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Original Research

Entropy Generation and Performance Assessment of Finned Tubes in Double Pipe Heat Exchangers

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ABSTRACT

Double-pipe heat exchangers (DPHE's) are widely used in thermal systems, and enhancing their performance is critical for energy efficiency. This study examines the impact of fins installation on heat transfer (HT) and entropy generation in a counterflow DPHE using CFD simulation. The inner pipe is made of copper with an inner diameter of 35 mm, outer diameter of 38.1 mm, and 1000 mm in length. The outer pipe is made of PVC with an inner diameter of 70.26 mm and an outer diameter of 74.66 mm, sharing the same length as the inner pipe. Numerical simulations were conducted to compare smooth and finned tubes under identical thermal and flow conditions. Entropy generation due to HT and fluid friction was analyzed. The results showed that finned tubes improved HT (Q) watts up to 22%, particularly at higher hot water temperatures of 70 ºC and 0.1 kg/s flow rate. However, this enhancement was accompanied by increased entropy generation up to 40%, indicating higher thermodynamic irreversibility. The study highlights the trade-off between thermal performance and entropy generation, providing valuable insight for optimizing heat exchanger design. The finned tube also showed good enhancement of Nusselt number by 6-8% and effectiveness by 20% compared to the smooth tube.

Keywords: Heat Exchanger, Double pipe, Thermal performance, Micro Fins, Entropy, CFD Simulation, Nusselt Number, Effectiveness

1. Introduction

Heat exchangers (HEs) are essential in residential and industrial applications. DPHEs are popular for their simplicity and thermal range. Passive techniques like fins, twisted tapes, nanofluids, and roughened surfaces enhance HT but also increase pressure loss, requiring a balance for optimal energy efficiency.[1]. The applications of heat exchangers have advanced significantly in engineering due to their inherent privileges, obtained through the execution of passive, active, or combined improvement techniques. **Al-Kayiem et al.**[2] reported that the HT enhancement is either (i) active methods, which use an external power source, or (ii) passive methods, which use a techniques without a power source, such as turbulence generators (iii) compound HT enhancement, when two passive or passive active methods are used simultaneously. The passive techniques are usually applied to the inner pipe of DPHEs using several inserts, such as twisted tapes, which disturb the boundary layers of flowing fluid inside the pipe. For better HT, the enhancement technique should also be in the annular region of the DPHEs, this improves the HT rate in the annular space by minimizing the overall thermal resistance between the two fluids that vary in their temperatures [3].Studies that involve enhancement techniques implied in the annular region are mainly discussed here.

Al-Kayiem et al. [4] An experimental study investigated artificial roughness using ribbed energy promoters in the cold annulus of a DPHE. Rib pitch-to-height ratios (p/e) of 10 and 15, and height-to-diameter ratios (e/D) of 0.0595, 0.083, and 0.107 were tested across Re = 2900–21000. Increasing e/D raised the friction ratio by 28–36%. At p/e = 10 and e/D = 0.107, Stanton number increased by 4.2×, and by 3.3× at p/e = 15. Smaller ribs (e/D

= 0.0595) showed better hydrothermal performance, leading to a recommendation of e/Dh ≈ 0.06 ± 0.005. **Kadhim and Mohammed** [5] numerically analyzed the effect of corrugated surfaces of inner tubes on the HT rate. The study showed a noticeable improvement in the rate of HT in comparison to the plain tube. An improvement of 56% to 60% was registered for different temperatures of inner fluid. **Sharma et al**. [6] made a numerical investigation to explain the effect of HT rate by installing circular fins having a 9.8% cut installed on the outer surface of inner tube. The study showed that using a fined tube increases the temperature difference of the fluid in the annular region by 2.9 times compared to the plain DPHE. This improvement is due to the increase in the fluid mixing in the annular region as it flows on the fined surface. **Mansour et al.** [7] performed a numerical analysis for a DPHE equipped with serrated finned in the annular region. The study showed an increase in Nusselt number as of 61.9% when using fourteen serrated fins. The simulation showed that serrated fins create intense turbulence, which resulted in better mixing of the fluid located in the annular region. **Mozafarie et al.** [8] performed a CFD study on circular finned DPHEs with varying fin pitches, heights, and turbulent Re showed that adding fins disrupts flow paths and breaks thermal boundary layers, boosting HT by 1.45× over smooth tubes. Fin pitch had a greater impact than fin height, as it more effectively enhanced turbulence and HT. **Farahani et al.** [9] A numerical study evaluated natural convection between two cylinders with various fin designs (linear, annular, twisting, conical) in the annular region. Adding fins enhanced thermal conductivity by up to 67% compared to a plain tube, with more fins yielding greater improvement. **Al-Zahrani** [10] performed a numerical study on counter-flow DPHEs compared circular wavy, plain oval, and oval wavy inner tubes to enhance HT and reduce pumping energy. Oval tubes increased Nu by up to 28% over circular ones, while oval wavy tubes reduced the friction factor by 0.5 compared to circular wavy tubes. **Saleh et al.** [11] conducted a numerical simulation of shell and tube heat exchangers equipped with helical tubes. This heat exchanger is designed to recover the heat that is lost to the atmosphere in the brick factory to reuse the lost energy to heat the fuel prior to use in the combustion chamber. The study revealed that an increase in helical tube curvature resulted in higher Nu by 21%. **Mohsen et al.** [12] conducted an experimental study on DPHEs with rectangular, circular, and helical rib fins under varying turbulent Re showed overall HT enhancement compared to smooth tubes. Rectangular fins gave the highest improvement up to 168% at high Re but also had the highest friction. Circular fins had the lowest enhancement but the lowest friction, with a reduction of 37% at low Re and 0.44% at high Re. **Hussein and Hameed** [13] An experimental study on air-water heat exchangers with semi-circular perforated baffles (20, 25, and 30 mm holes) showed up to 80% increase in overall HT coefficient using the smallest perforation, due to enhanced turbulence and vortex formation.. **Praveenkumara et al.** [14] experimentally examined the effect of integrated fins on the HT performance of a DPHE having different pitch values. The experiment showed that there was a significant increase in the performance of the DPHE when using an integrated fin with a maximum value of 65% for a fin pitch of 1.5 mm in comparison to the plain tube. These fins create noticeable turbulence in the flow region, leading to better mixing of fluid particles and breaking of thermal boundary layers. **Saleh et al.** [15] performed an experimental study on stainless steel DPHEs with helical fins in the annular region showed enhanced HT due to increased swirling flow and fluid mixing. Compared to smooth tubes, the maximum improvements were 15% in Nu, and increases of 26%, 15%, and 13% in overall HT coefficient, HT rate, and effectiveness, respectively. **Kumar and Abraham** [16] outlined a numerical and experimental study examined the effect of circular ribs with varying diameters and pitch ratios in a DPHE's annular air flow. The ribs caused flow separation and reattachment, enhancing mixing and turbulence, which increased Nu by 2.5% compared to a smooth tube.

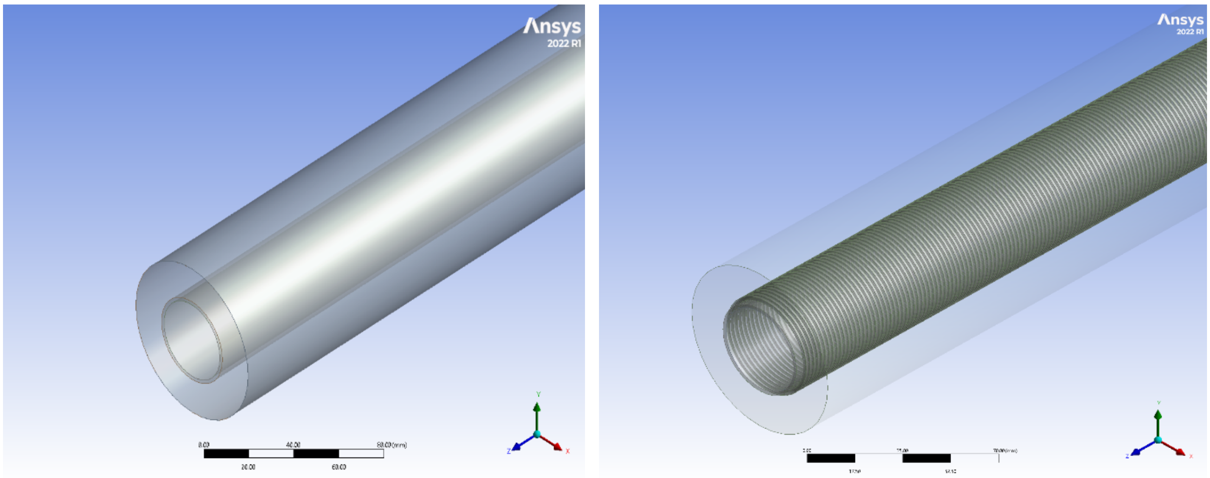
Due to their wide application, heat exchangers require enhanced thermal performance within cost and design limits. This study focuses on using integrated micro-circular fins to improve HT by disrupting the thermal boundary layer and promoting cold fluid mixing in the annulus. These enhancements are particularly critical in industries requiring high thermal efficiency, including power generation, chemical processing, oil refineries, and HVAC systems. Enhancing the design of the micro-integrated fins can substantially boost DPHE performance, resulting in greater energy efficiency and improved system operation.

2. Numerical methodology

2.1. Geometry

Two models of concentric DPHE's were Numerically modeled and analyzed using ANSYS FLUENT 2022 R1. As shown in Fig.1, The first model (a) has a smooth inner tube, while the second model (b) consists of an inner tube equipped with micro-integrated circular fins located on its outer surface. The inner pipe for both models is made of copper with an inner diameter D_{ii}=35 mm and outer diameter D_{io}= 38.1 mm and length of pipe

L=1000 mm. The spacing (S) of integrated circular fins on the outer surface of the inner pipe S = 1 mm and height h=1.5 mm. The outer pipe is made of PVC with an inner diameter D_{oi}=70.26 mm and outer diameter D_{oo}=74.66 mm and sharing the same length of the inner pipe. The outer pipe is considered adiabatic in the numerical solution. Therefore, no heat crosses from/to the outer pipe. A counterflow was achieved in this study, where cold water flows in the annular region, and the hot flow flows in the inner tube. These geometrical parameters listed in table 1 are used for both smooth and finned tubes.



(a) (b)

Figure 1. Schematic Views for The Two Models (a) Smooth DPHE and (b) Finned DPHE.

This research used several parameters, such as a mass flow rate ranging from 0.02 to 0.1 kg/s for both cold and hot water. The temperatures of hot water ranged from 50 °C to 70 °C, and cold water remained with a constant temperature of 25 °C. The smooth tube is designed to be the benchmark for heat performance comparison.

Table 1. Geometrical Parameters of DPHE.

Parameter	Outer Pipe	Inner Smooth Pipe	Inner Finned Pipe
Outer Diameter	74.66 mm	38.1 mm	74.66 mm
Inner Diameter	70.66 mm	35.0 mm	70.26 mm
Thickness	2.0 mm	1.55 mm	2.2 mm
Length	1000 mm	1000 mm	1000 mm
Distance Between Fins	NA	NA	1 mm
Fins Height	NA	NA	1.5 mm

2.2. Mesh generation and independence test with boundary condition of smooth and finned tubes and assumptions.

The smooth tube model was created in ANSYS FLUENT 2022 R1 with 3246739 elements and 804653 nodes. A grid independence test shown in Fig.2 confirmed this mesh size, with a 0.03% relative error. The unstructured Tetrahedral mesh had an average skewness of 0.2. Inlet conditions are shown in Table 2, and a pressure outlet was used. The SIMPLE scheme with second-order discretization was applied, using the RNG k-ε model (model constants $C_{1\varepsilon}$, $C_{2\varepsilon}$, C_{μ}, σ_k and σ_ε has the following default values: 1.44, 1.92, 0.09, 1.0 and 1.3, respectively [17]) with enhanced wall treatment. Convergence was achieved at 388 iterations with residuals below 10^{-3} as shown in Fig. 3.

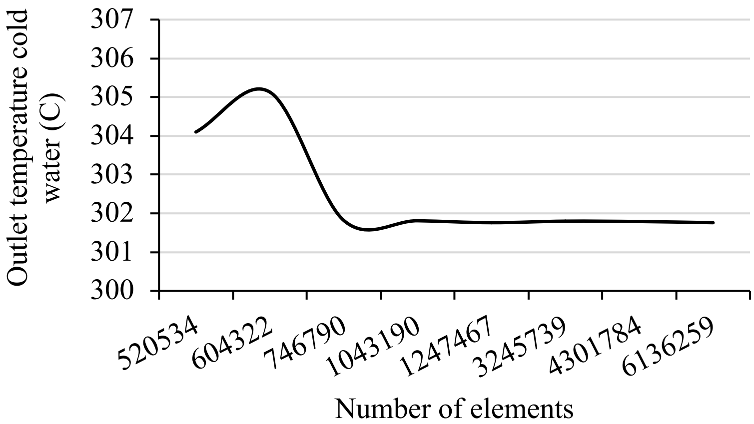


Figure 2. Grid Independence Test for Smooth DPHE With Respect to Outlet Temperature of Cold Water.

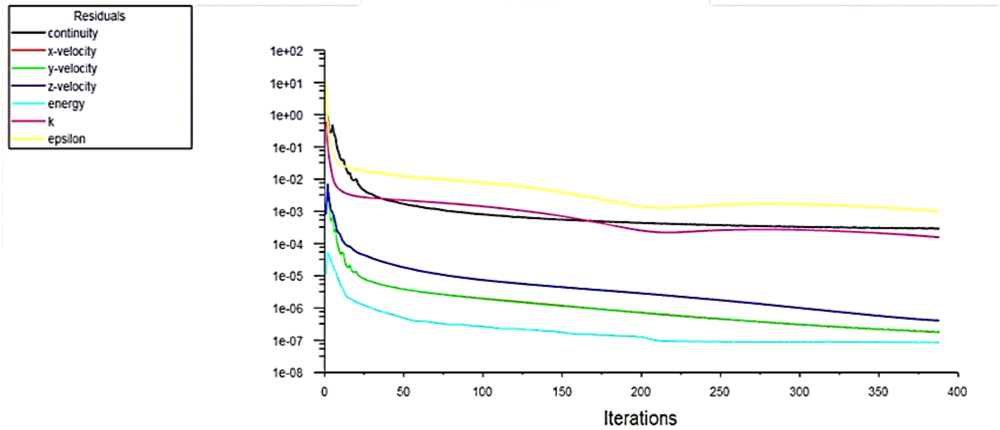


Figure 3. Convergence Simulation Plot for The Smooth DPHE.

Table 2. Boundary Condition for Smooth Tube Simulation.

Boundary condition	Parameters
Inlet - cold pipe For Smooth and Finned Tubes	Mass Flow Rate 0.02 – 0.1 kg/s Constant Temperature 25 °C
Inlet - hot pipe For Smooth and Finned Tubes	Mass Flow Rate 0.02 – 0.1 kg/s Temperatures 50, 60, 70 °C
Outlets (for all models)	Pressure Outlet
Walls (for all models)	Via System Coupling 0 W/m^2 (Adiabatic)

A concentric DPHE with circular fins was modeled in ANSYS FLUENT 2022 R1. The model used 3677164 elements and 797984 nodes, with grid independence verified in Fig. 4 and a maximum error of 0.03% achieved. Boundary conditions for this model matched those in Table 2. The energy model and RNG k-ε with enhanced wall treatment were used. Convergence was achieved at 550 iterations when residuals dropped below 10^{-4} as shown in Fig. 5. A schematic diagram showing the distribution of mesh network explained in Fig 6. A summarization of grid details listed in Table 3, for both smooth and finned tubes.

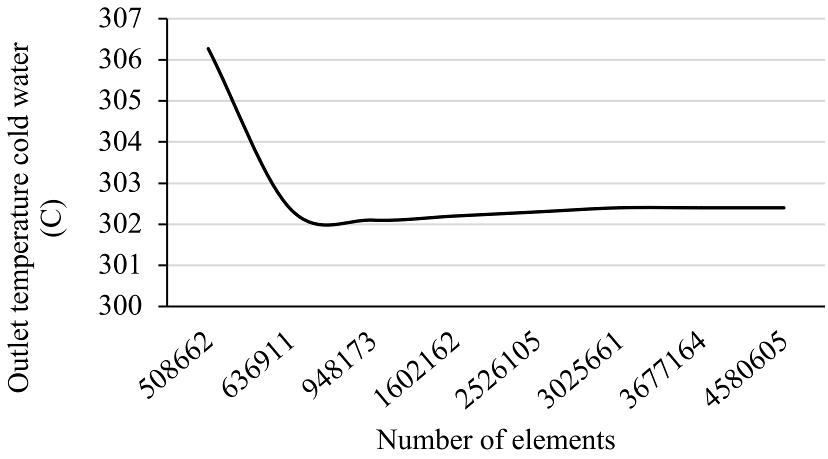


Figure 4. Grid Independence Test for Finned DPHE With Respect to Outlet Temperature of Cold Water.

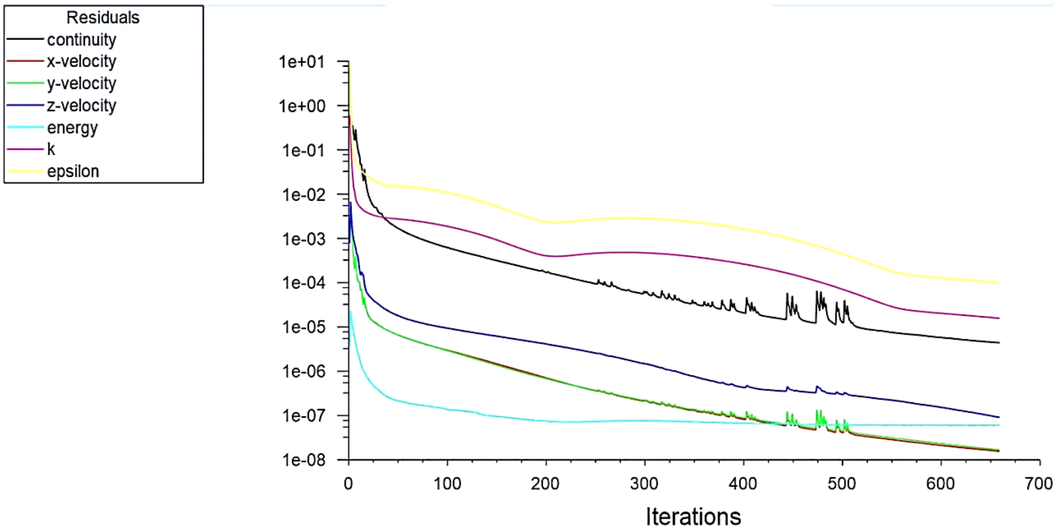
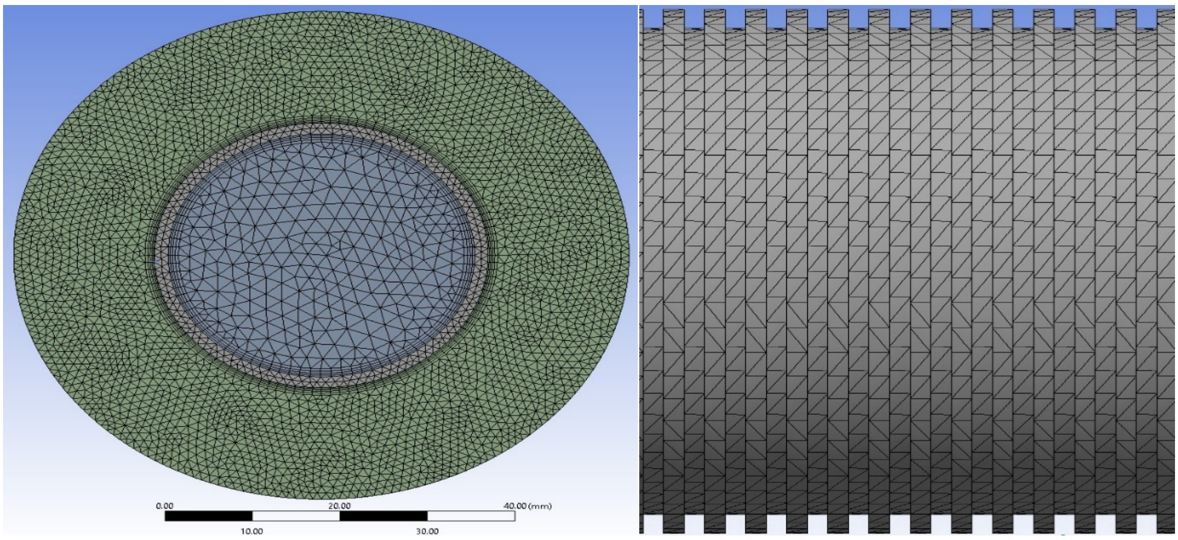


Figure 5. Sample Simulation Convergence Plot for the Finned DPHE.



(a) (b)

Figure 6. Finned Tube Schematic Diagram Showing The Distribution of Networks Mesh (a) Front View (b) Side View.

Both models are based on several assumptions during the solution process such as: Flow is incompressible, Steady state flow, Heat radiation neglected, Adiabatic wall condition, Newtonian flow, Single phase model, and constant physical properties with temperature.

Table 3. Mesh Details for Smooth and Finned Tubes.

	Smooth Tube	**Finned Tube**
Type of Mesh	Unstructured Tetrahedral Mesh	Unstructured Tetrahedral Mesh
Number of Iterations	388	550
Time Required for Each Case	29 mins	57 mins
Mesh Metric (Avg. Skewness)	0.197	0.266
Number of Nodes	804653	797984
Number of Elements	3246739	3677164

3. Data Reduction equations

In the design of any heat exchanger, the first law of thermodynamics must be upheld at both the macroscopic and microscopic levels. The total energy balance for a two-fluid heat exchanger is expressed in "Eq. (1)" as follows [18]:

$$Q_h = m_h C_{p,h}(T_{h,i} - T_{h,o}) \qquad 1$$

And from total energy balance ($Q_h = Q_c$), could be developed as in "Eq. (2)" as follows:

$$m_h C_{p,h}(T_{h,i} - T_{h,o}) = m_c C_{p,c}(T_{c,o} - T_{c,i}) \qquad 2$$

$m_h C_{p,h}$. fluid properties were calculated based on the average mean temperature (T_{avg}) given in "Eq. (3)" as follows:

$$T_{avg} = \frac{T_I + T_o}{2} \qquad 3$$

Nusselt Number (Nu) can be obtained for Re 600 - 3000 using "Eq. (4)" correlations [19]:

$$Nu = 4.36 + 5.36 \times 10^{-9} Re^{2.39} \qquad 4$$

Nu For turbulent flow can be calculated using Gnielinski (1976) given by "Eq. (5)" as follows [20]:

$$Nu = \frac{(f/8)(Re - 1000)\, Pr}{1 + 12.7(f/8)^{0.5}(pr^{2/3} - 1)} \qquad \left(\begin{matrix} 0.5 \le Pr \le 2000 \\ 3000 < Re < 5 \times 10^6 \end{matrix}\right) \qquad 5$$

Prandtl number (Pr) given by "Eq. (6)" as follows:

$$Pr = \frac{v}{\propto} \qquad 6$$

Reynold (Re) given by "Eq. (7)" as follows:

$$Re_D = \frac{4\dot{m}}{\pi D_h \mu} \qquad 7$$

Friction factor (f) given by "Eq. (8) and Eq. (9)" as follows:

$$f = 64/Re \ \ (Re < 2300) \qquad 8$$

$$f = 0.316\, Re^{-0.25} \ \ (Re > 2300) \qquad 9$$

The effectiveness (ε) for heat exchangers is given by "Eq. (10)" as follows:

$$\varepsilon = \frac{Q_{Act}}{Q_{max}} \qquad 10$$

Actual HT (Q_{Act}) is given by "Eq. (11)" as follows:

$$Q_{Act} = C_c(T_{c,o} - T_{c,i}) = C_h(T_{h,i} - T_{h,o}) \qquad 11$$

Where $C_c = \dot{m}_c c_{p,c}$ and $C_h = \dot{m}_h c_{p,h}$ are the heat capacity rates of cold and hot fluids, respectively.

Maximum HT (Q_{max}) rate is given by "Eq. (10)" as follows:

$$Q_{max} = C_{min}(T_{h,i} - T_{c,i}) \qquad 12$$

Where C_{min} is the smaller of C_c and C_h

The number of transfer units (NTU) can be calculated using "Eq. (12)" as follows:

$$NTU = A_s U / C_{min} \qquad 13$$

The total entropy generation of the heat exchanger ($S_{gen,total}$) is the sum of entropy generated by pressure drop ($S_{gen,f}$) and by HT ($S_{gen,ht}$) as per "Eq. (14)" [21]:

$$S_{gen,total} = S_{gen,ht} + S_{gen,f} \quad (14)$$

$S_{gen,total}$ The entropy generation by HT (at constant pressure) is given by "Eq. (15)" as follows:

$$S_{gen,ht} = m_h C_{p,h}\, ln(\frac{T_{h,o}}{T_{h,i}}) + m_c C_{p,c}\, ln(\frac{T_{c,o}}{T_{c,i}}) \quad (15)$$

While the entropy generation by pressure drop is given by "Eq. (16)" as follows:

$$S_{gen,f} = \frac{m_h \Delta P_{h,i}}{\rho_h T_{avg,h}} + \frac{m_c \Delta P_{c,o}}{\rho_c T_{avg,c}} \quad (16)$$

For calculating sources of irreversibility, Bejan number (*Be*) can be used, and it's given by "Eq. (17)":

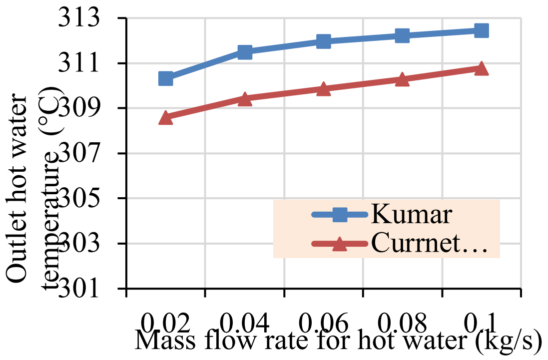
$$Be = \frac{S_{gen,ht}}{S_{gen,ht} + S_{gen,f}} \quad (17)$$

Bejan number is a dimensionless quantity ranging from 0-1. For Be ≈ 1 entropy dominated by HT, Be ≈ 0.5 has an equal contribution from both heat and friction and Be ≈ 0 entropy dominated by friction irreversibility.

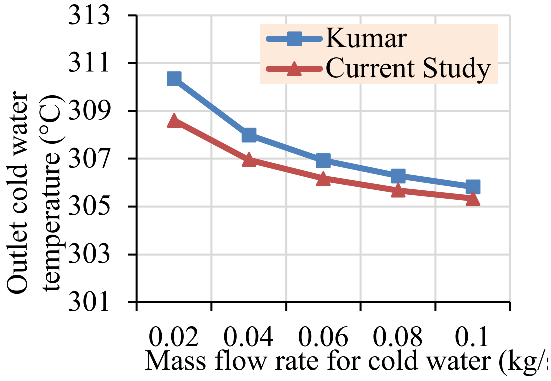
4. Result and Discussion

4.1. Validation work

To validate ANSYS FLUENT 2022 R1, results were compared with Kumar et al. [22] using flow rates from 0.02 to 0.1 kg/s. Inlet temperatures were 332 K (hot) and 303 K (cold), with tube dimensions matching the previous study. As shown in Fig. 7, the maximum relative error was 0.67%, confirming the software's accuracy.



(a)



(b)

Figure 7. Validation by Comparison of The Current Results With [22] Results. (a) At a Fixed Cold Water Flow Rate of 0.02 kg/s. (b) At Fixed Hot Water Flow Rate 0.02 kg/s

4.2. Impact of mass flow rate and inlet temperature of hot water on temperature difference

Fig. 8 compares cold-side temperature difference (ΔT_c) in finned and smooth DPHEs. Increasing hot water flow rate and inlet temperature (50 °C → 70 °C) raises ΔT_c due to greater thermal input and driving force. Finned tubes show stronger response, with a 40% ΔT_c increase from 50 °C to 60 °C and 29% from 60 °C to 70 °C, highlighting their effectiveness at higher temperature gradients.

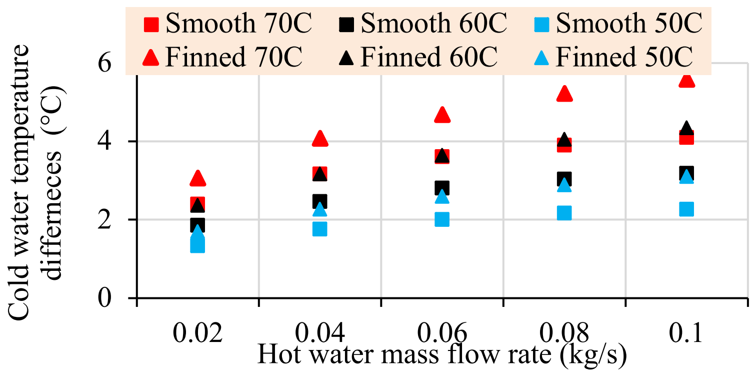


Figure 8. Simulation Results of Mass Flow Rate and Inlet Temperature of Hot Water Effect on Cold Water Temperature Difference at Cold Water Flow Rate 0.1 (kg/s) at 25 °C.

4.3. Impact of mass flow rate and inlet temperature of hot water on the heat transfer rate

Fig. 9 shows the effect of hot water temperature on heat transfer, comparing smooth and finned tubes under similar conditions. Hot water temperature varied from 50–70 °C, with cold water fixed at 25 °C and flow rates from 0.02 to 0.1 kg/s. Cold flow was kept constant at 0.1 kg/s for peak heat transfer. Increasing hot water temperature and flow rate boosts heat transfer for both tube types, due to a larger temperature difference, enhanced convective coefficients, and stronger fluid mixing at higher flow rates. Heat transfer rises 40% from 50 °C to 60 °C, and another 28% from 60 °C to 70 °C. At 70 °C for maximum flow, fins enhance heat transfer by 22% due to the increased surface area and turbulence that promote better cold water mixing and more efficient thermal exchange.

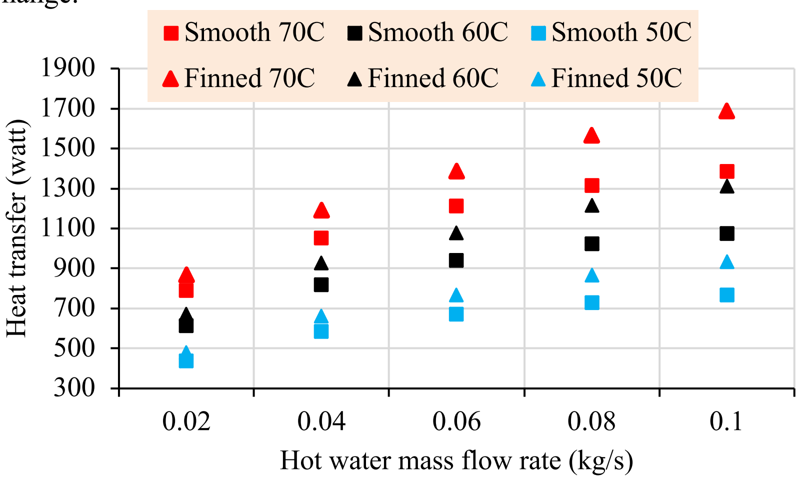
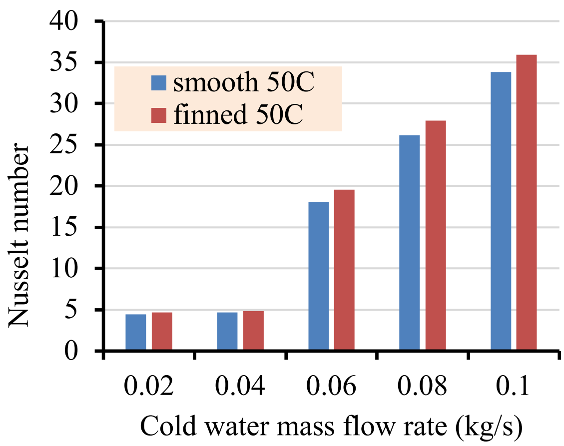


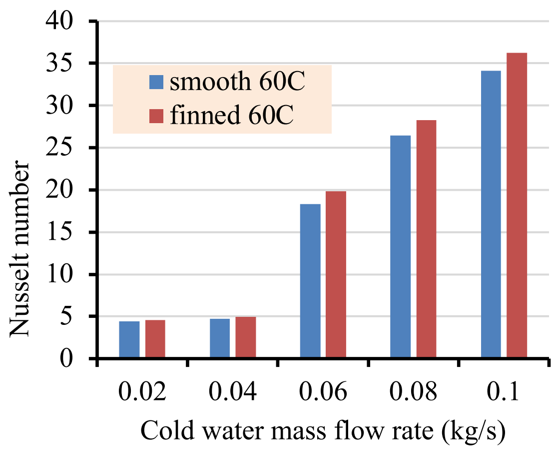
Figure 9. Simulation Results of Inlet Hot Water Temperature and Flow Rate Effect on Heat Transfer Rate at Cold Water Flow Rate 0.1 kg/s at 25 °C.

4.4. Nusselt number behavior in smooth and finned double-pipe heat exchangers

The Nusselt number (Nu) is crucial for the analysis of DPHE's, as it correlates with convective and conductive heat transfer. Fig. 10 shows that Nu for the smooth tube increases with cold fluid flow rate due to higher Re and enhanced forced convection. Thinner thermal boundary layers and greater turbulence improve heat transfer. Hot fluid temperature (50–70°C) has little effect, indicating flow dynamics dominate over temperature variations. The finned tube also shows rising Nu with increasing $\dot{m}_c$, consistently 6-8% higher than the smooth tube. This is due to added surface area and boundary layer disruption. As with the smooth tube, hot fluid temperature has little impact compared to flow velocity.



(a)



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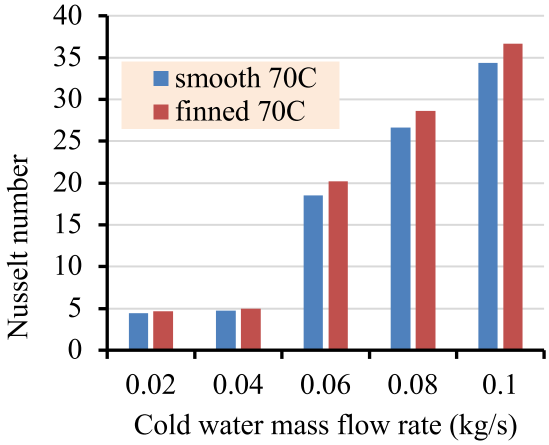

(c)

Figure 10. Simulation Results of Nusselt Number Behavior in Double-Pipe Heat Exchangers (a) at 50 °C, (b) at 60 °C, (c) at 70 °C.

4.5. Effectiveness-NTU behavior in smooth and finned double-pipe heat exchangers

The effectiveness-NTU method is widely used when outlet temperatures are unknown. As shown in Fig. 11, finned tubes show 10–20% higher effectiveness and 11–25% higher NTU than smooth tubes at 50°C, depending on flow rate. This is due to fins increasing surface area, boundary layer disruption and enhanced convective heat transfer provided by the fins which boost heat transfer.

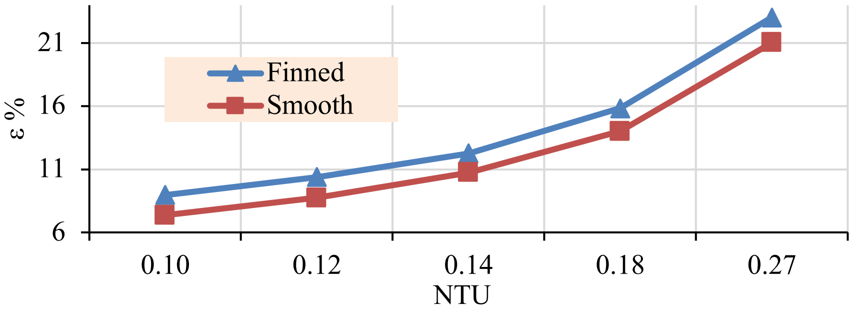

Figure 11. Simulation Results of Effectiveness-NTU Behavior in Smooth and Finned DPHE's.

4.6. Entropy generation in smooth and finned double-pipe heat exchangers

Entropy generation indicates exergy destruction and reflects HT irreversibility. It guides designers in optimizing system parameters by quantifying energy losses from inefficiencies.[23]. This study used entropy analysis to assess the impact of fins on a counter-flow DPHE. Entropy generation from HT and friction were evaluated. Fig. 12 shows that finned tubes produce 30–40% more heat transfer entropy than smooth ones, due to enhanced mixing and boundary layer disruption. Entropy peaks at 70°C and increases with mass flow rate, as higher flow enhances convective transfer and irreversibility.

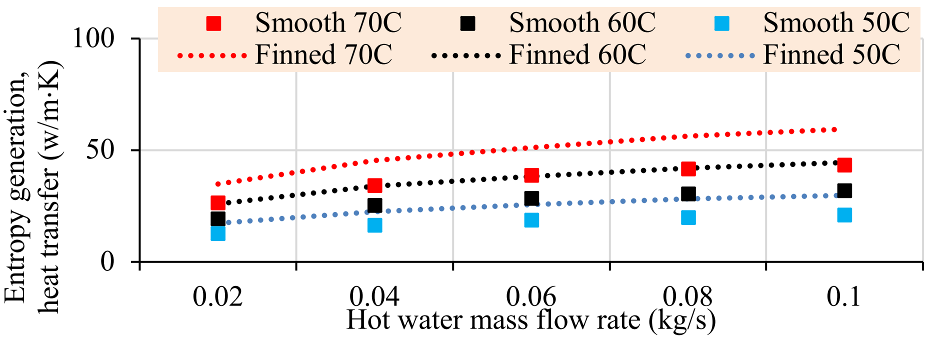

Figure 12. Simulation Results of Entropy Generation Due to Heat Transfer at Different Temperatures of Hot and Cold Waters with Constant Cold Water Flow Rate at 0.1 kg/s.

Fig. 13 shows that finned tubes generate 7–12% more frictional entropy than smooth ones. As hot water temperature rises, frictional entropy decreases due to lower viscosity, which reduces viscous dissipation. In some fluids, this trend may reverse. Fig. 13 confirms that increasing mass flow rate raises frictional irreversibility due to higher shear and turbulence.

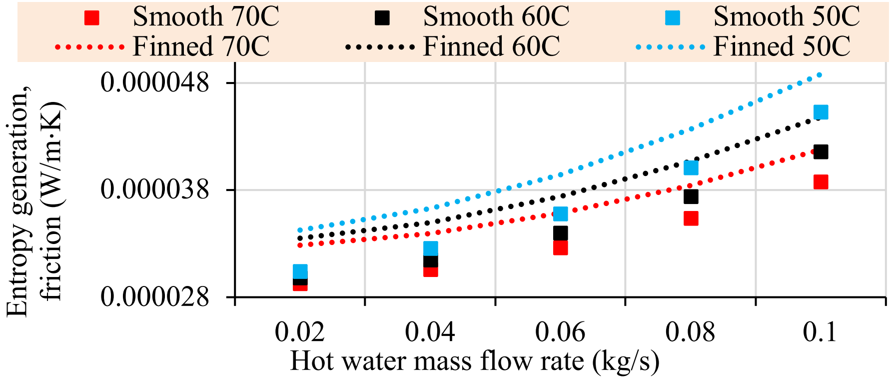

Figure 13. Simulation Results of Entropy Generation Due To Friction At Different Temperatures of Hot and Cold Waters with Constant Cold Water Flow Rate at 0.1 kg/s.

From Fig. 14, total entropy in the heat exchanger rises with flow rate and combines heat transfer and friction effects. Finned tubes generate 30–40% more entropy than smooth ones.

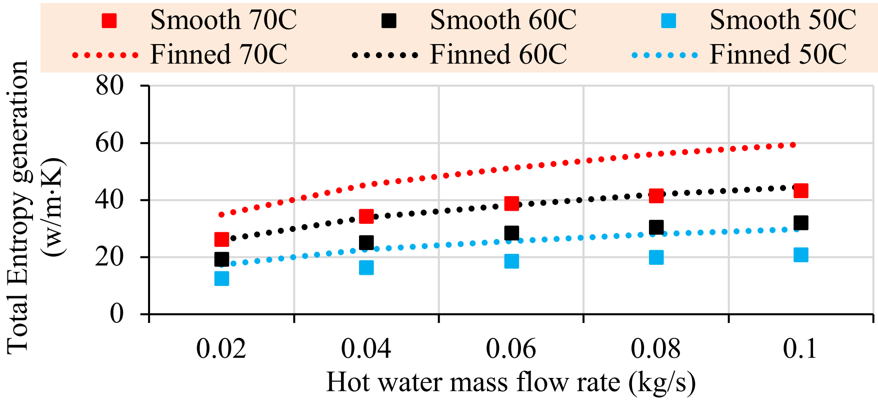

Figure 14. Total Entropy Generation at Different Temperatures of Hot Water and cold waters with constant cold water flow rate at 0.1 kg/s.

From Fig. 15, the Bejan number for both finned and smooth tubes peaks around 0.9, indicating that heat transfer irreversibility dominates over frictional losses. Finned tubes show slightly higher values due to enhanced heat transfer and reduced friction. As hot water flow increases, the *Be* rises, peaking at 0.06 kg/s, then slightly declines due to growing frictional irreversibility at higher flow rates.

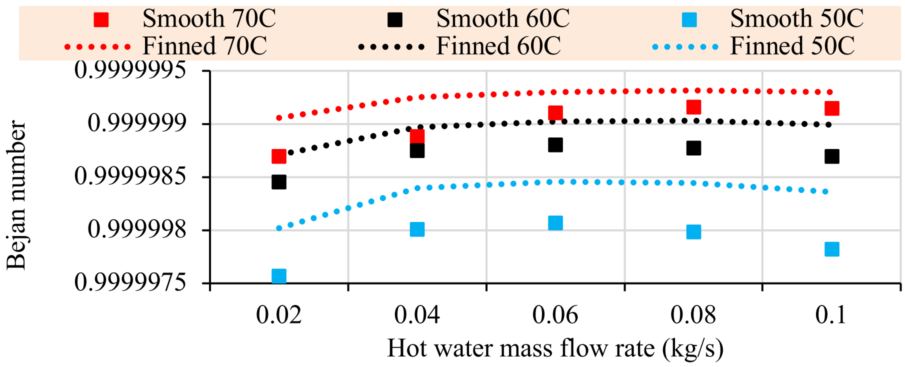

Figure 15. Bejan Number at Different Temperatures of Hot Water and Cold Waters with Constant Cold Water Flow Rate at 0.1 kg/s.

5. Conclusion

In this research, a CFD approach used to analyze the thermal performance and entropy generation of counter flow DPHE. This study closely examined how micro fins affect the heat transfer rate and entropy generation when installed on the outer surface of the inner pipe for a DPHE. The following conclusions can be drawn:

1. Finned tubes have a 22% enhanced heat transfer rate relative to smooth tubes, demonstrating their better thermal efficiency. The Finned tube outperforms the smooth tube by attaining a 46% higher Nu, hence illustrating its enhanced convective heat transfer efficiency.
2. Finned tubes deliver 10–20% higher effectiveness and 11–25% higher NTU than smooth tubes for a variation of flow rates.
3. Finned tube generates 30% to 40% higher total entropy compared to the smooth tube.
4. Bejan number analysis confirms that heat transfer irreversibility dominates over frictional losses in both finned and smooth tubes as its value reaching approximately 0.9. Finned tubes exhibit slightly higher Bejan number than smooth tube.

Declaration of Competing Interest

The authors declare that there are no conflicts of interest regarding the publication of this manuscript.

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Author Contributions

The study problem was proposed by Zena Khalefa Kadhim. Ali Arkan and Shatha Ali gathered up-to-date articles and arranged them in straightforward formats. Hussain H. Al-Kayiem verified the proposed work and contributed with research writing structure. The design, findings, and completed version of this work were all done by Ali Arkan Alwan. All the authors discussed the results and the final version of this paper.

References

[1] S. M. Mousavi, A. Sheikhi Azizi, M. Razbin, A. A. Rabienataj Darzi, and M. Li, “Optimized design of Helical-Finned Double Pipe heat exchangers via numerical simulation and Artificial Intelligence,” *Appl. Therm. Eng.*, vol. 258, p. 124605, Jan. 2025, doi: 10.1016/j.applthermaleng.2024.124605.

[2] H. H. Al-Kayiem, M. S. Kassim, and S. T. Taher, “Applications of Compound Nanotechnology and Twisted Inserts for Enhanced Heat Transfer,” in *Inverse Heat Conduction and Heat Exchangers*, S. Bhattacharya, M. Moghimi Ardekani, R. Biswas, and R. C. Mehta, Eds., IntechOpen, 2020. doi: 10.5772/intechopen.93359.

[3] M. S. Kumar and S. Abraham, “Experimental and numerical studies of detailed heat transfer and flow characteristics in the rib turbulated annulus of a double pipe heat exchanger,” *Int. J. Therm. Sci.*, vol. 207, p. 109382, Jan. 2025, doi: 10.1016/j.ijthermalsci.2024.109382.

[4] H. H. Al-Kayiem and M. F. El-Rahman, “RIBBED DOUBLE PIPE HEAT EXCHANGER: ANALYTICAL ANALYSIS,” *J. Eng. Sci. Technol.*, vol. 6, no. 2011, pp. 39–49, 2011.

[5] Z. K. Kadhim and S. A. Mohammad, “CFD Study to Enhance the Heat Transfer in Heat Exchanger by Change the Outer Surface of the Inner Tube and Use Nano Fluid”, doi: 10.11648/j.es.20170203.12.

[6] Ashish sharma, A. Tyagi, and A. Kumar, “Thermal Analysis of Double Pipe Heat Exchanger,” *Res. India Publ.*, vol. Volume 13, no. Number 6, pp. 299–302, 2018.

[7] A. J. Mansour, Z. K. Kadhim, and K. A. Hussein, “CFD study of Heat Transfer Characteristics for Annular Serrated Finned-Tube Heat Exchanger,” *Nabu Res. Acad.*, vol. 5, no. 1, p. pp 77-87, 2018.

[8] S. S. Mozafarie, K. Javaherdeh, and O. Ghanbari, “Numerical simulation of nanofluid turbulent flow in a double-pipe heat exchanger equipped with circular fins,” *J. Therm. Anal. Calorim.*, vol. 143, no. 6, pp. 4299–4311, Mar. 2021, doi: 10.1007/s10973-020-09364-w.

[9] S. D. Farahani, R. Sheikhi, and M. S. Kisomi, “Natural convection heat transfer in the annular space by using novel fins and water droplets injection,” *Braz. J. Chem. Eng.*, vol. 39, no. 2, pp. 441–454, June 2022, doi: 10.1007/s43153-021-00123-4.

[10] S. Al-Zahrani, "Heat transfer characteristics of innovative configurations of double pipe heat exchanger," *Heat Mass Transf.*, vol. 59, no. 9, pp. 1661–1675, Sept. 2023, doi: 10.1007/s00231-023-03360-0.

[11] Safaa A. Saleh, Zena K. Kadhim, and Kamil Abdulhusein Khalaf, "CFD simulation of helical coil heat exchanger with Different coil pitch to heating heavy fuel oil," *Wasit J. Eng. Sci.*, vol. 11, no. 3, pp. 120–139, Dec. 2023, doi: 10.31185/ejuow.Vol11.Iss3.472.

[12] O. A. Mohsen, M. A. R. Muhammed, and B. O. Hasan, "Heat Transfer Enhancement in a Double Pipe Heat Exchanger Using Different Fin Geometries in Turbulent Flow," *Iran. J. Sci. Technol. Trans. Mech. Eng.*, vol. 45, no. 2, pp. 461–471, June 2021, doi: 10.1007/s40997-020-00377-2.

[13] M. A. Hussein and V. M. Hameed, "Experimental Investigation on the Effect of Semi-circular Perforated Baffles with Semi-circular Fins on Air–Water Double Pipe Heat Exchanger," *Arab. J. Sci. Eng.*, vol. 47, no. 5, pp. 6115–6124, May 2022, doi: 10.1007/s13369-021-05869-0.

[14] P. B M, B. Sadashive Gowda, and N. H V, "Experimental Investigation of the Effect of Integrated Fins on Heat Transfer Rate of Double Pipe Heat Exchanger," *J. Mines Met. Fuels*, pp. 396–404, Mar. 2023, doi: 10.18311/jmmf/2022/32939.

[15] Safaa Abdul Adheem Saleh, Zena Khalefa Kadhim, and Kamil Abdulhusein Khalaf, "An Experimental Study of Heating Heavy Fuel Oil by Hot Air using Helical Fins in a Double-Pipe Heat Exchanger," *J. Adv. Res. Fluid Mech. Therm. Sci.*, vol. 112, no. 2, pp. 102–115, Jan. 2024, doi: 10.37934/arfmts.112.2.102115.

[16] M. S. Kumar and S. Abraham, "Experimental and numerical studies of detailed heat transfer and flow characteristics in the rib turbulated annulus of a double pipe heat exchanger," *Int. J. Therm. Sci.*, vol. 207, p. 109382, Jan. 2025, doi: 10.1016/j.ijthermalsci.2024.109382.

[17] *ANSYS Fluent Theory Guide*, 2021 R2. Canonsburg, PA, USA: ANSYS, Inc., 2021. [Online]. Available: https://www.ansys.com

[18] K. Thulukkanam, "Heat Exchanger Design Handbook, Second Edition," *CRC Press Taylor Francis Group*, p. 1186, 2013.

[19] A. I. Bashir, M. Everts, R. Bennacer, and J. P. Meyer, "Single-phase forced convection heat transfer and pressure drop in circular tubes in the laminar and transitional flow regimes," *Exp. Therm. Fluid Sci.*, vol. 109, p. 109891, Dec. 2019, doi: 10.1016/j.expthermflusci.2019.109891.

[20] Y. A. Çengel and A. J. Ghajar, *Heat and mass transfer: fundamentals & applications*, 5th edition. New York, NY: McGraw-Hill Education, 2015.

[21] G. S. Dhumal and S. N. Havaldar, "Experimental investigation of entropy generation in a pipe heat exchanger with turbulence generator inside and on the outside," *Case Stud. Therm. Eng.*, vol. 51, p. 103464, Nov. 2023, doi: 10.1016/j.csite.2023.103464.

[22] S. Kumar, K. V. Karanth, and K. Murthy, "Numerical study of heat transfer in a finned double pipe heat exchanger".

[23] L. N. Thanh, L. M. Nhut, and A.-Q. Hoang, "The heat transfer and entropy generation of fin and inclined flat tube heat exchanger," *Case Stud. Therm. Eng.*, vol. 56, p. 104202, Apr. 2024, doi: 10.1016/j.csite.2024.104202.