

Comparative study of shell-and-tube heat exchanger designs for improved thermal performance and energy efficiency

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Abstract

Heat exchangers are fundamental components in numerous industrial processes. This study presents a new shell-and-tube heat exchanger (STHX) design that significantly improves cooling efficiency. The proposed design, featuring a double shell, elliptical tubes at a 90° angle of attack, and flower baffles (DSTHX-90°-F), achieves a 121% increase in temperature drop in the shell side relative to conventional heat exchangers. A computational analysis was performed using ANSYS-Fluent 2021-R2. The numerical model was validated against experimental data from both a conventional single-shell configuration and a prototype of the novel double-shell design, confirming the model's reliability across different geometries. The investigation was conducted in three stages. Initially, a comparative analysis was conducted on a heat exchanger with a 25% baffle cut-off, examining circular (STHX-C-S), elliptical tubes at 90° (STHX-90°-S), and 0° (STHX-0°-S) angles of attack. The elliptical tube at 90° demonstrated the greatest temperature reduction in the shell zone. In the second stage, incorporating flower baffles with elliptical tubes at 90° (STHX-90°-F) resulted in a 69% reduction in pressure drop compared to STHX-90°-S. The third stage developed the DSTHX-90°-F design, combining the double shell, elliptical tubes, and flower baffles for superior performance. This study provides comprehensive flow visualizations using temperature contours, streamlines, and velocity distributions, illustrating substantial enhancements in heat exchanger performance.

Keywords

heat exchanger, CFD, cooling system, temperature drop, pressure drop

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Introduction

Heat exchangers are devices that transfer heat between two fluids at different temperatures without mixing them. They are widely used in various industries, including waste heat recovery, power generation, and air conditioning systems.¹ Their significance is evident in processes such as condensation, heating, and cooling. Shell-and-tube heat exchangers (STHXs) are commonly used due to their ease of installation and robust fabrication.² The effectiveness of heat transfer in this

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system is influenced by factors such as tube arrangement, baffle shape, and shell structure.

Cross-flow heat exchangers with elliptical tubes can provide a greater heat transfer surface area than those with circular tubes. As a result, researchers have explored the impact of tube shape on STHXs. Li et al.³ investigated the friction factor and heat transfer in an elliptical tube and found that it exhibited higher heat transfer performance than a circular tube. Similarly, Matos et al.⁴ evaluated 12 elliptical and circular tube configurations and discovered that elliptical tubes were more efficient, resulting in a 20% increase in heat transfer.

A recent study by Khaled and Mushatet⁵ showed that using double elliptical twisted tubes with twisted tape resulted in a 75% enhancement in heat transfer compared to a plain double-tube design. The twisted tape increased fluid mixing and centrifugal forces, leading to improved thermal-hydraulic performance. Abdul Razzaq and Mushatet⁶ further investigated twisted tube designs, and their study revealed that twisted tubes with oval dimples exhibited a thermal performance factor of 1.38 compared to straight tubes. This highlights the effectiveness of such tubes in disrupting the thermal boundary layer.

Mohanty et al.⁷ conducted a study comparing the heat transfer and pressure drop of plain and ellipsoidal tubes, and the results showed that ellipsoidal tube bundles outperformed plain tubes. He et al.⁸ studied the flow characteristics on the shell side of an STHX using helical baffles and elliptical tubes. Their study revealed that elliptical tubes exhibited higher heat transfer rates, Nusselt numbers, and thermal performance factors compared to circular tubes, with increases of 14.7%–16.4%, 11.4%–16.6%, and 30%–35%, respectively. Saffarian et al.⁹ discovered that heat exchangers with elliptical tubes at a 90° orientation angle near the shell side, combined with circular tubes at the core, exhibit a higher heat transfer rate. Yogesh et al.¹⁰ proposed a three-dimensional method for investigating the impact of the aspect ratio and direction of an elliptical tube on its thermal-hydraulic performance. In a study of STHXs with twisted oval tubes, Zhu et al.¹¹ numerically analyzed the impact of tube geometry on performance. They found that increasing the tube's aspect ratio led to a significant rise in pressure drop of up to 21.8% and a reduction in the heat transfer coefficient. A study published by Kaood et al.¹² examined the performance of convergent tubes with varying dimple shapes and found that stepped-conical dimples exhibited the most favorable thermal enhancement factor, with a 7.81% improvement over smooth tubes and a 36.92% reduction in entropy generation.

The zigzag flow path created by segmental baffles in an STHX has significant drawbacks, including a high pressure drop and the formation of stagnant zones,

which can lead to increased fouling. Consequently, changes in baffle geometry strongly influence the flow and heat transfer on the shell side, prompting extensive research in this area.

Several researchers have explored innovative baffle configurations to address these limitations. Marzouk et al.¹³ found that a design with circular rings and holes was superior to a conventional segmental baffle, improving thermal performance with a 166% increase in effectiveness and a 142% increase in the heat transfer coefficient. Ambekar et al.¹⁴ examined the impact of various baffle configurations on STHX efficiency and found that shorter baffle spacing resulted in better heat transfer performance. Additionally, Yang and Liu¹⁵ performed a simulation comparing plate baffles and rod baffles, revealing that the STHX with plate baffles exhibited significantly better performance (115%–122% heat transfer enhancement) than the rod baffle design. Chen et al.¹⁶ conducted an experimental analysis of an STHX with trisection helical baffles at varying inclination angles. The results showed a decrease in the shell-side heat transfer coefficient and pressure drop as the baffle inclination angle increased. In another study, Gao et al.¹⁷ conducted experiments on discontinuous helical baffles with helix angles of 8°, 12°, 20°, 30°, and 40°. Their results showed that the baffle with a 40° helix angle outperformed the others.

Through a combined numerical and experimental study, Marzouk et al.¹⁸ demonstrated the potential of helical tube heat exchangers, showing that a two-pathway helical configuration enhanced the overall heat transfer coefficient by 125%–185% compared to conventional tube arrangements. Wang et al.¹⁹ investigated a new type of heat exchanger with branch baffles and found that it offered improved pressure loss characteristics compared to traditional designs.

Based on their research, Lei et al.²⁰ conducted a numerical and experimental study on three STHXs with different baffle types, and their analysis revealed that the two-layer helical baffle outperformed the other types. In a separate study, Lei et al.²¹ found that a newly suggested STHX with louver baffles had a higher ratio of heat transfer coefficient to pressure drop than the standard STHX with segmental baffles. Li et al.²² also investigated louvered baffles and showed that they significantly reduced the shell-side pressure drop, increasing the heat transfer coefficient per unit of pressure drop by 20%–125.9%. In their experimental study, Wang et al.²³ demonstrated the effectiveness of a flower-baffle STHX compared to a standard segmental baffle. Chen et al.²⁴ proposed an innovative design with triple-layer flower-shaped baffles. Experimental data showed that the heat transfer coefficients for triple-layer and double-layer flower-shaped baffles are 31.7% and 14.3% greater, respectively, than for conventional segmental baffles. Al-darraj et al.²⁵ compared several

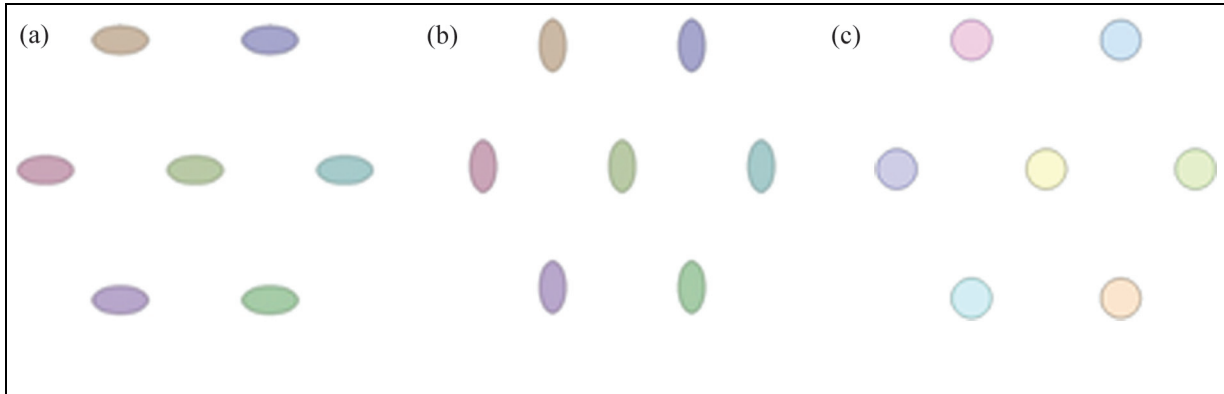


Figure 1. (a) Elliptical tubes with an angle of attack of 0° , (b) elliptical tubes with an angle of attack of 90° , and (c) circular tubes.

baffle designs with air bubble injection and determined that disc and ring baffles with holes were the superior configuration, enhancing effectiveness by 202%–231%. In a different study, El-Said et al.²⁶ showed that using convex peripheral and concave core baffles improved the overall heat transfer coefficient by up to 51.31% while reducing pressure drop by up to 12.40%.

Recently, configurations that optimize thermal performance have been examined, with one promising approach being the integration of an additional shell into the heat exchanger system. This design enhances heat transfer by introducing an additional layer for improved fluid interaction. Bařal and Ünal²⁷ conducted a CFD investigation of a triple concentric tube heat exchanger that used a phase change material to increase the heat transfer area significantly. Other researchers have focused on modeling these systems. Pătrăscioiu and Rădulescu²⁸ created a numerical method to predict the outlet temperature of a triple-tube heat exchanger, while Ünal^{29,30} presented an analytical expression for the temperature variations of the three fluid streams. Similarly, Touatit and Bougriou³¹ presented a method for assessing the temperature profiles and frictional energy consumption in a concentric triple-tube heat exchanger.

In addition to modifying the physical geometry, a widely explored method of enhancing thermal performance is improving the thermophysical properties of the working fluid. Nanofluids, which are engineered suspensions of nanoparticles (e.g. Al_2O_3 , CuO , TiO_2) in a base fluid such as water, have received significant attention. These nanoparticles increase the fluid's thermal conductivity, thereby boosting the convective heat transfer coefficient. Numerous studies have demonstrated that using nanofluids in STHXs can considerably increase the overall heat transfer rate and effectiveness. However, this enhancement often comes at a cost: increased fluid viscosity and higher pressure drop. These factors are important

considerations in practical applications, as reported in several studies.^{32–37}

Despite these advances, a notable gap remains in the literature regarding the synergistic effect of combining multiple advanced components. Most studies focus on modifying either the tubes or the baffles, while the potential of integrating these enhancements with novel shell designs is often overlooked. This work aims to address this gap by proposing and numerically investigating a novel design pathway. The main objective is to maximize thermal efficiency while managing the associated pressure drop. This is achieved by first replacing conventional internals with elliptical tubes and flower baffles, and then integrating this optimized configuration into a unique three-fluid, double-shell heat exchanger. By systematically analyzing these combined geometries, this research provides a comprehensive evaluation of a configuration that has not been explored in previous literature, establishing a new benchmark for enhanced STHX performance.

Methodology

This section describes the geometric configurations, problem formulation, turbulence modeling, boundary conditions, data reduction, validation, and grid independence study.

Geometry overview

The configurations of the shell-and-tube heat exchangers (STHXs) were created using ANSYS Design Modeler. The study investigated STHXs with varying tube geometries, baffle shapes, and shell structures. Three tube variations were considered: elliptical tubes with an angle of attack of 0° (Figure 1(a)), elliptical tubes with an angle of attack of 90° (Figure 1(b)), and conventional circular tubes (Figure 1(c)). In addition, the design incorporated two types of baffles: segmental

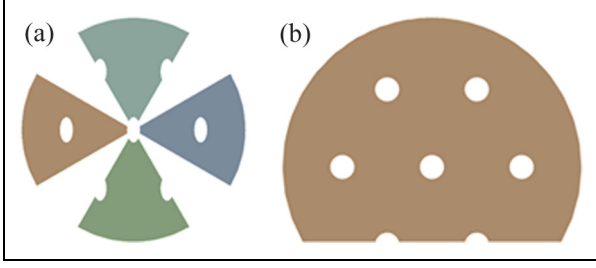


Figure 2. (a) Four-layer flower baffles and (b) segmental baffles with a 25% baffle cut.

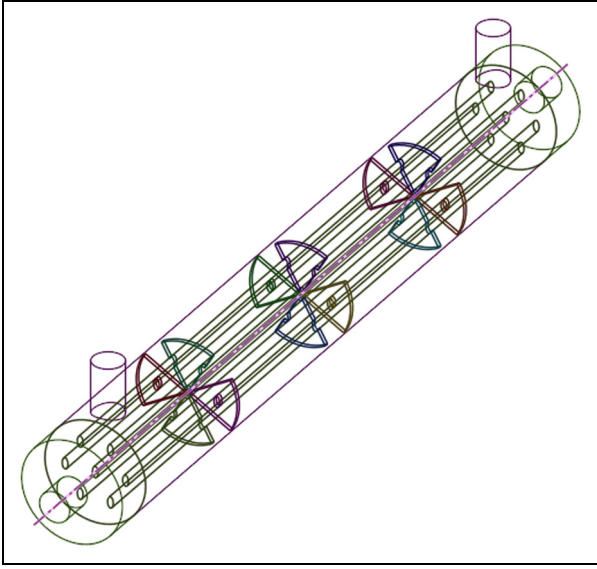


Figure 3. Shell with four-layer flower baffles and elliptical tubes with an angle of attack of 90° (STHX-90°-F).

baffles with a 25% baffle cut (Figure 2(a)) and four-layer flower baffles (Figure 2(b)). The shell structures varied between a standard single shell and a novel double shell.

Consequently, the research comprised five distinct geometric configurations. Three of these designs utilized a single shell with segmental baffles (25% baffle cut) and tubes of varying cross-sections: (1) STHX-0°-S, featuring elliptical tubes at a 0° angle of attack; (2) STHX-90°-S, with elliptical tubes at a 90° angle of attack; and (3) STHX-C-S, with standard circular tubes. The fourth design (STHX-90°-F) integrates a single shell with four-layer flower baffles and elliptical tubes at a 90° angle of attack (Figure 3). The final design is a double-shell heat exchanger (DSTHX-90°-F) featuring four-layer flower baffles and elliptical tubes at a 90° angle of attack. This design includes an inner shell with a diameter of D_1 and an outer shell

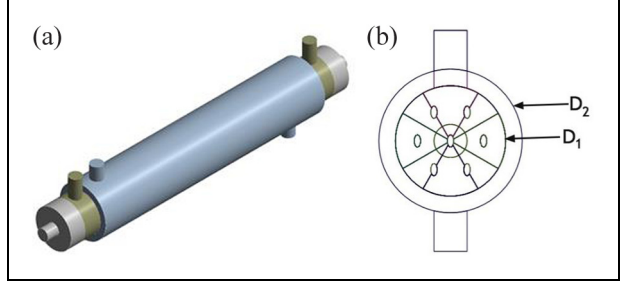


Figure 4. Double shell with four-layer flower baffles and elliptical tubes with an angle of attack of 90° (DSTHX-90°-F).

with a diameter of D_2 (Figure 4). Detailed specifications for each design parameter are provided in Tables 1 and 2.

Problem formulation

This subsection details the assumptions and governing equations used in this investigation.³⁸

The assumptions

1. Fluid flow and heat transfer are characterized by steady, turbulent, three-dimensional, and incompressible conditions.
2. The system comprises a single-phase fluid.
3. Constant thermophysical properties are assumed for fluids and solid materials.
4. The impact of thermal radiation and magnetic force is disregarded.
5. Gravitational effects (body forces) are neglected, as the system is dominated by forced convection.
6. The external walls are considered adiabatic (no heat loss). These idealizations are appropriate for a comparative study focused on the relative performance of different internal geometries.

Principal equations. Continuity equation:

$$\frac{\partial(\rho \bar{u}_i)}{\partial x_i} = 0 \quad (1)$$

Momentum equation:

$$\begin{aligned} & \frac{\partial(\rho \bar{u}_i \bar{u}_j)}{\partial x_i} \\ &= -\frac{\partial \bar{P}}{\partial x_i} + \frac{\partial}{\partial x_j} \left[\mu \left(\frac{\partial \bar{u}_i}{\partial x_j} + \frac{\partial \bar{u}_j}{\partial x_i} \right) - \frac{2}{3} \mu \frac{\partial \bar{u}_i}{\partial x_i} \delta_{ij} - \rho \bar{u}_i' \bar{u}_j' \right] \end{aligned} \quad (2)$$

Table 1. The geometric characteristics of STHX-C-S, STHX-0°-S, STHX-90°-S, and STHX-90°-F.

STHX	STHX-C-S	STHX-0°-S	STHX-90°-S	STHX-90°-F
Shell side (stainless steel)				
D_s	50 mm			
Tube side (stainless steel)				
d_t	4 mm	2.8–1.42 mm	1.42–2.8 mm	1.42–2.8 mm
L	353 mm			
N_t	7			
Layout pattern	60°			
Baffles (stainless steel)				
B_{th}	2 mm			
Number	3	3	3	12

Table 2. The geometric characteristic of DSTHX-90°-F.

Inner shell—the annulus side (stainless steel)	
D_1	50mm
Outer shell (stainless steel)	
D_2	55mm
Tube side (stainless steel)	
d_t	1.42–2.8mm
L	353mm
N_T	7
Layout pattern	60°
Baffles	
B_{th}	2mm
Number	12

Energy equation:

$$\frac{\partial}{\partial x_j} (\rho \bar{u}_j C_P \bar{T}) = \frac{\partial}{\partial x_i} \left(\lambda \frac{\partial \bar{T}}{\partial x_i} + \frac{\mu_t}{Pr_t} \frac{\partial (C_P \bar{T})}{\partial x_i} \right) \quad (3)$$

Turbulence modeling

The Realizable k - ε turbulence model was selected based on its superior performance in accurately predicting the complex flow and heat transfer characteristics within STHXs. This model offers enhanced accuracy, particularly for flows exhibiting high levels of strain and swirl, which are prevalent in heat exchanger applications. In comparison to the standard k - ε and RNG k - ε models, it has demonstrated superior predictive capabilities in these contexts. Studies by El-Said et al.²⁶ and Fetuga et al.³⁹ have confirmed the effectiveness of this model for accurately modeling turbulent flow in similar systems.

Furthermore, this model offers an excellent balance between computational efficiency and accuracy, making it an ideal choice for the simulations conducted in this research. Although more advanced models like the Reynolds stress model (RSM) provide greater accuracy, they are considerably more computationally demanding. The SST k - ω model was also considered; however, based on preliminary trials and a review of similar

studies, the Realizable k - ε model provided the best balance of accuracy and computational cost for this study. Unlike the standard k - ε model, the realizable model features a modified transport equation for the dissipation rate (ε) and calculates the turbulent viscosity coefficient (C_μ) as a variable, which improves accuracy for complex flows.

The equations for the Realizable k - ε turbulence is shown below^{40,41}:

K equation:

$$\frac{\partial}{\partial x_j} (\rho k \bar{u}_j) = \frac{\partial}{\partial x_j} \left[\left(\mu + \frac{\mu_t}{\sigma_k} \right) \frac{\partial k}{\partial x_j} \right] + P_K - \rho \varepsilon \quad (4)$$

ε equation:

$$\frac{\partial}{\partial x_j} (\rho \varepsilon \bar{u}_j) = \frac{\partial}{\partial x_j} \left[\left(\mu + \frac{\mu_t}{\sigma_\varepsilon} \right) \frac{\partial \varepsilon}{\partial x_j} \right] + C_1 \rho S \varepsilon - C_2 \rho \frac{\varepsilon^2}{k + \sqrt{\varepsilon \nu}} \quad (5)$$

Where:

$$C_1 = \max[0.43, \eta/(\eta + 5)] \text{ and } \eta = Sk/\varepsilon$$

The production of turbulent kinetic energy equation:

$$P_k = \mu_t S^2 \quad (6)$$

The turbulent viscosity equation:

$$\mu_t = \rho C_\mu \frac{k^2}{\varepsilon} \quad (7)$$

The model employs the following fixed constants:

$$C_2 = 1.9, \quad \sigma_k = 1.0, \quad \sigma_\varepsilon = 1.2$$

Boundary conditions

This investigation analyzes the heat transfer coefficient and pressure drop in STHXs using a water/water fluid system. The flow within the heat exchanger is turbulent and incompressible.

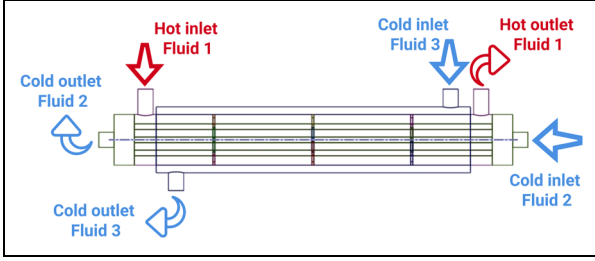


Figure 5. The flow dynamics of the three fluids within the heat exchanger for the DSTHX-90°-F.

In this study, the hot water is supplied to the shell inlet at a temperature of 338K, while the cold water enters the tubes at 283K. For the double-shell design (DSTHX-90°-F), a second cold water stream is also introduced into the outer shell at 283 K (Figure 5). A zero-gauge pressure condition is set at all fluid outlets, establishing the outlet pressure as the reference point for accurately calculating the pressure drop across the system. For the external walls of the heat exchanger, no-slip and adiabatic boundary conditions are applied. The no-slip condition ensures that the fluid velocity at the wall is zero, representing realistic physical interaction, while the adiabatic condition assumes that no heat is lost from the system to the surroundings. Crucially, all internal surfaces, such as tubes and baffles, are treated as coupled thermal boundaries, allowing for the direct calculation of conjugate heat transfer between the fluid streams. The initial comparative study is conducted with a fixed mass flow rate of 0.05 kg/s for all inlets. Subsequently, a parametric analysis examines the impact of varying the hot fluid mass flow rate from 0.02 to 0.05 kg/s.

ANSYS Fluent 2021 R2 was used to conduct the simulations. The Second-Order Upwind scheme was employed for energy and momentum discretization, utilizing the finite volume method. The COUPLED algorithm was applied for pressure-velocity coupling. The convergence criteria of 10^{-7} and 10^{-8} were set for the flow and energy equations, respectively. These strict thresholds were chosen to minimize numerical errors and ensure the solution is fully converged, which provides high confidence in the accuracy of the heat transfer and pressure drop results. These precise computational methods guarantee dependable and accurate results when analyzing the heat transfer and fluid dynamics within the system being investigated.

The thermal-physical characteristics of fluids at an average temperature of 42.5 °C are shown in Table 3.⁴²

Table 3. Thermophysical properties relating to fluids in the shell and tube at atmospheric pressure.

Thermo-physical characteristics	Hot water	Cold water	Unit
Dynamic viscosity, μ	0.00035	0.00035	kg/m.s
Density, ρ	990	997	kg/m ³
Specific heat, C_p	4180	4178	J/kg · K
Thermal conductivity, λ	0.673	0.607	W/m · K

Data reduction

Shell-and-tube heat exchanger. For single-phase heat exchange, the heat balance equation relates the heat transfer rates for each fluid:

$$\phi = \phi_1 = \phi_2 \quad (8)$$

$$\phi_1 = m_1 C_{p,1} (T_{1,in} - T_{1,out}) \quad (9)$$

$$\phi_2 = m_2 C_{p,2} (T_{2,out} - T_{2,in}) \quad (10)$$

For counter-current flow, the logarithmic mean temperature difference (LMTD) is given by:

$$\Delta T_{lm} = \frac{(T_{1,in} - T_{2,out}) - (T_{1,out} - T_{2,in})}{\ln \frac{(T_{1,in} - T_{2,out})}{(T_{1,out} - T_{2,in})}} \quad (11)$$

The overall heat transfer coefficient (U) from the shell side to the tube side, based on the outer surface area of the tubes, is calculated as:

$$U = \frac{\phi}{N_t \pi d_t L \Delta T_{lm}} \quad (12)$$

The shell-side heat transfer coefficient h_1 is defined as⁴³:

$$h_1 = \frac{Nu \cdot \lambda_1}{d_e} = \frac{k_{f,1}}{d_e} J_h Re_1 Pr_1^{1/3} \left(\frac{\mu}{\mu_w} \right) \quad (13)$$

Where Nu is the Nusselt number, λ_1 is the thermal conductivity of the hot fluid, d_e is the equivalent shell diameter, J_h is the heat transfer factor, Re is the Reynolds number, Pr is the Prandtl number, and $\frac{\mu}{\mu_w}$ is the viscosity correction factor, where μ is the dynamic viscosity at the bulk temperature, and μ_w is the dynamic viscosity at the wall temperature.

The pressure drop on the shell side can be calculated using the following equation⁴³:

$$\Delta P_1 = 8 j_f \frac{D_1}{d_e} \frac{L}{B_s} \frac{\rho u_1^2}{2} \left(\frac{\mu}{\mu_w} \right) \quad (14)$$

Where j_f is the friction factor, D is the shell diameter, L is the tube length, l_B is the baffle spacing, and u is the velocity of the fluid.

Double shell and tube heat exchanger. The energy balance of the double-shell heat exchanger (DSTHX) implies that the heat lost by the hot fluid is absorbed by the two cooling streams:

$$\phi = \phi_1 = \phi_2 + \phi_3 \quad (15)$$

$$\phi_1 = m_1 C_{P,1} (T_{1,in} - T_{1,out}) \quad (16)$$

$$\phi_2 = m_2 C_{P,2} (T_{2,out} - T_{2,in}) \quad (17)$$

$$\phi_3 = m_3 C_{P,3} (T_{3,out} - T_{3,in}) \quad (18)$$

The temperature difference for counter-current flow between the fluid on the annulus side and the outer shell can be calculated using the logarithmic temperature difference described below⁴⁴:

$$\Delta T_{lm1-3} = \frac{(T_{1,in} - T_{3,out}) - (T_{1,out} - T_{3,in})}{\ln \frac{(T_{1,in} - T_{3,out})}{(T_{1,out} - T_{3,in})}} \quad (19)$$

The expression below represents the LMTD between the tube face and the annular side:

$$\Delta T_{lm1-2} = \frac{(T_{1,in} - T_{2,out}) - (T_{1,out} - T_{2,in})}{\ln \frac{(T_{1,in} - T_{2,out})}{(T_{1,out} - T_{2,in})}} \quad (20)$$

The total heat transfer coefficient between the annulus side and the outer shell (U_{1-3}) is defined as:

$$U_{1-3} = \frac{\phi}{\pi D_1 L \Delta T_{lm1-3}} \quad (21)$$

Similarly, the total heat transfer coefficient between the annulus side and the tube side (U_{1-2}) is given by:

$$U_{1-2} = \frac{\phi}{N_t \pi d_t L \Delta T_{lm1-2}} \quad (22)$$

Heat exchanger effectiveness. The maximum possible heat transfer rate (ϕ_{max}) is defined as:

$$\phi_{max} = C_{min} (T_{h,in} - T_{c,in}) \quad (23)$$

Where:

$$C_{min} = \min(C_h, C_c)$$

$$C_h = m_h \cdot C_{P,h} \text{ (Heat capacity rate of the hot fluid)} \quad (24)$$

$$C_c = m_c \cdot C_{P,c} \text{ (Heat capacity rate of the cold fluid)} \quad (25)$$

Finally, the effectiveness (ε_H) is calculated as:

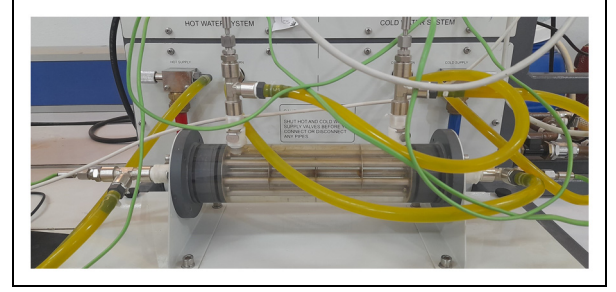


Figure 6. Bench-top heat exchanger system (TecQuipment).

$$\varepsilon_H = \frac{\phi}{\phi_{max}} \quad (26)$$

Mesh independence and baseline experimental validation

Experimental setup and procedure. To establish the accuracy of the numerical model, a validation process was performed by comparing simulation results with experimental data. Simultaneously, the grid independence of the numerical solution was verified. As the DSTHX-90°-F is a novel design with no pre-existing literature data, the validation of the numerical methodology and the grid independence study were first conducted on the conventional STHX-C-S configuration.

The experimental benchmark was performed using a bench-top STHX system developed by TecQuipment (Figure 6), which replicates the dimensions of the STHX-C-S model. The test bench consisted of a hot water system and a cold water circuit. The hot water system comprised a tank with a PID-controlled electric heater, a pump, and a tank with safety features to ensure stable flow rates and temperatures. The cold water circuit was equipped with a flow regulator, connections for an external mains water supply, precision valves, and flow meters for accurate control and measurement. The accuracy of the flow meter used is $\pm 0.5\%$ of full scale, while temperature measurement is achieved using a thermocouple with a typical error range of $\pm 0.5^\circ\text{C}$. The overall uncertainty of the temperature measurement was determined to be $\pm 2\%$.

During the experimental procedure, temperature readings were obtained by inserting thermocouples into pre-designated points on both the shell and tube sides. The flow rates were measured continuously using inline flow meters that were positioned before the inlet of both circuits. Data were recorded in real-time, thereby ensuring consistent monitoring of the temperature and flow rate changes. Before each test, all instruments were

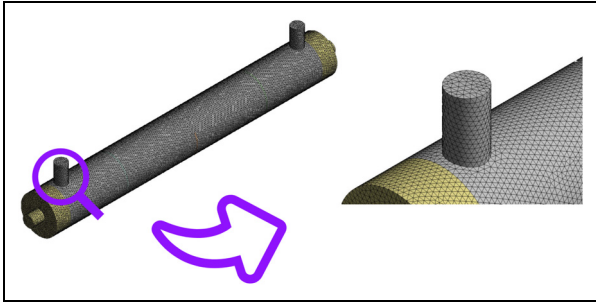


Figure 7. Unstructured tetrahedral mesh in the employed meshed model.

calibrated to guarantee accuracy. Each experiment was conducted under steady-state conditions, with the flow rates adjusted while the inlet temperatures were kept constant. The setup was allowed to stabilize before each measurement to minimize transient effects.

The experiment involved changing the flow rates in the shell while keeping the tube flow rates and temperatures constant. Specifically, the flow in the shell was adjusted from 0.02 to 0.05 kg/s while maintaining a temperature of 338 K. Meanwhile, the tubes remained at a constant flow rate of 0.02 kg/s and a temperature of 293 K under realistic operating conditions. To ensure data precision and reliability, each measurement was repeated three times and averaged.

Grid independence study. To ensure the simulation results were independent of mesh resolution, a grid

independence test was conducted on the STHX-C-S model. An unstructured tetrahedral mesh was generated to accurately capture the complex geometry (Figure 7). Various mesh configurations were evaluated, ranging from 89,800 to 1,654,555 elements. The criteria used to determine independence were the stability of the shell-side pressure drop and heat transfer coefficient. As shown in Figure 8, these parameters began to plateau as the element count increased. The mesh consisting of 890,888 elements was selected as the optimal choice, as further refinement resulted in negligible changes to the solution (1.8% for pressure drop and 1.4% for the heat transfer coefficient). Consequently, the meshing strategy (element sizing and topology) established in this step was consistently applied to all subsequent designs (STHX-90°-F and DSTHX-90°-F) to ensure numerical consistency.

Numerical validation results. Using this selected grid, a direct comparison was made between the numerical predictions and the experimental data, focusing on the shell-side temperature drop (Figure 9). The results showed a good agreement, with the error between the simulation and the experiment ranging from 2.4% to 5.1%. This minor deviation is acceptable and can be attributed to several factors, including the idealizations required in the numerical model, such as the assumption of perfectly smooth surfaces, and uncertainties in fluid material properties. The close correlation between the experimental data and the numerical predictions confirms the reliability of the grid and the overall

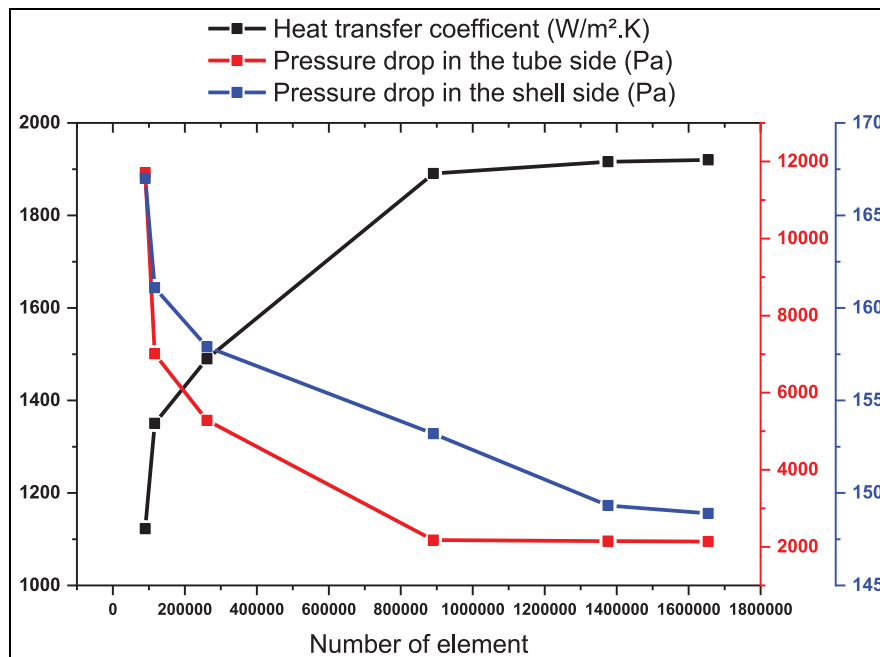


Figure 8. The heat transfer coefficient and pressure drop for different numbers of grids.

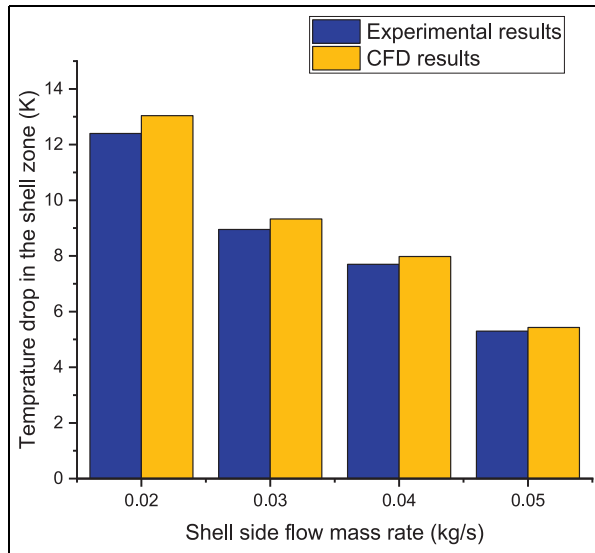


Figure 9. Comparison between simulation and experimental findings.

solution method for analyzing the performance of the STHXs in this study.

Experimental validation of the double-shell configuration

Prototype description and operation. To build confidence in the numerical model for the novel double-shell geometry, an experimental study was conducted on a fabricated prototype of the double-shell heat exchanger. This prototype was constructed with standard circular tubes and flower baffles (Figure 10). The principal geometric dimensions, including the tube and shell diameters, remained consistent with those in the numerical investigation. The only difference between the prototype and the model is the tube length, which was set to 1000mm.

The prototype was designed for an air-water working fluid system. During operation, hot air was directed into the inner shell. Cold water from a single source was split into two streams: one flowed through the tubes and the second flowed through the outer shell.

Experimental procedure and conditions. Turbine flowmeters were used to measure the mass flow rates of the air and water streams. K-type thermocouples were positioned at each inlet and outlet to measure temperatures. Each instrument was calibrated before the tests, with typical accuracies of $\pm 0.5\%$ for flowmeters and $\pm 0.5^\circ\text{C}$ for thermocouples, contributing to an overall experimental uncertainty of $\sim 2\%$.



Figure 10. Fabricated prototype of the double-shell heat exchanger with internal components.

The objective was to measure the air outlet temperature as its mass flow rate was varied. The following boundary conditions were maintained:

- *Hot air inlet temperature:* Fixed at 473 K.
- *Cold water inlet temperature:* Fixed at 293 K for both streams.
- *Cold water mass flow rate:* A total of 20 L/min ($\sim 0.33 \text{ kg/s}$) was split evenly between the tubes and outer shell.
- *Hot air mass flow rate:* Varied from 0.01. to 0.04 kg/s.

CFD replication and qualitative comparison. A corresponding CFD simulation was performed to replicate the exact geometry and air-water conditions of the prototype. The simulation used the same meshing techniques, turbulence model (Realizable $k-\epsilon$), and solver settings as described in the main numerical study.

The goal of this comparison is a qualitative validation of the performance trends. Figure 11 compares the experimental and numerical results for the hot air outlet temperature. As shown, the air outlet temperature increases with increasing mass flow rate. This is physically consistent, as a higher flow rate reduces the residence time of the air in the exchanger, limiting the time available for heat transfer.

The CFD model successfully predicts this trend with good accuracy, showing a maximum deviation of only 5.3%. This minor deviation between the experimental and numerical results is expected and can be attributed to several factors, including idealizations in the CFD model (such as assuming perfectly smooth surfaces), minor heat losses to the ambient environment in the experimental setup, and the inherent measurement

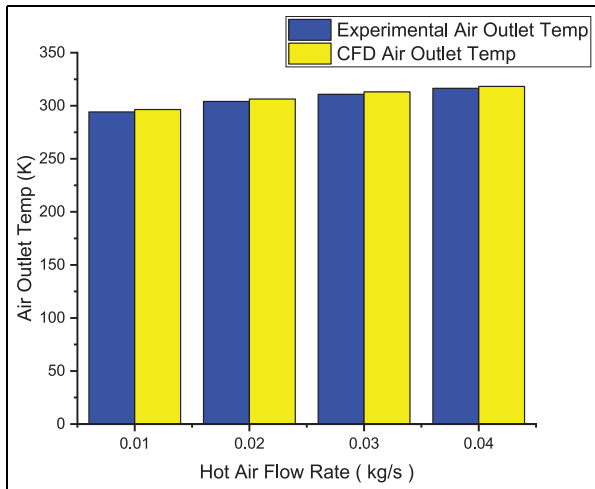


Figure 11. Comparison of experimental and numerical air outlet temperatures for the double-shell prototype.

uncertainty of the instruments. The strong agreement on the performance trend provides significant additional confidence in the numerical model's ability to capture the complex physics within the novel double-shell configuration.

Results

CFD analysis results and performance comparison

This study was divided into three parts to achieve the optimal design and development for the heat exchanger cooling system.

Comparison of tube geometries. The initial computational fluid dynamics (CFD) analysis revealed important insights when comparing a standard shell and tube heat exchanger (STHX) with a 25% baffle cut and tubes of different cross-sections: circular (STHX-C-S), elliptical with an angle of attack of 90° (STHX- 90° -S), and 0° (STHX- 0° -S). Table 4 presents the CFD findings.

The STHX- 90° -S showed the highest temperature drop within the shell, highlighting its superior thermal

performance, while the baseline STHX-C-S had the lowest. In contrast, the highest pressure drop was observed in the STHX- 0° -S. Compared to STHX-C-S, the temperature drop in STHX- 90° -S and STHX- 0° -S was 23% and 14% greater, respectively. Additionally, the pressure drop was 13% and 15% higher in STHX- 90° -S and STHX- 0° -S, respectively.

These performance variations are based on the way the tube geometry interacts with the shell-side flow. At a 90° angle of attack, the elliptical tubes present a greater frontal area to the cross-flow, which more successfully breaks up the thermal boundary layer and creates more turbulence. This enhanced mixing is the primary reason for its superior heat transfer. The higher pressure drop is also a direct result of this disruption, which creates higher form drag and flow resistance compared to the more streamlined circular tubes.

Comparison of baffle configurations. In heat exchanger design, pressure drop is a crucial factor. To address the increased pressure drop from the elliptical tubes, a different type of baffle, known as a flower baffle, was tested. This second phase of the research compared the STHX- 90° -S (with segmental baffles) with the modified configuration, STHX- 90° -F, which features four-layer flower baffles. Table 4 demonstrates a significant improvement in hydraulic performance. Specifically, STHX- 90° -F exhibited a substantial 69% decrease in pressure drop compared to STHX- 90° -S.

The two baffle types induce fundamentally different flow paths. Conventional segmental baffles create large recirculation zones (dead zones) and high-velocity jets that dissipate energy by forcing fluid into an abrupt zigzag crossflow. In contrast, flower baffles guide the fluid around the tube bundle along a smoother, helical path. This minimizes dead zones and sudden changes in direction, significantly reducing the pressure drop while maintaining effective flow distribution.

Performance of the double-shell configuration. In the third and final phase of this investigation, the optimal features from the previous stages were combined in a

Table 4. CFD analysis results: STHX-C-S, STHX- 0° -S, STHX- 90° -S, STHX- 90° -F.

STHX	STHX-C-S	STHX- 0° -S	STHX- 90° -S	STHX- 90° -F
Shell side				
Temperature drop (K)	5.2	6.65	6.91	6.93
($T_{li} = 338$ K)				
Pressure drop (Pa)	146.65	151.67	149.42	88.46
Tube side				
Temperature rise (K)	6.49	6.36	6.2	6.25
($T_{2i} = 293$ K)				
Pressure drop (Pa)	2114.09	2152.3	2125.23	2049.4

Table 5. CFD analysis results: DSTHX-90°-F.

STHX	DSTHX-90°-F
Inner shell—the annulus side	
Temperature drop (K)	11.54
($T_{li} = 338$ K)	
Pressure drop	88.61
Outer shell	
Temperature rise (K)	4.67
($T_{3i} = 293$ K)	
Pressure drop (Pa)	89.67
Tube side	
Temperature rise (K)	6.31
($T_{2i} = 293$ K)	
Pressure drop (Pa)	2049.81

novel double-shell heat exchanger (DSTHX-90°-F), as detailed in Table 5. This design includes four-layer flower baffles and elliptical tubes positioned at a 90° angle of attack. When comparing the thermal performance of DSTHX-90°-F to its single-shell predecessor (STHX-90°-F), a significant 67% increase in temperature drop is observed. Furthermore, when compared to the original baseline design (STHX-C-S), under the same inlet mass flow rate conditions (0.05 kg/s), DSTHX-90°-F demonstrates a remarkable 121% increase in temperature drop.

The superior performance of the DSTHX-90°-F is the result of a strong synergistic effect between its three key components. First, the elliptical tubes at 90° enhance the local heat transfer at the tube surfaces. Second, the flower baffles provide an efficient, low-pressure-drop flow path that ensures the entire tube bundle is effectively utilized. Finally, the double-shell structure introduces a secondary cooling mechanism, where the inner shell fluid is cooled simultaneously by both the tubes and the outer shell fluid. This combination of enhanced local heat transfer, efficient global flow, and an additional cooling surface enables this design to achieve thermal performance far superior to any of the prior configurations.

Temperature contour analysis

Figure 12(a) to (e) shows the temperature distribution across the five STHX configurations: STHX-C-S, STHX-0°-S, STHX-90°-S, STHX-90°-F, and DSTHX-90°-F. The contour plots illustrate the thermal profiles along the central plane of each system. A consistent trend is observed in all configurations: the shell-side fluid temperature decreases from the inlet to the outlet while the tube-side fluid temperature increases. This confirms the fundamental counter-current heat exchange process.

Upon examining the temperature contours, a critical distinction becomes apparent in the DSTHX-90°-F

model. The contours demonstrate a steeper and more uniform thermal gradient throughout the heat exchanger volume compared to the single-shell designs. Notably, the baseline STHX-C-S and STHX-0°-S designs exhibit “hot spots,” or regions of sustained high temperature, along the shell side, especially in the recirculation zones immediately downstream of the segmental baffles. These stagnant regions inhibit effective mixing and indicate poor local heat transfer.

By contrast, the DSTHX-90°-F configuration shows a quick and steady temperature drop throughout the shell area, with far fewer localized hot spots. This uniform appearance demonstrates the effectiveness of the flower baffles in eliminating stagnation points. The enhanced turbulent mixing generated by the elliptical tubes (oriented at 90°) and the flower baffles is evident in the more dispersed and less stratified thermal layers. In the STHX-90°-S and STHX-90°-F configurations, the flow structures around the elliptical tubes disrupt the thermal boundary layer. This prevents the formation of large temperature gradients and promotes thorough interaction between the hot and cold streams.

Furthermore, the contribution of the double-shell structure is clearly visible. In the DSTHX-90°-F contours, the outer boundary of the inner shell appears as a region of lower temperature, indicated by cooler colors. This confirms that the fluid in the inner shell is cooled by both the internal tube bundle and the secondary coolant stream in the annulus. This dual-cooling mechanism effectively increases the active heat transfer surface area, directly explaining the 121% improvement in temperature drop reported in the quantitative analysis.

Streamlines and velocity distribution

Figure 13(a) and (b) provides a detailed representation of the streamlines for STHX-C-S and STHX-90°-F, respectively.

As shown in Figure 13(a), the streamlines on the shell side of the STHX-C-S model exhibit a complicated zigzag flow pattern. This flow, forced by the segmental baffles, creates large recirculation zones (dead zones) behind each baffle. These stagnant areas lead to poor heat transfer and contribute to a high overall pressure drop.

In contrast, Figure 13(b) shows a much more uniform, helical flow path for the STHX-90°-F. This streamlined pattern, created by the flower baffles, significantly reduces the size of the dead zones and ensures a more effective distribution of the fluid across the entire tube bundle. The direct modification of the flow path from a zigzag pattern to a helical pattern is the fundamental reason for the 69% decrease in pressure drop observed in the STHX-90°-F compared to the STHX-90°-S. This is because less energy is dissipated

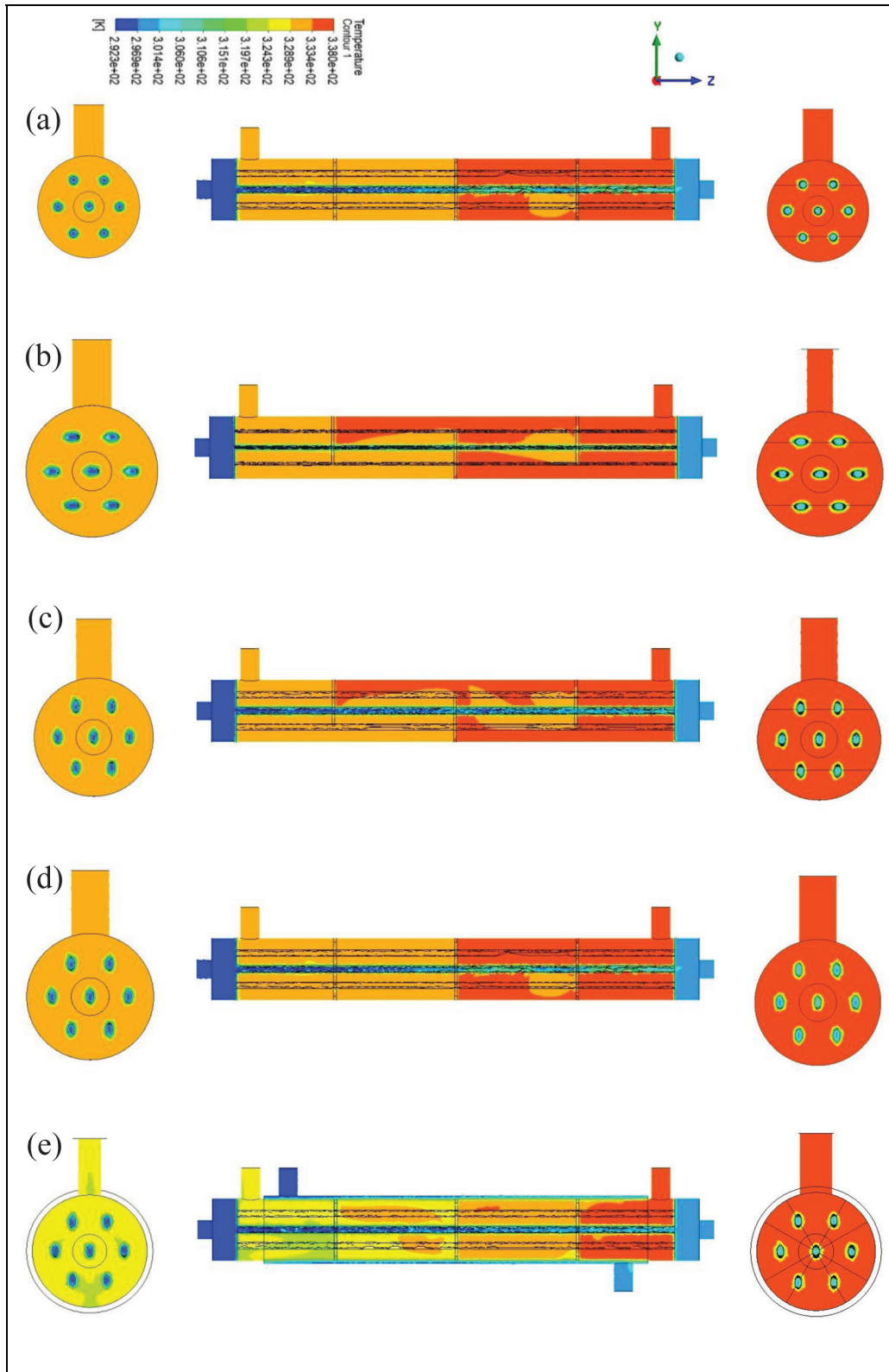


Figure 12. Temperature distribution through the y-z axis at the center of the heat exchanger: (a) STHX-C-S, (b) STHX-0°-S, (c) STHX-90°-S, (d) STHX-90°-F and (e) DSTHX-90°-F.

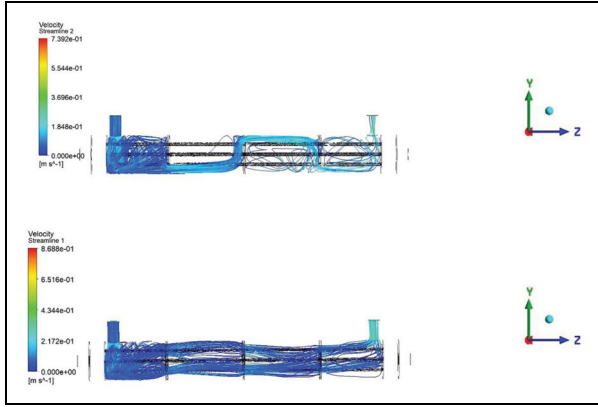


Figure 13. Streamlines and velocity distribution through the y - z axis of the heat exchanger: (a) STHX-C-S and (b) DSTHX-90°-F.

through abrupt changes in direction and the generation of turbulence in stagnant zones.

Figure 14 provides a visual representation of the DSTHX-90°-F streamlines. This illustration shows how the third fluid is introduced into the outer shell, creating an additional cooling domain. The addition of this third fluid stream improves overall thermal performance by providing a secondary cooling mechanism, which utilizes the full volume of the heat exchanger more effectively.

The impact of varying flow rates

To determine the impact of varying flow rates on temperature drop, pressure drop, and heat exchanger effectiveness, a comparison was made between the conventional STHX-C-S and the double-shell DSTHX-90°-F. For the STHX-C-S, the tube-side flow was fixed at 0.05 kg/s while the shell-side flow was varied from 0.02 to 0.05 kg/s. For the DSTHX-90°-F, the tube-side and outer-shell flows were fixed

at 0.05 kg/s, while the inner-shell flow was varied from 0.02 to 0.05 kg/s.

Effect on hot fluid temperature drop. Figure 15 shows that as the hot fluid mass flow rate decreases, the temperature drop of the hot fluid increases for both designs. This is because a lower flow rate increases the fluid's residence time, allowing for more complete heat transfer. Across the entire tested range, the temperature drop in the DSTHX-90°-F is 67%–121% greater than in the STHX-C-S, confirming its superior cooling performance.

Effect on shell-side pressure drop. Figure 16 shows that the pressure drop decreases significantly as the flow rate decreases for both models. Crucially, the use of flower baffles in the DSTHX-90°-F results in a 76%–123% reduction in pressure drop compared to the segmental baffles in the STHX-C-S across the entire range of flow rates. This demonstrates the hydraulic advantage of the flower baffle design.

Effect on heat exchanger effectiveness. A series of tests were conducted to assess the performance of two heat exchanger designs (Figure 17), the STHX-C-S, and the DSTHX-90°-F, across a range of shell-side mass flow rates from 0.02 to 0.05 kg/s. A summary of the findings is shown below:

- At a shell-side mass flow rate of 0.02 kg/s, the STHX-C-S had an effectiveness of 0.23, whereas the DSTHX-90°-F had a higher effectiveness of 0.39.
- As the mass flow rate increased to 0.03 kg/s, the STHX-C-S's effectiveness decreased to 0.20. Similarly, the DSTHX-90°-F showed a decrease in effectiveness, with a value of 0.30.

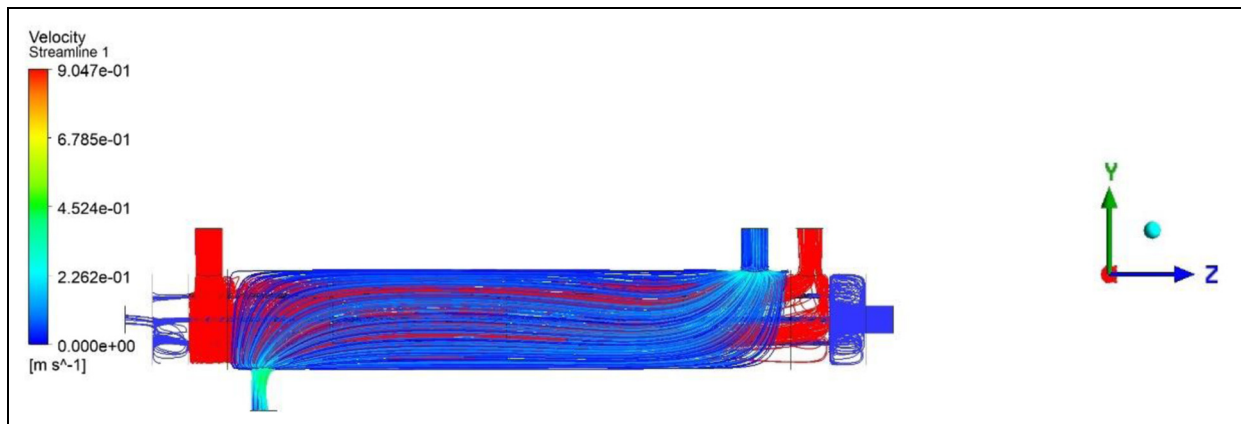


Figure 14. Streamlines and velocity distribution through the y - z axis of the heat exchanger DSTHX-90°-F.

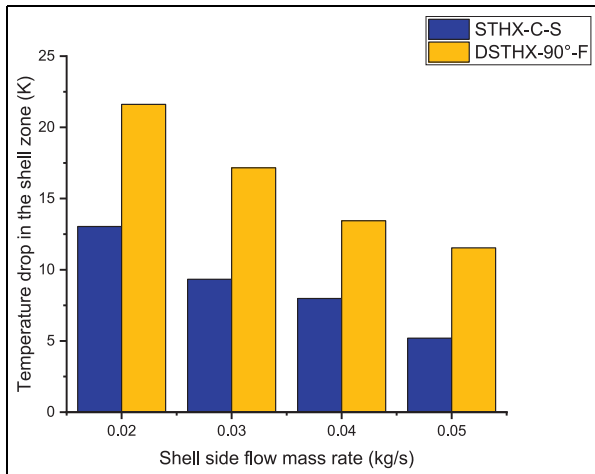


Figure 15. The effect of modifying the flow on the cooling of the hot water.

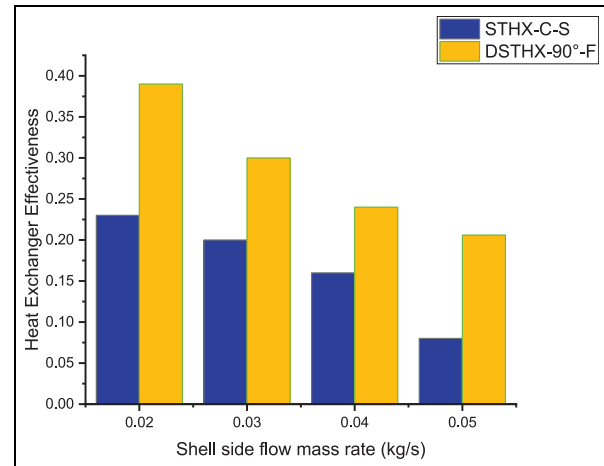


Figure 17. Comparison of heat exchanger effectiveness between STHX-C-S and DSTHX-90°-F at different shell-side mass flow rates.

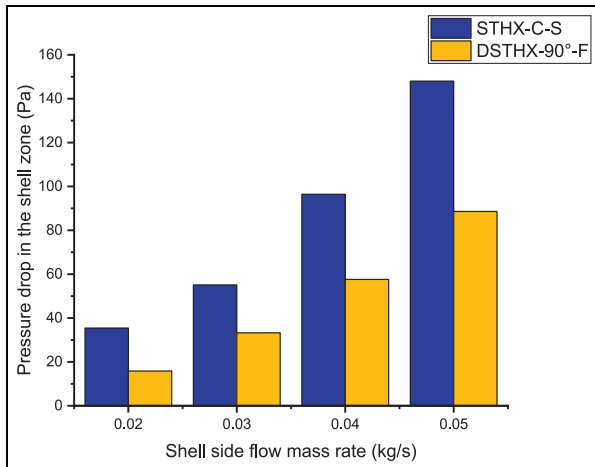


Figure 16. Effects of changing flow rate on pressure loss.

- Increasing the mass flow rate to 0.04kg/s further reduced the STHX-C-S effectiveness to 0.16. In contrast, the DSTHX-90°-F maintained an effectiveness of 0.24.
- At the maximum evaluated mass flow rate of 0.05kg/s, the effectiveness of the STHX-C-S decreased significantly to 0.08. In comparison, the DSTHX-90°-F showed a far higher effectiveness of 0.206.

The results demonstrate that the DSTHX-90°-F heat exchanger was consistently more effective than the STHX-C-S at all tested shell-side mass flow rates. It is worth noting that the difference in effectiveness between the two designs was most noticeable at lower flow rates, suggesting that the DSTHX-90°-F design is more beneficial in low-flow applications. As the mass

flow rate increased, both designs' effectiveness decreased; nonetheless, the DSTHX-90°-F outperformed the STHX-C-S. This demonstrates that the DSTHX-90°-F design is more robust to flow rate variations, making it a superior choice for applications with variable or lower mass flow rates on the shell side.

Experimental demonstration of the double-shell effect

To demonstrate the practical significance of the double-shell configuration, a comparative test was conducted using the fabricated prototype described in the validation section. The objective of this experiment was to isolate the direct impact of the third fluid stream on cooling performance.

Two tests were performed under the same inlet conditions (Table 6). First, the prototype was operated as a standard two-fluid heat exchanger by closing the cold water inlet of the outer shell. In this configuration, the single stream of cold water had limited cooling capacity, resulting in an elevated outlet temperature of 302.9K for the hot air. Second, the test was repeated with the outer shell inlet opened to enable full three-fluid operation. With both cooling streams active, the cooling capacity was significantly increased, and the outlet temperature of the hot air decreased to 291.4K. These results confirm that the second cooling stream in the double-shell design significantly enhances the overall heat transfer.

It is essential to highlight the practical implications of the proposed double-shell, three-fluid design. This configuration is not intended as a universal replacement for traditional STHXs, but rather as a specialized solution for specific industrial applications. For example, it

Table 6. Experimental comparison of prototype operating modes.

Parameter	Test 1: single-shell mode	Test 2: double-shell mode
Outer shell condition	Inactive (no flow)	Active (flow enabled)
Hot air inlet temperature	473K	473K
Cold water inlet temperature	293K	293 K (for both streams)
Result: air outlet temperature	302.9K	291.4K
Experimental observation	Rapid thermal saturation of coolant	Sustained cooling capacity
Performance outcome	Standard cooling	Significantly enhanced cooling

is well-suited for processes that require the simultaneous cooling of a primary hot stream by two separate coolants, or for systems that need to precisely control the temperature of an intermediate fluid. Therefore, this design represents a flexible and compact solution for complex thermal management tasks.

Conclusion

This study optimized the performance of a shell-and-tube heat exchanger (STHX) by systematically investigating various tube cross-sections, baffle configurations, and shell structures. The study resulted in a novel double-shell design (DSTHX-90°-F). The key findings are summarized below.

- The use of elliptical tubes significantly improved thermal performance. Compared to the conventional STHX-C-S, the STHX-90°-S and STHX-0°-S designs showed a 23% and 14% increase in shell-side temperature drops, respectively.
- While enhancing heat transfer, the STHX-90°-S and STHX-0°-S elliptical tube configurations also resulted in higher shell-side pressure drops of 13% and 15%, respectively, compared to the STHX-C-S configuration.
- The implementation of flower baffles successfully minimized the rise in pressure losses. The STHX-90°-F design reduced the pressure drop by 69%, demonstrating its hydraulic advantage over the segmental baffle STHX-90°-S design.
- The DSTHX-90°-F was developed by combining these optimal features. This design demonstrated a remarkable 67% increase in temperature drop compared to the single-shell STHX-90°-F predecessor and a 121% increase compared to the baseline STHX-C-S, confirming the synergistic effect of its components.
- A parametric analysis of the DSTHX-90°-F revealed that decreasing the flow rate of the hot fluid in the inner shell significantly increased its temperature drop and reduced the pressure drop. These results suggest that optimal performance

is achieved by combining slower flow in the inner shell with faster flow in the tubes and outer shell.

- The reliability of the numerical model was confirmed through experimental validation against both a conventional STHX and a fabricated prototype of the novel double-shell configuration. The results showed strong agreement with the CFD predictions.

The findings offer practical pathways to improve the efficiency of industrial processes that rely on heat exchangers. This could lead to energy savings, reduced costs, and a smaller environmental footprint. However, future advancements in manufacturing and sensor integration are necessary to realize the full potential of these optimized designs.

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Appendix

Nomenclature

Latin characters

B_c	Baffle cut, %
B_s	Baffle spacing, mm
B_{th}	Baffle thickness, mm
C	Heat capacity rate, W/K
C_P	Specific heat capacity, J/(kg · K)

C_1

C_2

C_μ

d_e

d_t

D_s

D_1

D_2

h

j_f

j_h

k

L

m

N_t

Nu

P

ΔP

P_k

Pr

Pr_t

Re

S

T

ΔT_{lm}

u

$\overline{u_i}, \overline{u_j}$

U

x_i, x_j

x, y, z

Δ

∇

δ_{ij}

ε

ϕ

η

ε_H

λ

μ

μ_t

ν

ρ

$\sigma_\varepsilon, \sigma_k$

Δ

∇

δ_{ij}

ε

ϕ

η

ε_H

λ

μ

μ_t

ν

ρ

$\sigma_\varepsilon, \sigma_k$

Δ

∇

δ_{ij}

ε

ϕ

Realizable k – ε model coefficient (variable)

Realizable k – ε model constant

Realizable k – ε model coefficient (variable)

Equivalent shell diameter, mm

Tube diameter, mm

Shell diameter,

Inner shell diameter, mm

Outer shell diameter,

Heat transfer coefficient,

Friction factor

Heat transfer factor

Turbulent kinetic energy,

Tube length, mm

Mass flow rate, kg/s

Number of tubes

Nusselt number

Pressure, Pa

Pressure drop, Pa

Production of turbulent kinetic energy, kg/m · s³

Prandtl number

Turbulent Prandtl number

Reynolds number

Rate of strain tensor

Temperature, K or °C

Logarithmic mean temperature difference, K

Velocity, m/s

Time-averaged velocity components in directions i, j , m/s

Overall heat transfer coefficient, $\frac{W}{m^2 \cdot K}$

Spatial coordinates in directions i, j , m

Cartesian coordinate axes, m

Greek symbols

Δ

∇

δ_{ij}

ε

ϕ

η

ε_H

λ

μ

μ_t

ν

ρ

$\sigma_\varepsilon, \sigma_k$

Δ

∇

δ_{ij}

ε

ϕ

η

ε_H

Change

Gradient operator

Kronecker delta

Dissipation rate of turbulence, m²/s³

Heat transfer rate, W

Intermediate variable for calculating C_1

Heat exchanger effectiveness

Thermal conductivity, W/(m · K)

Dynamic viscosity, kg/(m · s)

Turbulent viscosity, kg/(m · s)

Kinematic viscosity (molecular), m²/s

Density, kg/m³

Turbulent model constants

Subscripts

1

Fluid or wall relative to the inner shell (the annulus side)

2

Fluid or wall relative to the tube side

3	Fluid or wall relative to the outer shell	<i>Abbreviations</i>	
<i>c</i>	Cold fluid	0°	Elliptical tubes with an angle of attack of 0°
<i>h</i>	Hot fluid	3D	Three-dimensional
<i>i,j</i>	Tensor index (for spatial direction or component)	90°	Elliptical tubes with an angle of attack of 90°
<i>in</i>	Inlet condition	C	Circular tubes
<i>max</i>	Maximum	CFD	Computational fluid dynamics
<i>min</i>	Minimum	DSTHX	Double-shell heat exchanger
<i>num</i>	Numerical	F	Flower baffle
<i>out</i>	Outlet condition	LMTD	Log mean temperature difference
<i>w</i>	Wall condition	NTU	Number of transfer units
<i>Superscripts</i>		PID	Proportional-integral-derivative controller
–	(Overbar) time-average quantity	RNG	Re-normalization group
‘	(RANS)	RANS	Reynolds-averaged Navier–Stokes
	(Prime) fluctuating component	RSM	Reynolds stress model
	(RANS)	SST	Shear stress transport
		STHX	Shell-and-tube heat exchanger