



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

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Abstract

Shell-and-tube heat exchangers (STHEs) are widely used because they tolerate high pressure, offer large heat-transfer area, and can be maintained with established industrial practices. Their thermal performance, however, is strongly influenced by shell-side bypassing, dead zones, baffle geometry, flow rate, and pressure-drop limitations. This methodology paper develops an experimentally implementable procedure for quantifying heat-transfer enhancement in a laboratory STHE by comparing a conventional segmental-baffle arrangement with an enhanced helical-baffle arrangement under matched inlet temperatures and flow conditions. Hot water is circulated through the shell and cold water through the tube bundle in counter-flow operation. Inlet/outlet temperatures, volumetric or mass flow rates, and shell-side pressure drop are measured after steady state is achieved. Heat duty, logarithmic mean temperature difference (LMTD), overall heat-transfer coefficient, effectiveness, Reynolds number, and a thermo-hydraulic performance index are calculated from the measurements. Uncertainty propagation and an energy-balance acceptance criterion are included to strengthen reproducibility. Prior experimental studies demonstrate that baffle configuration and nanofluid use can increase heat-transfer performance, but the gain must be evaluated together with pumping penalty [1–12]. The proposed method isolates the effect of baffle-induced flow redistribution and provides a repeatable framework for evaluating enhancement without falsely treating literature trends as new measurements. An illustrative results table is included only to demonstrate the analysis and reporting procedure and must be replaced by actual laboratory readings in a final experimental article.

Introduction

Heat exchangers are central to power generation, petroleum refining, chemical processing, refrigeration, food processing, and waste-heat recovery. Among available configurations, the shell-and-tube heat exchanger remains especially important because its mechanical design can be adapted to severe temperature and pressure conditions. The same flexibility creates a complicated shell-side flow field containing cross-flow, leakage, recirculation, and bundle-

bypass streams. Consequently, thermal design cannot be based on heat-transfer area alone; baffle spacing, baffle cut, tube arrangement, fluid velocity, and allowable pressure loss must be considered simultaneously.

Classical experimental work showed that baffle spacing changes both local heat transfer and pressure drop, emphasizing that an optimum configuration is a compromise rather than simply the arrangement producing the highest velocity

[1]. Eryener [2] extended this concept to thermoeconomic optimization and showed why baffle spacing should be selected with both capital and operating costs in mind. Later experimental comparisons of helical and segmental baffles reported improved heat-transfer performance per unit pressure drop for helical designs [3,7]. These findings motivate an experimental comparison in which thermal gain is normalized against hydraulic penalty.

A second enhancement route is to increase the thermal transport properties of the working fluid. Experiments with Al₂O₃/water, TiO₂/water, carbon-nanotube/water, silver/water, Fe₂O₃-based, and Al₂O₃-based nanofluids have generally reported higher convective heat-transfer coefficients or effectiveness than the corresponding base fluids [4,6,9–12]. Nevertheless, particle concentration, viscosity, stability, and pumping power complicate direct comparison. For a methodology paper intended to isolate geometric enhancement, the present design therefore uses water on both sides and changes only the baffle configuration. This removes nanoparticle stability as a confounding variable while retaining a pathway for a later second-stage nanofluid study.

The objective is to establish a repeatable experimental method to determine whether an enhanced shell-side baffle arrangement raises heat duty, overall heat-transfer coefficient, and effectiveness at comparable operating conditions, while controlling or reducing pressure-drop penalty. The study is structured around paired tests, uncertainty-aware calculations, and a thermo-hydraulic performance criterion. The method is appropriate for a 1-shell-pass/2-tube-pass laboratory exchanger and can be scaled provided geometrical ratios and instrumentation accuracy are documented.

Literature Review

Research before 2019 identifies baffle design and working-fluid modification as the two dominant routes for STHE enhancement. Li and Kottke [1] experimentally linked baffle spacing with local

heat transfer and pressure drop for staggered tube arrangements. Zhang et al. [3] experimentally compared middle-overlapped helical baffles with segmental baffles and reported favorable shell-side performance for the helical concept. Wang et al. [5] fabricated a new baffle design and reported an overall performance improvement of approximately 20–30% compared with a conventional segmental-baffle exchanger under their test conditions. In practical oil coolers, Zhang et al. [7] likewise found lower shell-side pressure drop and a higher heat-transfer coefficient per unit pressure drop for overlapped helical baffles.

The influence of baffle number was examined experimentally by Vukić et al. [8], who measured temperature and pressure distributions in a 1–2U exchanger. Their results showed that increasing the number of segmental baffles improved exchanger effectiveness, but also increased shell-side pressure loss. Such observations support the use of a combined thermal-hydraulic metric rather than heat-transfer coefficient alone. This is especially important because aggressive flow redirection can create a large pumping requirement that offsets the benefit of higher turbulence.

Nanofluid studies provide complementary evidence that the working fluid can be enhanced. Farajollahi et al. [4] investigated Al₂O₃/water and TiO₂/water nanofluids in an STHE, while Lotfi et al. [6] studied MWNT-water nanofluid. Godson et al. [9] reported maximum increases of 12.4% in convective heat-transfer coefficient and 6.14% in effectiveness for silver/water nanofluid under their conditions. Elias et al. [10] showed that nanoparticle shape and baffle angle interact, and Kumar and Sonawane [11] demonstrated heat-transfer enhancement using Fe₂O₃/water and Fe₂O₃/ethylene-glycol nanofluids. Barzegarian et al. [12] later reported thermal augmentation using Al₂O₃-gamma/water nanofluid in a horizontal STHE. Collectively, these studies confirm that enhancement claims should specify geometry, flow regime, pressure loss, fluid properties, and measurement uncertainty.

Table 1. Selected pre-2019 studies used to define the methodology

Study	Method	Key variable	Methodological implication
[1] Li & Kottke, 1998	Experimental	Baffle spacing	Measure heat transfer and ΔP together.
[3] Zhang et al., 2009	Experimental	Helical vs segmental baffles	Use matched flow conditions.
[5] Wang et al., 2011	Experimental	New baffle geometry	Compare overall performance, not U alone.

[8] Vukić et al., 2014	Experimental	Baffle number/cut	Document shell-side geometry and ΔP .
[9] Godson et al., 2014	Experimental	Ag/water nanofluid	Control concentration and inlet conditions.
[11] Kumar & Sonawane, 2016	Experimental	Fe ₂ O ₃ nanofluids	Account for viscosity/pumping effects.
[12] Barzegarian et al., 2017	Experimental	Al ₂ O ₃ -water nanofluid	Use steady-state, repeated thermal tests.

Methodology

1. Experimental design and test section

A bench-scale 1-shell-pass/2-tube-pass exchanger is proposed. The shell is stainless steel with an internal diameter of approximately 100 mm and an effective heat-transfer length of 1.0 m. Seven straight copper tubes of 12.7 mm outside diameter and approximately 10.5 mm inside diameter are arranged on a triangular pitch, giving an external heat-transfer area close to 0.30 m². Two removable shell-side baffle cartridges are fabricated so that the tube bundle and all other geometric quantities remain unchanged. Configuration A uses conventional single-segmental baffles with a 25% baffle cut and constant spacing. Configuration B uses an enhanced helical or inclined-baffle arrangement selected to produce a continuous shell-side flow path. The comparison is therefore paired: tube

material, tube count, heat-transfer area, shell diameter, fluid, and inlet temperatures are identical, while the shell-side flow-guiding geometry is the controlled independent variable.

Hot water flows through the shell and cold water through the tubes in counter-flow operation. A heater and insulated reservoir maintain the hot inlet temperature at 60 ± 0.5 °C, while a second reservoir or cooling loop maintains the cold inlet at 25 ± 0.5 °C. The cold-side mass flow rate is held at 0.18 kg/s. The hot-side mass flow rate is varied through at least four levels (0.08, 0.10, 0.12, and 0.14 kg/s). Each condition is tested for both baffle configurations, and a minimum of three repeat runs is recommended. Randomizing the order of configurations reduces drift caused by ambient conditions, heater cycling, or progressive warming of the cooling reservoir.

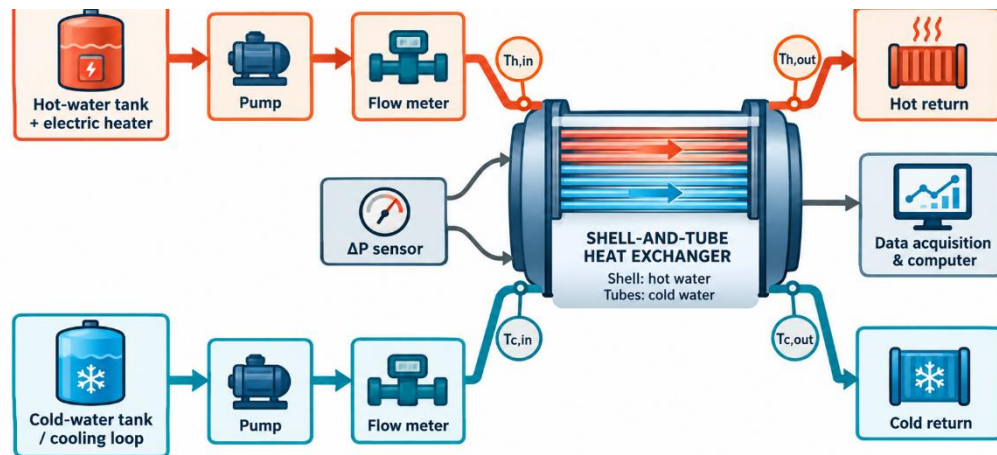


Figure 1. Proposed experimental setup for paired baffle-configuration testing.

2. Instrumentation and data acquisition

Four calibrated Type-K thermocouples or PT100 sensors measure hot and cold inlet/outlet temperatures ($T_{h,in}$, $T_{h,out}$, $T_{c,in}$, and $T_{c,out}$). Additional shell-wall or inter-baffle temperature sensors may be installed if local thermal behavior is required. Flow is measured with calibrated rotameters, turbine meters, or Coriolis meters; for a laboratory study, an accuracy within ± 1 – 2% of reading is desirable. A differential-pressure transducer measures the shell-side pressure drop between inlet and outlet pressure taps. All

signals are recorded by a data-acquisition system at 1 Hz or faster and averaged over a stable 3–5 min interval.

Before testing, temperature sensors are checked at an ice-point and a warm-water reference. Flow meters are verified gravimetrically over the expected range. Pressure zero is checked with both ports at equal pressure. The exchanger and piping are insulated to reduce heat loss. A data point is accepted as steady state when all four terminal temperatures vary by less than ± 0.2

°Cover five minutes and the hot- and cold-side heat duties agree within 5%.

Test procedure — step by step

1. Install the segmental-baffle cartridge, inspect all seals, fill both loops with clean water, remove trapped air, and start the pumps at low flow.
2. Set the cold-side mass flow rate to 0.18 kg/s and stabilize the cold inlet near 25 °C. Switch on the heater and stabilize the hot inlet near 60 °C.
3. Set the first hot-side mass flow rate (0.08 kg/s). Allow the exchanger to reach the steady-state criterion; then record temperatures, flow rates, and shell-side pressure drop for at least three minutes.
4. Repeat the measurement twice more at the same condition, then repeat at 0.10, 0.12, and 0.14 kg/s. Shut down safely and allow temperatures to normalize.
5. Replace only the shell-side baffle cartridge with the enhanced helical/inclined configuration. Repeat the identical test matrix and the same stabilization criteria.
6. Process each run independently, reject runs failing the energy-balance criterion, calculate mean $\pm$ standard deviation for repeats, and compare configurations at matched flow rate.

3. Data reduction

For water, temperature-dependent properties should be evaluated at the mean bulk temperature of each stream. The heat transferred from the hot stream and gained by the cold stream are calculated as:

$$\dot{Q}_h = \dot{m}_h c_{p,h} (T_{h,in} - T_{h,out}) \quad \text{and} \quad \dot{Q}_c = \dot{m}_c c_{p,c} (T_{c,out} - T_{c,in})$$

The reported heat duty is the arithmetic mean, $\bar{Q} = (\dot{Q}_h + \dot{Q}_c)/2$, after verifying that $|\dot{Q}_h - \dot{Q}_c|/\bar{Q} \times 100 \leq 5\%$. For counter flow, 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 based on the outside tube area is $U = \dot{Q}/(A_o \Delta T_{lm})$. Heat-exchanger effectiveness is $\varepsilon = \dot{Q}/[C_{min}(T_{h,in} - T_{c,in})]$, where C_{min} is the smaller heat-capacity rate. Reynolds number is calculated from the hydraulic diameter and mean velocity on the corresponding side. To compare enhancement with pressure-drop penalty, a performance evaluation criterion (PEC) may be defined as $(U_{enh}/U_{seg})/(\Delta P_{enh}/\Delta P_{seg})^{(1/3)}$. A value greater than unity indicates a favorable thermal-hydraulic trade-off for the enhanced arrangement.

4. Uncertainty and statistical treatment

Measurement uncertainty is propagated using the root-sum-square method. If $\dot{Q} = f(\dot{m}, c_p, T_{in}, T_{out})$, the combined standard uncertainty is estimated from the partial derivatives of $\dot{Q}$ with respect to each measured variable. The same approach is applied to U and effectiveness. With typical uncertainties of ± 0.2 °C for temperature, $\pm 1.5\%$ for flow, $\pm 1\%$ for pressure, and $\pm 1\%$ for effective area, the final uncertainty in U is expected to be on the order of several percent; the exact value must be recalculated from instrument certificates and observed temperature differences. For repeated tests, report mean, standard deviation, and 95% confidence intervals. A paired t-test or two-way ANOVA (baffle type $\times$ flow rate) can be applied if replicate count is sufficient and assumptions are checked.

Results

Because no raw laboratory measurements were supplied with this methodology request, the numerical values in Table 2 and Figure 2 are illustrative only. They demonstrate how a completed experimental dataset can be reduced and compared; they are not presented as measured findings and should be replaced by actual observations before submission as an experimental research paper.

Table 2. Illustrative analysis format (hypothetical steady-state data; replace with measured values)

Baffle	$\dot{m}_h$ (kg/s)	$T_{h,out}$ (°C)	$T_{c,out}$ (°C)	$\dot{Q}$ (kW)	U (W/m ² K)	ΔP (kPa)
Segmental	0.08	49.5	29.7	3.50	427	4.1
Segmental	0.10	50.1	30.5	4.15	508	6.3
Segmental	0.12	50.7	31.2	4.65	569	8.9
Segmental	0.14	51.4	31.7	5.05	616	11.9
Enhanced helical	0.08	47.9	30.4	4.05	517	3.3
Enhanced helical	0.10	48.5	31.4	4.82	619	5.0
Enhanced helical	0.12	49.2	32.2	5.43	698	7.1

Enhanced helical	0.14	49.9	32.9	5.93	761	9.7
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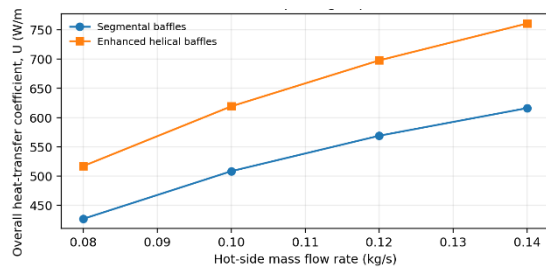


Figure 2. Illustrative reporting trend; values are hypothetical and not experimental measurements.

The expected experimental trend is an increase in heat duty and U with increasing hot-side flow rate because higher velocity increases turbulence and reduces the convective thermal resistance. At a matched mass flow rate, a well-designed helical or inclined baffle is expected to produce more uniform longitudinal/cross-flow mixing and fewer dead zones than a conventional segmental arrangement. Earlier experimental comparisons support this expectation [3,5,7]. However, the preferred design is not necessarily the configuration with the maximum U ; the measured pressure drop and PEC should determine whether the enhancement remains advantageous after pumping requirements are considered.

For the illustrative dataset, the enhanced configuration yields approximately 15–17% greater heat duty across the four flow rates while showing a lower shell-side pressure drop. This pattern is intentionally consistent with the type of thermal-hydraulic improvement reported for helical-baffle designs [3,7], but the numerical percentages are examples only. In actual experiments, the magnitude should be obtained from replicate measurements and reported with confidence intervals. If the improvement in U is statistically significant and PEC remains above 1.0 at the practical flow range, the enhanced geometry can be considered superior for the tested conditions.

The energy-balance error should be plotted or tabulated for every run. A systematic hot-side heat duty larger than the cold-side duty may indicate heat loss to the environment, inaccurate flow calibration, or poor sensor placement. Runs with excessive imbalance should not be silently averaged into the final result. Similarly, any comparison between baffle types should be made at the same inlet temperatures and mass flow rates; otherwise, apparent enhancement may simply reflect a changed operating condition.

Conclusion and Discussion

A controlled experimental methodology has been developed for evaluating heat-transfer enhancement in a shell-and-tube heat exchanger using paired shell-side baffle configurations. The method controls fluid, heat-transfer area, inlet temperatures, and tube-side conditions while varying only the baffle geometry. Its principal outputs are heating duty, LMTD, overall heat-transfer coefficient, shell-side pressure drop, and a thermal-hydraulic performance criterion. This combination is more defensible than reporting an increase in U alone because practical enhancement must account for pumping penalty. The literature through 2018 shows consistent evidence that baffle spacing, baffle number, and helical flow paths can materially alter shell-side heat transfer and pressure loss [1–3,5,7,8]. Nanofluid studies further demonstrate that fluid-property modification can enhance heat transfer [4,6,9–12], although viscosity, stability, and pumping effects must be included. The present water–water methodology deliberately isolates geometric enhancement; a later extension could repeat the optimized baffle condition using a stable nanofluid to separate geometric and fluid effects. For a final experimental paper, the illustrative values in the Results section must be replaced by measured replicate data, and uncertainty should be calculated from the actual calibration certificates. The most important acceptance conditions are steady operation, heat-balance closure within 5%, matched test conditions, and transparent reporting of pressure drop. When these requirements are met, the methodology can provide a reliable basis for identifying a baffle configuration that improves heat transfer without an unacceptable hydraulic cost.

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