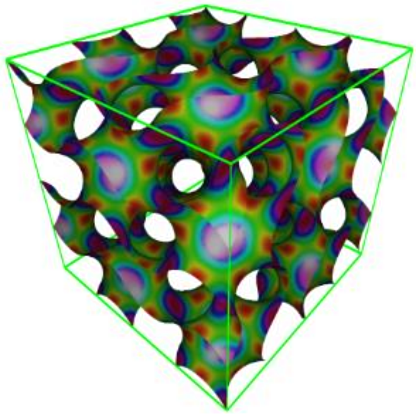

Figure 6. Gyroid lattice.

The disadvantages of using these structures in heat exchangers include increased pressure losses due to hydraulic resistance, as well as more difficult maintenance compared to classical heat exchangers. The maintenance of such heat exchangers will have certain difficulties since such curved channels will be more difficult to clean from solid particles by means of purging. These difficulties, in turn, lead to increased requirements for the preparation of working bodies, and this suggests the need for the use of various devices such as air cleaners, ash collectors, and so on. Also, in the case of the use of such heat exchangers in gas turbine engines, it leads to the requirement of higher quality fuel in order to reduce the content of particulate matter in the working body.

In the article [9], a computational study of a pipe section was performed using a TPMS lattice [9]. Figure 7 shows the computational model they used.

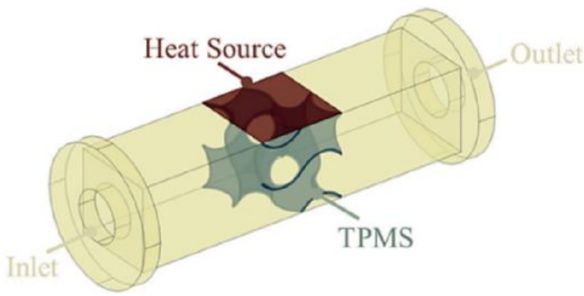

Figure 7. Model of the study [9].

This study was carried out in the COMSOL calculation software. As a result of the calculations, the plots of pressure drop, heat transfer coefficient, and Nusselt number dependence on the gyroid lattice wall thickness and Reynolds number were obtained as shown in Figure 8.

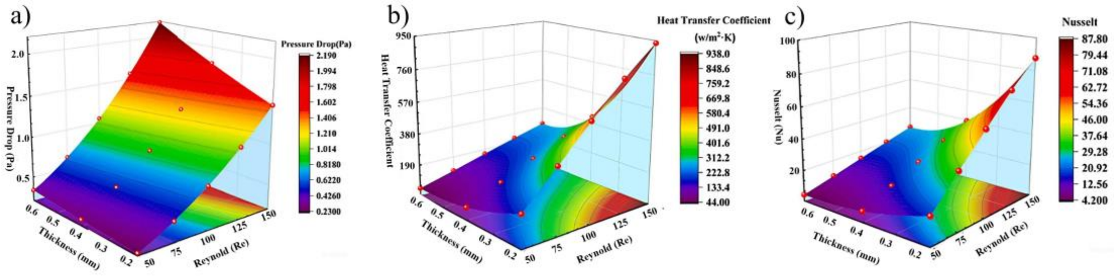

Figure 8. Plots of pressure drop (**a**), heat transfer coefficient (**b**) Nusselt number (**c**) dependence on gyroid lattice wall thickness and Reynolds number.

The use of lattice structures in heat exchangers is devoted to the works of other authors [10–12]. However, despite new studies of lattice structures, it would be extremely useful to estimate the increase in convective heat transfer due to their introduction into the heat exchanger. Thus, this study will allow us to apply the classical methodology for evaluating the effectiveness of fundamentally new flow turbulators in lattice structure form for heat exchangers.

2. Methodology and Methods

This study was conducted using computer-aided engineering software Ansys 19 R2 A block diagram of the algorithm for conducting the study is presented in Figure 9.

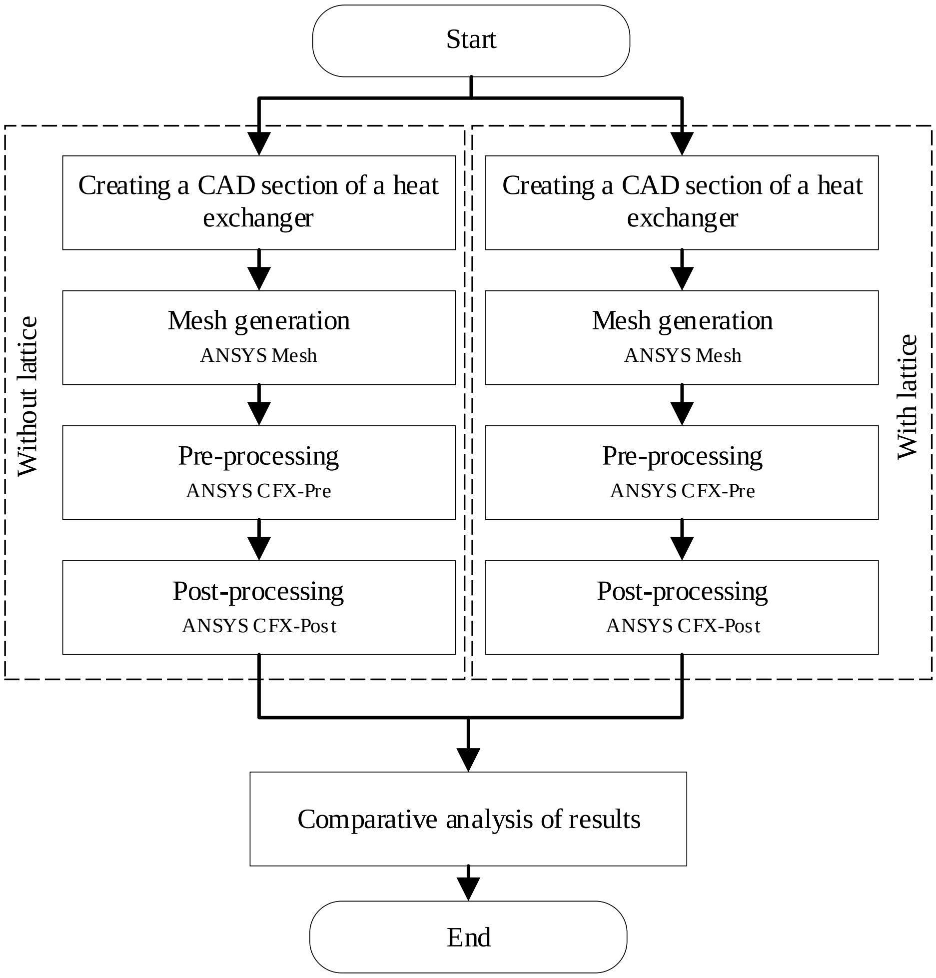

Figure 9. Block diagram of the study of heat transfer intensification through the use of gyroid lattice structures.

To investigate the intensification of heat transfer by introducing a gyroid lattice structure, two small sections of a countercurrent heat exchanger of the "pipe-in-pipe" type were modeled, in which the hot coolant flows in the inner circuit and the cold coolant flows in the outer circuit. In the outer loop of one of the investigated heat exchanger sections, a lattice gyroid structure was introduced in its central part. A sketch of the heat exchanger is shown in Figure 10.

With the advent of metal printing capabilities, groundbreaking software has begun to emerge that automates the process of creating such complex structures. This approach to design allows the creation of complex and optimized lattice structures with a high degree of automation, which saves design time and resources. Today, there are a number of programs that allow the design of lattice structures, including nTopology, Triangulatica, Gen3D Sulis, Autodesk Fusion 360, Siemens NX, and others [13].

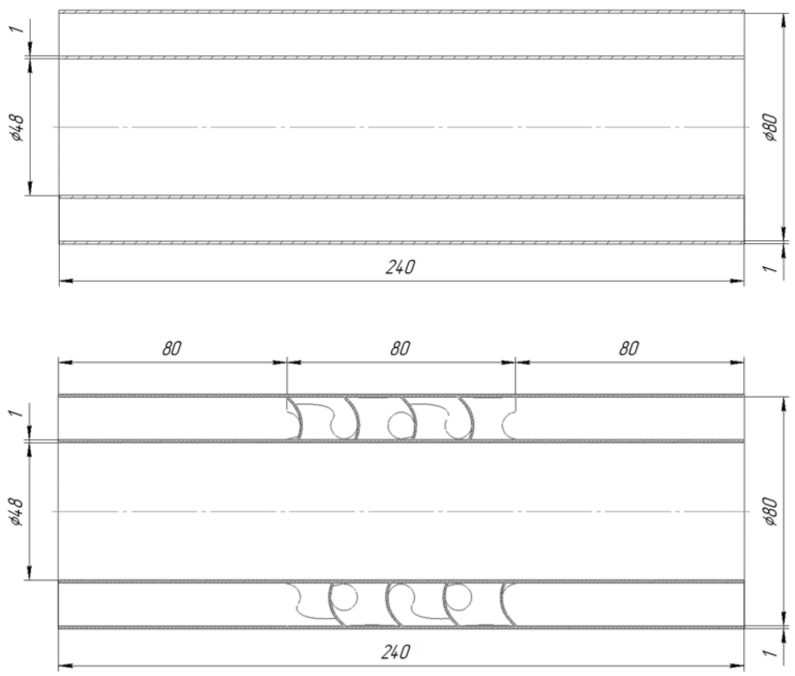

Figure 10. Sketch of heat exchanger sections.

The nTopology 4.18.2 program was used to create a heat exchanger with a gyroid lattice structure in the outer loop. The periodicity cell size of the gyroid lattice was 40 mm and the wall thickness was 1 mm.

The geometric models of both heat exchanger sections are shown in Figure 11.

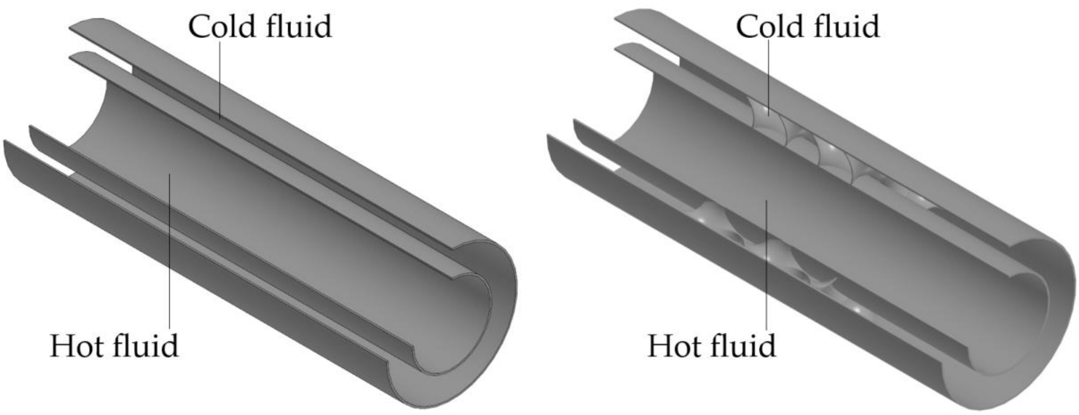

Figure 11. Geometric models of the studied heat exchanger sections in cross-section.

The geometric properties of the heat exchanger sections are presented in Table 1.

The material properties of the heat exchanger and the fluid (air) are shown in Table 2. The table shows the parameters of the heat exchanger material and for air at temperatures of 500 and 1000 K. When specifying the material properties in ANSYS CFX, temperature nonlinearity was taken into account. The data array for each of the parameters was taken at different temperatures with a small step for approximation by a fifth-order polynomial.

Table 1. The geometric properties of the heat exchanger sections.

Parameter	Units of Measurement	Value
Inner diameter of the pipe	mm	48
Outer diameter of the pipe	mm	80
Thickness of all walls	mm	1
Length of the heat exchangers sections	mm	240
Length of the lattice section	mm	80
Periodicity size of the gyroid lattice	mm	40

Table 2. The material properties of the heat exchanger and the fluid.

Parameter	Units of Measurement	Value at Temperature					
		500 K	600 K	700 K	800 K	900 K	1000 K
Thermal conductivity of the air λ	$\frac{W}{m \times K}$	0.041	0.048	0.054	0.058	0.064	0.071
Heat capacity of the air c_p	$\frac{J}{kg \times K}$	1032	1053	1076	1097	1119	1141
Dynamic air viscosity μ	$Pa \times s$	26.8×10^{-6}	30.6×10^{-6}	33.8×10^{-6}	37.0×10^{-6}	39.9×10^{-6}	42.5×10^{-6}
Thermal conductivity of the steel λ_s	$\frac{W}{m \times K}$	66.6	57.0	50.5	44.0	37.5	31.0
Heat capacity of the air c_{p_s}	$\frac{J}{kg \times K}$	481	498	517	545	572	600
Steel density ρ_s	kg/m^3	7791	7755	7720	7681	7642	7605

The working body was formed by filling the space inside the pipes to create computational models.

Then the mesh was formed by finite volume method with the addition of densification near the walls to model the boundary layer. In creating the computational mesh, a mesh independence study was also conducted to determine the optimal number of computational mesh cells at which the independence of the results obtained from the number of elements is achieved. The grid independence study was performed for the flow regime with $Re = 7500$ for both sections of the heat exchanger. Thus, it was found that for the grid section, 27.5 million cells were sufficient to obtain a reliable result, further reduction of the cell size did not affect the result. Similarly, 10.65 million cells were sufficient for the computational grid without an integrated grid. A graph with the results of testing the mesh model for mesh independence is shown in Figure 12.

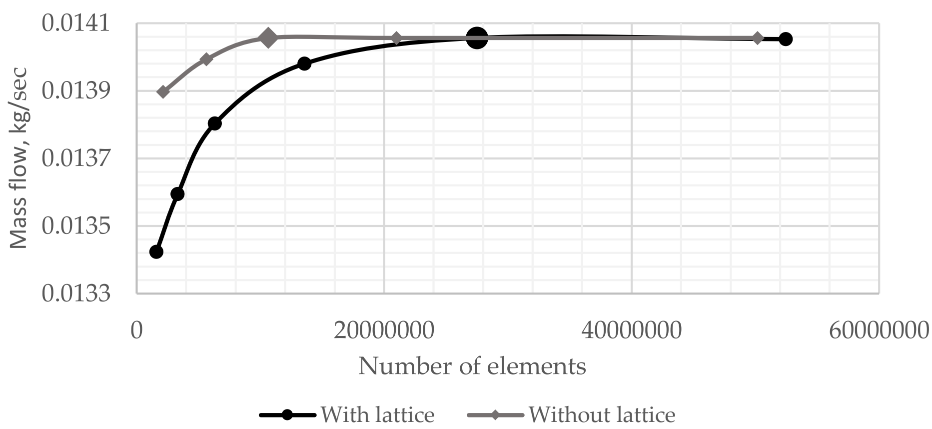

Figure 12. Mesh independence test results.

The total number of elements of the grid is 27,502,826 cells for a mesh for a section with a lattice structure and 10,656,831 for a mesh for a section without a lattice structure. The appearance of the mesh is shown in Figure 13.

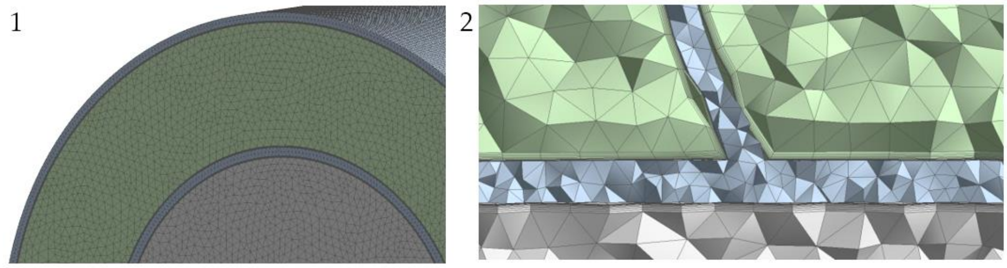

Figure 13. Mesh model: (**1**)—view from the flow inlet side; (**2**)—cross-sectional view.

The quality plot (Aspect Ratio) is shown in Figure 14. The mesh model for a heat exchanger section with a lattice structure is more complex than the model for a smooth section. When building the mesh for this model, it was important to make the correct settings to obtain high quality. The quality of the mesh can be estimated by the Aspect Ratio plot, ideally most of the elements should have this parameter in the region 1–3.

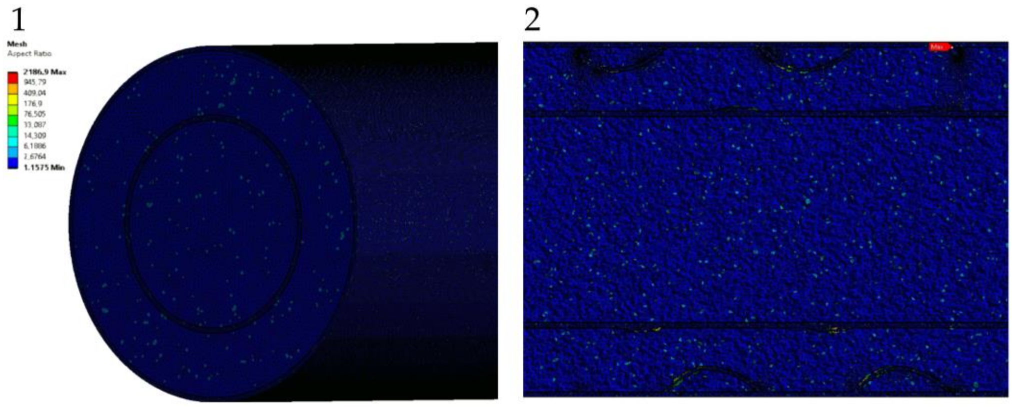

Figure 14. Aspect Ratio: (**1**)—view from the flow inlet side; (**2**)—cross-sectional view.

The graph of the distribution of percentages of the total number of elements is shown in Figure 15. As can be seen from Figure 15, more than 99.9 percent of the elements have an Aspect Ratio of no more than 3.

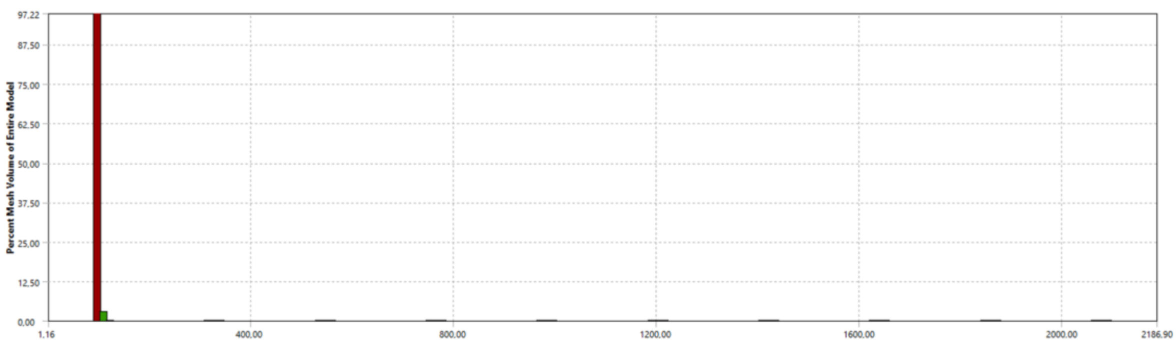

Figure 15. Aspect Ratio graph.

A similar analysis was performed when analyzing a simpler calculation mesh for a smooth section of the heat exchanger

After that, computational models were created in ANSYS CFX preprocessor by creating domains, transfer boundaries (interfaces) considering heat exchange between different media, and setting the boundary conditions of the working bodies. Three domains were created for each of the calculations: the domain of the coolant, the cooler, and the heat exchanger walls. Customization of the calculation model implied the selection of the turbulence model, energy equation, and interface parameters. For the working bodies, the parameter for solving the heat transfer equation "Total Energy" was selected, which takes into account the kinetic energy of the flow in heat transfer. For the body of the heat exchanger walls the standard parameter "Thermal Energy" was chosen, excluding the possibility of using the separating walls of the heat exchanger as adiabatic. A two-parameter SST k-ω Menter turbulence model was also selected, taking into account the compressibility effect. This model combines the advantages of the standard k-ε and k-ω models, allowing a sufficient level of point and physicality to perform calculations both near the boundary layer and away from the walls. It is also possible to take into account the laminar-turbulent transition, and the Gamma-theta transition model was chosen to take it into account. The General Grid Interface (GGI) model with the activated heat transfer function was used for the possibility of interfacing the contacting boundaries of different bodies.

It is worth noting that the object of research is thermally insulated from the environment. Ideal gas air was chosen as a working body and structural steel was chosen as the heat exchanger material. In the settings of the working bodies, domains for both calculation variants identical parameters were set for the heat transfer fluid and flow cooler which are presented in Table 3. The velocity of the working bodies was the chosen characteristic for different types of flows: laminar flow ($Re = 1000$) and turbulent ($Re = 7500$ и $Re = 40,000$).

Table 3. Boundary Conditions.

Parameter	Units of Measurement	Cold Fluid	Hot Fluid
Inlet velocity v_{in}	m/s	0.87/6.5/34.7	0.87/6.5/34.7
Inlet temperature T_{in}	K	500	1000
Outlet pressure P_{out}	Pa	101,325	101,325

3. Results

Next, a coupled gas dynamic calculation was performed considering the heat transfer between the coolants through the steel wall, and the results are presented below in Figures 16–19.

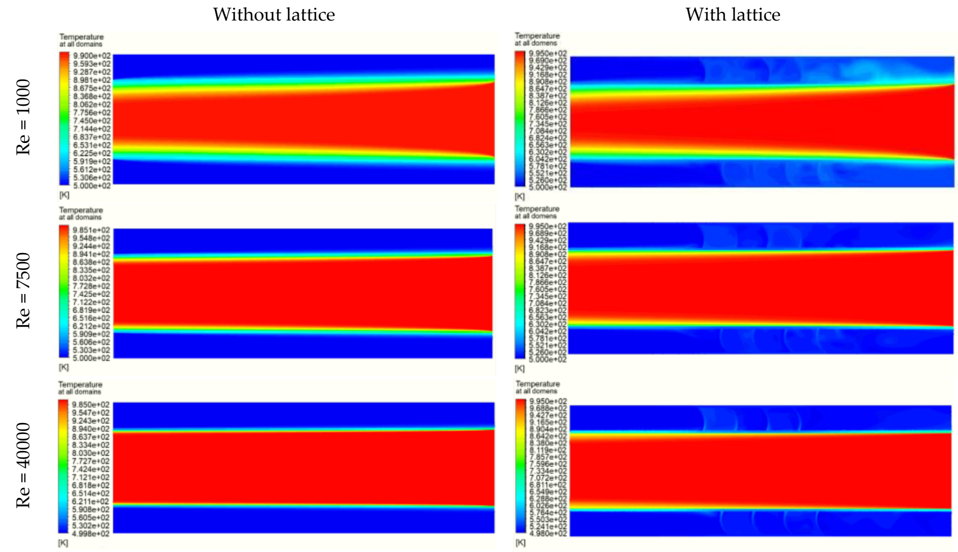

Figure 16. Temperature distribution in all domains of the heat exchanger for different calculation variants.

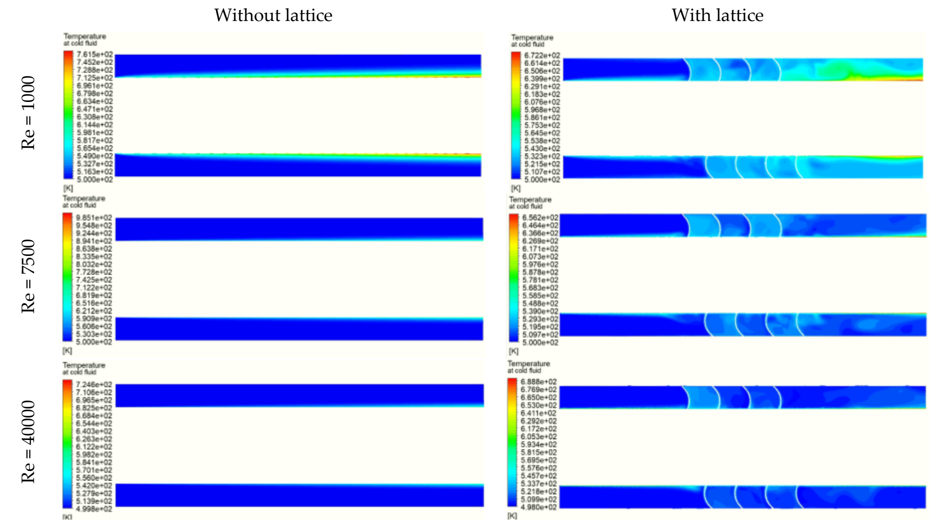

Figure 17. Temperature distribution in the cold domain for different calculation variants.

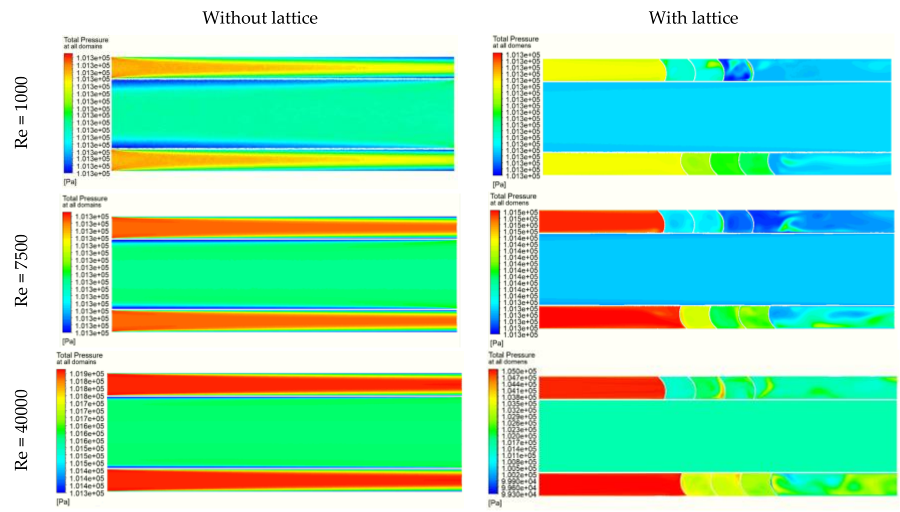

Figure 18. Full pressure distribution in all domains of the heat exchanger for different calculation variants.

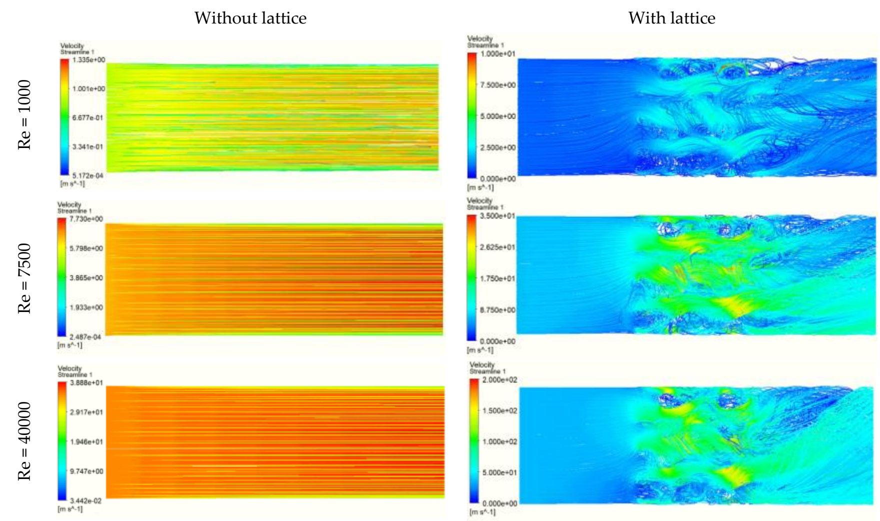

Figure 19. Current lines (velocities) in the cooler domain for different calculation variants.

After that, the CFX-Post was used to determine the flow parameters necessary for further comparative analysis. Static parameters were averaged by area and volume; dynamic parameters were averaged by mass flow rate.

Returning to the previously mentioned criteria for evaluating the efficiency of convective heat transfer, the classical formula for calculating the Nusselt number was used:

$$Nu = \frac{h \times l}{\lambda}, \tag{3}$$

where λ—thermal conductivity of the air, l—characteristic size (hydraulic diameter, for annular channel $d_h = \frac{4F}{P} = D - d$), h—heat transfer coefficient, which in turn, according to the Newton-Richman equation, is equal to:

$$h = \frac{Q}{A \times \left(t_w - t_f\right)} = \frac{q}{t_w - t_f}, \tag{4}$$

where Q—heat flux, A—heat transfer area, t_w—wall temperature, t_f—fluid temperature.

Also, the coefficient of local pressure losses was determined (friction losses were not taken into account in the study because local pressure losses play a major role):

$$\zeta = \frac{P^*_{in} - P^*_{out}}{\rho \times c^2/2}. \tag{5}$$

The parameters determined from the data obtained from the calculations are presented in Table 4.

Table 4. Calculation results.

Parameter	Unit	Variants					
		Re = 1000		Re = 7500		Re = 40,000	
		Without Lattice	With Lattice	Without Lattice	With Lattice	Without Lattice	With Lattice
Inlet temperature of the cold fluid T_{in}	K	500.0	500.0	500.0	500.0	500.0	500.0
Outlet temperature of the cold fluid T_{out}	K	539.4	549.4	516.2	517.7	506.8	510.6
Inlet velocity of the cold fluid v_{in}	m/s	0.87	0.87	6.50	6.50	34.67	34.67
Outlet velocity of the cold fluid v_{out}	m/s	1.00	1.33	6.88	9.53	35.52	49.74
Total inlet pressure of the cold fluid P^*_{in}	Pa	101,326	101,331	101,346	101,484	101,856	105,006
Total outlet pressure of cold fluid P^*_{out}	Pa	101,325	101,325	101,342	101,362	101,778	102,290
Wall temperature on the cooler side $T_{wall\ cold}$	K	678.6	542.9	667.2	531.8	612.6	538.3
Wall temperature on the hotter side $T_{wall\ hot}$	K	678.7	574.5	667.2	572.7	613.0	605.2
Average cold fluid temperature T_{cf}	K	520.8	524.1	507.6	509.0	502.8	504.8
Average hot fluid temperature T_{hf}	K	913.2	893.5	961.3	958.9	978.1	980.5
Average cold fluid density ρ	kg/m^3	0.706	0.706	0.706	0.706	0.706	0.706
Characteristic size (hydraulic diameter) d_h	m	0.03	0.03	0.03	0.03	0.03	0.03
Specific heat flux q	$\frac{W}{m^2}$	1602.6	576.1	4321.3	1702.4	10,439	5378
Heat transfer coefficient h	$\frac{W}{m^2 \times K}$	10.16	30.72	27.08	74.65	95.1	160.7
Nusselt number Nu	–	4.29	12.98	31.54	11.44	40.16	67.89
Local pressure loss coefficient ζ	–	3.778	21.571	0.268	8.188	0.184	6.403
Nusselt number ratio $\frac{Nu}{Nu_{smooth}}$	–	3.03		2.76		1.69	
Ratio of local pressure loss coefficients $\frac{\zeta}{\zeta_{smooth}}$	–	5.71		30.5		34.82	
Ratio of Nusselt increment to local resistance increment $\frac{Nu/Nu_{smooth}}{\zeta/\zeta_{smooth}}$	–	0.53		0.09		0.049	

4. Discussions

Based on the obtained calculation results, a comparative analysis of the two variants of the heat exchanger section was carried out. Figure 20 shows the dependences of temperature change and the value of relative total pressure loss as the studied heat exchanger section passes, calculated at different Reynolds numbers. Dashed lines on the graphs show the boundaries of the section with a turbulent grid.

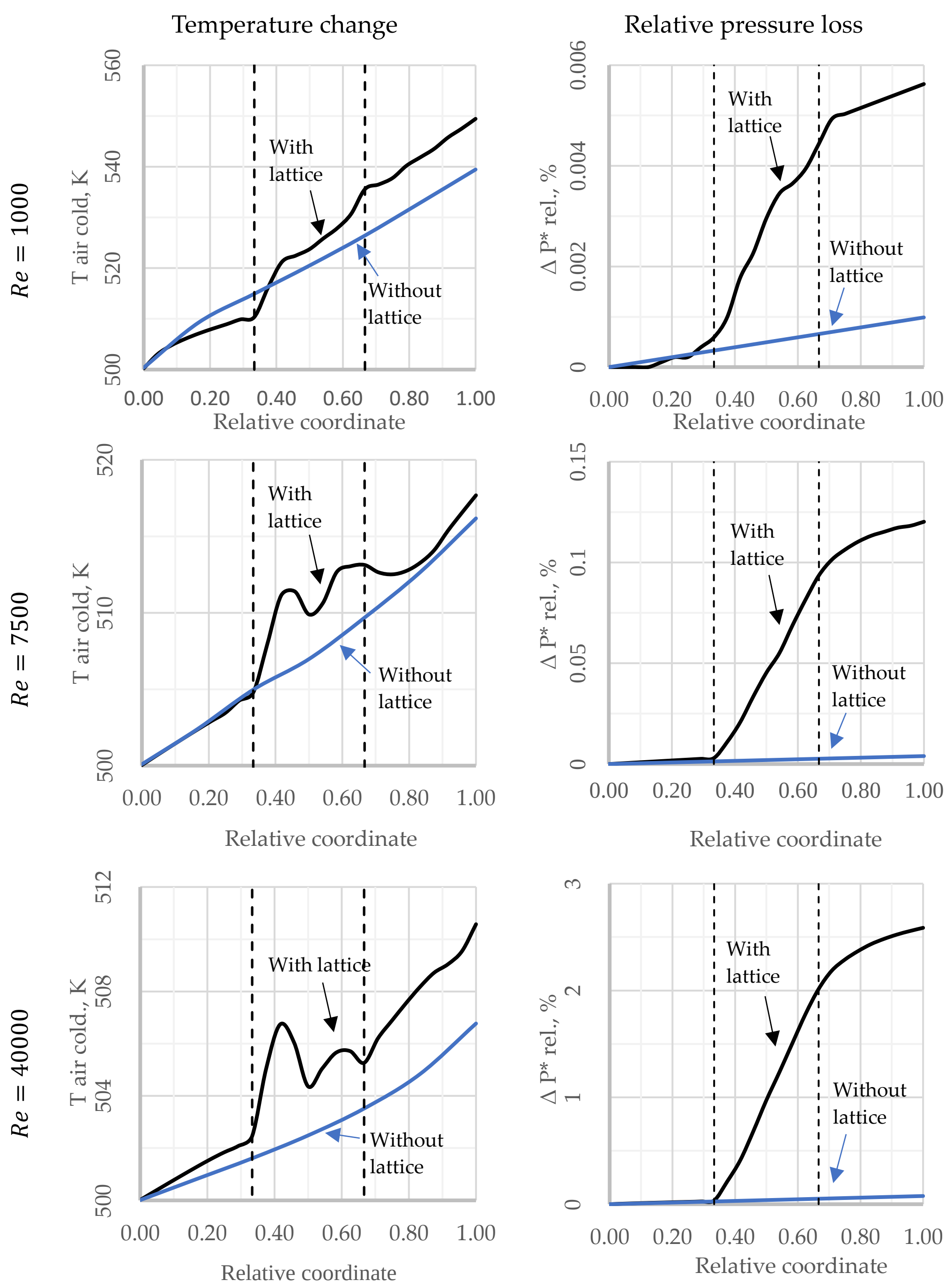

Figure 20. Plots of temperature change and value of relative losses of total pressure as the heat exchanger section passes through for heat exchanger sections with and without grids at different Reynolds numbers.

Analyzing these graphs of dependence of parameters in the coolant flow along the length of the studied section of the heat exchanger, it can be seen that the temperature growth of total pressure loss on smooth straight sections is less intensive than on sections with turbulent lattice and after it. It is possible to evaluate the influence of flow turbulization by means of a lattice structure at the middle section and after it.

Figure 21 shows the dependence of the ratio of Nusselt number, hydraulic loss coefficient, and Nusselt increment to the increment of hydraulic losses.

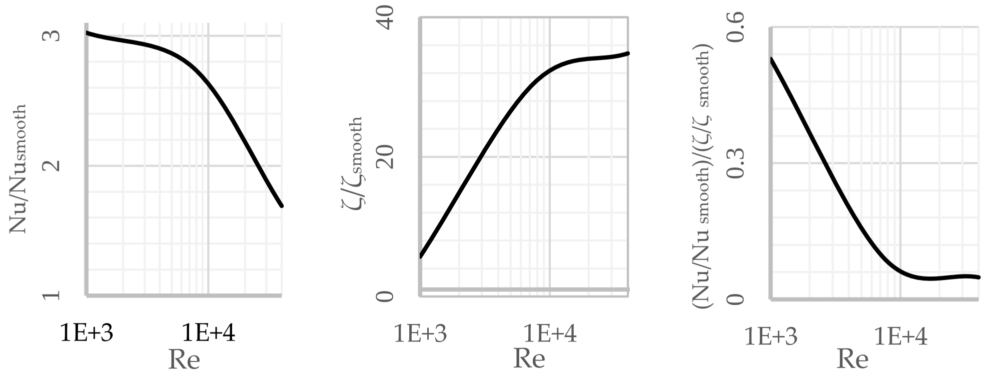

Figure 21. Plots of dependence of parameters Nu/Nu_{smooth}, ζ/ζ_{smooth} and $\frac{Nu/Nu_{smooth}}{\zeta/\zeta_{smooth}}$ of Re number for the investigated model of the heat exchanger section with gyroid lattice.

The graphs show that the highest increase in convective heat transfer due to the addition of a turbulator lattice is observed at lower Reynolds numbers. The increase in the local pressure loss coefficient increases with increasing Reynolds number.

If we turn to a parameter that evaluates not the intensity of the heat exchange process, but the efficiency of the heat exchanger, then such a parameter will be the degree of heat exchanger recovery. In theory, this number can be from 0 to 100 percent, in practice, the most efficient heat exchangers have this figure in the region of 95 percent. It is worth taking into account that in this study small sections of heat exchangers were studied, so the efficiency obviously cannot be high. However, this will give an idea of increasing the efficiency of the heat exchanger itself when introducing lattice structures into it. This coefficient shows the actual transferred amount of heat to the theoretically possible one and is calculated using the following formula:

$$R = \frac{T_{cold\ out} - T_{cold\ in}}{T_{hot\ in} - T_{cold\ in}} \times 100\% \tag{6}$$

Thus, within the studied heat exchanger sector, the relative increase in the degree of regeneration was up to 56 percent. Having proved its effectiveness in a small sector of the heat exchanger, this modification can later be implemented in bigger heat exchangers.

5. Conclusions

Thus, this study was carried out to improve the efficiency of heat exchangers manufactured using additive technologies with the use of lattice turbulators. According to the results obtained during numerical simulation, it was possible to compare a simple heat exchanger section with and without a lattice structure acting as a flow turbulator. The ratio of the Nusselt number obtained from the calculation with turbulator to the Nusselt number obtained from the calculation of smooth pipes depending on the Reynolds number varied in the range from 1.69 to 3.03 among the studied regimes. If we compare with the classical flow turbulators considered earlier, this parameter is higher in the lattice turbulator. However, the similar ratio of hydraulic resistance coefficients varies from 5.71 to 34.82 among the studied modes. This indicates that when using such turbulators, the rate of heat transfer growth is slower than the rate of hydraulic resistance growth. It is worth paying attention to the fact that the possibility of obtaining an advanced growth of heat transfer relative to the increase in hydraulic resistance compared to a similar smooth channel is of great scientific interest, but does not always lead to the most effective heat transfer intensification [14].

It is also worth noting that the implementation of such turbulators in grid form does not require additional operations during fabrication and can have a fairly high degree of automation in design. Classical flow turbulators are often a separate body from the heat exchanger itself, which means that the turbulator itself needs additional fabrication, or they are quite simple, which sometimes does not allow to achieve a balance between high efficiency of heat transfer intensification and simplicity of fabrication. Unlike classical turbulators, lattice structures have a complex shape and can be a single monolithic structure with a heat exchanger. Also, modern software allows their implementation in heat exchangers with a high degree of automation.

Another advantage of using lattice structures is the improved strength characteristics. Such lattices can act as stiffeners, thereby increasing both static strength and vibration characteristics.

Thus, we can say that despite certain disadvantages such as increased hydraulic resistance, possible difficulties in repair and operation, as well as high technological development costs, turbulators have potential. They allow for high performance in heat exchange intensification and optimize heat exchanger manufacturing because it does not require additional separate operations.

Further research development in this area could include validation of the results obtained using an experimental stand, which would confirm the effectiveness of such modifications in the design and would provide a field for further research. After that, it would be possible to investigate the influence of various parameters of different types of grating on heat transfer intensification. Further research in this direction will allow the use of these structures in heat exchangers, thereby increasing their effectiveness and reducing the mass-dimensional characteristics in theory. In the future, with cheaper metal printing technology, such heat exchangers will probably become cheaper than existing heat exchangers. This will also affect the efficiency of energy machines that use heat exchangers, such as gas turbines. This will reduce the cost of energy production, which will reduce the production price as well as reduce emissions into the atmosphere.

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