

Experimental Investigation of Heat Transfer Enhancement in Double-Pipe Heat Exchangers Using Al₂O₃/Water Nanofluid and Segmental or Flower Baffles

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Abstract: This study investigates the hydrothermal performance of a horizontal double-pipe heat exchanger with a counter-flow arrangement. The system incorporates flower baffles at a relative angle of $\gamma = 90^\circ$ and segmental baffles, while Al₂O₃/water nanofluid is used in the annular side. The experiments evaluate the effects of nanoparticle concentration (0–1.5% by volume), annulus Reynolds number ($2620 \leq Re_{an} \leq 13110$), and Prandtl number ($3.91 \leq Pr_{an} \leq 7.48$) under steady operating conditions. Hot water at 70°C is maintained at a constant flow rate in the inner tube, while cold water circulates in the annular region. Key findings demonstrate significant enhancements in the Nusselt number (Nu_{an}) and friction factor (f_{an}) with the addition of baffles and Al₂O₃ nanoparticles. Flower baffles outperformed segmental baffles, achieving increases in Nu_{an} and f_{an} by 343.1% and 190.3%, respectively, at a nanoparticle concentration of 1.5%. The Heat Transfer Performance Index (HTPI) consistently exceeded unity across all test parameters, with the highest HTPI of 3.24 observed for flower baffles with $\delta = 16.7\%$ and $\lambda = 8.3\%$. Furthermore, empirical correlations for Nu_{an} , f_{an} , and HTPI are proposed. The results highlight the combined potential of nanoparticle-enhanced fluids and innovative baffle geometries to improve heat exchanger performance, offering valuable insights for industrial applications.

Keywords : Double-pipe heat exchanger; Flower baffles; Al₂O₃ nanofluid; Heat Transfer Performance Index; Compound heat transfer enhancement.

1. INTRODUCTION

Double Pipe Heat Exchangers (DPHEs) are extensively utilized in various industries for effective heat transfer due to their superior thermal efficiency. They consist of two concentric pipes, where the inner and outer fluids flow in opposite directions, allowing heat exchange between the hot and cold fluids. These exchangers are especially favored for their ability to handle high-temperature and high-pressure applications, owing to their compact design. Although they may occupy more physical space compared to some other heat exchanger types, their simplicity and reliability make them an enduring choice for engineers seeking an optimal balance between cost efficiency and thermal effectiveness [1]. To address growing energy demands, extensive research

has focused on optimizing heat exchangers through innovative designs and materials. In particular, passive heat transfer techniques, such as the use of fins, baffles, and nanofluids, have been identified for their ability to enhance thermal performance without increasing operating costs [2–5]. The main challenge with traditional heat exchangers is the use of fluids with low thermal conductivity, limiting the heat transfer rates. Introducing nanofluids suspensions of nanoparticles (sized between 1 and 100 nm) within base fluids enhances thermophysical properties, leading to improved heat transfer performance. This advancement enables the design of more efficient and compact heat exchangers [6]. Baffle-walled channel heat exchangers have emerged as another promising design. Baffles help enhance

the turbulence of the fluid flow, improving heat transfer. However, the inclusion of baffles increases pressure drops and can lead to thermal stress at the baffle root. Research has focused on optimizing baffle geometry to balance heat transfer improvement and pressure drop minimization [7-10]. Studies by Shahmohammadi and Jafari [11] demonstrated that nanofluids outperform baffled tubes in terms of heat transfer within a Reynolds number range of 600 to 2100. Similarly, Bahiraei et al. [12] used numerical evaluations to examine water/alumina nanofluid in heat exchangers with helical baffles. Their findings suggested that higher nanoparticle concentrations and smaller helix angles could optimize heat transfer. Other studies, such as those by Shahmohammadi and Beiki [13], Akyürek et al. [14], and Ahmed and Mohammed [15], also highlighted the positive effects of nanofluids on heat transfer, particularly in systems equipped with different baffle configurations. Several studies have investigated the impact of baffles and nanofluids on heat exchanger performance under various conditions. Magalhães et al. [16] formulated a differential transient model for a helical shell-and-tube heat exchanger to analyze temperature distribution, heat transfer, and pressure losses. Utilizing Python and SiO_2 nanofluid, their study aimed to address a gap in transient differential analysis. Ghobadi et al. [17] found that using rectangular baffles significantly improved heat transfer, while Alazwari and Safaei [18] showed that increasing nanoparticle concentrations and Reynolds numbers positively affected heat exchange rates. Phu et al. [19] explored the effect of baffle height on nanofluid-based heat exchangers, highlighting its considerable influence on pressure loss. Meanwhile, research by Bahiraei et al. [20] and Javaherdeh et al. [21] revealed that nanofluids, particularly with platelet-shaped nanoparticles, showed higher heat transfer rates than conventional fluids, especially when combined with baffles. The studies of Bellahcene et al. [22] and Arani [23] corroborated these findings, showing that higher nanoparticle concentrations and optimal baffle configurations enhance heat transfer and minimize entropy generation. Baffle geometry and the type of nanofluid used also affect thermal performance. Researchers have explored the effect of different baffle configurations, including twisted helical baffles [24], as well as the impact of nanoparticle shape on heat transfer efficiency.

To further refine the design of heat exchangers, researchers have examined various nanoparticle shapes, such as platelet-shaped nanoparticles, which are known to significantly improve thermal efficiency compared to other shapes. The integration of nanofluids with baffle systems has become a key focus for optimizing heat exchanger performance [25-32]. Gugulothu and Sanke [26] found that using Al_2O_3 -water nanofluid in combination with helical baffles improved heat transfer coefficients. Other studies, such as those by Sameer et al. [28] and Almulla et al. [29], demonstrated the advantages of combining baffles, nanofluids, and semicircular tubes for enhanced thermal performance. Recent investigations have also focused on the use of novel baffle designs, such as flower-shaped baffles, which show significant improvements in thermal performance compared to traditional designs. The unique geometry of flower baffles,

with their petal-like structure, enhances fluid mixing and heat transfer [5, 33-35]. Mahendran [33] explored the impact of flower baffles and found that they improve thermal behavior compared to helical baffles. Vivekanandan et al. [34] also studied the performance of heat exchangers with helical wire inserts and flower-patterned baffles, showing that the combination of these designs results in better heat transfer rates. Chen et al. [35] experimented with different baffle configurations and found that triple-layer flower baffles offered the best thermal performance. Furthermore, Salem et al. [5] conducted a comparative experimental study between segmental and flower baffles, concluding that flower baffles outperformed segmental ones under all operating conditions in the annular side.

Researchers are actively exploring ways to improve the thermal performance of heat exchangers due to increasing industrial demands and rising energy costs. While individual enhancement techniques may not be sufficient on their own, combining multiple methods has proven effective. Baffles have been shown to play a key role in improving heat exchange performance, and the use of nanoparticles in the annulus side further enhances heat transfer. The concentration of nanoparticles has a substantial impact on the performance of nanofluids. However, the combined effect of floral-designed baffles and nanofluids has not been extensively studied. This research aims to experimentally analyze the thermal performance of a horizontal counterflow double-pipe heat exchanger with segmental or flower baffles and Al_2O_3 /water nanofluid under various operating conditions and nanoparticle concentrations.

2 EXPERIMENTAL SETUP

2.1 Experimental Apparatus Specifications

The experimental setup for this study features a heat exchange system, as depicted in Salem et al. [5], specifically designed for accurate temperature regulation. It comprises two interconnected circuits: a heating loop and a cooling loop. The heating circuit incorporates key components such as a heating unit, a centrifugal pump, ball valves, an internal copper tube, and a rotameter. Similarly, the cooling circuit consists of a centrifugal pump, ball valves, the outer pipe (annular section), and an additional rotameter. Figures 1 and 2 provide a visual representation of both circuits and the complete system. To achieve the desired water temperature (70°C), the heating tank employs two 5 kW electric heaters. Simultaneously, the cooling tank integrates vapor compression refrigeration systems with a total capacity of 10.5 kW to cool the nanofluid to the desired temperature. Digital thermostats regulate and maintain stable temperatures for the fluids circulating through the heat exchanger. Each tank is equipped with top and bottom ports: the top ports receive the working fluid from both the heat exchanger and the bypass line, while the bottom ports direct the liquids toward the heat exchanger and the drain line. Additionally, two centrifugal pumps, each rated at 1.5 hp, facilitate fluid circulation within their respective loops. The annular main line is linked to a differential pressure transducer via PVC and flexible nylon connections.

The volume flow rates of both hot and cold fluids within the heat exchanger were quantified using two calibrated rotameters (1.7–18 L/min) with an accuracy of $\pm 5\%$. Temperature measurements at the inlet and outlet of the fluids in both the annular region and the inner tube were obtained using four K-type thermocouples, positioned approximately 5 cm from the heat exchanger ports. These temperature readings were displayed on a digital thermometer with a resolution of 0.1°C . Furthermore, a calibrated digital transducer, operating within a pressure range of 0–103.4 kPa and an accuracy of $\pm 1\%$, was employed to determine the differential pressure across the annulus. Three concentric tube heat exchangers with counter-flow configurations were designed and fabricated, including one without baffles, one incorporating conventional segmental baffles, and another utilizing flower baffles. The baffles were configured with a fixed pitch of 150 mm, consisting of 12 baffles, and featured cut ratios of 8.3% and 16.7%, respectively. To optimize performance, flower baffles were arranged at a relative angle of 90° , as recommended by Salem et al. [5], with pure water serving as the working fluid in the annular section.

In the experimental setup, the baffles were fabricated from 2 mm-thick acrylic sheets and precisely cut using a laser to form circular shapes that align with the diameter of the heat exchanger's annular pipe. This annular pipe comprises a seamless black steel tube with an internal diameter of 56 mm and a wall thickness of 4 mm, with PVC caps sealing its ends. Each heat exchanger incorporates an internal copper tube with an effective length of 1800 mm, featuring external and internal diameters of 22 mm and 20 mm, respectively. To minimize thermal losses, the exterior of the annular pipe is insulated with multiple layers of glass wool, asbestos rope, and ceramic fiber. Figure 3 illustrates the cross-sectional and 3D representations of the baffle arrangement within the annulus. The cut-off ratio (δ), which is a critical parameter, is determined using Eq. (1) based on the total unoccupied area (A_{cut}) in each baffle, calculated via SolidWorks. Additionally, the baffle pitch ratio (λ) is defined by Eq. (2) as the ratio of the baffle pitch (p_{baffle}) to the tube length (L_t). The relative angle (γ) between consecutive baffles, as depicted in Fig. 4, also influences the heat exchanger's performance. The impact of these parameters on heat transfer efficiency in the annular section with pure water flow has been previously analyzed by Salem et al. [5].

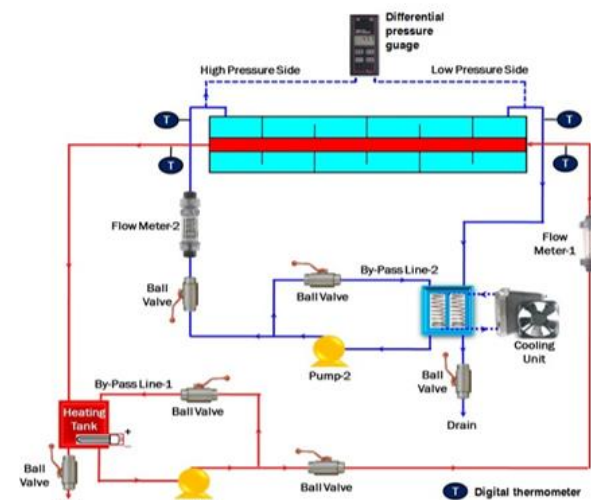


FIG 1. Layout of the setup.



FIG 2. A snapshot of the existing setup.

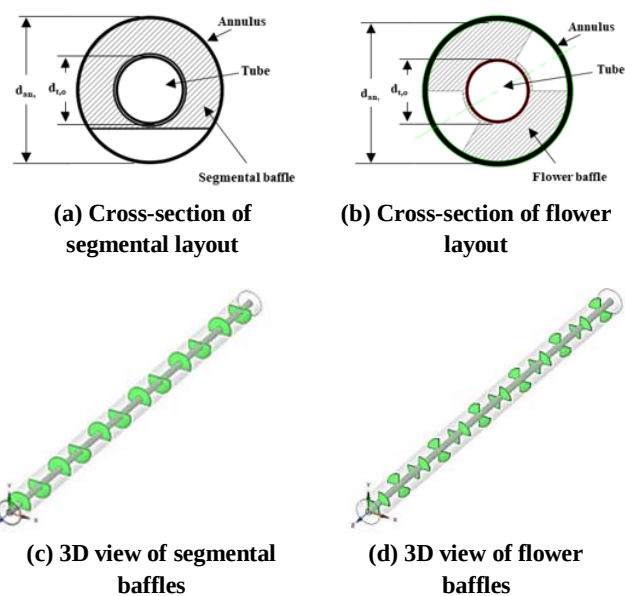


FIG 3. Demonstration of the baffles

$$\delta = \frac{4A_{\text{cut}}}{\pi d_{\text{an}}^2} \quad (1)$$

$$\lambda = \frac{p_{\text{baffle}}}{L_t} \quad (2)$$

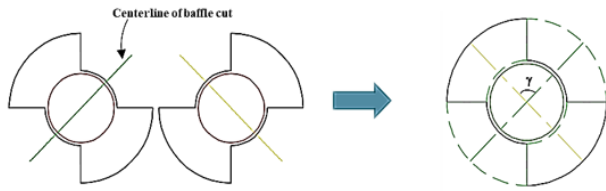


FIG 4. Illustration of the relative angle.

2.2 Nanofluid Preparation

In the context of this study, the preparation of a stable nanoparticle suspension within the base liquid serves as the initial step in utilizing nanofluids as heat transfer working fluids. Specifically, alumina nanopowder supplied by Inframat® Advanced Materials Corporation, USA, are employed. These nanoparticles exhibit key characteristics and relevant thermophysical properties, which are detailed in Table 1. The focus lies on an alumina/water nanofluid, prepared with varying volume concentrations of nanoparticles (ϕ): 0.25%, 0.5%, 0.75%, 1%, 1.25%, and 1.5%. Achieving uniform dispersion and stable suspension is critical. To this end, the designed concentration of alumina is mixed in a measuring flask. Subsequently, an agitator bath operating at 3450 rpm runs continuously for 36 hours over four sequential days, ensuring thorough dispersion. Additionally, ultrasonic vibration techniques maintain homogeneity. The mixture undergoes 5 hours of ultrasonic vibration at room temperature using an ultrasonic mixing system (with 420 W ultrasonic pulses at 50 ± 5 kHz). Furthermore, a subsequent 3-hour ultrasonic vibration occurs at 85°C . Remarkably, even after 45 days of static conditions, no noticeable settlement of nanopowder is observed by visual

inspection. The volume concentration (ϕ) of nanoparticles is typically calculated as:

$$\phi = \frac{V_{\text{np}}}{V_{\text{total}}} = \frac{V_{\text{np}}}{V_{\text{np}} + V_{\text{bf}}} = \frac{\frac{m_{\text{np}}}{\rho_{\text{np}}}}{\frac{m_{\text{np}}}{\rho_{\text{np}}} + \frac{m_{\text{bf}}}{\rho_{\text{bf}}}} \quad (3)$$

TABLE 1. Characteristics of alumina nanoparticles.

Purity	99.9%
Average particle size (nm)	40
Specific surface area (m^2/g)	> 200
Thermal conductivity ($\text{W}/\text{m} \cdot ^\circ\text{C}$)	36
Density (kg/m^3)	3600
Specific heat ($\text{J}/\text{kg} \cdot ^\circ\text{C}$)	773

3 EXPERIMENTAL PROCEDURES

In this meticulously designed study, strategically positioned thermocouples monitor temperature variations at the annulus inlet, exit, and tube sides. The experimental setup comprises essential components: a double-pipe heat exchanger, pumps, pipelines, heating and cooling units, flow meters, and a differential pressure transducer. Meticulously, the heating and cooling tanks are filled with water and a precisely prepared $\gamma\text{-Al}_2\text{O}_3/\text{water}$ nanofluid, respectively. Upon activation, the pumps, cooler, and heater orchestrate a comprehensive series of 378 experiments across three distinct heat exchangers. Steady-state conditions prevail when nanofluid intake and exit temperatures exhibit minimal fluctuations; less than 0.1°C . These meticulously recorded values serve as crucial data for subsequent analysis. For reference, Table 2 succinctly outlines the operational conditions.

TABLE 2. Range of the working circumstances during the experiments.

Parameters	Range or value
Annulus-side flow rate, L/min	4.1–16.3 ($2620 \leq \text{Re}_{\text{an}} \leq 13110$)
Annulus-side inlet temperature, $^\circ\text{C}$	15, 20, 25 ($3.91 \leq \text{Pr}_{\text{an}} \leq 7.48$)
Tube-side flow rate, L/min	8.1 ($\text{Re}_t \cong 19300$)
Tube-side inlet temperature, $^\circ\text{C}$	70 ($\text{Pr}_t \cong 2.76$)
Al_2O_3 volume concentration, %	0, 0.25, 0.5, 0.75, 1, 1.25, 1.5

4 CALCULATION METHODOLOGY

In the present analysis, the thermophysical properties of the cooling and heating fluids in the annulus and the tube are calculated at the mean temperatures, $T_{\text{an,m}}$ and $T_{\text{t,m}}$, respectively. Besides, the thermophysical properties of alumina/water nanofluid are assessed using Eqs. (6) to (16) [36–40].

$$T_{\text{an,m}} = \frac{T_{\text{an,o}} + T_{\text{an,i}}}{2} \quad (4)$$

$$T_{\text{t,m}} = \frac{T_{\text{t,o}} + T_{\text{t,i}}}{2} \quad (5)$$

$$k_{\text{nf}} = k_{\text{static}} + k_{\text{Brownian}} \quad (6)$$

$$k_{\text{static}} = k_{\text{bf}} \left[\frac{k_{\text{np}} + 2k_{\text{bf}} + 2\phi(k_{\text{np}} - k_{\text{bf}})}{k_{\text{np}} + 2k_{\text{bf}} - \phi(k_{\text{np}} - k_{\text{bf}})} \right] \quad (7)$$

$$k_{\text{Brownian}} = 5 * 10^4 \Gamma \xi \varphi \rho_{\text{bf}} C_{\text{pbf}} \sqrt{\frac{\kappa T}{\rho_{\text{np}} d_{\text{np}}}} \quad (8)$$

$$\Gamma = (1722.3\varphi - 134.63) - (6.04\varphi - 0.4705)T \quad (9)$$

$$\xi = 0.0017(100\varphi)^{-0.0841} \quad (10)$$

$$\mu_{\text{nf}} = \mu_{\text{static}} + \mu_{\text{Brownian}} \quad (11)$$

$$\mu_{\text{static}} = \frac{\mu_{\text{bf}}}{[1 - \varphi]^{2.5}} \quad (12)$$

$$\mu_{\text{Brownian}} = 5 * 10^4 \xi \Gamma \varphi \rho_{\text{bf}} \sqrt{\frac{\kappa T}{\rho_{\text{np}} d_{\text{np}}}} \quad (13)$$

$$\rho_{\text{nf}} = \varphi \rho_{\text{np}} + (1 - \varphi) \rho_{\text{bf}} \quad (14)$$

$$C_{\text{pnf}} = \frac{\varphi(\rho_{\text{np}} C_{\text{pnp}}) + (1 - \varphi)(\rho_{\text{bf}} C_{\text{pbf}})}{\rho_{\text{nf}}} \quad (15)$$

$$\text{Pr}_{\text{nf}} = \frac{\mu_{\text{nf}} C_{\text{pnf}}}{k_{\text{nf}}} \quad (16)$$

In the realm of heat transfer calculations, six fundamental variables are pivotal: the flow rates and temperatures at both the inlet and outlet of the heat exchanger. Eqs. (17) and (18) are utilized to calculate the heat transfer rates, designated as Q_t and Q_{an} , on the tube and annulus sides, respectively. In an ideal scenario where measurements are flawless and there are no heat gains or losses, the energy balance would equate these two rates ($Q_t = Q_{\text{an}}$). However, practical experiments often reveal minor discrepancies between them. As a result, the average value, Q_{ave} , is used for the calculations. Notably, the heating and cooling loads on the hot and cold sides differ by no more than $\pm 3.9\%$ across all experimental trials. This average heat transfer rate is then employed to determine the overall thermal conductance ($U_i A_{t,i}$).

$$Q_t = \dot{m}_t C_{\text{pt}} (T_{t,i} - T_{t,o}) \quad (17)$$

$$Q_{\text{an}} = \dot{m}_{\text{an}} C_{\text{pan}} (T_{\text{an},o} - T_{\text{an},i}) \quad (18)$$

$$Q_{\text{ave}} = \frac{|Q_t| + |Q_{\text{an}}|}{2} \quad (19)$$

$$\Delta Q_{\text{ave}}(\%) = \frac{|Q_t| - |Q_{\text{an}}|}{Q_{\text{ave}}} * 100 \quad (20)$$

$$U_i A_{t,i} = \frac{Q_{\text{ave}}}{\Delta T_{\text{LM}}} \quad (21)$$

$$A_{t,i} = \pi d_{t,i} L_t \quad (22)$$

$$\Delta T_{\text{LM}} = \frac{(T_{t,i} - T_{\text{an},o}) - (T_{t,o} - T_{\text{an},i})}{\ln \left[\frac{T_{t,i} - T_{\text{an},o}}{T_{t,o} - T_{\text{an},i}} \right]} \quad (23)$$

The tube's length 1800 mm is 90 times its hydraulic diameter ($d_{t,i} = 20$ mm), indicating that the water flow within the tube is fully developed. Utilizing the Dittus-Boelter correlation [41], as expressed in Eq. (24), the average Nusselt number on the tube side ($\overline{\text{Nu}}_t$) is determined. This correlation is applicable for $L_t/d_t \geq 60$, $0.7 \leq \text{Pr}_t \leq 100$, and $\text{Re}_t \geq 10000$. Subsequently, Eq. (25) is utilized to estimate the heat transfer coefficient on the annular side. Furthermore, the average Nusselt number and the Fanning friction factor for

the annulus side can be calculated using Eqs. (26) and (31), respectively.

$$\overline{\text{Nu}}_t = 0.023 * \text{Re}_t^{0.8} * \text{Pr}_t^{0.4} \quad (24)$$

$$\bar{h}_t = \frac{\overline{\text{Nu}}_t k_t}{d_{t,i}} \quad (25)$$

$$\frac{1}{U_i A_{t,i}} = \frac{1}{\bar{h}_{\text{an}} A_{t,o}} + \frac{1}{\bar{h}_t A_{t,i}} \quad (26)$$

$$\overline{\text{Nu}}_{\text{an}} = \frac{\bar{h}_{\text{an}} d_{\text{an},h}}{k_{\text{an}}} \quad (27)$$

$$d_{\text{an},h} = d_{\text{an},i} - d_{t,o} \quad (28)$$

$$\text{Re}_t = \frac{4\dot{m}_t}{\pi d_{t,i} \mu_t} \quad (29)$$

$$\text{Re}_{\text{an}} = \frac{4\dot{m}_{\text{an}}}{\pi d_{\text{an},h} \mu_{\text{an}}} \quad (30)$$

$$f_{\text{an}} = \frac{\Delta P_{\text{an}} \pi^2 \rho_{\text{an}} d_{\text{an},h}^5}{32 L_{\text{an}} \dot{m}_{\text{an}}^2} \quad (31)$$

5 UNCERTAINTY ANALYSIS

This study utilizes laboratory data and physical characteristics to assess the experimental findings. To accurately calculate the uncertainty factor, it is essential to consider measurement errors. Achieving precision in sensor reading involves adhering to manufacturer standards and calibration procedures. The internal tube diameter has a manufacturer's margin of error of ± 0.05 mm, while the uncertainty for other specified dimensions is ± 0.5 mm. Additionally, a tolerance of $\pm 0.1\%$ is applied to the thermophysical properties of the water. Table 3 outlines the maximum uncertainties in dominant parameters assessed using method of Kline and McClintock [42] while Appendix A introduces further details.

TABLE 3. Maximum uncertainties.

Parameter	Uncertainty (%)
Annulus-side Reynolds number	± 1.72
Tube-side Reynolds number	± 1.91
Annulus-side average Nusselt number	± 3.38
Annulus-side average heat transfer coefficient	± 3.12
Tube-side average Nusselt number	± 1.35
Tube-side average heat transfer coefficient	± 1.71
Overall heat transfer coefficient	± 4.51
Annulus-side Fanning friction factor	± 5.72
Hydrothermal performance index	± 5.97

6 APPARATUS VALIDATION AND DATA VERIFICATION

To validate the accuracy of the heat transfer and friction factor measurements, the flow and thermal parameters in the annular region are continuously observed. The average Nusselt number on the annulus side, denoted as $\overline{Nu}_{an}$, and the friction factor, f_{an} , are evaluated against well-known empirical relationships, as detailed in Table 4. Specifically, $\overline{Nu}_{an}$ is benchmarked against the equation provided by Gnielinski [43], Eq. (32), and f_{an} is assessed using the Filonenko's correlation [44], Eq. (33). These assessments

yielded congruent outcomes for both $\overline{Nu}_{an}$ and f_{an} , exhibiting maximum deviations of 7.2% and 5.5%, respectively, as depicted in Fig. 5. Such findings affirm the reliability of the experimental setup, measurement protocols, and computational procedures.

$$\overline{Nu}_{an} = \frac{f_{an} (Re_{an} - 1000) Pr_{an}}{1 + 12.7 \sqrt{f_{an}} (Pr_{an}^{2/3} - 1)} \left[1 + \left(\frac{d_{an,h}}{L_{an}} \right)^{2/3} \right] \quad (32)$$

$$f_{an} = 0.25 (1.82 \log Re_{an} - 1.64)^{-2} \quad (33)$$

TABLE 4. Conditions of the validation process.

Parameters	Range or value
Annulus-side flow rate, L/min	4.1–16.3 ($2985 \leq Re_{an} \leq 12835$)
Annulus-side inlet temperature, °C	15, 20, 25 ($5.59 \leq Pr_{an} \leq 7.48$)
Tube-side flow rate, L/min	8.1 ($Re_t \approx 19715$)
Tube-side inlet temperature, °C	70 ($Pr_t \approx 2.69$)

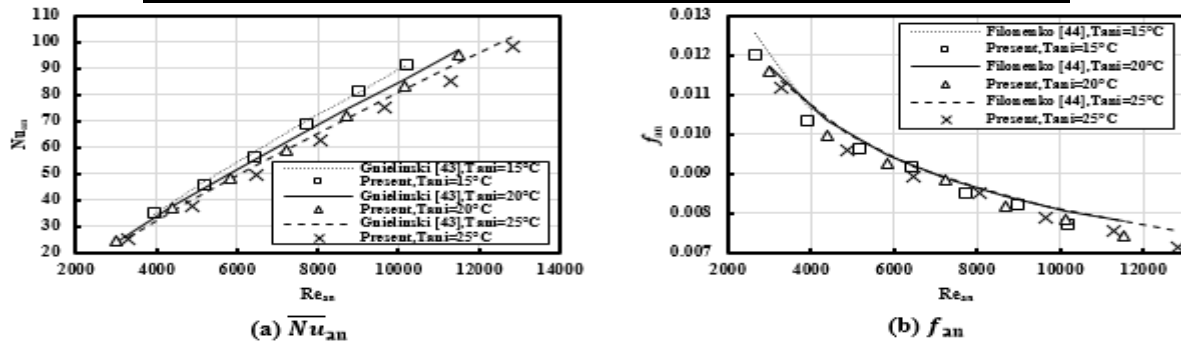


FIG 5. The results of the validation process.

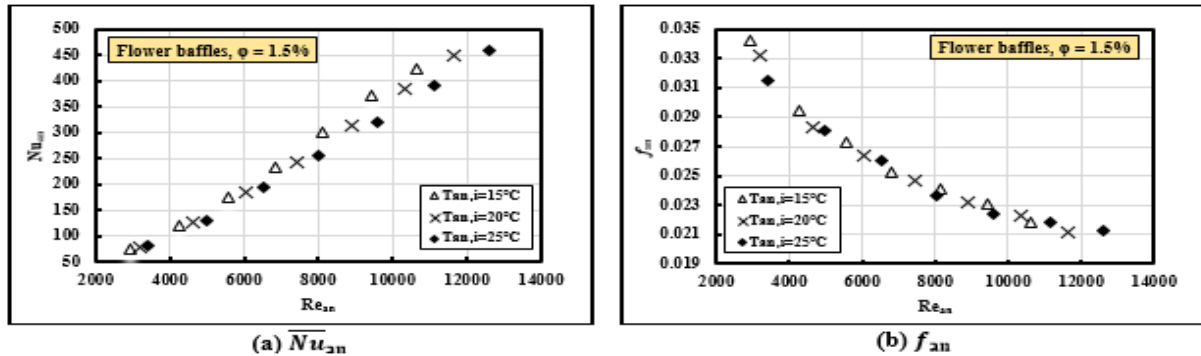


FIG 6. Effect of annulus-side operating conditions on performance parameters.

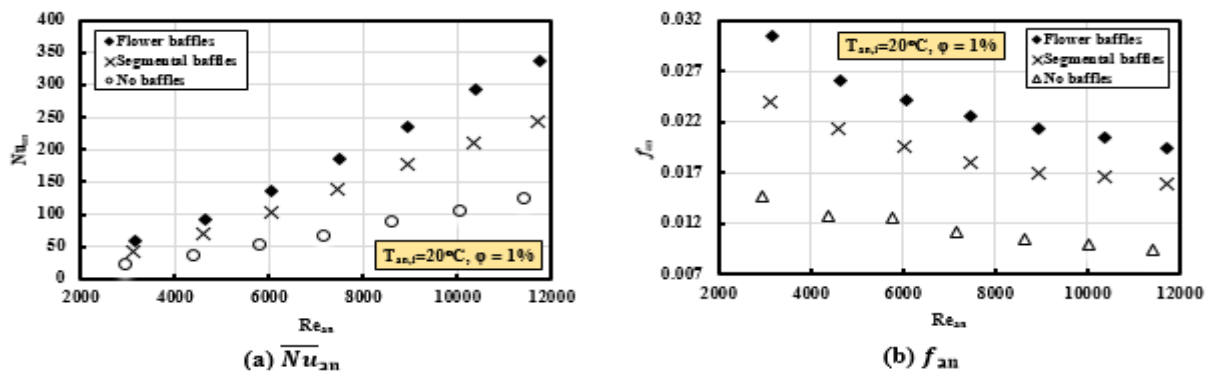


FIG 7. Effect of baffle layout on heat exchanger performance.

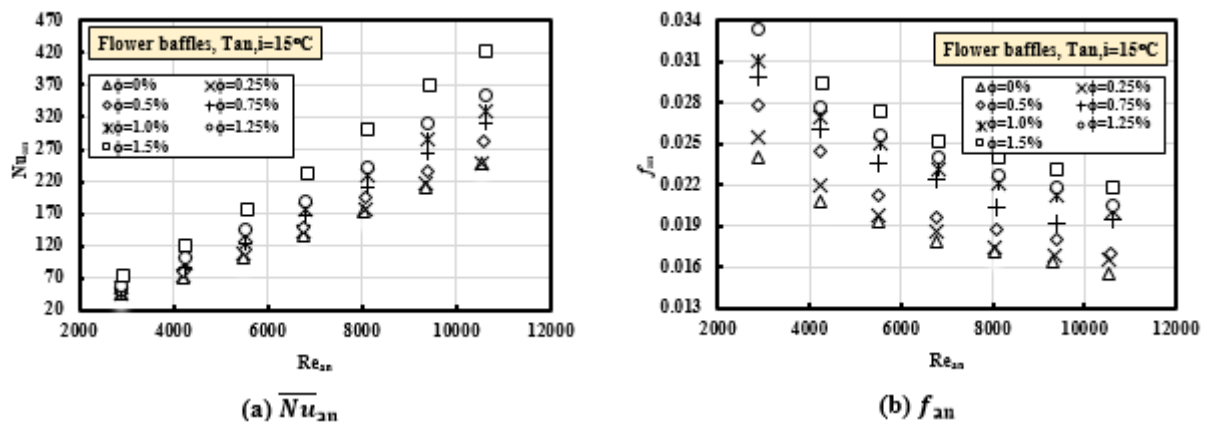


FIG 8. Effect of alumina nanoparticles loading on heat exchanger performance

7 RESULTS AND ANALYSIS

7.1 Effect of Annulus-Side Operating Conditions

In this analysis, seven flow rates in annulus side over three inlet temperature values are considered for the three tested heat exchangers. Figure 6 provides a sample of the results, focusing on key parameters; $\overline{Nu}_{an}$ and f_{an} using flower baffles and alumina nanoparticles with concentration 1.5% in annulus side. As the inlet temperature of the annulus side decreases, observe that the $\overline{Nu}_{an}$ experiences slight increase. This behaviour can be attributed to the decline in the annular-side Prandtl value with rising fluid temperature. Furthermore, an increase in the Reynolds number (Re_{an}) substantially enhances these parameters. The higher flow velocity leads to increased turbulence, promoting better fluid mixing. Improved mixing disrupts the thermal boundary layer, resulting in augmented heat transfer rates. Interestingly, the influence of flow inlet temperature on flow resistance can be disregarded. Viscosity changes have minimal effects compared to inertia forces. Notably, increasing Re_{an} leads to a notable reduction in f_{an} , emphasizing the dominance of momentum forces over viscous forces.

7.2 Effect of Baffle Layout

In this analysis, the impact of incorporating segmental or flower baffles with/without adding nanoparticles in the annular tube is examined under various operational conditions and compared to the scenario without baffles. Figure 7 shows sample results at $T_{an,i}=20^{\circ}\text{C}$ and $\phi = 1\%$. It is clear that the heat transfer rate in the heat exchanger increases with the use of baffles. However, these improvements come with a significant rise in the pumping power needed to counteract the increased pressure loss in the annular tube fluid. Across all tests using water or nanofluid in the annulus side, the findings show that the average enhancements in $\overline{Nu}_{an}$, compared to no baffles, are 165.9% and 100.7% with flower and segmental layouts, respectively. These conditions also correspond to average increases of 95.5% and 64.7%, respectively, in the annulus-friction factor. The enhanced performance of the flower-shaped baffles can

be attributed to their unique design, which effectively reduces the wake zone behind them. Conversely, conventional single-segmental baffles, characterized by a cut on one edge, tend to generate a larger wake zone, leading to reduced fluid mixing around the internal tube. Salem et al. [5] and Sahel [45] showed also that using flower baffles make great enhance for heat exchanger performance for fluid without nanoparticles also.

7.3 Effect of Alumina Nanoparticles Loading

In this study, six different volume concentrations of alumina nanoparticles, ranging from 0.25% to 1.5%, were investigated under different operating conditions. These conditions included setups both with and without segmental or flower baffles, as detailed in Table 2. Figure 8 illustrates sample results when flower baffles are used at an inlet temperature of 15°C. Significant increases in both the $\overline{Nu}_{an}$ and f_{an} are observed, correlating with the nanoparticle concentration. Specifically, at a 0.25% concentration, the average increases are 3.7% and 11.8%, 99.0% and 76.8%, and 168.7% and 113.9% for heat exchangers with plain annulus, segmental baffles, and flower baffles, respectively. At a 1.5% concentration, the increases are 62.3% and 46.4%, 223.1% and 142.1%, and 343.1% and 190.3%, respectively. The enhancement in heat transfer can be attributed to several factors. First, the addition of alumina nanoparticles to the working fluid improves its thermal conductivity which lead to significant increase in Nusselt number. This enhancement occurs because the nanoparticles have high thermal conductivity, which promotes more effective heat transfer between the fluid and the heat exchanger surfaces. Second, the enhanced Brownian motion of the nanoparticles boosts micro-convection within the fluid. As the nanoparticles move randomly, they stir the fluid, facilitating better mixing and improving overall heat transfer. However, it's crucial to acknowledge that adding nanoparticles also increases the fluid's viscosity. This higher viscosity leads to greater flow resistance, particularly on the annulus side of the heat exchanger which lead to significant increase in friction factor. The increased resistance results in higher pressure

drops, which in turn requires more pumping power to maintain a constant flow rate. Thus, although the use of nanoparticles can significantly improve heat transfer, it also presents challenges, such as increased flow resistance and potential energy costs. Overall, the study demonstrates that the integration of alumina nanoparticles in heat exchangers, particularly in conjunction with segmental or flower baffles, results in considerable improvements in heat transfer performance. These findings highlight the potential of nanofluids to enhance the efficiency of thermal systems, although careful consideration must be given to the associated increases in flow resistance and energy requirements.

7.4 Hydrothermal Performance Index

The analyses in the previous sections demonstrate that both baffles and nanofluids independently enhance the heat transfer rate. The improvement hierarchy, from lowest to highest, is as follows: conventional segmental baffles, flower-shaped baffles, and nanofluids at a 1.5% concentration. Additionally, the simultaneous use of baffles and nanofluids results in even greater improvements. Specifically, combining nanofluids with flower-shaped baffles yields more significant enhancements compared to nanofluids with conventional segmental baffles. Furthermore, these improvements in the heat exchange rate are associated with an increase in pressure drop. The trend of increasing flow resistance on the annulus side corresponds to the trend of improving the heat exchange rate, the greater the heat transfer rate, the greater the flow resistance. Therefore, a standard is needed to evaluate the performance of the method used, whether it involves baffles alone, a single nanofluid, or the simultaneous use of both. Eq. 34 is applied to compare the percentage increase in the heat transfer rate with the accompanying increase in flow resistance. This equation determines the HTPI by assessing the St_{an} and f_{an} ratios [5,8,46].

$$HTPI = \frac{St_{an,modified}/St_{an,no\ modification}}{(f_{an,modified}/f_{an,no\ modification})^{1/3}} \quad (34)$$

For heat exchangers with segmental baffles ($\delta=16.7\%$, $\lambda=8.3\%$) and flower baffles ($\delta=16.7\%$, $\lambda=8.3\%$, $\gamma=90^\circ$), the HTPI is calculated for various concentrations of alumina nanoparticles. The results are presented in Fig. 9. Under varying working conditions and nanoparticle concentrations, the HTPI for heat exchangers with flower baffles consistently exceeded unity, indicating superior performance compared to conventional segmental baffles. Specifically, the highest average HTPI is observed at 3.24 for the flower baffles heat exchanger at a nanoparticle concentration of $\phi=1.5\%$. For the segmental baffles heat exchanger, the maximum average

HTPI is 2.51 also at $\phi=1.5\%$. These findings highlight the significant enhancement in heat transfer performance achieved by adding nanoparticles to the working fluid. Furthermore, the results underscore the remarkable effectiveness of flower baffles in improving heat exchanger performance, demonstrating their potential to outperform conventional segmental baffles in similar conditions.

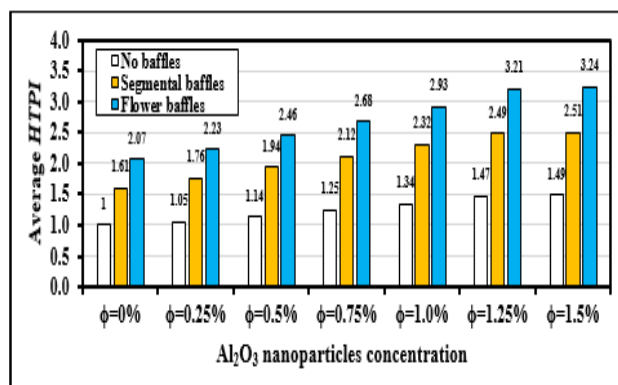


FIG 9. The average HTPI for heat exchangers with flower or segmental baffles

8. CORRELATIONS

In the present study, a series of correlations have been developed to predict essential parameters, including the average Nusselt number ($\overline{Nu}_{an}$), friction factor (f_{an}), and heat transfer performance index (HTPI). These correlations are formulated for the simultaneous application of alumina/water nanofluid and baffles in a double-pipe heat exchanger, covering a broad range of operating conditions on the annular side. The baffles are of cut ratio of 16.7% and pitch ratio of 8.3% and they are either conventional segmental ($\gamma=180^\circ$) or flower design ($\gamma=90^\circ$). Correlations (Eqs. 35 to 37) for tested heat exchangers are applicable within the ranges of $2620 \leq Re_{an} \leq 13110$, $3.91 \leq Pr_{an} \leq 7.48$, and Al_2O_3 /water nanofluid of concentration $0 \leq \phi \leq 1.5$. All correlations are also applicable to double-pipe heat exchangers without baffles, where the condition constant (C) is set to 0, while values of $C = 1$ or $C = 2$ correspond to configurations incorporating segmental or flower baffles, respectively. The maximum variances of $\pm 7.6\%$, $\pm 6.8\%$, and $\pm 7.8\%$ for $\overline{Nu}_{an}$, f_{an} , and HTPI, respectively, the suggested correlations anticipate the current experimental findings as demonstrated in Fig. 10.

$$\overline{Nu}_{an} = 0.00024 Re_{an}^{1.2671} Pr_{an}^{0.5791} (1 + C)^{0.8764} (1 + \phi)^{36.4234} \quad (35)$$

$$f_{an} = 0.15952 Re_{an}^{-0.3268} (1 + C)^{0.6567} (1 + \phi)^{24.6397} \quad (36)$$

$$HTPI = 0.97982 Re_{an}^{0.0258} Pr_{an}^{-0.1227} (1 + C)^{0.6956} (1 + \phi)^{30.4955} \quad (37)$$

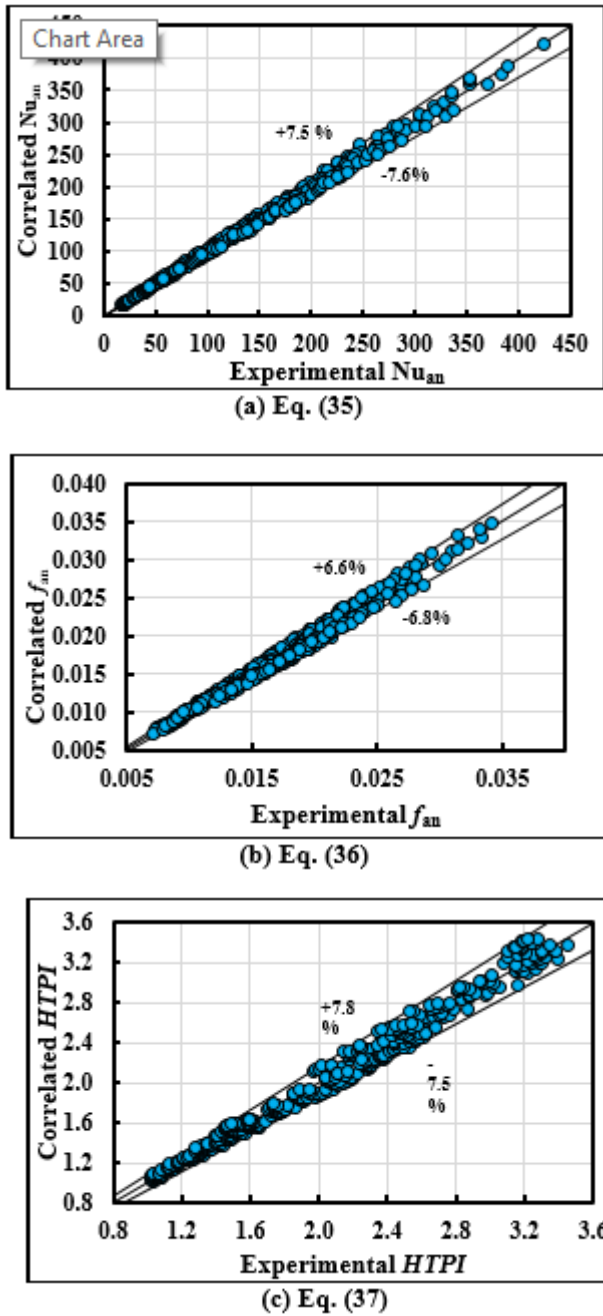


FIG 10. Comparisons between the correlated outputs and the current experimental results for conventional segmental baffles.

9. CONCLUSIONS

This research experimentally evaluates the hydrothermal performance of a horizontal double-pipe heat exchanger configured in a counter-flow arrangement. The investigation incorporates either flower baffles, arranged at a relative angle of $\gamma = 90^\circ$, or segmental baffles, while employing

Al_2O_3 /water nanofluid with nanoparticle volume concentrations ranging from 0% to 1.5% in the annular side. During the tests, hot water at $70^\circ C$ flows at a constant flow rate in the internal tube, while cold water flows in the annular pipe within the ranges of $2620 \leq Re_{an} \leq 13110$, $3.91 \leq Pr_{an} \leq 7.48$. The baffles have geometric specifications of $\delta = 16.7\%$ and $\lambda = 8.3\%$. The key findings are summarized as follows:

- The $\overline{Nu}_{an}$ and f_{an} increase significantly when baffles are inserted in the annular side of the heat exchanger and when Al_2O_3 nanoparticles are used.
- By increasing the Al_2O_3 nanoparticle concentration, $\overline{Nu}_{an}$ and f_{an} increase by 223.1% and 142.1%, respectively, for the segmental baffles, and by 343.1% and 190.3%, respectively, for the flower baffles.
- The temperature does not affect the friction factor but significantly reduces $\overline{Nu}_{an}$ as the annulus-fluid inlet temperature rises.
- Increasing Re_{an} results in higher Nusselt numbers and lower friction factors.
- The HTPI is evaluated to assess the heat exchanger's performance with Al_2O_3 /water nanofluid and baffles. Across the parameter ranges:
 - HTPI consistently exceeds unity.
 - Flower-design baffles show better HTPI compared to the conventional segmental layout, especially with the addition of Al_2O_3 nanoparticles.
 - HTPI increases with increasing nanoparticle concentration.
- The highest average HTPI of 3.24 is achieved for the flower baffle design with $\delta = 16.7\%$, $\lambda = 8.3\%$, and $\gamma = 90^\circ$ at $\phi = 1.5\%$.
- Experimental correlations for $\overline{Nu}_{an}$ and f_{an} , and HTPI predictions are suggested.

These findings highlight the effectiveness of combining Al_2O_3 /water nanofluid and innovative baffle geometries in enhancing heat transfer performance, providing valuable insights for optimizing heat exchanger designs in practical applications.

10 APPENDIX A

In accordance with Kline and McClintock's methodology [42], we systematically evaluate the impact of each variable to quantify the uncertainty associated with computed parameters. By doing so, we gain a deeper understanding of the reliability of the results.

$$\frac{\omega_\delta}{\delta} = \pm \sqrt{\left(\frac{\omega_{A_{cut}}}{A_{cut}}\right)^2 + \left(\frac{2 * \omega_{d_{an}}}{d_{an}}\right)^2} \quad (A.1)$$

$$\frac{\omega_\lambda}{\lambda} = \pm \sqrt{\left(\frac{\omega_{p_{\text{baffle}}}}{p_{\text{baffle}}}\right)^2 + \left(\frac{\omega_{L_t}}{L_t}\right)^2} \quad (\text{A.2})$$

$$\omega_{\Delta T} = \pm \sqrt{(\omega_{T_i})^2 + (\omega_{T_o})^2} \quad (\text{A.3})$$

$$\omega_{\Delta T_{LM}} = \pm \frac{\omega_T \sqrt{2}}{\ln \left[\frac{\Delta T_1}{\Delta T_2} \right]} \sqrt{2 - 2\Delta T_{LM} \left(\frac{1}{\Delta T_1} + \frac{1}{\Delta T_2} \right) + \Delta T_{LM}^2 \left(\frac{1}{\Delta T_1^2} + \frac{1}{\Delta T_2^2} \right)} \quad (\text{A.4})$$

$$\omega_{\dot{V}} = \pm \sqrt{\left(\frac{\omega_V}{t}\right)^2 + \left(\frac{-V}{t^2} \omega_t\right)^2} \quad (\text{A.5})$$

$$\frac{\omega_{\dot{m}_t}}{\dot{m}_t} = \pm \sqrt{\left(\frac{\omega_{\rho_t}}{\rho_t}\right)^2 + \left(\frac{\omega_{\dot{V}_t}}{\dot{V}_t}\right)^2} \quad (\text{A.6})$$

$$\frac{\omega_{Re_t}}{Re_t} = \pm \sqrt{\left(\frac{\omega_{\dot{m}_t}}{\dot{m}_t}\right)^2 + \left(\frac{-\omega_{d_{t,i}}}{d_{t,i}}\right)^2 + \left(\frac{-\omega_{\mu_t}}{\mu_t}\right)^2} \quad (\text{A.7})$$

$$\omega_{d_{an,h}} = \pm \sqrt{(\omega_{d_{an,i}})^2 + (\omega_{d_{t,o}})^2} \quad (\text{A.8})$$

$$\frac{\omega_{Re_{an}}}{Re_{an}} = \pm \sqrt{\left(\frac{\omega_{\dot{m}_{an}}}{\dot{m}_{an}}\right)^2 + \left(\frac{-\omega_{d_{an,h}}}{d_{an,h}}\right)^2 + \left(\frac{-\omega_{\mu_{an}}}{\mu_{an}}\right)^2} \quad (\text{A.9})$$

$$\frac{\omega_{Q_t}}{Q_t} = \pm \sqrt{\left(\frac{\omega_{\dot{m}_t}}{\dot{m}_t}\right)^2 + \left(\frac{\omega_{Cp_t}}{Cp_t}\right)^2 + \left(\frac{\omega_{\Delta T_t}}{\Delta T_t}\right)^2} \quad (\text{A.10})$$

$$\frac{\omega_{Q_{an}}}{Q_{an}} = \pm \sqrt{\left(\frac{\omega_{\dot{m}_{an}}}{\dot{m}_{an}}\right)^2 + \left(\frac{\omega_{Cp_{an}}}{Cp_{an}}\right)^2 + \left(\frac{\omega_{\Delta T_{an}}}{\Delta T_{an}}\right)^2} \quad (\text{A.11})$$

$$\omega_{Q_{ave}} = \pm \frac{1}{2} \sqrt{(\omega_{Q_t})^2 + (\omega_{Q_{an}})^2} \quad (\text{A.12})$$

$$\frac{\omega_{\overline{Nu}_t}}{\overline{Nu}_t} = \pm \sqrt{\left(\frac{0.8 \omega_{Re_t}}{Re_t}\right)^2 + \left(\frac{0.4 \omega_{Pr_t}}{Pr_t}\right)^2} \quad (\text{A.13})$$

$$\frac{\omega_{\overline{h}_t}}{\overline{h}_t} = \pm \sqrt{\left(\frac{\omega_{\overline{Nu}_t}}{\overline{Nu}_t}\right)^2 + \left(\frac{\omega_{k_t}}{k_t}\right)^2 + \left(\frac{-\omega_{d_{t,i}}}{d_{t,i}}\right)^2} \quad (\text{A.14})$$

$$\frac{\omega_{A_{t,i}}}{A_{t,i}} = \frac{\omega_{A_{t,o}}}{A_{t,o}} = \pm \sqrt{\left(\frac{\omega_{d_{t,i}}}{d_{t,i}}\right)^2 + \left(\frac{\omega_{L_t}}{L_t}\right)^2} \quad (\text{A.15})$$

$$\frac{\omega_{U_i}}{U_i} = \pm \sqrt{\left(\frac{\omega_{Q_{ave}}}{Q_{ave}}\right)^2 + \left(\frac{-\omega_{A_{t,i}}}{A_{t,i}}\right)^2 + \left(\frac{-\omega_{\Delta T_{LM}}}{\Delta T_{LM}}\right)^2} \quad (\text{A.16})$$

$$\omega_{\overline{h}_{an}} = \pm \sqrt{\left(\frac{\partial \overline{h}_{an}}{\partial U_i} \omega_{U_i}\right)^2 + \left(\frac{\partial \overline{h}_{an}}{\partial A_{t,o}} \omega_{A_{t,o}}\right)^2 + \left(\frac{\partial \overline{h}_{an}}{\partial A_{t,i}} \omega_{A_{t,i}}\right)^2 + \left(\frac{\partial \overline{h}_{an}}{\partial \overline{h}_t} \omega_{\overline{h}_t}\right)^2} \quad (\text{A.17})$$

$$\frac{\omega_{\overline{Nu}_{an}}}{\overline{Nu}_{an}} = \pm \sqrt{\left(\frac{\omega_{\overline{h}_{an}}}{\overline{h}_{an}}\right)^2 + \left(\frac{\omega_{d_{an,h}}}{d_{an,h}}\right)^2 + \left(\frac{-\omega_{k_{an}}}{k_{an}}\right)^2} \quad (\text{A.18})$$

$$\frac{\omega_{St_{an}}}{St_{an}} = \pm \sqrt{\left(\frac{\omega_{\overline{Nu}_{an}}}{\overline{Nu}_{an}}\right)^2 + \left(\frac{\omega_{Re_{an}}}{Re_{an}}\right)^2 + \left(\frac{\omega_{Pr_{an}}}{Pr_{an}}\right)^2} \quad (\text{A.19})$$

$$\frac{\omega_{f_{an}}}{f_{an}} = \pm \sqrt{\left(\frac{\omega_{\Delta P_{an}}}{\Delta P_{an}}\right)^2 + \left(\frac{5\omega_{d_{sh,h}}}{d_{an,h}}\right)^2 + \left(\frac{\omega_{\rho_{an}}}{\rho_{an}}\right)^2 + \left(\frac{-\omega_{L_{an}}}{L_{an}}\right)^2 + \left(\frac{-2\omega_{\dot{m}_{an}}}{\dot{m}_{an}}\right)^2} \quad (\text{A.20})$$

$$\frac{\omega_{HTPI}}{HTPI} = \pm \sqrt{\left(\frac{\omega_{St_{an,baffle}}}{St_{an,baffle}}\right)^2 + \left(\frac{\omega_{St_{an,no\ baffle}}}{St_{an,no\ baffle}}\right)^2 + \left(\frac{\frac{1}{3}\omega_{f_{an,baffle}}}{f_{an,baffle}}\right)^2 + \left(\frac{\frac{1}{3}\omega_{f_{an,no\ baffle}}}{f_{an,no\ baffle}}\right)^2} \quad (A.21)$$

REFERENCES

- [1] M.A. Mehrabian, M. Hemmat, The overall heat transfer characteristics of a double pipe heat exchanger: Comparison of experimental data with predictions of standard, Transactions on Modelling and Simulation, Vol. 30, pp. 607-618, 2001.
- [2] A.N. Mahureand, V.M. Kriplani, Review of heat transfer enhancement techniques, International Journal of Engineering Research and Technology, Vol. 5(3), pp. 241-249, 2012.
- [3] M.R. Salem, K.M. Elshazly, R.Y. Sakr, R. K. Ali, Experimental study on convective heat transfer and pressure drop of water-based nanofluid inside shell and coil heat exchanger, PhD Dissertation, Faculty of Engineering at Shoubra, Benha University, 2014.
- [4] S.M. Elshamy, M.T. Abdelghany, M.R. Salem, O.E. Abdellatif, Energy and exergy analysis of shell and coil heat exchanger using water based Al₂O₃ nanofluid including diverse coil geometries: An experimental study, Journal of Nanofluids, Vol. 9(1), pp. 13-23, 2020.
- [5] M.R. Salem, M.M. Ellaban, R.K. Ali, A.E. Elmohlawy, Experimental investigation of the performance attributes of a double pipe heat exchanger equipped with baffles of conventional or flower layouts, Applied Thermal Engineering, Vol. 253, 123771, 2024.
- [6] M.R. Salem, Experimental investigation on the hydrothermal attributes of MWCNT/water nanofluid in the shell-side of shell and semi-circular tubes heat exchanger, Applied Thermal Engineering, Vol. 176, article no. 115438, 2020.
- [7] M.R. Salem, M.K. Althafeeri, K.M. Elshazly, M.G. Higazy, M.F. Abdrabbo, Experimental investigation on the thermal performance of a double pipe heat exchanger with segmental perforated baffles, International Journal of Thermal Sciences, Vol. 122, pp. 39-52, 2017.
- [8] M.R. Salem, M. Eltoukhey, R. Ali, K. Elshazly, Experimental investigation on the hydrothermal performance of a double-pipe heat exchanger using helical tape insert, International Journal of Thermal Sciences, Vol. 124, pp. 496-507, 2018.
- [9] A. Colaço, V.C. Mariani, Mohamed R. Salem, L. Coelho, Maximizing the thermal performance index applying evolutionary multi-objective optimization approaches for double pipe heat exchanger, Applied Thermal Engineering, Vol. 211, article no. 118504, 2022.
- [10] N.M. Almulla, M.A. Moawed, M.A. Abd Elrahman, M.R. Salem, Effect of baffle configuration on the thermal performance attributes of shell and semi-circular tube heat exchangers, Engineering Research Journal (Shoubra), Vol. 53(1), pp. 279-291, 2024.
- [11] A. Shahmohammadi, A. Jafari, Application of different CFD multiphase models to investigate effects of baffles and nanoparticles on heat transfer enhancement, Frontiers of Chemical Science and Engineering, Vol. 8(3), pp. 320-329, 2014.
- [12] M. Bahiraei, M. Hangi, M. Saeedan, A novel application for energy efficiency improvement using nanofluid in shell and tube heat exchanger equipped with helical baffles, Energy, Vol. 93, pp. 2229-2240, 2015.
- [13] P. Shahmohammadi, H. Beiki, A numerical investigation of γ -Al₂O₃-water nanofluids heat transfer and pressure drop in a shell and tube heat exchanger, Transport Phenomena in Nano and Micro Scales, Vol. 4(1), pp. 29-35, 2016.
- [14] E.F. Akyürek, K. Gelis, B. Sahin, E. Many, Experimental analysis for heat transfer of nanofluid with wire coil turbulators in a concentric tube heat exchanger, Results in Physics, Vol. 9, pp. 376-389, 2018.
- [15] M.A. Ahmed, M.K. Mohammed, Numerical Study on Thermal-Hydraulic Performance and Entropy Generation of Nanofluid Flow in Channel with Oval Baffles, 2018 11th International Conference on Developments in eSystems Engineering (DeSE), Cambridge, UK, 2018, pp. 312-317.
- [16] M.L. Magalhães, A.S. Pereira, S.J.M. Cartaxo, Development of a transient model for evaluation of shell and tube heat exchanger with helical baffles applied for SiO₂ nanofluid, Journal of Advanced Thermal Science Research, Vol. 5, pp. 9-16, 2019.
- [17] B. Ghobadi, F. Kowsary, F. Veysi, The influence of geometric parameters of baffle on the flow and heat transfer of Al₂O₃/water nanofluid in a tube with rectangular baffle, International Nano Letters, Vol. 11, pp. 395-404, 2021.
- [18] M.A. Alazwari, M.R. Safaei, combination effect of baffle arrangement and hybrid nanofluid on thermal performance of a shell and tube heat exchanger using 3-D homogeneous mixture model, Mathematics, Vol. 9, article no. 881, 2021.
- [19] N. Phu, P. Thao, D. Truyen, Heat and fluid flow characteristics of nanofluid in a channel baffled opposite to the heated wall, CFD Letters, Vol. 13(1), pp. 33-44, 2021.
- [20] M. Bahiraei, M. Naseri, A. Monavari, A CFD study on thermohydraulic characteristics of a nanofluid in a shell-and-tube heat exchanger fitted with new unilateral ladder type helical baffles, International Communications in Heat and Mass Transfer, Vol. 124, article no. 105248, 2021.
- [21] K. Javaherdeh, M. Moslemi, M. Mahmoudnezhad, Numerical analysis of the effect of opposite baffles on the heat transfer enhancement of non-Newtonian nano fluid through a backward facing step, Big Data and Computing Visions, Vol. 1(4), pp. 216-231, 2021.
- [22] L. Bellahcene, D. Sahel, A. Yousfi, Numerical study of shell and tube heat exchanger performance enhancement using nanofluids and baffling technique, Journal of Advanced Research in Fluid Mechanics and Thermal Sciences, Vol. 80(2), pp. 42-55, 2021.
- [23] A.A.A. Arani, Shell and tube heat exchanger thermal-hydraulic analysis equipped with baffles and corrugated tubes filled with non-Newtonian two-phase nanofluid, International Journal of Numerical Methods for Heat & Fluid Flow, Vol. 31(4), 1214-1244, 2021.
- [24] F. Sharifat, A. Marchitto, M.S. Solari, D. Toghraie, Analysis, prediction, and optimization of heat transfer coefficient and friction factor of water-Al₂O₃ nanofluid flow in shell-and-tube heat exchanger with helical baffles (using RSM), The European Physical Journal Plus, Vol. 137, article number 991, 2022.
- [25] M. Bahiraei, M. Naseri, A. Monavari, Thermal-hydraulic performance of a nanofluid in a shell-and-tube heat exchanger equipped with new trapezoidal inclined baffles: Nanoparticle shape effect, Powder Technology, Vol. 395, pp. 348-359, 2022.
- [26] R. Gugulothu, N. Sanke, Effect of helical baffles and water-based Al₂O₃, CuO, and SiO₂ nanoparticles in the enhancement of thermal performance for shell and tube heat exchanger, Heat Transfer, Vol. 51, pp. 3768-3793, 2022.
- [27] S. Akcay, U. Akdag, Effect of baffle angles on flow and heat transfer in a circular duct with Nanofluids, International Advanced Researches and Engineering Journal, Vol. 6(03), pp. 176-185, 2022.
- [28] S. Sameer, S. Prakash, S.V. Shaik, N.S. G, Experimental study on effectiveness in segmental baffled shell and tube heat exchanger using Al₂O₃ nanofluid at variable concentrations. Proceedings of the Institution of Mechanical Engineers, Part E: Journal of Process Mechanical Engineering, Vol. 237(6), pp.2349-2361, 2023.

- [29] N.M. Almulla, M.A. Moawed, M.A. Abd-Elrahman, Mohamed R. Salem, Study of the thermal performance characteristics of shell and semi-circular tube heat exchanger using both baffles and nanofluid, Engineering Research Journal (ERJ), Vol. 53(2), pp:68-81, 2024.
- [30] R. Ahamed, M. Salehin, M.M. Ehsan, Thermal-hydraulic performance and flow phenomenon evaluation of a curved trapezoidal corrugated channel with E-shaped baffles implementing hybrid nanofluid, Heliyon, Vol. 10, article no. e28698, 2024.
- [31] M. Anbupala, A. Sundaresan, B. Balasubramanian, C. Senjimala, Heat transfer enhancement analysis using TiO₂-water nanofluid in shell and tube heat exchanger with doughnut and flower segmental baffles, AIP Conference Proceedings, Vol. 2986(1), 030063, 2024.
- [32] A. Khadang, M. Nazari, H. Maddah, M.H. Ahmadi, M. Sharifpur, Experimental study and neural network-based prediction of thermal performance of applying baffles and nanofluid in the double-pipe heat exchangers, Vol. 149, pp. 4239-4259, 2024.
- [33] J. Mahendran, Experimental analysis of shell and tube heat exchanger using flower baffle plate configuration, Materials Today: Proceedings, Vol. 21(Part 1), pp. 419-424, 2020.
- [34] M. Vivekanandan, R. Venkatesh, R. Periyasamy, S. Mohankumar, L. Devakumar, Experimental and CFD investigation of helical coil heat exchanger with flower baffle, Materials Today: Proceedings, Vol. 37(Part 2), pp. 2174-2182, 2021.
- [35] J. Chen, P. Zhao, Q. Wang, M. Zeng, Experimental investigation of shell-side performance and optimal design of shell-and-tube heat exchanger with different flower baffles, Heat Transfer Engineering, Vol. 42(7), pp. 613-626, 2021.
- [36] J.C. Maxwell, Treatise on electricity and magnetism, 1st ed., Clarendon Press, Oxford, UK, 1873.
- [37] J. Koo, C. Kleinstreuer, A new thermal conductivity model for nanofluids, Journal of Nanoparticle Research, Vol. 6, pp. 577-588, 2004.
- [38] J. Koo and C. Kleinstreuer, Laminar nanofluid flow in microheat-sinks, International Journal of Heat Mass Transfer, Vol. 48, pp. 2652-2661, 2005.
- [39] H.C. Brinkman, The viscosity of concentrated suspensions and solution, The Journal of Chemical Physics, Vol. 20(4), pp. 571-581, 1952.
- [40] Y. Xuan, W. Roetzel, Conceptions for heat transfer correlation of nanofluids, International Journal of Heat and Mass Transfer, Vol. 43, pp. 3701-3707, 2000.
- [41] F.W. Dittus, L.M.K. Boelter, Heat transfer in automobile radiators of the tubular type, University of California Publications in Engineering, 1930, 2,443461. Lawrence, A.E.
- [42] S.J. Kline, F.A. McClintock, Describing uncertainties in single-sample experiments, Mechanical Engineering, Vol. 75(1), pp. 3-8, 1953.
- [43] V. Gnielinski, New equations for heat and mass transfer in turbulent pipe and channel flow, International Chemical Engineering, Vol. 16, pp. 359-368, 1976.
- [44] G.K. Filonenko, Hydraulic resistance of pipes (Hydraulischer widerstand von rohrlleitungen), Teploenergetika, Vol. 1(4), pp. 40-44, 1954.
- [45] D. Sahel, Thermal performance assessment of a tubular heat exchanger fitted with flower baffles, Journal of Thermophysics and Heat Transfer, Vol. 35(4), pp. 726-734, 2021.
- [46] H.H. Al-Kayiem, A. Bin Ekhwan, L.N. Muhi, Augmentation of ribs turbulators height on the hydrothermal performance of double pipe heat exchanger, J. Eng. Sci. Tech., Vol. 12(2), pp. 548-563, 2017.

NOMENCLATURES

A	Area, m ²
C_p	Specific heat, J/kg. °C
d	Diameter, m
f	Fanning friction factor
$\bar{h}$	Convection heat transfer coefficient, W/m ² . °C
k	Thermal conductivity, W/m. °C
L	Length, m
$\dot{m}$	Mass flow rate, kg/s
N	Number of baffles
P	Pressure, Pa
p	Baffle pitch, mm
Q	Heat transfer rate, W
T	Temperature, °C or K
t	Time, s
U	Overall heat transfer coefficient, W/m ² . °C
V	Volume, m ³
$\dot{V}$	Volume flow rate, m ³ /s

DIMENSIONLESS GROUPS

$\overline{Nu}$	Average Nusselt number
Pr	Prandtl number
Re	Reynolds number
St	Stanton number

GREEK LETTERS

λ	Baffle pitch ratio
ϕ	Volume concentration of nanoparticles
Δ	Differential
ω	Uncertainty
γ	Baffle relative angle, °
δ	Baffle cut-off ratio
π	Pi $\equiv$ A mathematical constant $\cong 3.1416$
ρ	Density, kg/m ³

SUPERSCRIPTS AND SUBSCRIPTS

an	Annulus
ave	Average
h	Hydraulic
i	Inlet/Inner/Internal
LM	Logarithmic mean
o	Out/Outer
t	Tube

ACRONYMS AND ABBREVIATIONS

HTPI	Hydrothermal Performance Index
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