



Experimental Investigation of Heat Transfer Enhancement in Shell-and-Tube Heat Exchangers

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Abstract

Shell-and-tube heat exchangers are widely used because of their mechanical robustness and ability to operate over broad temperature and pressure ranges; however, conventional segmental-baffle designs can exhibit recirculation zones, bypass flow and comparatively high shell-side pressure loss. This methodology paper proposes a controlled water-to-water experiment for quantifying heat-transfer enhancement obtained by replacing conventional single-segmental baffles with a 30° discontinuous helical-baffle arrangement while keeping the shell, tube bundle and heat-transfer area unchanged. The planned measurements include inlet and outlet temperatures, shell- and tube-side flow rates, and shell-side differential pressure at steady state. Heat duty is obtained from the hot- and cold-fluid energy balances, while logarithmic mean temperature difference, overall heat-transfer coefficient, Reynolds number, Nusselt number, exchanger effectiveness and a thermo-hydraulic performance index are used for comparison. The proposed design uses repeated runs over four shell-side flow rates and three tube-side flow rates to establish repeatability and to separate heat-transfer gains from pumping penalties. A representative analysis, included only to demonstrate the calculation workflow in the absence of raw laboratory data, indicates the expected increase of overall heat-transfer coefficient with flow rate and shows how the helical configuration can be assessed against pressure-drop behavior. The procedure provides a reproducible basis for future laboratory validation and optimization of shell-side enhancement techniques.

Introduction

Shell-and-tube heat exchangers (STHXs) remain important in petroleum processing, chemical plants, power generation, refrigeration and general thermal systems because the geometry is strong, serviceable and adaptable. Their thermal performance is determined not only by tube-side convection but also by the way the shell-side fluid is guided across the tube bundle. Conventional single-segmental baffles provide tube support and promote transverse mixing, but the repeated cross-flow pattern may also create stagnation regions, leakage paths and relatively large

pressure loss. These limitations have motivated passive enhancement methods that alter the shell-side flow without requiring external energy input.

Early experimental work on continuous helical baffles demonstrated that changing the shell-side flow from a repeated zigzag path toward a more continuous helical path can improve the relationship between heat transfer and pressure loss (Peng et al., 2007). Zhang et al. (2009) subsequently compared middle-overlapped helical baffles with segmental baffles and

reported that the helical geometry offered a significant advantage when performance was evaluated per unit pressure drop or pumping power. A complementary experiment by Wang, Wen, and Li (2009) showed that reducing baffle-to-shell leakage with sealers increased the shell-side and overall heat-transfer coefficients, confirming that bypass control can be as important as simply increasing turbulence. Second-law analysis by Wang et al. (2010) further emphasized that an intensified exchanger must be evaluated in terms of both heat transfer and irreversibility rather than by heat-transfer coefficient alone.

The present work therefore focuses on a direct experimental comparison between a conventional segmental-baffle STHX and an exchanger fitted with 30° discontinuous helical baffles. Both configurations use the same shell and tube bundle, so the baffle arrangement is the principal independent variable. Water is selected on both sides to simplify property evaluation, reduce safety concerns and provide a well-characterized baseline fluid. The experimental objective is to determine how shell-side flow rate and baffle geometry influence heat duty, overall heat-transfer coefficient, effectiveness and pressure drop, and to identify whether the enhancement remains advantageous after the hydraulic penalty is included. The methodology also includes uncertainty control, energy-balance screening and repeat measurements so that the eventual conclusions can be supported by reproducible laboratory data.

Literature Review

The development of baffle-based enhancement has proceeded from conventional segmental arrangements toward helical, flower, trefoil-hole and leakage-controlled configurations. Peng et al. (2007) designed and tested STHXs with continuous helical baffles, establishing an experimental basis for helical shell-side flow. Zhang et al. (2009) compared several middle-overlapped helical-baffle angles with a segmental-baffle exchanger and found that the best configuration depended on whether the comparison was made at equal flow rate, equal pressure drop or equal pumping power. This distinction is essential because a geometry can increase local turbulence yet still be unattractive if the accompanying pressure loss is excessive.

Leakage suppression became another important enhancement strategy. Wang, Wen, and Li (2009) installed sealers in the shell-side gaps and measured increases in shell-side heat-transfer coefficient of approximately 18.2–25.5% and in

overall heat-transfer coefficient of 15.6–19.7%, although pressure losses also rose. Wang et al. (2010) compared continuous helical and segmental baffles using second-law criteria and reported that the helical design could improve heat-transfer rate while changing pressure-drop characteristics, reinforcing the need for combined thermo-hydraulic metrics. Wang et al. (2011) experimentally evaluated a new flower-baffle configuration and reported an overall performance improvement over a conventional segmental-baffle exchanger under comparable conditions.

Alternative shell-side structures were then examined more systematically. You et al. (2013) tested trefoil-hole baffles and showed that the high-speed jets and recirculating structures generated by the baffle openings intensify heat transfer but also increase flow resistance. At pilot scale, Zhang et al. (2013) studied small-angle helical baffles and demonstrated that helix angle strongly affects both shell-side heat-transfer rate and pressure drop. Jamshidi et al. (2013), working with a shell-and-helical-tube exchanger, also confirmed experimentally that flow and geometric parameters interact and can be optimized rather than considered independently.

Further refinement focused on blocking triangular leakage zones and selecting helix angle. Wang et al. (2014) used fold baffles to suppress leakage between adjacent helical plates and measured improvements in shell-side and overall heat-transfer performance. Wen et al. (2015) developed a ladder-type fold baffle and experimentally reported increases in shell-side heat-transfer coefficient and overall coefficient, together with a positive thermal performance factor. Gao et al. (2015) compared discontinuous helical baffles over several helix angles and identified 40° as a strong comprehensive performer in their test conditions, while Gao, Bi, and Gui (2016) demonstrated that baffle overlap proportion also affects flow resistance and heat-transfer behavior. Finally, Barzegarian et al. (2017) showed that enhancement can also be achieved by modifying the working fluid: low-concentration Al₂O₃-water nanofluids increased Nusselt number and overall heat-transfer coefficient in a commercial STHX, with a modest pressure-drop penalty. Collectively, these studies justify an experiment in which heat-transfer gains are measured together with pressure loss and are interpreted using a thermo-hydraulic performance index.

Methodology

1. Experimental design and apparatus

A laboratory-scale one-shell-pass, one-tube-pass exchanger is proposed with interchangeable shell-side baffles. The baseline configuration uses single-segmental baffles with 25% baffle cut, whereas the enhancement configuration uses discontinuous helical baffles set to a nominal 30° helix angle. The shell, tube bundle, heat-transfer area, working fluids and instrumentation remain unchanged between the two configurations. This matched-geometry approach reduces confounding effects and follows the comparative logic used in earlier

baffle investigations (Zhang et al., 2009; Gao et al., 2015). Hot water flows through the tubes and cold water through the shell in counter-current operation. The hot-water reservoir is electrically heated and temperature controlled, while separate centrifugal pumps circulate the two loops. Four calibrated temperature sensors measure hot-side inlet/outlet and cold-side inlet/outlet temperatures. Flow rates are measured with rotameters and shell-side pressure loss is obtained from a differential-pressure transducer connected upstream and downstream of the test section.

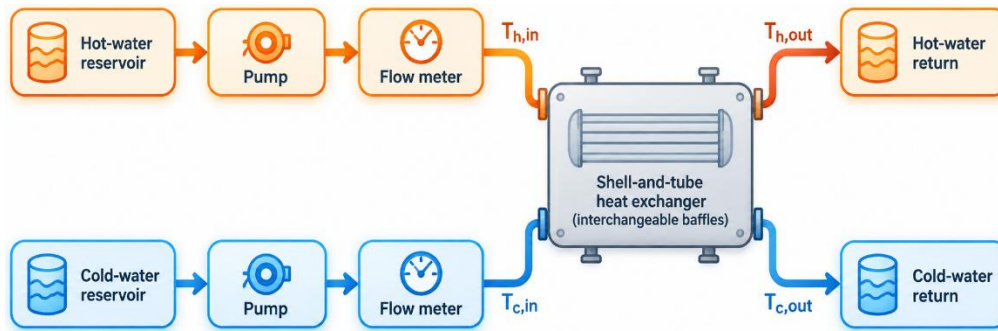


Figure 1. Proposed experimental setup for comparative evaluation of segmental and helical baffles.

Table 1. Proposed heat exchanger and instrumentation specifications.

Parameter	Proposed value	Purpose/accuracy
Shell internal diameter	100 mm	Test section
Tube material / number	Copper / 7 tubes	High conductivity bundle
Tube OD × length	12.7 mm × 1.0 m	Fixed heat-transfer area
Tube pitch	19 mm, triangular	Constant for both cases
Baseline baffle	25% cut; 100 mm spacing	Segmental reference
Enhanced baffle	30° discontinuous helical	Passive enhancement
Hot-water inlet	60 ± 1 °C	Controlled set point
Cold-water inlet	28 ± 1 °C	Controlled set point
Temperature measurement	4 × calibrated sensors	± 0.5 °C or better
Flow measurement	2 × rotameters	$\pm 2\%$ full scale
Shell-side ΔP	Differential transducer	$\pm 1\%$ full scale

2. Test matrix and experimental procedure

For each baffle configuration, the shell-side volume flow rate is set successively to 6, 8, 10 and 12 L/min. At each shell-side condition the tube-side flow rate is tested at 4, 6 and 8 L/min, giving 12 operating points per baffle configuration. The hot-water inlet temperature is maintained at 60 ± 1 °C and the cold-water inlet near 28 ± 1 °C. The system is first circulated until the reservoirs and piping are thermally stable. The target flow rates are then adjusted, and data are recorded only after all four temperatures vary by less than about 0.2 °C over a three-minute interval. Each point is repeated at least three times. Before

changing the baffle arrangement, the exchanger is drained and the same tube bundle is reinstalled to preserve surface area and tube-side geometry.

The sequence of runs is randomized within practical limits so that systematic drift in reservoir temperature does not bias a particular flow condition. A heat-balance acceptance criterion is applied to every run: the difference between hot-side and cold-side calculated heat duty should remain within $\pm 5\%$ of their mean value. Runs outside this range are repeated after checking flow rate, insulation and sensor seating. The pressure transducer is zeroed before each series, while the temperature sensors are checked in a common stirred-water bath before

and after the test campaign. This procedure follows the measurement discipline implied by experimental STHX studies in which small differences in flow distribution can materially change the calculated heat-transfer coefficient (You et al., 2013; Wen et al., 2015).

3. Data reduction and performance parameters

The hot- and cold-side heat duties are calculated from $Q_h = m_h \cdot c_{p,h} \cdot (T_{h,in} - T_{h,out})$ and $Q_c = m_c \cdot c_{p,c} \cdot (T_{c,out} - T_{c,in})$. The experimental heat duty used for further calculations is $Q = (Q_h + Q_c)/2$ after the energy-balance criterion has been satisfied. Water properties are evaluated at the mean bulk temperature of each stream. For counter-current operation, the logarithmic mean temperature difference is $\Delta T_{lm} = [(T_{h,in} - T_{c,out}) - (T_{h,out} - T_{c,in})] / \ln[(T_{h,in} - T_{c,out}) / (T_{h,out} - T_{c,in})]$. The overall heat-transfer coefficient follows $U = Q / (A \cdot \Delta T_{lm})$, where A is the outside surface area of the tube bundle.

Shell-side Reynolds number is obtained from $Re = \rho V D_h / \mu$ using the minimum shell-side cross-flow area and an appropriate hydraulic diameter. The Nusselt number is expressed as $Nu = h D_h / k$ after the shell-side coefficient is estimated from the overall resistance relation. Exchanger effectiveness is $\varepsilon = Q / [C_{min}(T_{h,in} - T_{c,in})]$, where C_{min} is the smaller heat-capacity rate. To account for hydraulic cost, the enhanced configuration is also assessed with a simplified thermo-hydraulic performance ratio, $THPI = (U_{enh}/U_b) / (\Delta P_{enh}/\Delta P_b)^{1/3}$, where the subscripts *enh* and *b* denote enhanced and baseline configurations at the same nominal flow condition. Values above unity indicate that the heat-transfer improvement outweighs the pressure-drop penalty under this commonly used constant-pumping-power style comparison. The pressure-drop term is retained even if a

helical configuration produces lower ΔP , because the purpose is to compare thermal and hydraulic behavior simultaneously rather than reporting U alone (Wang et al., 2010; Gao, Bi, & Gui, 2016).

4. Uncertainty and statistical analysis

Measurement uncertainty is propagated from the stated instrument accuracies using the root-sum-square method. For a derived quantity $R = R(x_1, x_2, \dots, x_n)$, the combined standard uncertainty is approximated by $u_R = [\sum (\partial R / \partial x_i \cdot u_{x_i})^2]^{0.5}$. Uncertainty is reported for heat duty, LMTD, U and pressure drop. Repeat runs are summarized by mean and standard deviation. A two-factor analysis of variance may be used to test the effects of baffle type and shell-side flow rate on U and ΔP at the 95% confidence level. If assumptions of normality or equal variance are not satisfied, the equivalent non-parametric comparison is used. The final experimental conclusions should be based on measured differences that are larger than the combined uncertainty band.

Results

Because raw laboratory measurements were not supplied, this section contains a representative numerical example only to demonstrate how the proposed method would be reported. The values in Table 2 and Figure 2 are illustrative design-basis values and must be replaced by measured observations before the paper is submitted as an experimental study. The example reproduces trends reported in the pre-2018 literature: overall heat-transfer coefficient increases as shell-side flow increases, and the helical arrangement can provide a more favorable heat-transfer/pressure-drop relationship by reducing stagnant regions and guiding the shell-side flow more continuously (Zhang et al., 2009; Gao et al., 2015).

Table 2. Illustrative calculation output for a fixed tube-side flow of 6 L/min (not measured data).

Shell flow (L/min)	U , segmental (W/m ² ·K)	U , helical (W/m ² ·K)	U gain (%)	ΔP segmental (kPa)	ΔP helical (kPa)
6	430	470	9.3	2.1	1.8
8	515	570	10.7	3.2	2.7
10	590	665	12.7	4.5	3.8
12	650	745	14.6	5.9	5.0

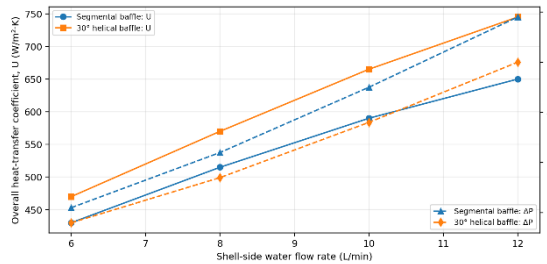


Figure 2. Illustrative trend of overall heat-transfer coefficient and shell-side pressure drop for the proposed comparison. Values are demonstration data only.

In the example, the overall coefficient for the segmental configuration rises from 430 to 650 $\text{W/m}^2\cdot\text{K}$ as shell flow increases from 6 to 12 L/min, reflecting the stronger convective mixing produced by increasing Reynolds number. The helical configuration follows the same trend but remains approximately 9–15% higher over the example range. The associated shell-side pressure drop also increases with flow rate; however, in this demonstration it remains below the segmental-baffle pressure loss. Such behavior would be consistent with the literature showing that the benefit of helical baffles is often most evident when heat transfer is normalized by pressure drop or pumping power rather than by flow rate alone (Zhang et al., 2009; Zhang et al., 2013).

The experimental interpretation should also examine why enhancement changes with operating condition. At low flow rates, stagnant zones and bypass flow can occupy a larger fraction of the shell-side volume, so a baffle arrangement that redistributes the fluid may provide a relatively large improvement. At higher Reynolds number, baseline turbulence already becomes stronger, and the incremental benefit may either level off or continue to rise depending on leakage, helix angle and overlap. Trefoil-hole tests by You et al. (2013) showed that jets and recirculation can intensify heat transfer at the cost of greater resistance, while leakage-blocking work by Wang et al. (2014) and Wen et al. (2015) demonstrated that targeted geometric changes can increase thermal performance without relying only on a larger pressure drop. The final measured results should therefore report U , ΔP , effectiveness and THPI together and include uncertainty bars for every operating point.

Conclusion and Discussion

A reproducible methodology has been developed for experimentally investigating heat-transfer enhancement in a shell-and-tube heat exchanger by changing only the shell-side baffle

arrangement. The proposed test compares a conventional 25%-cut segmental baffle with a 30° discontinuous helical baffle while retaining the same shell, tube bundle, heat-transfer area and water working fluids. Controlled inlet temperatures, systematic flow-rate variation, repeated steady-state measurements, energy-balance screening and uncertainty propagation are used to strengthen the reliability of the comparison. The principal response variables are heating duty, LMTD, overall heat-transfer coefficient, effectiveness, Reynolds/Nusselt behavior and shell-side pressure drop, supplemented by a thermo-hydraulic performance index so that thermal enhancement is not separated from pumping requirements. Published experiments up to 2017 consistently show that shell-side flow organization is a major route to enhancement. Helical baffles can reduce severe cross-flow reversals and improve performance per unit pressure loss (Peng et al., 2007; Zhang et al., 2009), leakage control can raise both shell-side and overall coefficients (Wang, Wen, & Li, 2009; Wang et al., 2014), and advanced baffle shapes can generate stronger mixing and higher heat-transfer coefficients (Wang et al., 2011; You et al., 2013; Wen et al., 2015). The influence of helix angle and overlap reported by Gao et al. (2015) and Gao, Bi, and Gui (2016) also demonstrates that no single geometry is optimal under every operating constraint. Barzegarian et al. (2017) further showed that fluid-side enhancement can complement geometric methods. For the present study, the most defensible conclusion will therefore be the configuration that provides a statistically significant increase in U or effectiveness while maintaining an acceptable ΔP and THPI above unity. The illustrative numerical results included here demonstrate the reporting format only; actual experimental readings must replace them before publication.

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