Numerical Analysis for Enhancing the Performance of a Shell and Helical Coiled Tube Heat Exchanger by Adding Continuous Fin, Discontinuous Fins and Annular Fins on the Coiled Tube

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A Thesis submitted in partial fulfillment of the requirement for the degree of Bachelor of Science in Mechanical Engineering

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Candidate's Declaration

This is to certify that the work presented in this thesis, titled, "Numerical Analysis for Enhancing the Performance of a Shell and Helical Coiled Tube Heat Exchanger by Adding Continuous Fin, Discontinuous Fins and Annular Fins on the Coiled Tube", is the outcome of the investigation and research carried out by us under the supervision of Dr. Arafat Ahmed Bhuiyan, Associate Professor, Department of Mechanical and Production Engineering, Islamic University of Technology.

It is also declared that neither this thesis nor any part of it has been submitted elsewhere for the award of any degree or diploma.

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Abstract

Heat exchangers are broadly used in various forms in domestic and industrial arena. Shell and helical tube heat exchangers have proved to be performing better than the other kinds for its improved fluid mixing and turbulence. In the present study, a baffled shell and helical coil tube heat exchanger with a total of three fin types (continuous, discontinuous and annular) are analysed keeping water at two different inlet temperatures as the working fluid. To investigate the hydrothermal impacts of the fluid at the tube side, the flow rate of the shell side fluid is changed across various examples, while tube side fluid is maintained at a constant rate. According to the findings, the average rate of heat transmission that the heat exchanger is capable of doing sees a significant boost if there is a rise in the flow rate. The heat transfer rate in a baffled shell and helical tube heat exchanger may be increased from 6% to a maximum of 14% by including fins in the tubes. In addition, based on the shapes of the tube side outlet, it was discovered that the temperature at the tube outlet falls with the rising flow rate of fluid on the shell side. This finding demonstrates that with an increase in flow rate, the total heat transfer is enhanced. In addition, the incorporation of fins into the tube side brings about a maximum temperature reduction of 4K at the exit.

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Nomenclature

Re	: Reynolds number
Pr	: Prandtl number
HTC	: Heat transfer coefficient
Nu	: Nusselt number
PEC	: Performance Evaluation Criterion
v	: Velocity (m/s)
T	: Temperature (K)
Dh	: Hydraulic diameter
q	: Heat transfer rate (W/m^2)
C_p	: Specific heat of fluid (J/kg K)
P	: pressure (N/m^2)
k	: Thermal conductivity ($W\ m^{-1}\ K^{-1}$)
μ	: Dynamic viscosity ($(Ns)/m^2$)
vol. %	: volume concentration (%)
wt%	: weight concentration (%)
ΔP	: Pressure Drop
ρ	: Density
φ	: Volume fraction (%)
A	: Area

Subscripts

c	Cold
h	Hot
i	Inner
o	Outer
m	Mean
s	Surface
eff	effective

Chapter 1: Introduction

1.1 Objectives of the Study

Shell and helical coiled tube heat exchangers have widespread use in commercial and industrial settings. Several distinct kinds of shell and tube heat exchangers are used in a variety of applications, including condensers, evaporators, energy conversion, power utility systems, and HVAC (heating, ventilation, and air conditioning) engineering [1-3]. The conventional heat exchangers, which employ straight tubes but are inconvenient and inefficient, have inspired the proposal of an alternative design using helical tubes to improve heat transfer characteristics, increase surface area to volume ratio, and eliminate dead zones.

The helical coiled tube heat exchanger is one kind of exchange that has been surface-modified to decrease in size while simultaneously increasing hydrothermal performance thanks to the existence of coils, which boost heat transmission. Cooling processes, chemical reactors, maritime cooling systems, air conditioning, and heating systems are just some of the numerous uses for vertical helical tube heat exchangers [4].

The heat exchanger with the shell and the helical coiled shell has higher heat transfer coefficients than the heat exchanger with the straight tube because of the greater fluid mixing that is achieved in the former. Coiled tubes have a higher pressure drop than straight tubes, but the working fluid's flow generates a secondary flow due to centrifugal force, which improves the heat transfer properties.

Active, passive, and compound heat transfer methods may all be used to improve heat exchanger performance. The electric field, magnetic field, surface vibration, etc., needed for the active approaches come from outside sources. In order to be effective, passive approaches involve either fluid changes (e.g. nanoparticles) or tailored surface modifications (e.g. helical coil). Internal and exterior tube fins, inserts of twisted tape and wires, helically added baffles, and fluid additives are all examples of common traditional intensification methods utilized in the process industries.

Due to its space-saving coil tubes, coiled-tube heat exchangers are widely used. The heat transfer area may be expanded without substantially growing the heat exchanger by adding fins to the helical tube. Introducing fins to the coiled tube heat exchanger makes more surface area capable of transferring heat. Because of this, the efficiency of heat transmission is enhanced, which in turn makes it possible to carry out more efficient heating or cooling operations. Fins on the coiled tube provide more turbulence and alter the flow path of the fluids going through the tube. The increased turbulence aids in mixing and thermal efficiency. Due to the increased fluid mixing, thermal boundary development is suppressed, fouling is minimized, and heat transfer coefficients are improved. Fins may be developed and modified to meet the needs of a variety of applications and heat transfer circumstances. Efficiency of the heat transferring system can be raised by adjusting the fins' shape, height, thickness, and spacing. The finned coiled tube heat exchanger's adaptability means it may be optimized for a wide range of uses and environments without compromising on efficiency or effectiveness. Including fins on the coiled tube may improve heat transmission at a lower cost than other methods. Fins have a cheaper total cost of ownership due to their simpler design and installation process compared to alternatives like extended surfaces and secondary heat transfer surfaces. Thus, heat transmission in shell and helical coiled tube heat exchangers may be improved in an efficient and cost-effective manner by installing fins. In view of the increasing emphasis that is being put on energy systems that are more friendly to the environment, it is of the greatest importance to enhance the efficiency of heat transferring devices. Improving the heat transfer performance of heat exchangers like SHCTHE is a step in the right direction toward reaching the objective of energy efficiency and lowering the negative effect on the environment.

1.2 Structure of the Thesis

The ultimate goal of this study is to investigate the effect of multiple types of fins on the hydrothermal performance of a shell and helical tube HX coupled with baffles. The literature review section provides a summary of the works done by researchers on this relevant topic. For carrying out the simulations, the geometry of the HX was done and all the geometric parameters are added in the model description section. The case was validated against experimental outputs in the validation section. The computation methodology section contains all the CFD settings that was applied in the simulation process including the turbulent kinetic equations followed by the theoretical equations that are relevant to the present work. Lastly, the result section illustrates the findings of the study that contains effect of the fins on various parameters such as HTC, pressure loss and surface temperature.

Chapter 2: Literature Review

There have been a lot of studies published on heat exchangers with helical tubes, and there is currently ongoing research on this specific type of heat transfer device. Studies that are described here are among the most recent and relevant ones available.

Datta et al. [5] conducted a numerical investigation of the thermal performance (PEC; – i.e., Performance Evaluation Criteria) of laminar flow in a helical pipe with a circular cross-section at three different Prandtl numbers. The results shown that there is a little drop in the Nu value if there is a reduction in the low torsion Prandtl number. Over the 7191-64723 Re ranges, Niu et al. [6] employed a computational model that included a number of heating scenarios for helically coiled tubes to examine thermal hydraulic properties. The authors argue that, unlike in the case of homogenous heating, the outer and inner side temperature changes become even more pronounced. Li and Bai [7] found that the heat transfer coefficient of a vertical helically coiled tube is less when heated from half of its circumference than when heated uniformly.

Alimoradi and Veysi [8] proposed a correlation to forecast heat transfer rate numerically via a shell and helical tube heat exchanger, taking into account the effects of fluid characteristics, operating conditions, and geometrical parameters. Fouda et al. [9] performed a numerical analysis of the flow and PEC in a multi-tube-in-tube helically coiled configuration subject to turbulent flow conditions. Results revealed that a diameter increase from 100 to 250 mm resulted in a 60% improvement in coil performance. Alimoradi et al. [10] looked at how the addition of annular fins to the outside of a helical tube improves heat transfer in a turbulent system. Gholamalizadeh et al. [11] found numerically and experimentally that using coiled wire inserts with a circular cross section increased the heat transfer rate by 340.9% in a helical tube with coiled wires. Cancan et al. [12] used a helical coiled tube with spherical corrugation to perform a numerical analysis of heat transfer versus a smooth helical tube. Bozzoli et al. [13] conducted an experiment to see how a corrugated wall within a helically coiled tube affected the rate of heat transmission. The outcomes presented that the local heat flow rate was enhanced for the corrugated wall compared to the curved wall. Kannadasan et al. [14] conducted an experimental study to evaluate the differences between the vertical and horizontal impacts of helically coiled in terms of heat transfer characteristics. The outcomes revealed that heat transport and friction are unaffected by the helically coiled configuration. The thermal performance of a vertical shell and helical coiled tube heat exchanger was investigated numerically by Mirgolbabaei [15], who took into account the actual condition of the conjugate thermal boundary of the coil wall. This was done while varying shell-side's flow rate, diameters' ratio of coil and tube, and the coil pitch. According to the findings, raising the coil pitch had a negative impact on the efficiency of coiled tube heat exchanger. On the other hand, modifying the tube diameter only had a marginal bearing on the efficiency of the system. In addition, increment in coil-pitch resulted in a reduction in the shell side's heat transmission rate; nevertheless, coil-pitch's increment beyond that value enhanced the shell side's heat transmission rate. Ferng et al. [16] used a computer program to research the influence that the size of the pitch has on the fluid flow characteristics and the thermal performance of helical coiled tube heat exchanger. The findings provided a reasonable description of the complex fluid flow phenomena that took place in a coiled tube heat exchanger. These phenomena included the separation and acceleration of the shell side's flow, coiled tube's significant secondary flow, and two wake vortices near the coil's rear end. The effectiveness of a coiled tube heat exchanger equipped with a trilobal tube was investigated both experimentally and statistically by Wang et al. [17]. According to the findings, having a powerful capability to disrupt the boundary layer led to an increase in the radial velocity near to the tube's wall. In addition, the trilobal helical tube's hydro-thermal performance increased around 1.16–1.36 times higher than coiled plain tube's performance. Yet, the increment in friction factor was 0.96 to 1.10 times.

Omidi et al. [18] used numerical analysis of four cross-sections of helically coiled tubes filled with an Al2O3-water nanofluid to determine the PEC for a Reynolds number range of 1300-2500. According to their findings, the coil diameter is more important than any of the other geometrical characteristics considered. Three distinct helically coiled tubes and a straight tube were experimentally investigated by Ardekan et al. [19] to determine their heat transfer capabilities and the influence of geometric factors such pitch circle diameter and helical pitch. The experiments were conducted using Ag-water and SiO2-water across the Re in between 8900 to 11,970. The coil diameter made a bigger difference in the heat transmission rate than pitch coil's diameter did. Srinivas and Venu Vinod [20] conducted an experiment to study the heat transfer of a shell and coil heat exchanger while employing various weight concentrations of CuO/water nanofluid. Weight congregation of 0.3 wt%, 0.6 wt%, 1 wt%, 1.5 wt%, and 2 wt% were utilized respectively. The outcomes showed the significant increment of heat transmission rate for using nanofluids in comparison to water. This augmentation of heat transmission became better as the weight concentration and Dean number of the nanofluid got higher. Bhanvase et al. [21] conducted an experiment to test the thermal performance of a vertically coiled PANI (polyaniline) nanofluid heat exchanger. PANI nanofluids' volume congregation used in this experiment ranged from 0.1 vol% to 0.5 vol%. According to the findings, the heat transfer rate was enhanced by 10.52% and 69.62% with PANI nanofibers of 0.1 vol% and 0.5 vol% concentrations, respectively. These values were found to be significantly higher than those for distilled water. A numerical evaluation of the heat transfer behavior and pressure drop of a shell and helical coil heat exchanger using Al_2O_3/water nanofluid was carried out by Bahrehmand and Abbassi [22]. Nanoparticles ranging from 0.1 vol% to 0.3 vol% concentration were used, the Re were changed from 9000 to 36,000 in the coil, and shell's Reynolds numbers were adjusted from 600 to 2600. The use of nanofluid led to the increment of the heat transfer rate as well as the pressure drop, as shown by the findings. Additionally, raising tube and coil diameters as well as nanofluids' volume congregation, all contributed to the enhancement of the heat exchanger's efficiency; despite this, the efficiency saw a reduction when there was an increment in the flow rate. Rakhsha et al. [23] conducted a computational and experimental investigation regarding the thermal performance and pressure drop of a CuO/water nanofluid with a 0.1 vol% concentration while it was flowing through a horizontal helically coiled tube. According to the findings, the boosting of the Reynolds number as well as the curvature ratio of the coiled tube resulted in the increment of heat flow rate and pressure drop. Comparing to based fluid, the drop in pressure caused by utilizing nanofluid was found to be raised by 16–17%, and the coefficient of heat transfer was attained to be boosted by 14–16%. Both of these findings were based on the findings of experimental testing. Additionally, the pressure-drop and the coefficient of heat transfer increased for the rise in curvature ratio and Reynolds Number. Mahmoudi et al. [24] conducted an experimental and numerical investigation utilizing TiO_2/water nanofluid to investigate the heat transfer behavior and the pressure drop of a horizontally coiled tube. The effects of changing Reynolds number from 3,000 to 18,000 on the nanofluid curvature ratio and volume concentration were investigated. The reports showed that replacing regular water with nanofluid increased both the pressure drop and the rate at which heat was transferred.

Temperature distribution across a horizontal helical coil heat exchanger was studied experimentally and numerically by Hameed et al. [25]. A Perspex shell of 150 mm inside diameter and 1 m long houses the helically constructed copper tube. Both tube and shell sides utilized water heated reaching 65°C Celsius. The four operational rates at which mass flow for both tube and shell sides were determined by the author. Each of the flow rates ranged from 6 to 12 l/min. Excellent agreement was found in both experimental and theoretical findings. A heat exchanger of helical coiled tube wrapped with a wire buckle was used in experimental and computational examination, and the consequences were held against that of a plain heat exchanger. Experimental and computational research by Mhaske et al. [26] compared the performance of a helical tube heat exchanger that had a wire warp within the tube to that of a standard heat exchanger. The counter-flowing one-phase system was used. The tube at inner side carries the warm water of 60°C, while outside tube receives the cold water of 30°C. From 480 to 1200 liters per hour, water flowed through the pipes. The findings show that, compared to a standard heat exchanger of helical coiled tube, the tube wire wind significantly improves heat transfer efficiency. Nada et al. [27] analyzed the characteristics and efficacy of helical coil tubes with and without fins. Four distinct shell sizes ranging from 10 to 25 centimeters were used for completing the experiment. The refrigerant run through the coiled tube while the shell contained water. The tube's wall thickness was 0.5 mm, as well as it had an interior diameter of 8 mm. When compared to the case of an unfinned tube, the findings indicated the increment of Nusselt number, Reynolds number as well as Grashof number. Hameed et al. [28] tested and modeled a triangular finned tube heat exchanger. Copper tubing with inner diameter of 20 mm was utilized for two meters. The tube's outside included copper triangular fins. The fin's length, height, and thickness were 1 cm, 1 cm, and 1 mm. A 54 mm internal diameter Perspex shell was employed. Air and water were the system's operating fluids. Cold air flowed through the shell at 0.001875 to 0.003133 kg/s when hot water started to flow in tube. Comparisons were made between finned and unfinned heat exchangers. The fin on the tube increased heat transmission of the heat exchanger by 3.252 to 4.502 times compared to the unfinned tube. Experimental findings matched numerical findings. Amori et al. [29] tested a cold-water-immersed helical coil heat exchanger. Coiled heat exchangers were single and triple-helical. The flow rate of hot water in the tube ranged from 2.67 to 7.08 l/min. Temperatures of inlet were kept at 50°C, 60°C, 70°C, and 80°C, respectively. Heat transport and efficiency improved using a heat exchanger with triple helical coil. Second-case pressure declines were lower.

Chapter 3: Description of the model/System

In the present study, a baffled shell and helical tube heat exchanger was chosen on the basis of experimental work conducted by Güngör [30]. The geometry was generated using SolidWorks software and it has a cylindrical shell, a helical tube, 3 discs and 4 rings. The cold fluid enters into the shell of the heat exchanger and the hot fluid enters into the tube. Overall geometry of the heat exchanger consisted of shell and helical coiled tube is demonstrated by figure.

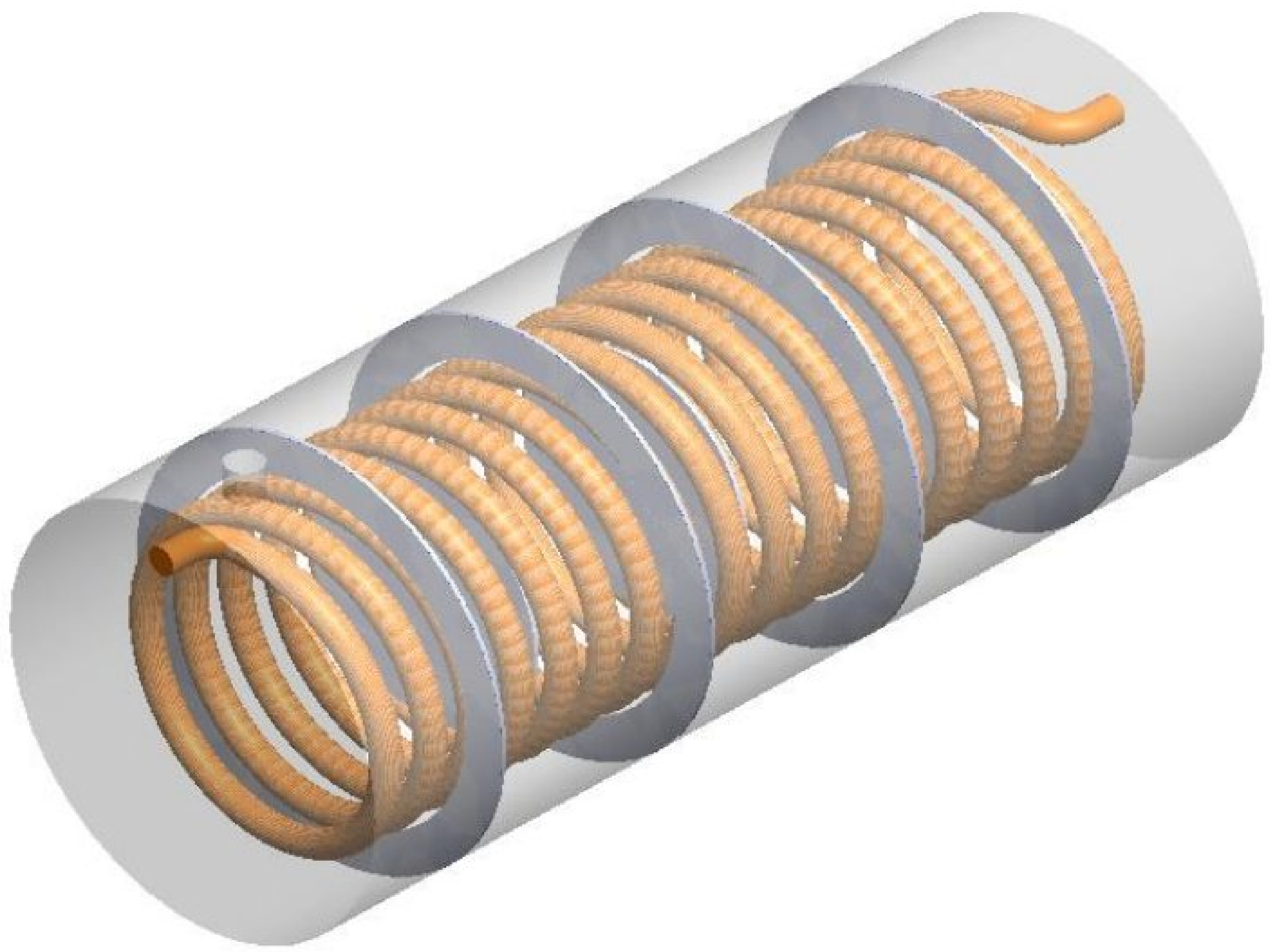

Fig 1 - Baffled shell and helical tube heat exchanger

The geometrical parameters and the dimensions were taken from the model used by Güngör [30]. The parameters are depicted in figure. For the baffles, 4 O-rings and 3 discs are used and their arrangements are shown in figure.

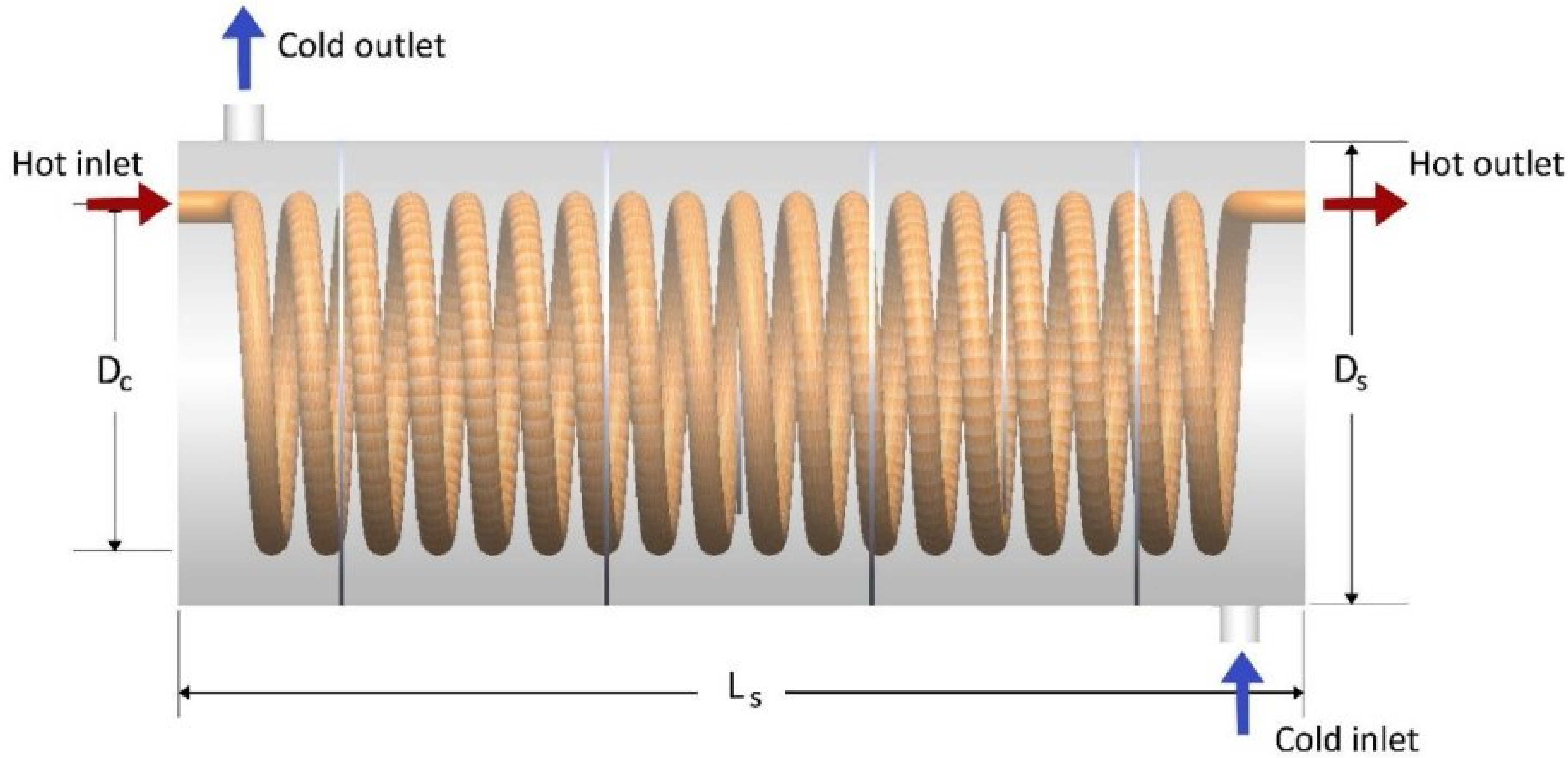

Fig 2 - Geometrical parameters of the SHCTHX

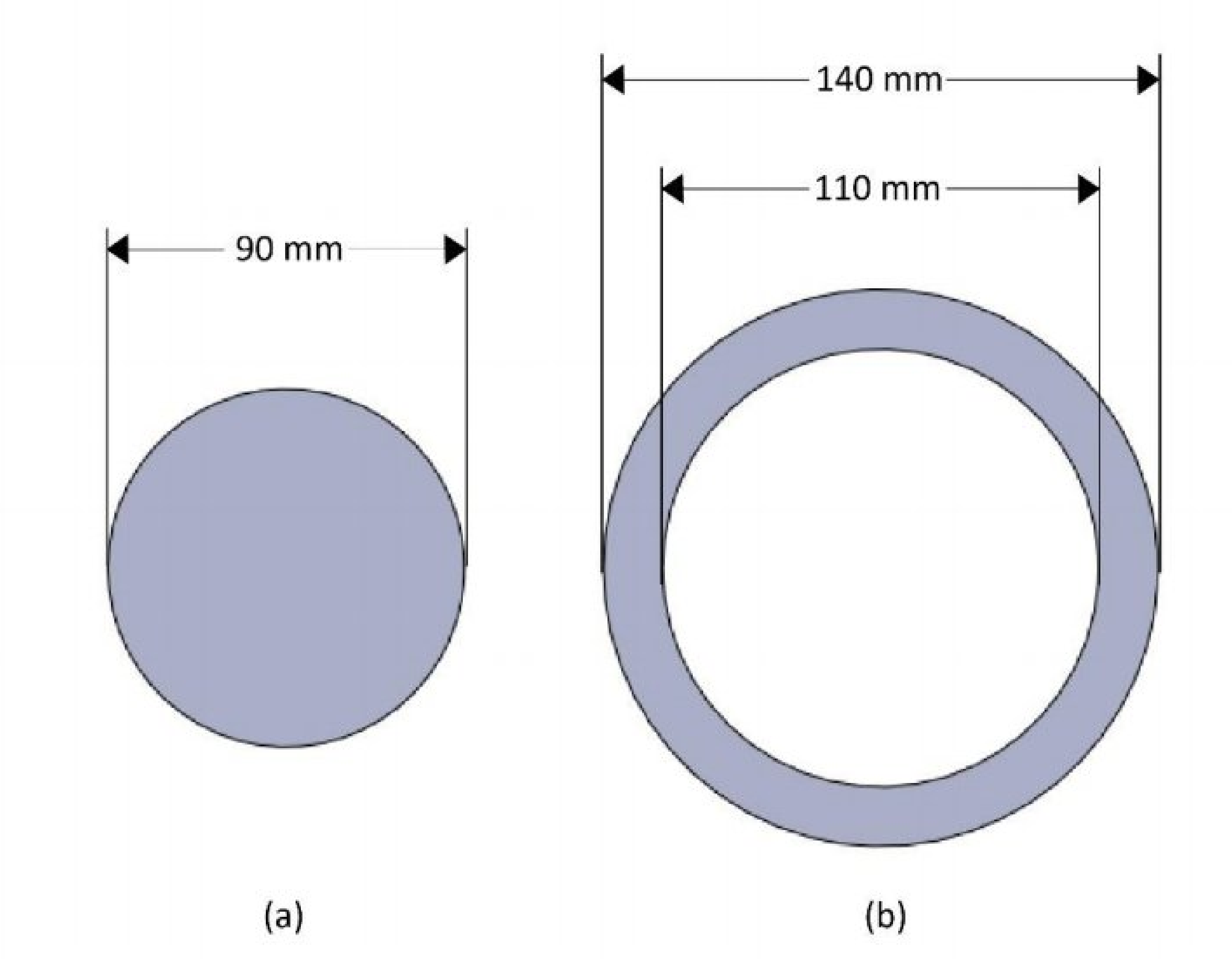

Fig 3 - Dimensions of (a) Discs (b) O-rings used for baffles

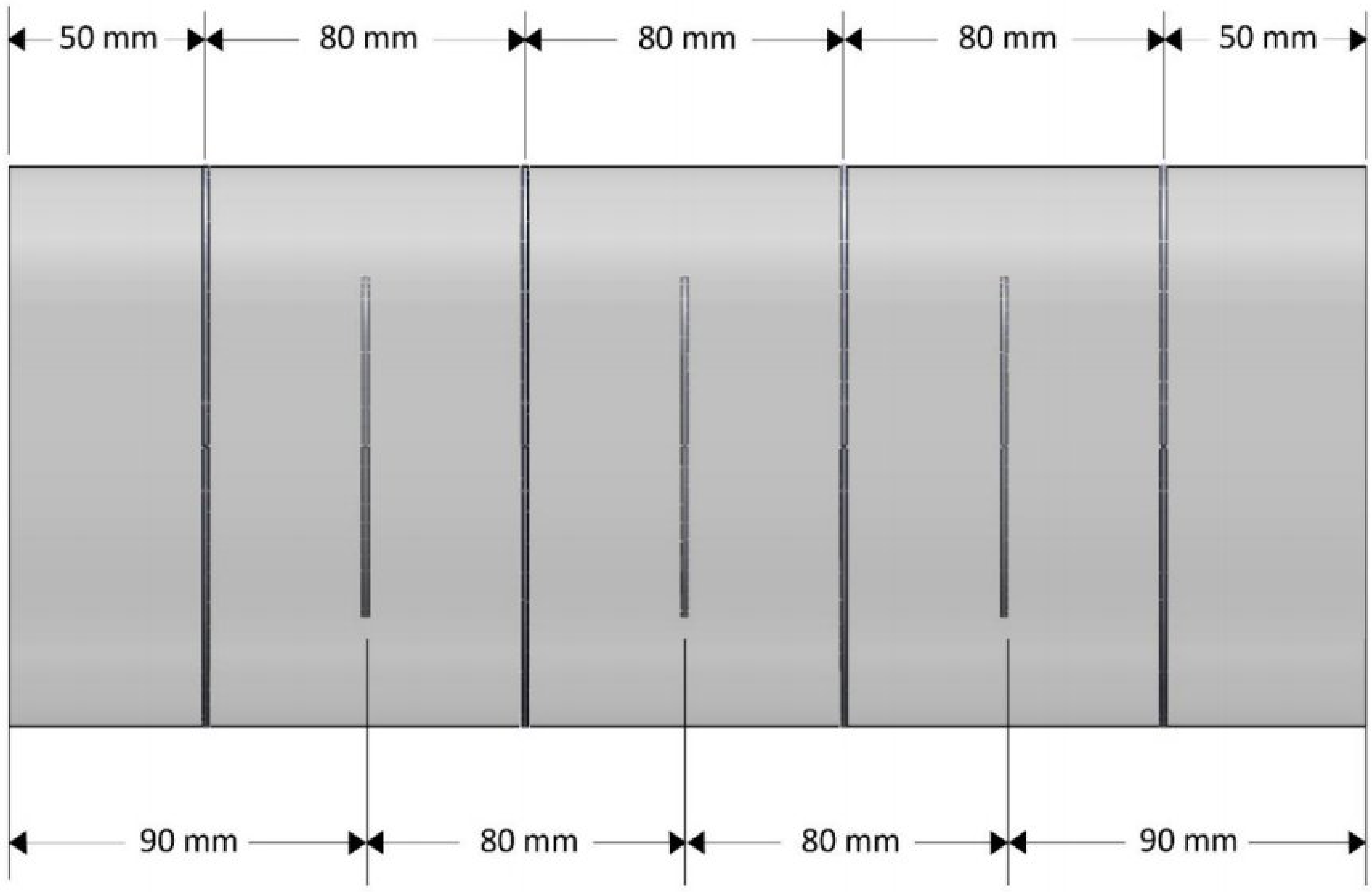

Fig 4 - Rings and discs arrangements

Shell's overall length (L_s) and diameter are 340 mm and 140 mm, respectively. The coil is having the diameter of 100 mm. The dimensional values of the helical tube are presented via table

Specifications	Dimensions
Pitch of helical coil	16.67 mm
No. of turns	18
Tube outer diameter	9.52 mm
Tube inner diameter	8.52 mm

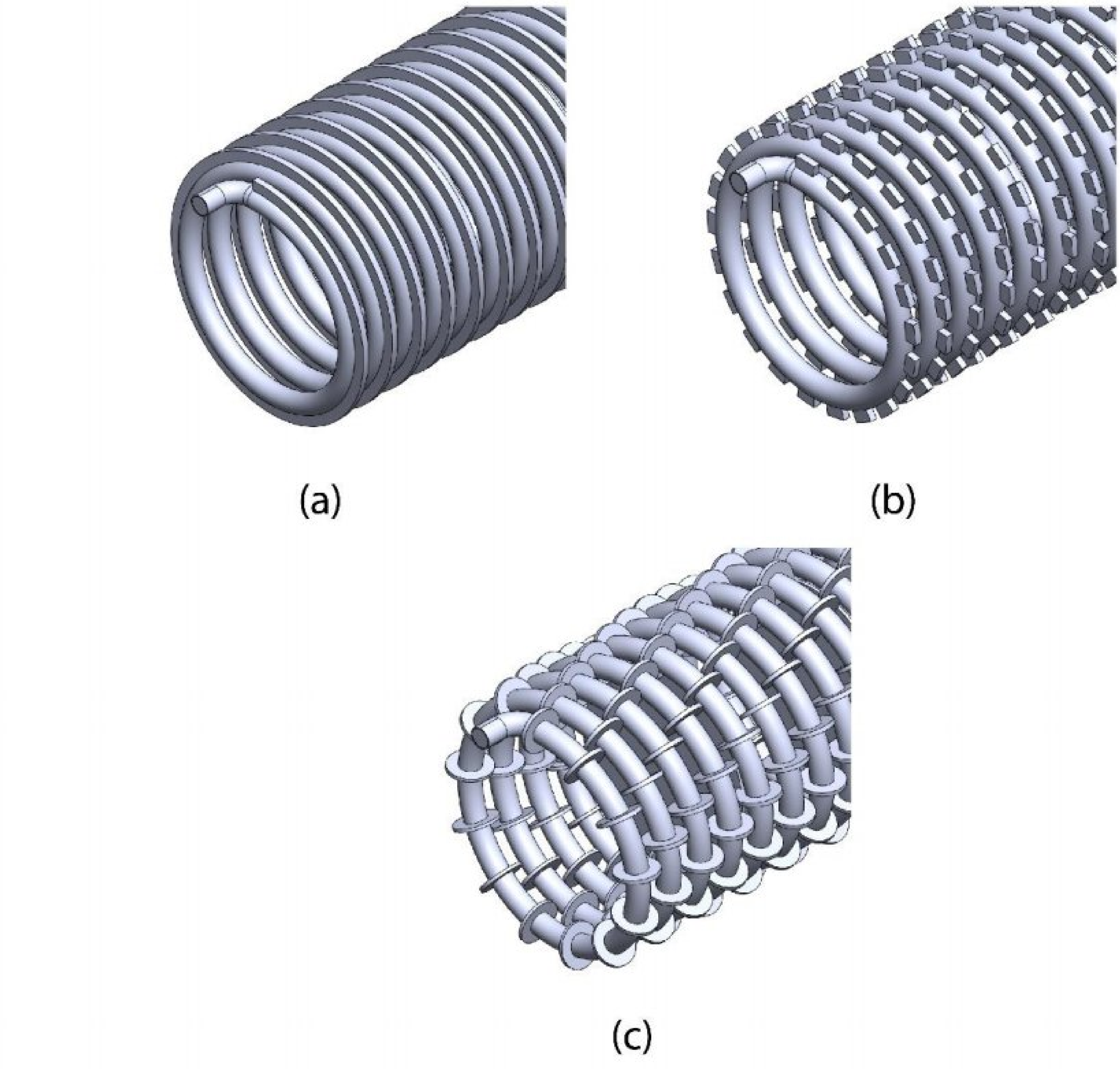

Fig 5 – Modified SHCTHXs with (a) Continuous Fin (b) Discontinuous Fin (c) Annular Fin

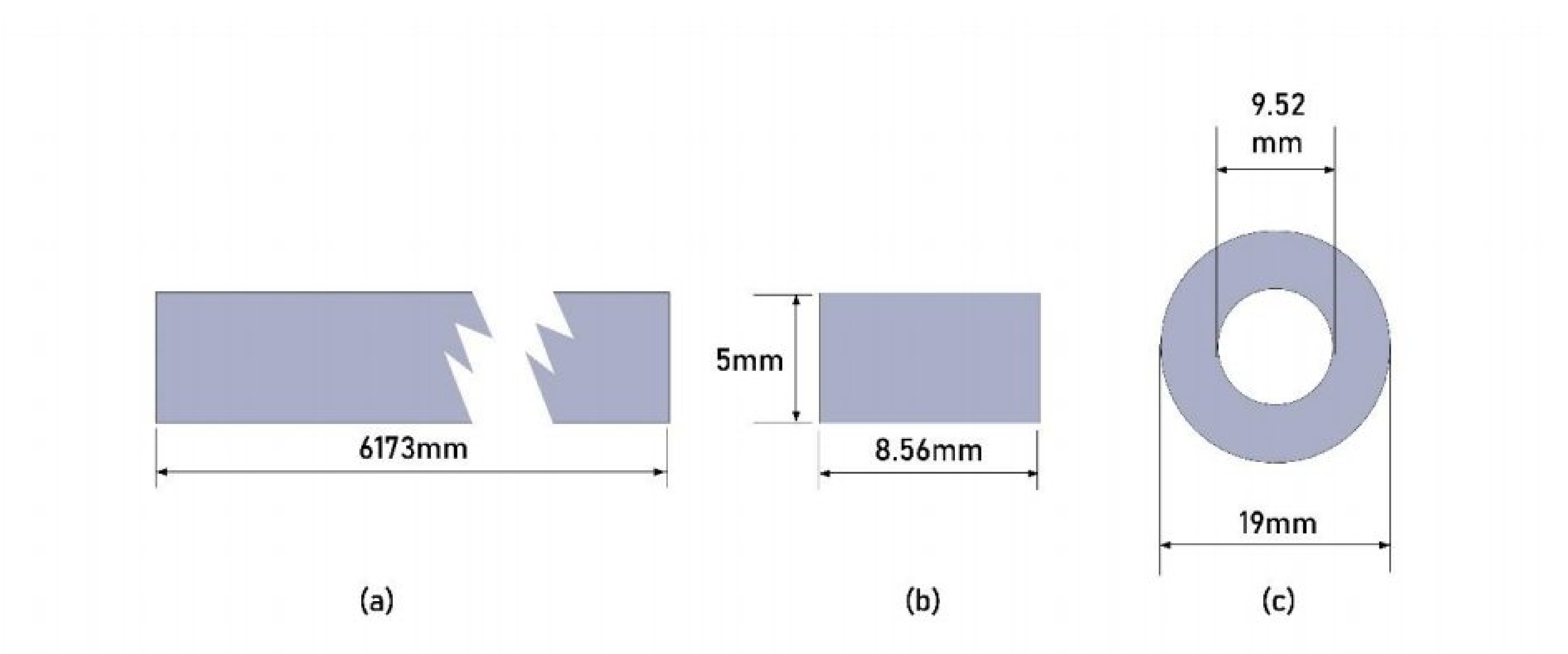

Fig 6 – Dimensions of (a) Continuous Fin (b) Discontinuous Fin (c) Annular Fin

The geometrical parameters of the modified shell and helical tube heat exchanger are shown in Figure 5 and Figure 6 where the thicknesses of continuous and discontinuous fins are kept at 3mm and the thickness of annular fins are 1.57mm.

There are a total of 40 discontinuous fins added in the spiral section while 26 annular fins were constructed on the tube

Chapter 4: Computational Methodology

As mentioned earlier, the model of SHTHX used in this study consists of fluids of two different inlet temperatures and baffles are used in the form of rings and discs. The numerical simulation was carried out using ANSYS 2020R1 which is a robust tool for CFD study. ANSYS Mesh module was employed for meshing the geometry and to ensure the integrity of the study and for accuracy verification of the numerical results, fine meshing was used in the geometry based on different body sizing. The mesh generated on the SHTHX is shown in figure.

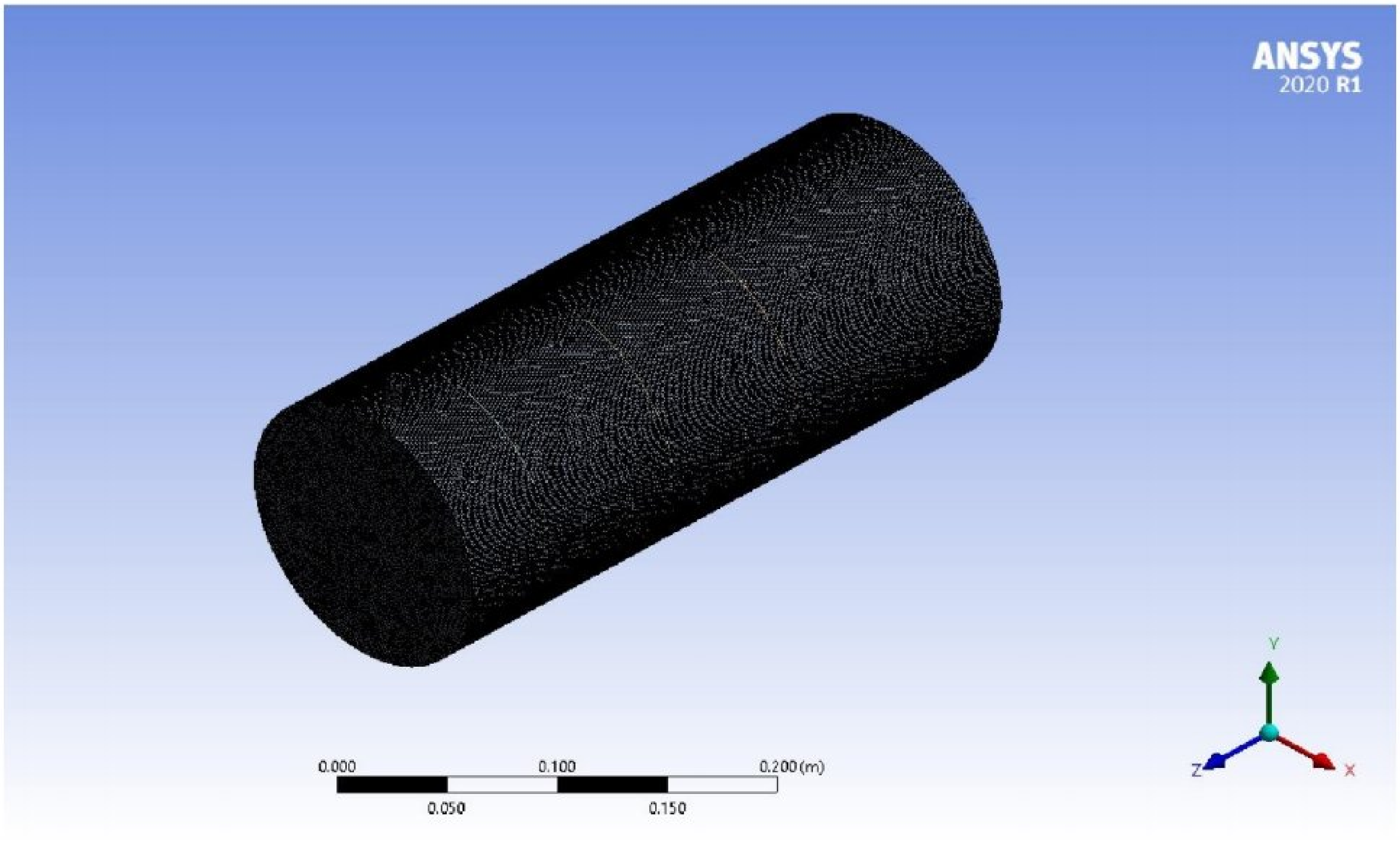

Fig 7 - Mesh structure of the SHTHX

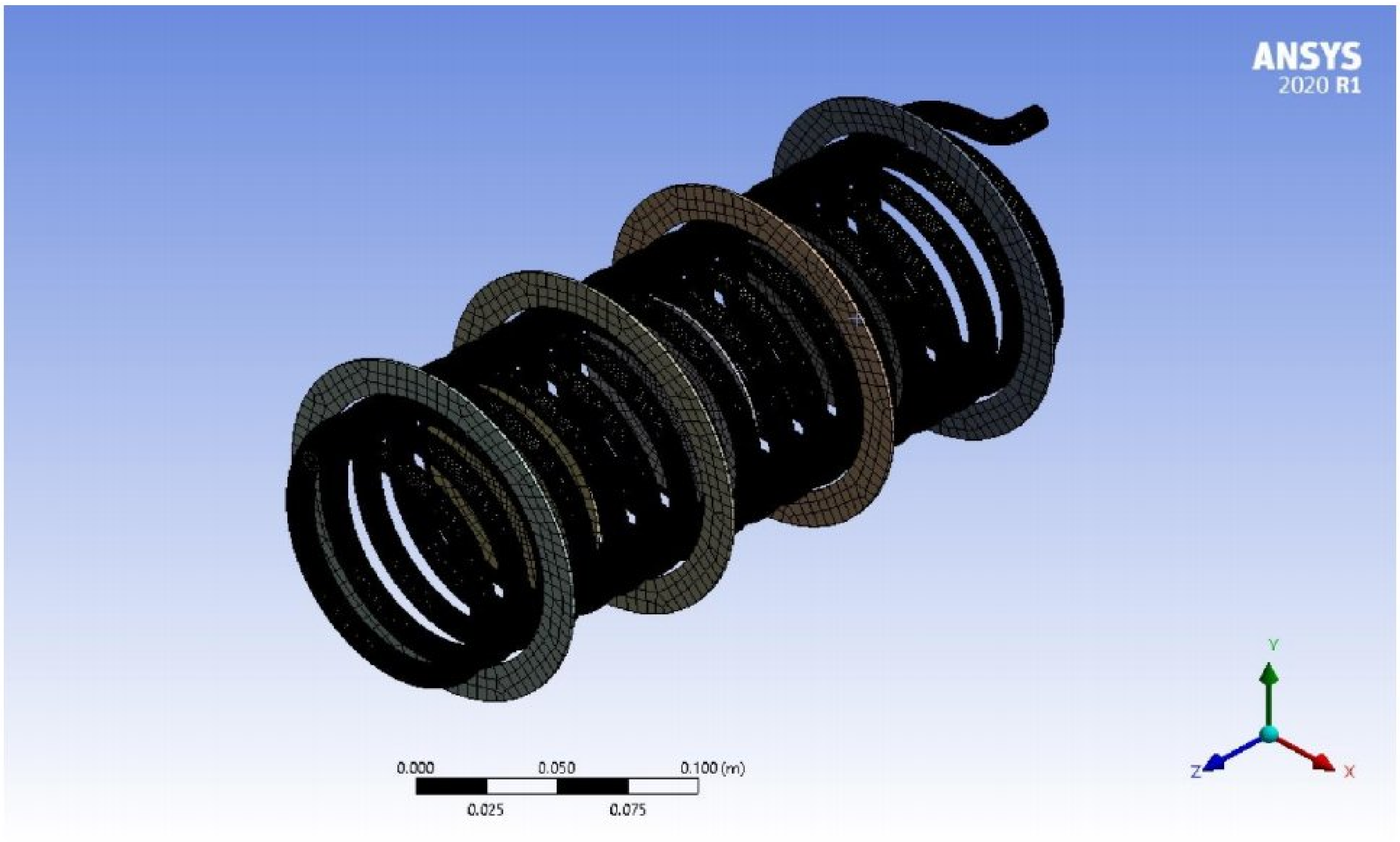

Fig 8 - Mesh Structure of the helical tube and baffles

The mesh skewness was kept at 0.85 which is below the recommended value of the skewness provided by the ANSYS Fluent guide. And for finding out the appropriate mesh structure, a mesh grid independency test was done which is shown in figure. The mesh element number was varied and the corresponding temperature of the hot fluid flow was measured. It is seen from the study, that when the element number of the mesh crosses 7.5 million, the varying of the outlet fluid temperature is very much negligible. As a result, the mesh structure with 7.5 million element number was selected for further simulations in this work.

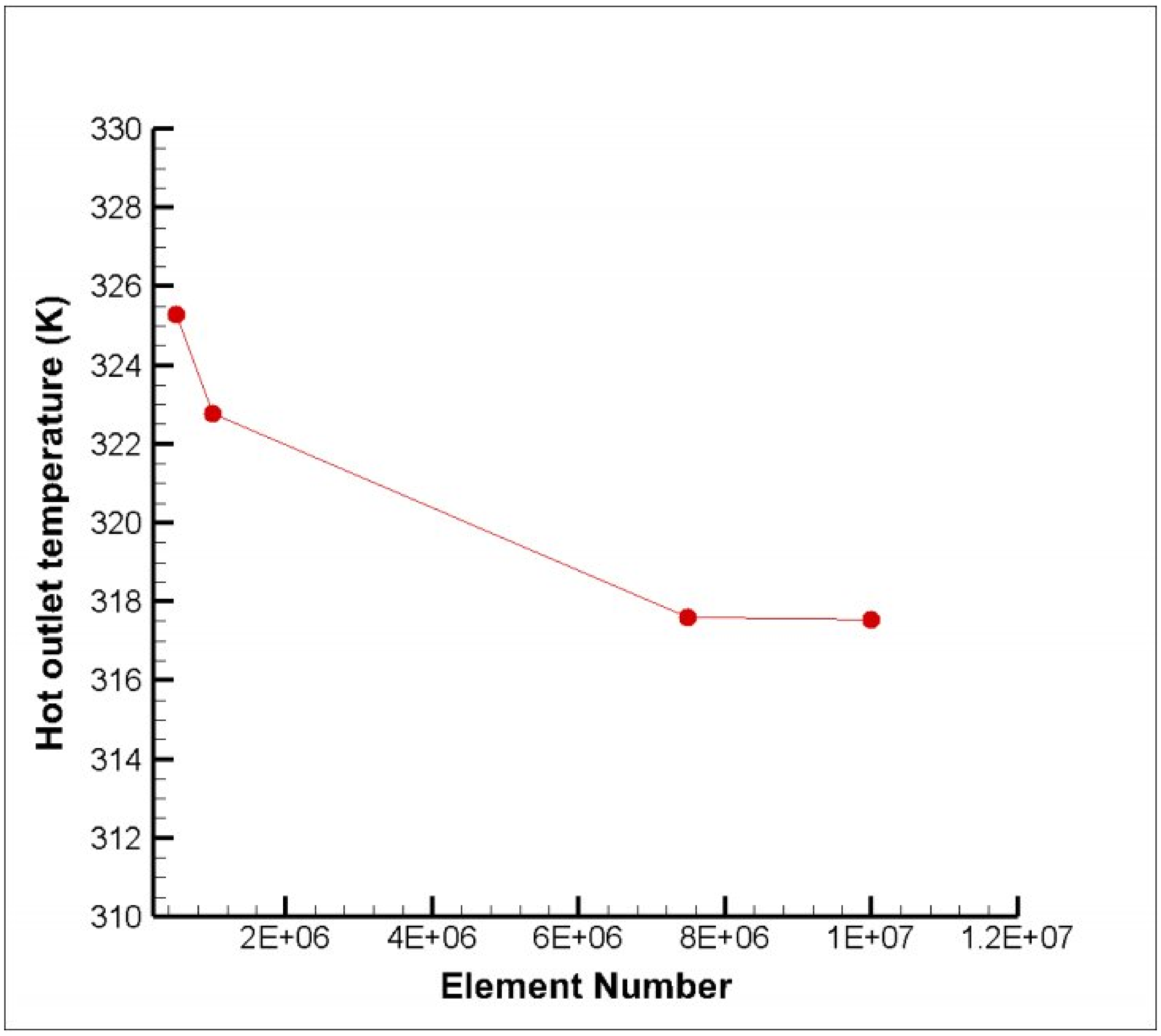

Fig 9 - Grid Independence study

4.1 CFD Study

CFD or computational fluid dynamics involves mathematical models to depict fluid flows. It is a numerical analysis to reveal fluid flow structures incorporating thermal behaviors in the flow. ANSYS Fluent is a widely used software for performing CFD analysis of different cases and due to its high accuracy, the numerical simulations in this study were conducted using this software. The governing equations that are related to this study are –

Continuity -

$$\nabla.(\rho.v) = 0 \tag{1}$$

Momentum –

$$\nabla.(\rho.\vec{v}.\vec{v}) = -\nabla p + \nabla.\left(\mu\left[(\nabla\vec{v} + \nabla\vec{v}^{T}) - \frac{2}{3}\nabla.\nabla\vec{v}I\right]\right) \tag{2}$$

Energy –

$$\nabla.\left(\vec{V}(\rho E + p)\right) = \nabla.k_{eff}\nabla T - h\vec{J} + \left(\mu\left[(\nabla\vec{v} + \nabla\vec{v}^{T}) - \frac{2}{3}\nabla.\nabla\vec{v}I\right]\right) \tag{3}$$

For the helical coil of the SHTHX, it is expected that turbulent fluid mixing will occur inside the heat exchanger and for this reason $k-\in$ turbulence model was employed in this study which ensures proper capturing of the thermal boundary layers in turbulent flows. The equations used in this turbulent model can be expressed as [32-34] –

$$\rho\frac{\partial}{\partial x_i}(kv_i) = \frac{\partial}{\partial x_j}\left[\left(\frac{\mu + \mu_t}{\sigma_k}\right)\frac{\partial k}{\partial x_j}\right] + G_k + G_h - \rho \in -Y_M + S_k \tag{4}$$

$$\rho\frac{\partial}{\partial x_i}(\in v_i) = \frac{\partial}{\partial x_j}\left[\left(\frac{\mu + \mu_t}{\sigma_\in}\right)\frac{\partial \in}{\partial x_j}\right] + C_{1\in}\frac{\in}{k}(G_k + C_{3\in}G_b) - C_{2\in}\rho\frac{\epsilon^2}{k} + S_\varepsilon \tag{5}$$

Where k stands for turbulent kinetic energy and $\in$ denotes the dissipation rate of the kinetic energy. The values of $C_{1\varepsilon}$, $C_{2\varepsilon}$, $C_{3\varepsilon}$, C_μ, σ_k and σ_ε are obtained from [27].

$C_{1\varepsilon} = 1.44$, $C_{2\varepsilon} = 1.92$, $C_{3\varepsilon} = 1.0$, $C_\mu = 0.09$, $\sigma_k = 1.0$ and $\sigma_\varepsilon = 1.3$

For the simulations carried out in this work, steady state approach was taken into consideration and for the pressure velocity coupling, SIMPLEC scheme was selected. Second order equations were used to solve the momentum and energy equations in the analysis and the residual criteria for the energy was kept at 10^{-7} while both the velocity and continuity were selected to be 10^{-5}.

Stainless steel was applied as the shell material of the heat exchanger and copper was used in the tube and the baffles i.e., discs and rings. As for the working fluid of the heat exchanger, water at two different temperatures were used in shell side and tube side. The boundary conditions applied in this study are presented in table

Description		Values
Hot fluid	Inlet temperature	333.15 K
	Flow rate	3 l/min
Cold fluid	Inlet temperature	291.65 K
	Flow rate	2,3,4,5,6 l/min

Theoretical Calculations

The general equation involving heat transfer employed in the present calculation is:

$$q = \dot{m}Cp(T_i - T_o) \tag{1}$$

Shell side heat transfer:

$$q_c = \dot{m}_c Cp_c\left(T_{c,o} - T_{c,i}\right) \tag{2}$$

Tube side heat transfer:

$$q_h = \dot{m}_h Cp_h\left(T_{h,i} - T_{h,o}\right) \tag{3}$$

Nusselt number calculations for the helical tube side:

$$Nu_i = \frac{d_i}{k_h} \tag{4}$$

Where (h_i) is obtained from newton's law of cooling:

$$h_i = \frac{q_h}{A_{si}(T_m - T_s)} \tag{5}$$

The surface area of the helical tube can be found from:

$$A_s = \pi d_i L_{coil} \tag{6}$$

For helical tube with added fins, (h_o) is calculated from:

$$h_o = \frac{q_{avg}}{A_{eff} \Delta T_{\log}} \tag{7}$$

$$A_{eff} = A_s + A_f \tag{8}$$

And the logarithmic mean temperature difference can be found from:

$$\Delta T_{\log} = \frac{\Delta T_1 - \Delta T_2}{\ln\left(\frac{\Delta T_1}{\Delta T_2}\right)} \tag{9}$$

where $\Delta T_1 = T_{h,i} - T_{c,o}$ and $\Delta T_2 = T_{h,o} - T_{c,i}$

4.2 Case Validation

To verify the accuracy of further simulations, the numerical simulations carried out on the baffled SHTHE was validated with the experimental works of Güngör [30]. Parameters such as the outlet side temperature of both cold and hot fluid and the average heat transfer rate of the SHTHE was calculated and compared with the experimental study.

Figure shows that the simulation output of the present work matches with the results obtained from experimental study with high precision and with a very low percentage of error.

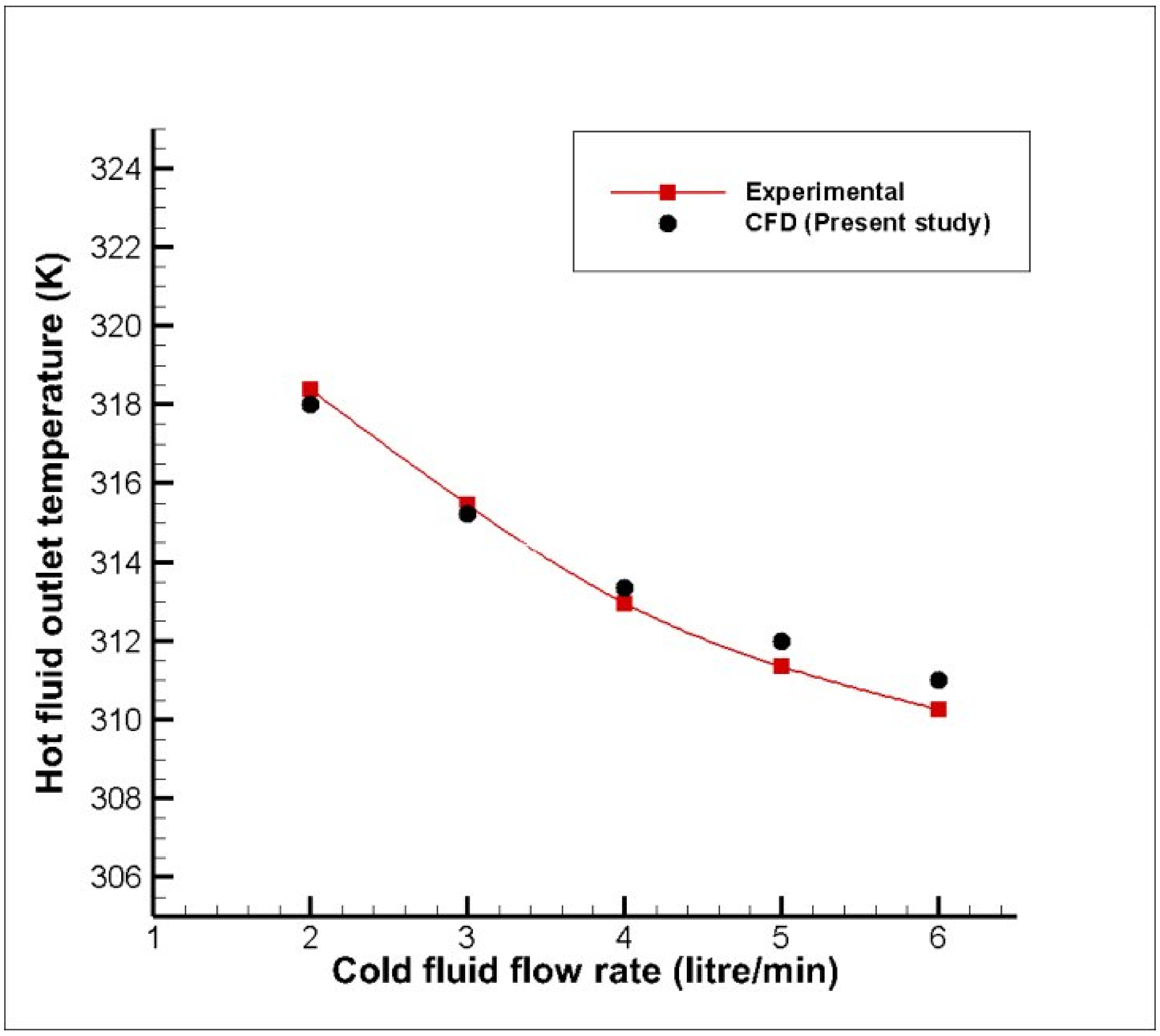

Fig 10 – Validation of outlet temperature of hot fluid at different cold fluid flow rates with experimental study

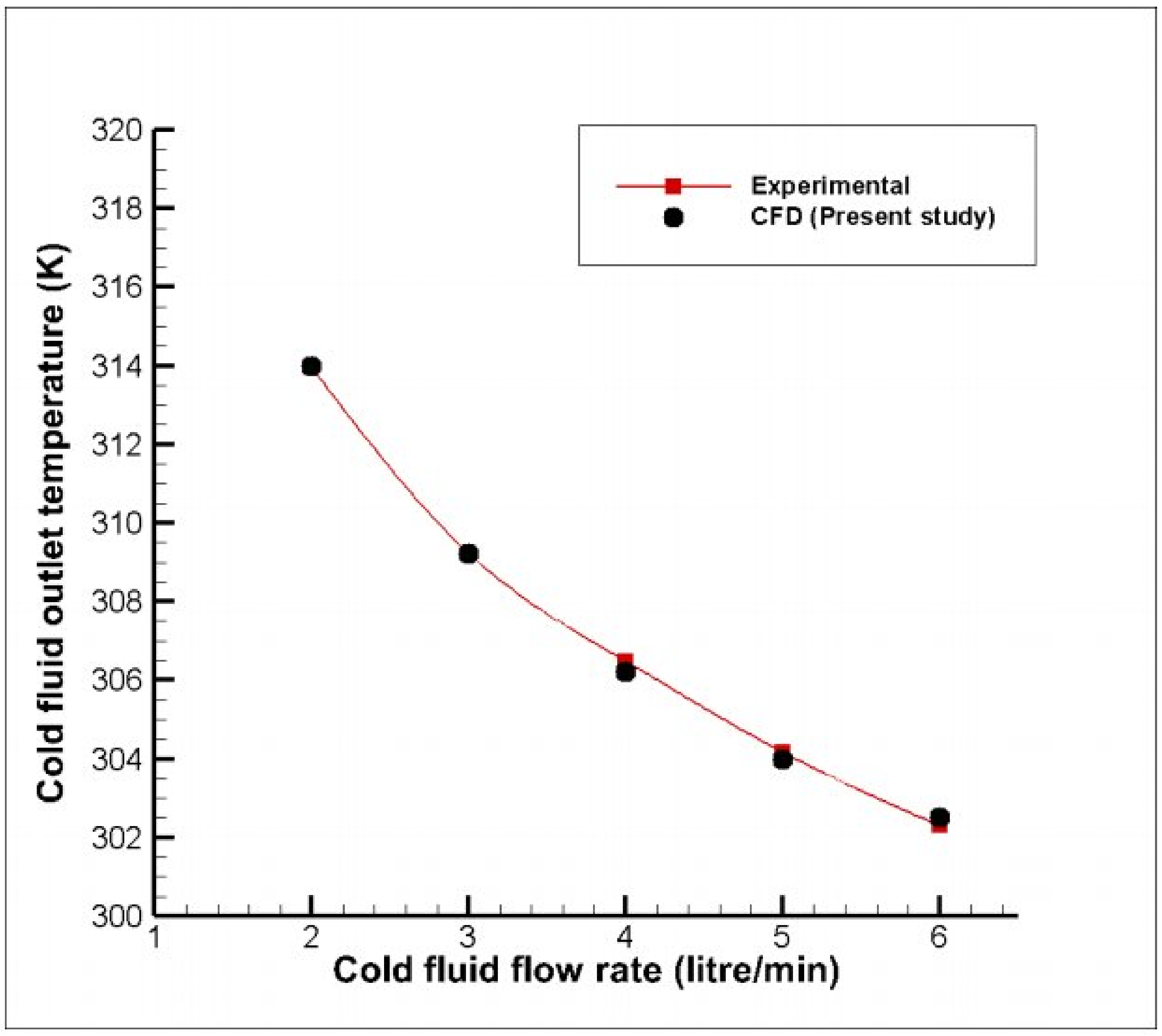

Fig 11 – Validation of outlet temperature of cold fluid at different cold fluid flow rates with experimental work

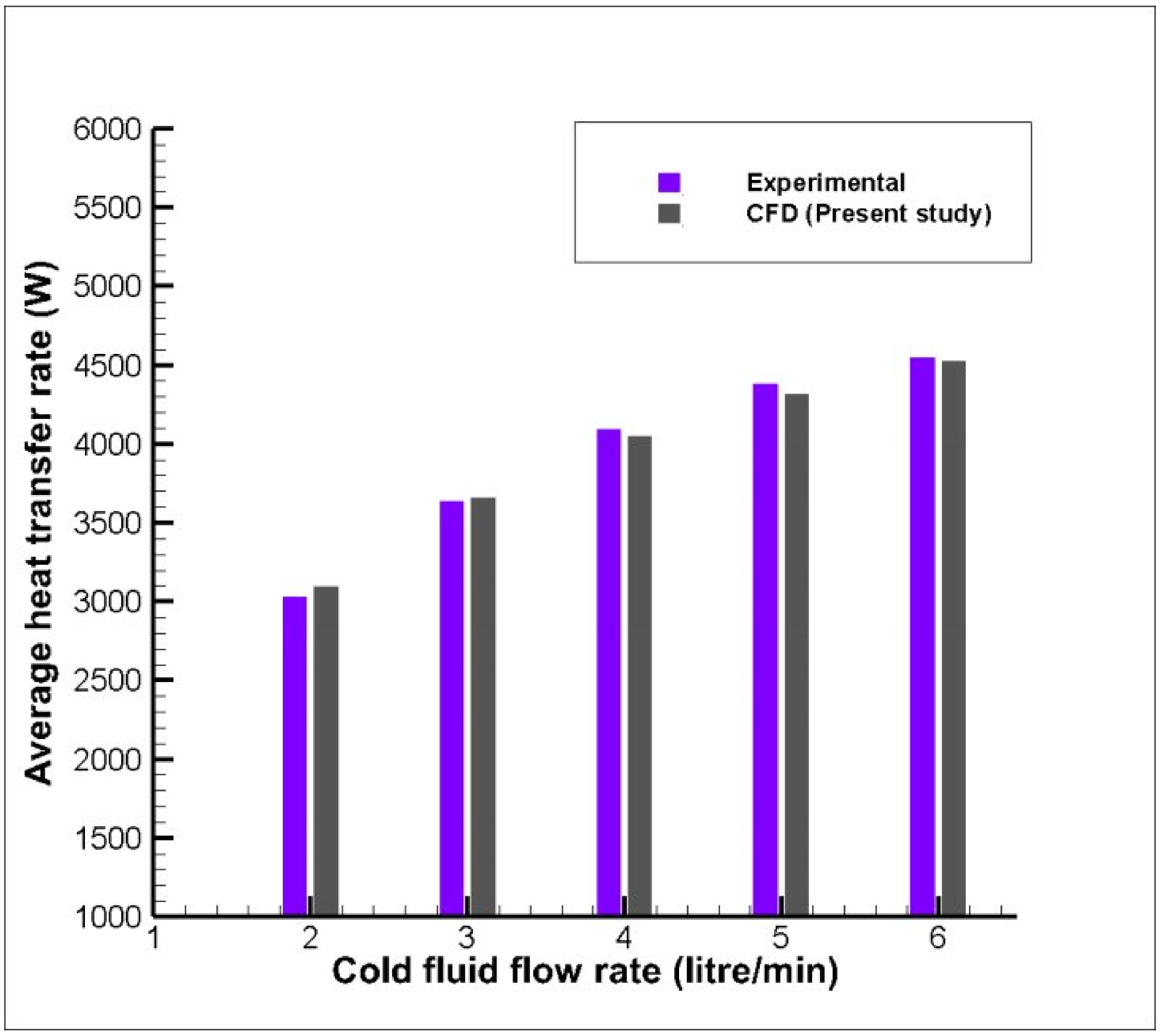

Fig 12 - Validation of heat transfer rate with experimental study

Chapter 5: Results and Discussions

5.1 Outlet Temperature

As mentioned before, numerical simulations of the SHTHX was carried out keeping the hot fluid side mass flow rate constant at 3 liter/min while varying the cold fluid flow rate to 2,3,4,5,6 liter/min. It is noted that when mass flow rate of cold fluid is raised, the outlet temperature of the hot fluid decreases and it occurs due to increased heat flow rate between the cold and hot streams. Figure shows the decrease in temperature of hot fluid side with the surge of cold fluid flow rate. Figure demonstrates the contour of the tube outlet wall temperature and its changes at cold fluid flowrates.

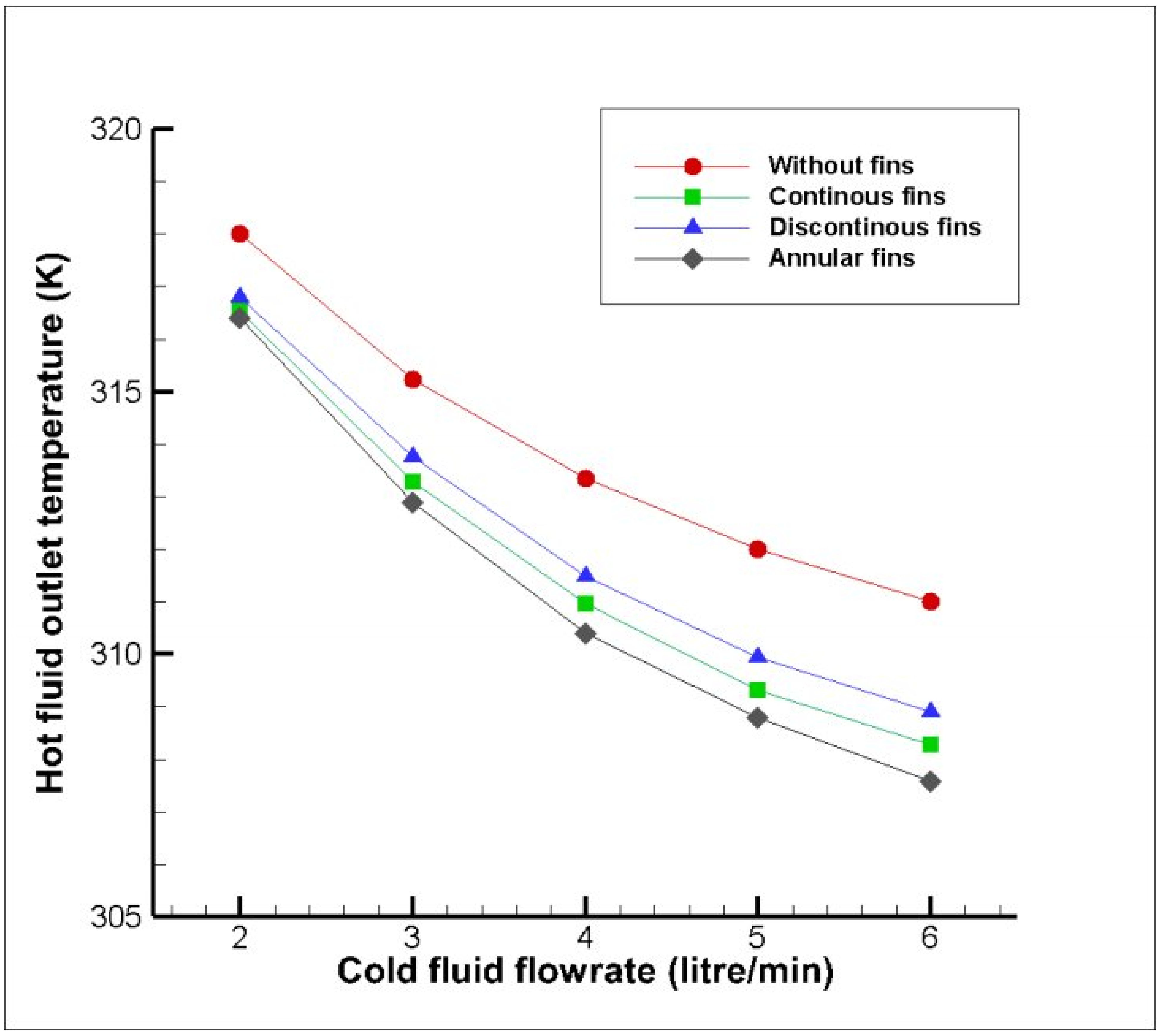

Fig 13 - Change of outlet temperature of hot fluid at different cold fluid flow rates

It is also observed that due to the geometry modification i.e adding fins of various types, the hot fluid side exit temperature drops by a noticeable number. Adding annular fins have the largest impact with a change of around 4K in the hot outlet side. This change is observed due to increased heat transfer area and better turbulent mixing.

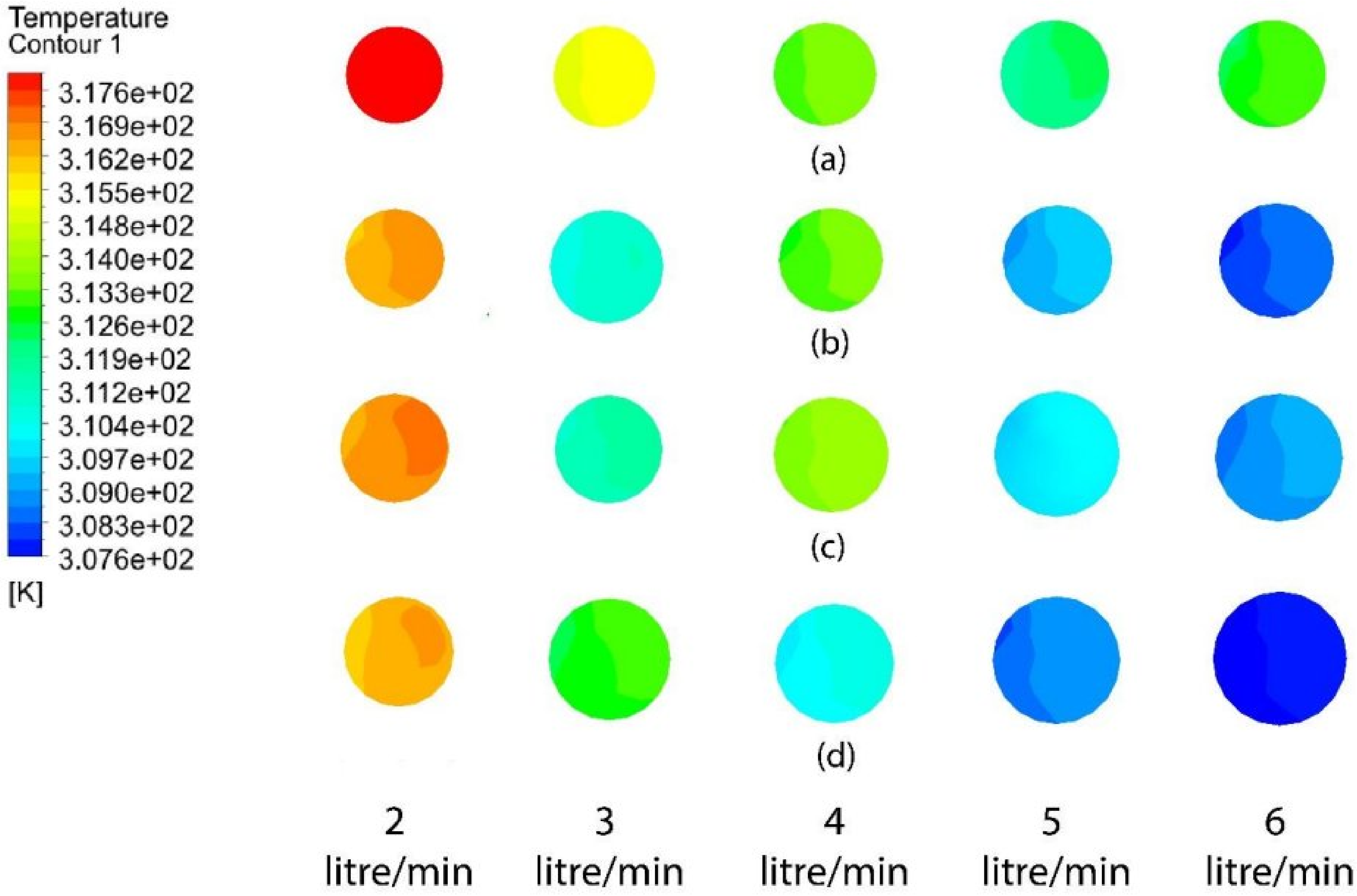

Fig 14 - Temperature contours of hot fluid outlet with varying flow rates (a) without fins (b) Continuous fins (c) Discontinuous fins (d) Annular fins

Figure 14 shows the temperature contour of hot fluid outlet with varying flow rates of all the modifications done in this study. It is clearly visible that the temperature at the outlet drops due to the effect of geometrical modification.

5.2 Average Heat Transfer Rate

The variation in the flow rate of the shell side fluid causes the rate of heat transmission to increase as well. The lowest heat transfer rate is found when the flow rate for the cold fluid is 2 litre/min, while the highest heat transfer rate is seen when the flow rate is kept at 6 litre/min. Figure shows the proportional relation of heat transfer rate with the flow rate.

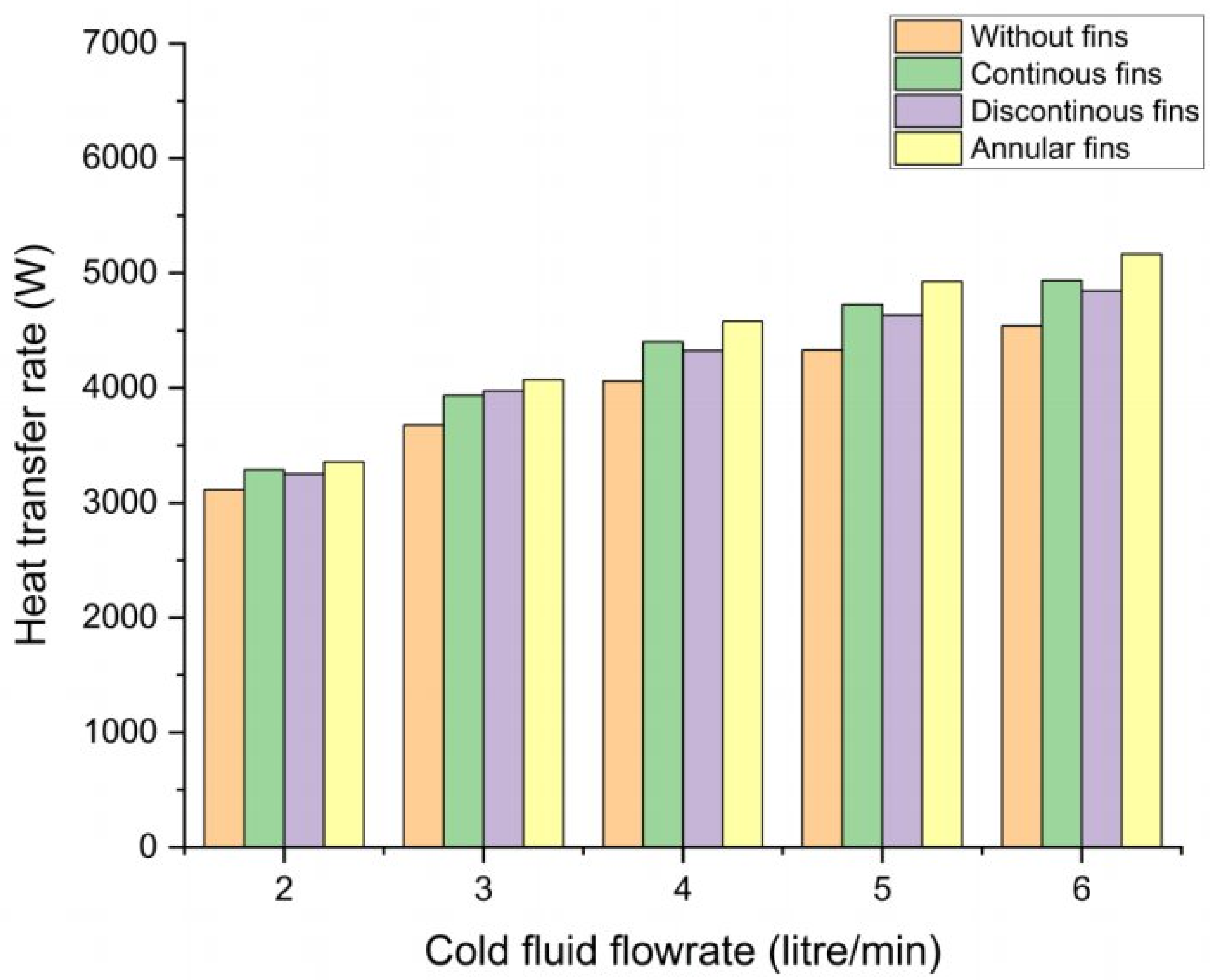

Fig 15 - Change of average heat transfer rate with varying flow rates of cold fluid

Figure 15 also shows the ramifications of geometrical modification on the heat transfer rate with the annular fins having the most amount of heat transfer rate. The rate of heat transfer is directly related to the amount of heat transferring area where convection may occur. Adding fins to the tube side expands the surface area, which is important since heat transfer is caused by convection. Due to this all the modified cases shows surge in the heat transmission rate with a maximal increase of 14% which is observed in the case of the annular fin design of the modified heat exchanger. The graph agrees with the statement properly.

5.3 Pressure Drop

Pressure drop is a significant parameter while calculating the heat transfer rate. The rise of pressure drop brings inefficiency in the system which reflects in the output power of the system. Figure 16 shows the change in pressure drop of the shell side at multiple mass flow rate of the cold side flow while the hot fluid side is kept constant at 3 liter/min.

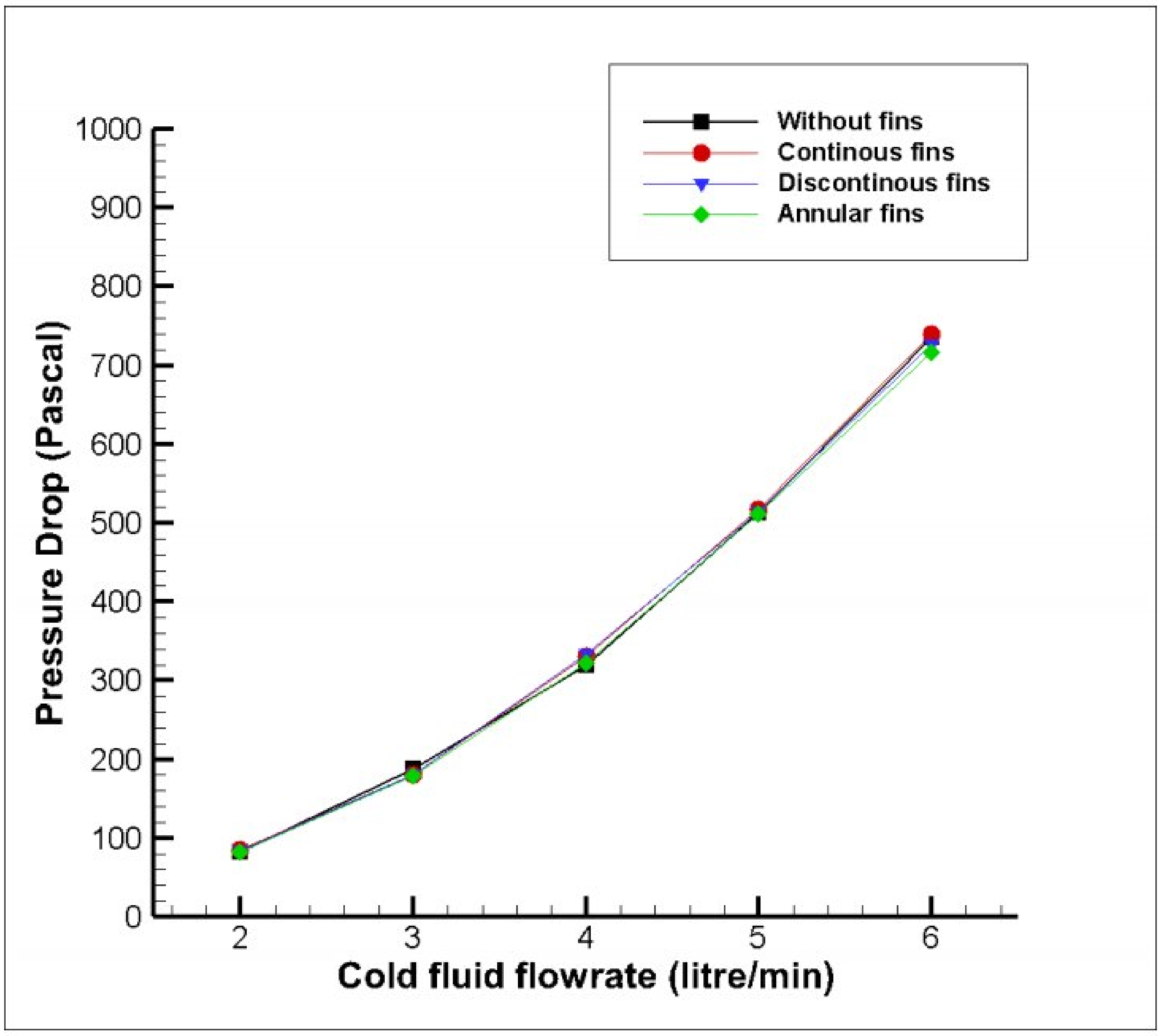

Fig 16 – Change in Pressure-drop with varying shell side fluid flowrate

It is observed from the graph that there is very negligible change in the pressure drop due the geometric modifications made in the present study. Also, the pressure-drop increases drastically due to increase of the flowrate which agrees with the analytical calculations and practical cases.

Chapter 6 Conclusion

In the present study thus far, the hydrothermal analysis involving a baffled shell and helical coil tube heat exchanger was performed varying the mass flow rate of one of two working fluids used. Based on the change of flow rates, variables such as outlet temperature and the average heat transfer rate was calculated.

- The flow rate of the cold fluid increases the rate of heat transfer.
- The outlet temperature of the hot side fluid decreases with the increasing flow rate of cold fluid flow.
- The outlet temperature of the hot fluid side decreases as a result of adding fins and maximum decrease of 4K is observed from the annular fins arrangement.
- The heat transfer rate is augmented with the change in geometry and an increase of up to 14% is reached on the annular fin cases.
- Pressure-loss increases with the increase of fluid flow rate but is very negligible when finned cases are compared to without fins arrangement of the heat exchanger.

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