



Novel Fin-Integrated Baffle Design for a Shell-and-Tube Heat Exchanger: An Experimental Study

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ABSTRACT

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Heat exchangers (HX) are used in many industrial processes. One of them is shell-and-tube heat exchanger (STHX). Many researchers have developed new models of STHX to improve its performance and effectiveness. This study evaluates the actual performance of an STHX featuring a novel baffle design that integrates the addition of fins with angle-cut baffles, aiming to characterize the effectiveness of these modifications for possible industrial applications. An experimental approach was conducted using a laboratory-scale STHX, with water employed as the working fluid for hot and cold fluids under controlled operating conditions. Parameters measured included pressure drop on both tube and shell sides of the exchanger to assess hydrodynamic behavior. The experimental results indicate that the pressure performance of the tested STHX yields a minimum pressure difference of 7.8 Pa (hot fluid) and 351.7 Pa (cold fluid). The corresponding maximum pressure differences reach 87.3 Pa for ΔP hot fluid and 4908.6 Pa for ΔP cold fluid. The highest effectiveness was obtained at the flow rate with variation 1.8 and 7 L/min of 87.33%, and the effectiveness for both the same flow rate investigated average 58.59%. These findings demonstrate the influence of the combined fin-and-angle-cut baffle configuration on the pressure drop characteristics of STHX and provide a basis for evaluating its potential effectiveness in practical applications.

1. INTRODUCTION

Many researchers aim to enhance the performance of the heat exchanger (HX) further. Lorenzini et al. [1] investigated and analysed the heat transfer of 128 horizontal tubes of a crossflow air-cooled HX with aluminium fins. Theodossiou et al. [2] computed and investigated the prediction of the flow field in an HX using a two-dimensional isothermal steady flow distribution case. Ozden and Tari [3] observed the shell side of the shell-and-tube heat exchangers (STHX) using numerical modelling in a small HX. Vukic et al. [4] discussed the effectiveness of STHX using different numbers of segmental baffles. Vukic et al. [5] also investigated the process in the STHX using 3D numerical simulations. Tanujaya and Sukania [6] investigated the performance and effectiveness of an STHX prototype using a stationary head with a ring rubber clamp using a single segmental baffle type. Tanujaya and Darmawan [7] also investigated the STHX with a disc-and-doughnut baffle type and 40% cut segmental baffles using a numerical method. Bayram and Sevilgen [8] investigated, using a numerical approach, the effect of variable baffle spacing on the thermal performance of an STHX in 8 fluid zones. Ghazanfari and Wahid [9] discussed the optimisation of a compact HX using a delta winglet type using numerical calculation. Sparrow and Reifschneider [10] discussed the effect of interbaffle spacing in the STHX to determine the response of the heat transfer and pressure drop. Kim and Aicher [11] conducted experimental research on heat transfer

in a shell and tube without the baffles. The conventional segmental baffles are the most widely used shell-side enhancement device; many studies show that increasing baffle spacing can reduce the shell-side heat transfer coefficient by almost 15% and cut the pressure drop by 40%, and that enlarging the baffle cut from 15% to 25% lowers the heat-transfer coefficient by 5% and the pressure drop by 26% [12]. A recent review shows that while traditional baffles enhance heat transfer, they also cause flow maldistribution and dead zones, which reduce thermal effectiveness and increase operating cost. Regarding the limitations, many modified baffle geometries have been proposed. Helical baffles create a continuous helical flow path that increases turbulence while reducing dead zones, fouling, and pressure drop relative to segmental baffles. Wang et al. [13] showed an extensive review of helixchangers. They reported superior flow and heat-transfer performance over conventional segmental designs in both experiments and simulations. İnan et al. [14] reviewed experimental comparisons of a new baffle geometry at different baffle intervals against a conventional design, confirming that baffle geometry remains among the most influential, and least systematically explored, shell-side variables.

The proposed research on fin-integrated baffle structure and its heat transfer enhancement mechanism, and flow distribution across the baffle therefore has direct applicability to a wide range of thermal engineering systems. In industrial HXs, baffles are used in industrial STHX to redirect the flow

and optimise the cross-flow heat transfer. Hamied et al. [15] also showed that integrating baffles with internal fins can raise the effectiveness of HX compared to a conventional configuration. The model of a fin-integrated baffle structure is a candidate for economizers and preheaters where compactness and high surface area density are critical, as explained by Wang et al. [16].

One problem is that the fluid flow inside the shell within the single segmental straight baffles of the STHX often causes dead-zone recirculation. This phenomenon can decrease both the heat transfer between the hot and cold fluids and the effectiveness of the system. One of our efforts in this research is to modify the baffle by integrating fins with an angle-cut baffle configuration. This research aims to solve the problem by decreasing the dead zone, which can increase the heat transfer and performance within an acceptable pressure drop range for the operation of the HX that has occurred in the system using an STHX laboratory [6].

2. MATERIALS AND METHODS

The design of fins fabricated through conventional routes using fiber and baffle assembly, topologically optimised layouts, and good mechanical properties. The fin position is considered for maintenance requirement which are the flow passages created by the fin baffle arrangement remain accessible for inspection, cleaning, and long-term operating reliability.

2.1 Thermal design

The total heat load of the STHX can be calculated using the heat balance equations for the hot and cold fluid streams, with the hot fluid flowing through the tube and the cold fluid through the shell, as given by Eqs. (1)–(3), assuming steady-state and adiabatic conditions:

$$Q_{cold} = Q_{hot} \quad (1)$$

$$Q_{cold} = [\dot{m} \cdot C_c (T_{c_{out}} - T_{c_{in}})]_{shell} \quad (2)$$

$$Q_{hot} = [\dot{m} \cdot C_h (T_{h_{in}} - T_{h_{out}})]_{tube} \quad (3)$$

The total heat load was calculated using the logarithmic mean temperature difference (LMTD) method, as shown in Eqs. (4)–(5):

$$Q = U \cdot A \cdot \Delta T_{LMTD} \quad (4)$$

$$\Delta T_{LMTD} = \frac{(T_{h_{in}} - T_{c_{out}}) - (T_{h_{out}} - T_{c_{in}})}{\ln \frac{(T_{h_{in}} - T_{c_{out}})}{(T_{h_{out}} - T_{c_{in}})}} \quad (5)$$

The total convective heat transfer coefficient (U), total heat transfer area (A), and ΔT_{LMTD} is the temperature difference between the two fluids at the two ends of the STHX. The temperature of the hot fluid inlet ($T_{h_{in}}$), the cold fluid inlet ($T_{c_{in}}$), and the temperature of the hot fluid outlet ($T_{h_{out}}$) and cold fluid outlet ($T_{c_{out}}$). The total heat transfer area (A) was calculated based on the tube diameter (d_{tube}), which is determined as the average of the inner (d_{intube}) and outer ($d_{outtube}$) diameters of the tube, tube length (L_{tube}), and number of tubes (N_{tube}), as shown in Eq. (6):

$$A = \pi \cdot \frac{(d_{intube} + d_{outtube})}{2} \cdot L_{tube} \cdot N_{tube} \quad (6)$$

Based on theoretical calculations, the fouling resistance or thermal energy resistance (R_{th}) is assumed negligible, so the fouling resistance in Eq. (7) can be written for the cylindrical tube and both overall convective heat transfer coefficients of hot (h_h) and cold (h_c) fluids, respectively. The thermal conductivity of the tube (k) is described in Eq. (8) for the clean surface of the total convective heat transfer coefficient (U) for the tube and shell sides.

$$R_{th} = \frac{1}{h_h \cdot A_{intube}} + \frac{\ln \left(\frac{d_{outtube}}{d_{intube}} \right)}{2\pi \cdot k \cdot L_{tube}} + \frac{1}{h_c \cdot A_{outtube}} \quad (7)$$

$$\frac{1}{U} = R_{th} \quad (8)$$

We use the Dittus-Boelter and Churchill-Bernstein approach to calculate the shell side of the convective heat transfer coefficient and the pressure drop. To solve the convective heat transfer coefficient of the tube side, the Nusselt number of the hot fluid (Nu_{hot}) is calculated using the internal-flow Dittus-Boelter approach, with $n = 0.4$ for heating of the fluid and $n = 0.3$ for cooling of the fluid [17]. This correlation is applicable only to the internal flow on the tube side and was therefore not applied to the shell side flow. The structural parameters such as baffle spacing, cut ratio, number of fins, and fin dimensions are adapted to laboratory equipment. The total number of tubes is 22. On the shell side (cold fluid), all the results are based on a constant baffle cut ratio of 20% with a spacing of 100 mm. The number of baffles is 8 baffles inserted in the shell. The Nusselt number of cold fluid (Nu_{cold}) is calculated using the Churchill and Bernstein approach for Reynolds number (Re) between $100 < Re < 10^7$. The tube's convective-heat-transfer-coefficient (hot fluid) was obtained using the appropriate method [18], as shown in Eqs. (9)–(12).

$$Re = \frac{\rho \cdot V \cdot d_{intube}}{\mu} \quad (9)$$

$$Nu_{hot} = 0.023 Re_{hot}^{4/5} Pr^n \quad (10)$$

$$h_{hot} = \frac{Nu \cdot k}{d_{intube}} \quad (11)$$

The convective heat transfer coefficient of the cold fluid (shell) and pressure drop were calculated using the Sieder and Tate correlation with parameters: the equivalent shell diameter for triangular pitch (d_e), tube pitch (P_T), cold fluid volumetric flow rate (Q_s), as shown in Eq. (12) and Eq. (13):

$$Nu_{cold} = 0.36 \left(\frac{d_e \cdot Q_s}{\mu} \right)^{0.55} \left(\frac{c \cdot \mu}{k} \right)^{0.33} \left(\frac{\mu}{\mu_w} \right)^{0.14} \quad (12)$$

$$d_e = \frac{(4P_T \cdot 6.88P_T - \pi d_{outtube}^2)}{\pi d_{outtube}} \quad (13)$$

The cold fluid volumetric flow rate (Q_s) was converted to the mass flow rate using $Q_s = \dot{m}/\rho$, where ρ is the fluid density evaluated at the mean fluid temperature. The Q_s was then obtained by substituting this mass flow rate, together with tube pitch, distance between tubes (C), inner diameter of shell (d_{shell}), and the distance between baffle (B), as shown in Eq.

(14):

$$Q_s = \frac{\dot{m} \cdot P_T}{d_s \cdot C \cdot B} \quad (14)$$

Kern method is used to solve the problem of pressure drop in the shell side (cold fluid) using Eq. (15). N is the number of baffles, μ is the dynamic viscosity of the fluid, μ_w is the dynamic viscosity of the tube-wall, and the friction factor (f) can be calculated considering the value of Re as shown in Eq. (16):

$$\Delta P_{cold} = \frac{f \cdot Q_s \cdot d_{shell} (N + 1)}{2 \cdot \rho \cdot d_e \cdot \left(\frac{\mu}{\mu_w}\right)^{0.14}} \quad (15)$$

$$f = \exp(0.576 - 0.19 \ln(Re_{cold})), 400 < Re_{cold} < 10^6 \quad (16)$$

In the tube side (hot fluid), the hot fluid volumetric flow rate (Q_t) is used to calculate the pressure drop. s is the specific gravity, V_{hot} is hot fluid velocity, and g is the acceleration of gravity, as shown in Eq. (17):

$$\Delta P_{hot} = \frac{f \cdot Q_t \cdot L_{tube} \cdot N}{5.22 \cdot 10^{10} \cdot d_e \cdot s \cdot \phi_t} + \frac{4 \cdot N \cdot V_{hot}^2}{s \cdot 2 \cdot g} \quad (17)$$

The shell is a cylindrical vessel in which the tube bundle is placed inside them. The thermal design of the STHX consists of the heat transfer area, number of tubes, tube length and diameter, tube pitch, number and type of baffles, and the pressure drops on both the shell and tube sides. The calculation of the heat transfer coefficient for the hot and cold fluids is assumed to be a single-phase flow without phase change occurring during the heat transfer process. This assumption simplifies the analysis and makes it easier to use conventional convective heat transfer coefficient correlations based on Re and Nu . The heat balance equation assumes steady-state conditions and adiabatic operation at the shell, consistent with classical STHX design assumptions for one shell pass and one tube pass [19]. This assumption is valid when heat losses to the environment are negligible due to the low thermal conductivity of the shell material and the large ratio of heat transfer surface area to external shell surface area.

The LMTD approach applied in Eq. (5) provides the actual mean driving force between the hot and cold fluids in counter-current flow [19]. Compared to the ε -NTU method, this approach is simpler and more intuitive when inlet and outlet fluid temperatures are known a priori from experimental data [6]. However, for multipass configurations, the LMTD must be corrected using a correction factor F , since the actual flow in multipass STHXs is never purely counter-current. Perry's Chemical Engineers' Handbook stipulates that an F value below 0.8 indicates thermodynamically inefficient design, and the HX Design Handbook sets a minimum threshold of $F = 0.75$ for economic feasibility [20]. With F values of 0.883–0.955 achieved in the present work, the fin-modified design demonstrates good thermohydraulic feasibility.

The series-resistance thermal model in Eq. (7) is a cylindrical geometry model that neglects fouling resistance, assuming clean tube surfaces during short-duration laboratory tests. Although this simplification is appropriate for the experimental conditions considered, fouling resistance must be included for industrial applications, as it significantly

degrades the overall heat transfer coefficient (U) over operating time [21]. The selection of stainless steel 304 for the tubes [$k = 16.2 \text{ W/(m}\cdot\text{K)}$] and PMMA (plexiglass) for the shell [$k = 0.24 \text{ W/(m}\cdot\text{K)}$] represents an interesting trade-off [6]. The high tube conductivity minimizes conductive wall resistance, while the low PMMA conductivity on the shell side acts as a natural insulator, approaching the assumed adiabatic shell condition. The choice of PMMA also enables visualization of flow patterns and bubble behavior during experimentation, providing additional diagnostic value. The Dittus-Boelter correlation in Eq. (10), with $n = 0.3$ for cooling and $n = 0.4$ for heating, applies within the fully turbulent regime ($Re > 10,000$, Pr between 0.7–16,700) [18]. However, this correlation has limitations in the transitional and laminar regimes, which are likely encountered at low flow rates (1.8 L/min) in the present work. These limitations must be addressed with regime-specific correlations in subsequent analyses.

The Kern method adopted for the shell side provides a simple, reliable, and widely applied STHX design approach. This method estimates the shell-side heat transfer coefficient using the shell diameter for triangular pitch (d_e) and the cold fluid velocity, accounting for tube pitch, clearance, and baffle spacing. However, the Kern correlation has well-known conservative limitations. It tends to underestimate the actual heat transfer coefficient because it neglects bypass-stream and leakage-stream effects, which are accounted for by the more accurate Bell–Delaware method [22, 23]. Comparative studies have shown that the discrepancy between the two methods grows with increasing number of baffles and variations in normalized baffle spacing [22]. The Sieder–Tate correlation in Eq. (12) introduces a viscosity-ratio correction $(\mu/\mu_w)^{0.14}$ to account for the effect of near-wall fluid viscosity variations with temperature, significantly improving predictive accuracy, particularly at low flow rates and large temperature differences.

For crossflow over the tube bundle, the Hilpert [24] and Zhukauskas [25] correlations were selected with $C = 0.9$ and $n = 0.4$ for staggered arrangements. This choice aligns with the Re range of 10–100 and Prandtl number (Pr) range of 0.7–500 from which the underlying experimental data were generated, matching the crossflow regime expected across small-diameter tube bundles [25].

This experimental study used an STHX type with the structure of one shell pass and one tube pass. The total number of tubes inside the shell was 20 tubes with a 12.7 mm outside diameter (BWG 20) and a length of 700 mm. The tube bundle was arranged in a 30° triangular pitch layout with a tube pitch of 17.8 mm. This model configuration was selected to provide a better heat transfer coefficient compared to a square pitch configuration due to increased turbulence and compact tube spacing. The shell had a total length of 700 mm, with an inner diameter of the shell is 110 mm. This experimental study used eight baffles with a baffle spacing of 100 mm, single segmental, and a cut-off of 20%. We determined that the baffle spacing was carefully selected to avoid excessive pressure drop. To increase the heat transfer performance, the baffles were modified by attaching fins to each baffle. Each fin had dimensions of 105 mm in length, 25 mm in width, and 1.5 mm in thickness, as illustrated and shown in Figure 1(a-c). The materials of the shell and tubes are polymethyl methacrylate (PMMA, plexiglass) and stainless steel 304, respectively. The thermal conductivity of plexiglass (PMMA) and stainless steel 304 are 0.24 W/(m·K) and 16.2 W/(m·K), respectively [6]. The selection of stainless steel 304 for the tube material

ensures good corrosion resistance and adequate thermal conductivity. Figure 2(a-c) shows the placement of fins inside the shell. The fins are placed between tubes, with a total of 7

fins as shown in Figure 2(c). The addition of fins will influence the effective heat transfer area and promote flow disturbance, which is expected to improve the thermal performance.

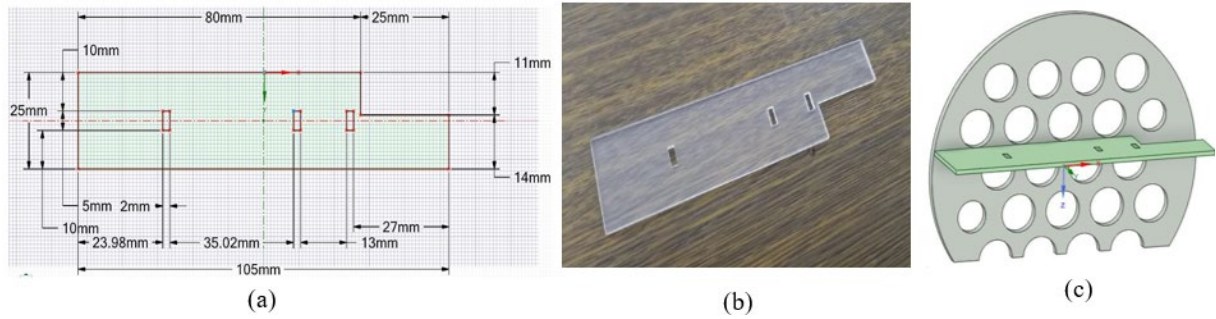


Figure 1. Design of finned baffle modified, (a) design of fin, (b) finished fin, and (c) fin on baffle

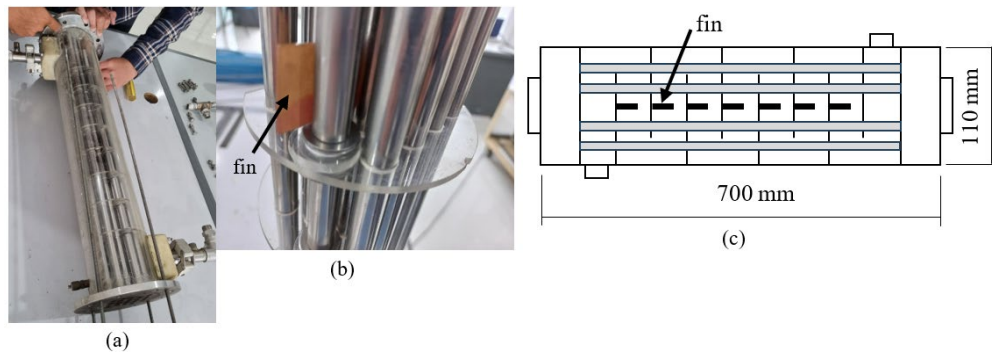


Figure 2. Placement of finned, (a) bundle of the tube, (b) fin on baffle, and (c) scheme of fin

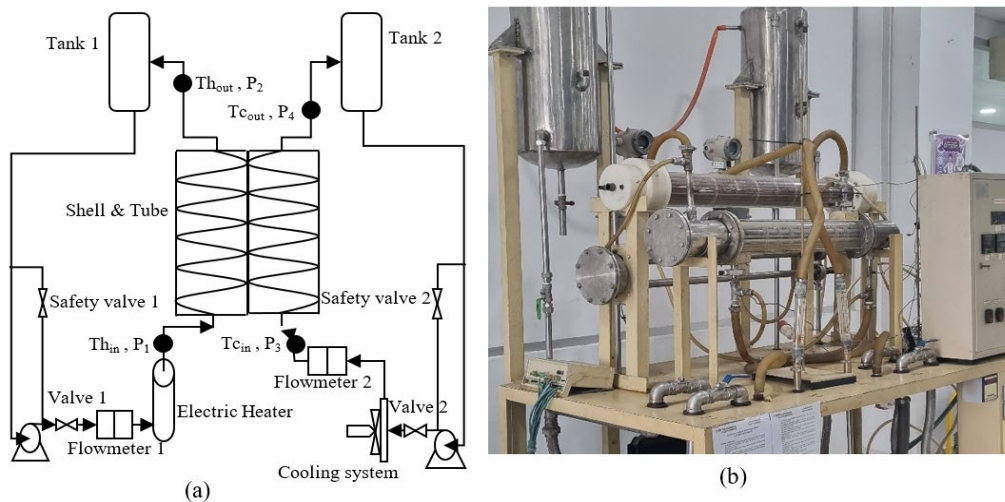


Figure 3. Experimental installation, (a) schematic diagram of shell-and-tube heat exchangers (STHX), and (b) equipment

STHX laboratory that are used in this study, as shown in Figure 3, had two tanks to store hot and cold fluids at two different temperatures. A controlled test facility using laboratory equipment with calibrated flow and differential-pressure instrumentation was constructed for this purpose, enabling repeatable comparison between the fin-integrated baffle configuration and the conventional baffle baseline under identical operating conditions. The equipment has two pumps, two flowmeters, and two valves to circulate and set the flow rate for each hot and cold fluid, as shown in Figure 3. The electric heater was installed as source heat generation set to 1.7 kW, and the radiator as a cooling system for the hot fluid. Pressure transducers were installed at the inlet and outlet of the

shell and tube using a differential pressure transmitter ST3000 model STD910. These apparatuses were used to investigate the fluid pressure distribution in the shell and tube sides. The measuring range of the pressure transmitter was -1000 to 1000 Pa, with accurate measurement of linear output based on the equipment of $\pm ((0.15 + 0.15 \times (1.0/(\text{greatest range value}/\text{lower range value}))) \%)$. The fluid temperatures were measured using a thermocouple (K type) [6, 7]. The fluid temperature and pressure data were collected using NI 9213 and NI 9203, respectively. Each operating condition was repeated 2 times on different days, and the maximum deviation of the measurements between repeated runs was below 1%.

The inlet and outlet temperatures of the shell and tube were

measured using calibrated thermocouples with an accuracy of ± 0.1 °C, flow rate with 10L/min ($\pm 1\%$ of reading), and differential pressure with a smart differential-pressure transmitter of the resonant-sensor class. Data were recorded only after temperature variations remained within ± 0.1 °C and flow/pressure variations within $\pm 1\%$ for at least 30 minutes.

The experimental study was conducted under the ambient temperature of 25 to 27 °C. Countercurrent flow was used in this experiment. We use 3 kinds of variations in mass flow rate, first using same mass flow rate between tube/hot fluid and shell/cold fluid. Second, using increasing the tube/hot fluid mass flow rate and fixed mass flow rate in shell/cold fluid. Third, using a fixed mass flow rate in tube/hot fluid and increasing the shell/cold fluid. The mass flow rate used in the experiment is set to 1.8, 3, 5, 7, and 9 L/min, for both shell and tube-side. To simplify the variation of mass flow rate name of shell and tube-sides, the first words are set to hot fluid mass flow rate, and the second words are set to cold fluid mass flow rate as $\dot{m}_{hot} - \dot{m}_{cold}$: 1.8-1.8; 3-3; 5-5; 7-7; 9-9 (the first variation of mass flow rate, $\dot{m}_{hot} - \dot{m}_{cold}$ with both hot and cold fluid in the same flow rate). $\dot{m}_{hot} - \dot{m}_{cold}$: 1.8-7; 3-7; 5-7 (the second variation of mass flow rate, $\dot{m}_{hot} - \dot{m}_{cold}$ with increasing the hot fluid and constant the cold fluid). $\dot{m}_{hot} - \dot{m}_{cold}$: 7-1.8; 7-3; 7-5 (the third variation of mass flow rate, $\dot{m}_{hot} - \dot{m}_{cold}$ with constant hot fluid and increasing the cold fluid). Using Eq. (6), the total heat transfer area is 0.558 m^2 . Thermophysical properties of saturated water are used in the experiment. Dimensions of the properties of saturated water: Temperature (T); $\mu(\text{Pa}\cdot\text{s})$; $k[\text{W}/(\text{m}\cdot\text{K})]$; $\rho(\text{kg}/\text{m}^3)$; $C_p[\text{kJ}/(\text{kg}\cdot\text{K})]$.

3. RESULTS AND DISCUSSION

The temperature distribution of $T_{c_{out}}$, $T_{c_{in}}$, $T_{h_{in}}$, and $T_{h_{out}}$ was analyzed to observe the heat transfer performance of HX. The results show that the hot fluid temperature decreased along the flow direction, while the cold fluid temperature increased due to heat change between the fluids. The temperature difference between the inlet and outlet of both the shell and tube sides indicates the effectiveness of the heat transfer. Higher ΔT corresponds to a higher heat transfer rate. The installation of fins on the baffles inside the shell enhances the turbulence of the shell-side fluid, which contributes to temperature distribution inside the shell. The performance and effectiveness of STHX using modified baffles with adding the fin can be seen have variation result depend on the variation of mass flow rate of $\dot{m}_{hot} - \dot{m}_{cold}$ (hot and cold fluid). The fluid flow of $\dot{m}_{hot} - \dot{m}_{cold}$ is set with countercurrent flow to maximize heat transfer. The LMTD is one of the key parameters used in this system calculation to determine the driving force for heat transfer. Figure 4 shows the correction factor of STHX calculation, which measures the deviation of STHX performance from the idealized flow which is related to LMTD. Perry's Chemical Engineers Handbook said that if LMTD correction factor is lower than 0.8 indicates that the design of the HX is inefficient. The HX Design Handbook said that the minimum value should be 0.75. The graph shows that the lowest and the highest correction factors are 0.883 and 0.955, respectively. This phenomenon indicates that the design and model of STHX with additional fins is an appropriate and good model to develop in these experiments.

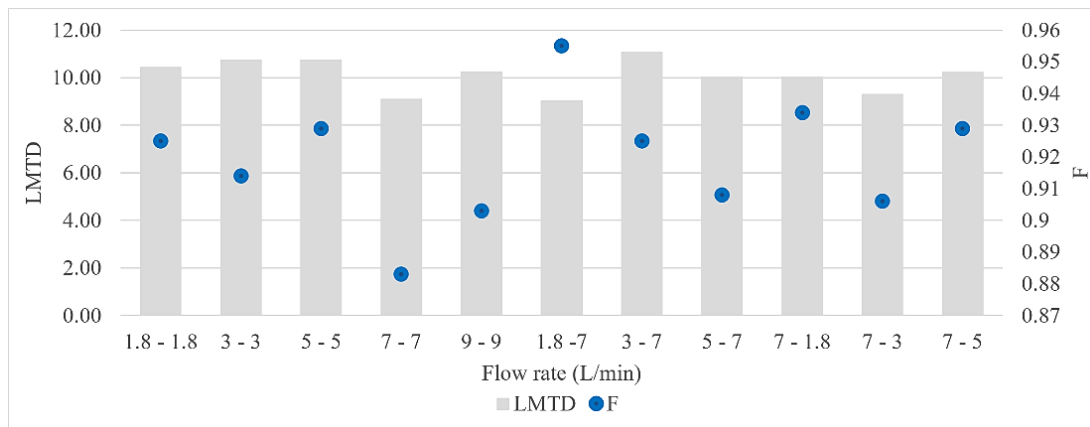


Figure 4. Logarithmic mean temperature difference (LMTD) and correction factor F

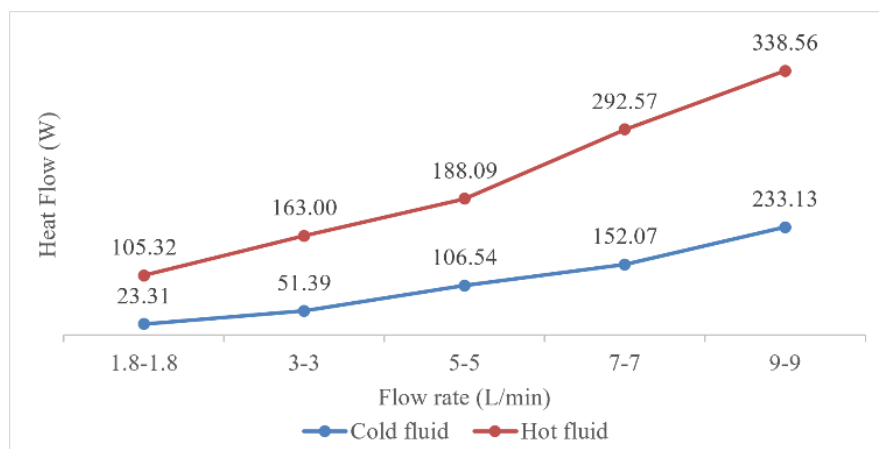


Figure 5. Heat flow with the same flow rate of hot and cold fluid

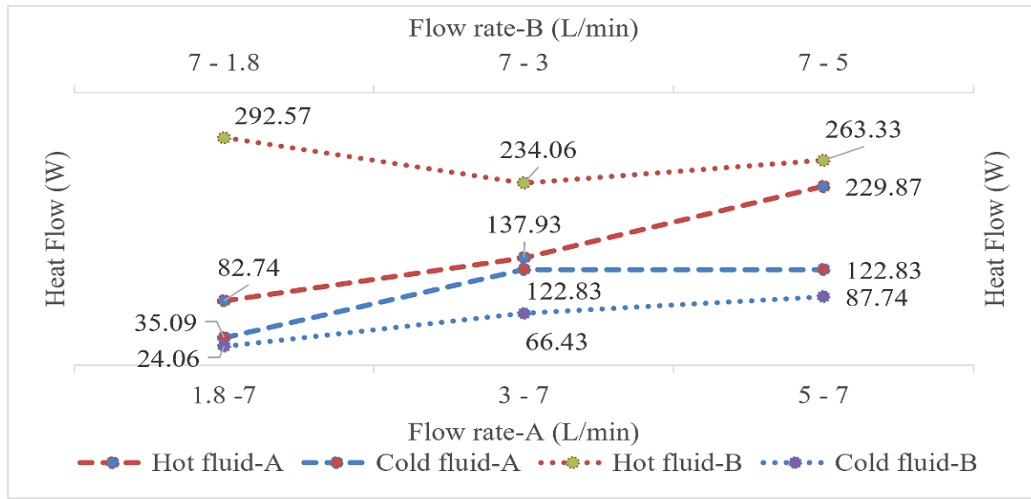


Figure 6. Heat flow with flow rate A (bottom) and B (top)

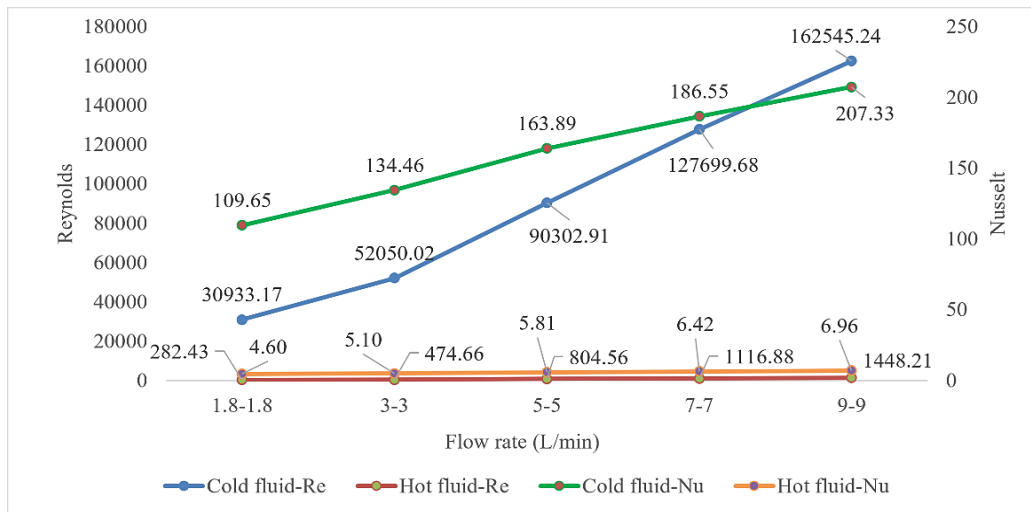


Figure 7. Reynolds number (Re)–Nusselt number (Nu) with the same flow rate of hot and cold

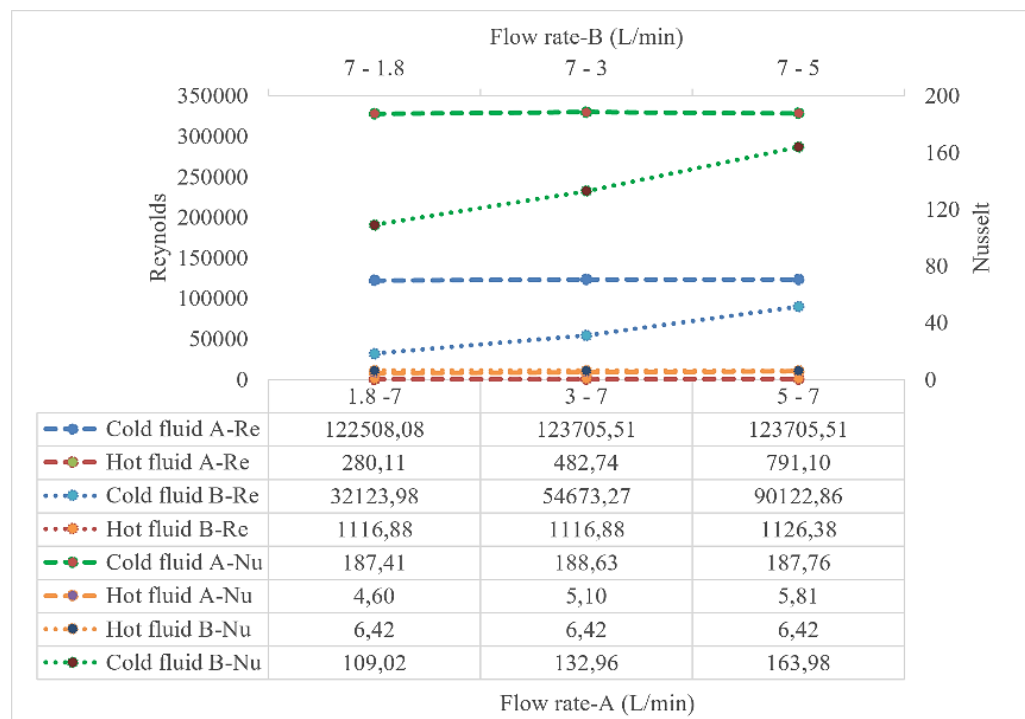


Figure 8. Reynolds number (Re)–Nusselt number (Nu) with flow rate A (bottom) and B (top)

The calculation of the overall heat flow shows that heat flow increases with increasing flow rate for both hot and cold fluids, as shown in Figures 5 and 6. The observed heat flow confirms that the channel geometries effectively facilitate thermal exchange by minimizing flow stagnation zones. Furthermore, the positive correlation between increased fluid velocity and higher exit temperatures suggests that maximizing turbulence at the pipe-fin interface is essential for enhancing overall system effectiveness.

Figures 7 and 8 show the relationship of the observed power law dependence between Re and Nu , where enhanced fluid turbulence leads to a proportional increase in the convective heat transfer coefficient. These data align with reported increases in cooling effectiveness from 58.59 % to 87.33 % as the Re increases numerically, demonstrating how thinner

thermal boundary layers directly augment efficiency. These performance gains necessitate a careful trade-off, as elevated turbulence concurrently induces substantial pressure drops and higher fan power requirements.

Figures 9 and 10 show a comparison of Re with the total convective heat transfer coefficient (U). Generally, the total U of the tube package inside the shell can be calculated using tube array in the mode of staggered arrangement. The fluid flows across the tube package composed of 20 rows with Re and Pr between 10 to 100 and 0.7 to 500, respectively. The shell Nusselt number (Nu_{Shell}) was calculated based on empirical correlations due to Hilpert [24], and Zhukauskas [25] correlation with $C = 0.9$ and $n = 0.4$ (staggered condition), respectively.

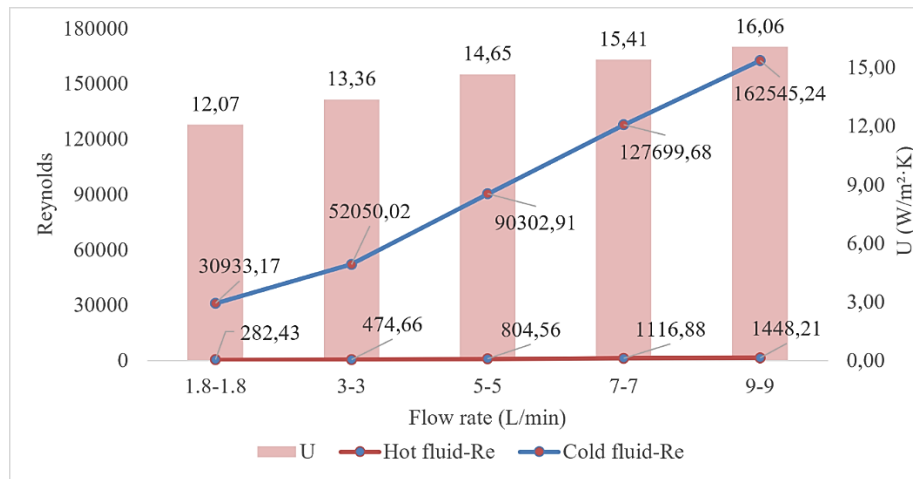


Figure 9. Reynolds number (Re) as a function of the overall heat transfer coefficient (U) at the same flow rate of hot and cold fluids

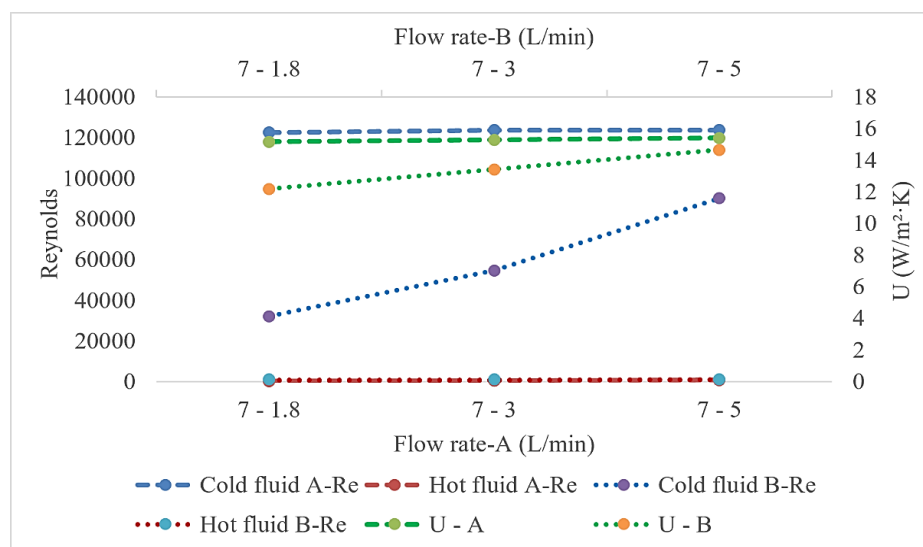


Figure 10. Reynolds number (Re) as a function of the overall heat transfer coefficient (U) at flow rate A (bottom) and B (top)

The heat transfer rate was investigated using the law of energy balance between the inlet and outlet of the hot and cold fluids. The highest and lowest heat (ΔQ) differences are investigated at flow rates variation 7-1.8 L/min of 268.5 W and 3-7 L/min of 15.1 W, respectively. The coefficients of convective heat transfer in the tube-side and shell-side are shown to be between 279.44 and 330.85 $W/(m^2 \cdot K)$ and 24.02 to 45.58 $W/(m^2 \cdot K)$, respectively. The difference in maximum

temperature between hot and cold fluids for the flow rates of 1.8, 3, 5, 7, and 9 L/min was indicated at 1.8 L/min as 14 °C (hot fluid) and 6.2 °C (cold fluid). The results indicate that the modified baffle configuration integrated with fins increased the overall heat transfer rate compared to the STHX without this configuration. This improvement occurs because the fins increase the effective heat transfer surface area and disturb the shell-side flow in the shell to increase turbulence. The

turbulence phenomenon improves the convective heat transfer coefficient and enhances the thermal performance of the HX. The convective heat transfer coefficients on both the tube and shell sides were evaluated using the calculated Nu . The shell-side heat transfer coefficient showed a noticeable increase due to the presence of fins attached to the baffles. The configuration of seven fins per baffle enhances heat-transfer effectiveness through increased local turbulence and added secondary heat-transfer surface area, without incurring a significant pressure-drop penalty. This is confirmed by the experimental data: hot fluid heat transfer coefficients between 279.76-347.68 W/(m²·K), cold fluid coefficients between 24.02-45.58 W/(m²·K), and a maximum hot-cold fluid temperature difference of approximately 14 and 6.2 °C at a flow rate of 1.8 and 9 L/min, respectively. The observed deviations between experimental results and theoretical

correlations can be investigated in three factors. First and most important, the modified fin integrated baffle geometry is not covered in the assumptions underlying the applied correlations, which were developed for fully developed flows in smooth and plain channels and ideal cross-flow configurations, even for plain tubes. Second, the test equipment is relatively short and creates an entrance effect in the flow that cannot be captured for developing flow, so the result measured values differ from the predicted ones. Third, every measuring instrument has uncertainty that propagates into the derived Nu and f , further widening the difference between the measured and predicted values. For these reasons, the correlations are positioned in this study as validation references rather than predictive tools for the fin-integrated configuration, and the experimental data for the modified geometry are reported directly [17].

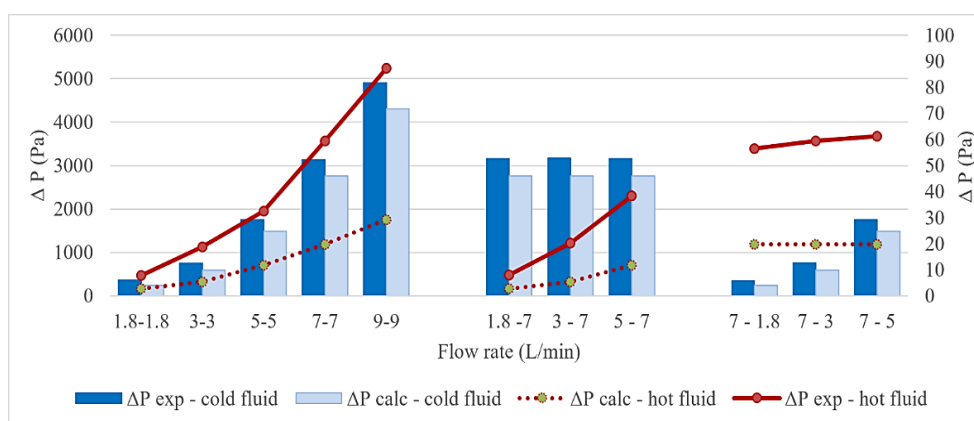


Figure 11. ΔP of hot and cold fluid at all flow rates

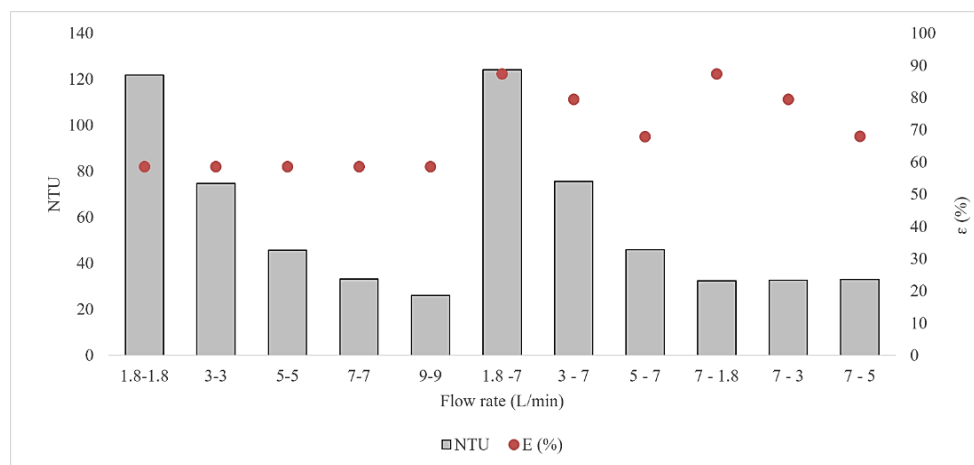


Figure 12. NTU - ϵ at all flow rates

Figure 11 shows the pressure drop (ΔP) of experimental results and calculations in the hot and cold fluids [26]. The result of pressure drops that are calculated using the Kern method are lower than the experimental methods for both hot and cold fluids. The difference of pressure drops for calculation and experimentally for hot and cold fluids was caused by leakage and fouling inside the tube and shell. The lowest and highest ΔP experimentally occurred at both flow rate 1.8 and 9 L/min of 7.8 Pa and 87.3 Pa for hot fluid, and 351.7 Pa and 4908.6 Pa for cold fluid, respectively. Although the addition of fins improves heat transfer performance, it also influences the hydraulic characteristics of the STHX. The pressure drop on both the shell and tube sides was theoretically

calculated and compared experimentally. The experimental results indicate that the shell-side pressure drop increased slightly due to the presence of fins, which create additional flow resistance. However, the increase in pressure drop remained within an acceptable range for the operation of the STHX. The both shell and tube-side pressure drop equations follow the classical Kern model, which combines frictional effects with direction-change contributions due to baffles. The presence of the factor $(N + 1)$ in the equation indicates that pressure drop scales proportionally with the number of flow traversals induced by the baffles. It is important to note that increasing the number of baffles to enhance the heat transfer coefficient always incurs a proportional increase in pressure

drop [27]. The STHX design employs a baffle cut of 20% and spacing of 100 mm, a small to conservative configuration. The results show that the theoretical calculation consistently underestimates pressure drop relative to experimental values. For the tube side, the discrepancies of the lowest and highest ΔP investigated at both flow rates of 1.8 and 9 L/min are 2.74 Pa and 29.3 Pa for hot fluid, and 244.5 Pa and 4306.3 Pa for cold fluid, respectively. For the cold fluid, the differences are larger. This is characteristic of the Kern method's tendency to underestimate pressure drop and overestimate heat transfer, particularly for STHXs with unconventional baffle modifications [22].

Figure 12 shows the NTU and effectiveness (ϵ) of the system. The highest effectiveness occurred at a flow rate with variation 1.8-7 L/min of 87.33%. The effectiveness for both investigated flow rates averaged 58.59%. The addition of fins measuring 105 mm $\times$ 25 mm $\times$ 1.5 mm, mounted between tubes, is intended to address the key weaknesses of conventional segmental baffles, namely the formation of dead zones and recirculation zones behind the baffle, as well as leakage streams through baffle-tube clearance [27]. Recent review evidence indicates that baffle modifications using novel geometries, including fins, consistently reduce dead-zone area and offer superior thermohydraulic trade-offs even though they increase the pressure drop [28].

Figure 13 shows that the heat capacity rate ratio between the hot and cold fluid streams significantly dictates the temperature effectiveness, where disparities in these values impose a thermal ceiling on the system's asymptotic performance limits. The configurations of tubes with fins consistently demonstrate superior performance compared to uniform distributions, suggesting that strategic flow path manipulation serves as a primary driver for optimizing the efficiency.

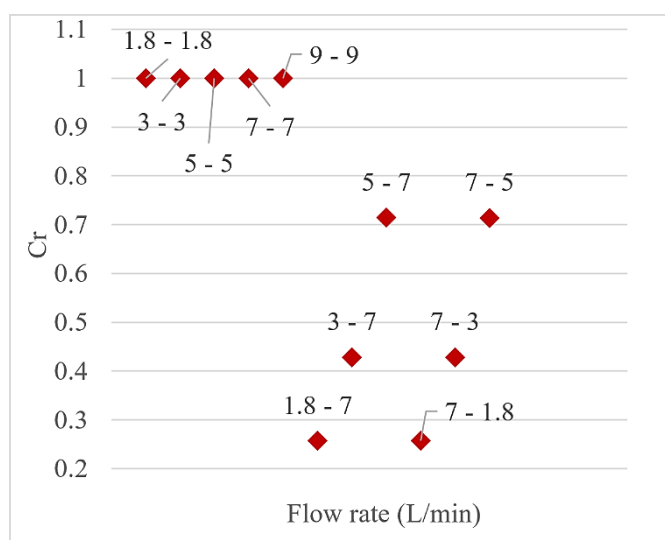


Figure 13. Heat capacity rate ratio (C_r) of the hot and cold fluids at all investigated flow rates

4. CONCLUSIONS

This study investigates the effect and characteristics of the finned baffle modification on the small STHX. As a result, the pressure drop increased with increasing volumetric flow rates. The results about the baffle-modified fin can be considered for industrial applications. The model of the baffle-modified fin

can also be recommended with some modifications to decrease the pressure drop significantly, so the parameters and variations should be manufactured for the best results of the pressure drop limit of the STHX. Numerical design can be used to predict the next. The highest and lowest heat (ΔQ) differences are investigated at flow rate variations of 7-1.8 L/min of 268.5 W and 3-7 L/min of 15.1 W, respectively. The coefficients of convective heat transfer in the tube-side and shell-side are shown to be between 279.44 and 330.85 W/(m²·K) and 24.02 and 45.58 W/(m²·K), respectively. The difference in maximum temperature between hot and cold fluids for the flow rates 1.8, 3, 5, 7, and 9 L/min was indicated at 1.8 L/min as 14 °C (hot fluid) and 6.2 °C (cold fluid). The lowest and highest ΔP experimentally occurred at both flow rates of 1.8 and 9 L/min of 7.8 Pa and 87.3 Pa for hot fluid, and 351.7 Pa and 4908.6 Pa for cold fluid, respectively. The highest effectiveness occurred at the flow rate with variation 1.8-7 L/min of 87.33%. The effectiveness for both at the same flow rates averaged 58.59%.

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NOMENCLATURE

Q	Total heat load
Q_{hot}	Heat load of hot fluid
Q_{cold}	Heat load of cold fluid
$\dot{m}$	Mass flow rate [kg/s]
C_c	Specific heat capacity of cold fluid [kJ/(kg·K)]
C_h	Specific heat capacity of hot fluid [kJ/(kg·K)]
$T_{c,out}$	Temperature of cold fluid outlet
$T_{c,in}$	Temperature of cold fluid inlet
$T_{h,in}$	Temperature of hot fluid inlet
$T_{h,out}$	Temperature of hot fluid outlet
U	Total convective heat transfer coefficient [W/(m ² ·K)]
A	Total heat transfer area [m ²]
$LMTD$	Logarithmic mean temperature difference
d_{shell}	Inner diameter of shell [m]
d_e	Shell diameter for triangular pitch [m]
d_{intube}	Inner tube diameter [m]
$d_{outtube}$	Outer tube diameter [m]
L_{tube}	Tube length [m]
N_{tube}	Number of tubes
R_{th}	Fouling resistance or thermal energy resistance

h_h	Convective heat transfer coefficient of hot fluid [W/(m ² ·K)]	μ_w	Dynamic viscosity of the tube-wall [kg/(m·s)]
h_c	Convective heat transfer coefficient of cold fluid [W/(m ² ·K)]	P_T	Tube pitch [m]
k	Thermal conductivity of tube [W/(m·K)]	Q_s	Cold fluid volumetric flow rate [kg/(m ² ·s)]
A_{intube}	Inner heat transfer area of tube [m ²]	Q_t	Hot fluid volumetric flow rate [kg/(m ² ·s)]
$A_{outtube}$	Outer heat transfer area of tube [m ²]	ΔT	Temperature difference between two fluids
Nu_{hot}	Nusselt number of hot fluid	ε	Effectiveness
Nu_{cold}	Nusselt number of cold fluid	NTU	number of transfer unit
Re	Reynolds number	B	distance between baffles [m]
Re_{hot}	Reynolds number of hot fluid	C	distance between tubes [m]
Re_{cold}	Reynolds number of cold fluid	s	specific gravity
ρ	Density [kg/m ³]	V	Velocity of fluid [m/s]
μ	Dynamic viscosity of the fluid [kg/(m·s)]	g	Acceleration of gravity [m/s ²]
		ΔP_{hot}	Pressure drop of hot fluid [Pa]
		ΔP_{cold}	Pressure drop of cold fluid [Pa]