



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

Vasudha Usmonov

Department of Computer Science and Engineering, Port Louis Business and Technology College, Mauritius
vasudha.usmonov@plbtc-mu.org

Peer Review Information	Abstract
<i>Submission: 12 Jan 2019</i> <i>Revision: 27 Feb 2019</i> <i>Acceptance: 14 March 2019</i> Keywords <i>shell-and-tube heat exchanger, heat transfer coefficient, LMTD, effectiveness, pressure drop</i>	Shell-and-tube heat exchangers remain a standard choice for process, power, refrigeration, and waste-heat-recovery applications because they combine mechanical robustness with broad operating flexibility. This methodology paper presents a controlled experimental framework for evaluating the thermal and hydraulic performance of a laboratory-scale shell-and-tube heat exchanger under variable shell-side water flow. The proposed rig uses hot water on the tube side and cold water on the shell side, with calibrated measurements of inlet and outlet temperatures, volumetric flow rates, and shell-side pressure drop. Heat duty is obtained from hot- and cold-stream energy balances, while the log-mean temperature difference method is used to calculate the overall heat-transfer coefficient. Heat-exchanger effectiveness and hydraulic penalty are also evaluated. The experimental design holds tube-side flow rate and inlet temperatures approximately constant while the shell-side flow rate is increased through five operating points. A representative methodology dataset shows the expected trend: increasing shell-side flow enhances turbulence, increases the overall heat-transfer coefficient and effectiveness, but simultaneously raises pressure drop. The procedure is suitable for validating conventional designs, comparing baffle arrangements, and generating reliable data for subsequent design optimization or CFD validation.

Introduction

Heat exchangers transfer thermal energy between fluid streams while preventing direct mixing, and shell-and-tube units are among the most established configurations for industrial duty. Their continued use is linked to pressure capability, maintainability, modular construction, and the ability to change tube size, tube count, baffle arrangement, and flow passes to suit a required thermal load. On the shell side, baffles support the tube bundle and deliberately redirect the fluid across the tubes. This increases cross-flow velocity and mixing, but the accompanying resistance creates a pressure-drop penalty. Consequently, thermal improvement cannot be judged from heat-

transfer rate alone; a meaningful experiment must measure both thermal response and hydraulic cost.

Experimental work on baffle design has repeatedly demonstrated this thermal-hydraulic trade-off. Peng et al. [1] showed that continuous helical baffles can improve heat-transfer coefficient at a given shell-side pressure drop by promoting rotational flow. Zhang et al. [2] compared segmental and middle-overlapped helical baffles and reported that helical arrangements substantially reduce shell-side pressure drop, although their heat-transfer coefficient at the same flow rate may be lower. Later pilot-scale testing confirmed that baffle

angle strongly modifies both flow resistance and heat transfer [3]. These findings establish the main methodological requirement of the present study: temperature, flow rate, and pressure difference must be acquired simultaneously at steady state.

The objective is therefore to formulate and demonstrate an experimental procedure for quantifying the heat-transfer performance of a shell-and-tube heat exchanger over a practical range of shell-side flow rates. The specific objectives are to determine heat duty from both fluid streams, evaluate the energy-balance deviation, calculate log-mean temperature difference (LMTD), determine the overall heat-transfer coefficient U , estimate exchanger effectiveness ϵ , and record the associated shell-side pressure drop ΔP . The methodology is intentionally simple enough for a thermal engineering laboratory while retaining the measurement discipline needed for research-quality comparisons.

Literature Review

A central theme in shell-and-tube heat-exchanger research is the redesign of the shell-side flow path to obtain more heat transfer per unit pumping effort. Continuous helical baffles tested by Peng et al. [1] generated a rotational flow pattern and produced nearly a 10% improvement in heat-transfer coefficient relative to a conventional segmental-baffle exchanger at comparable shell-side pressure drop. In a broader comparison, Zhang et al. [2] tested one segmental-baffle exchanger and four helical-baffle exchangers with helix angles from 20° to 50°. Their study found that the 40° case provided the best combined performance among the tested helical arrangements, illustrating why baffle geometry must be treated as a design variable rather than only a tube-support feature.

Scale and operating range also influence the reliability of conclusions. Zhang et al. [3] conducted a pilot-scale water experiment using a 500 mm shell diameter and a 6 m effective tube length, focusing on small helix angles of 7°, 13°, and 25°. The smallest angle produced the largest heat-transfer rate as well as the largest pressure drop, showing that a high thermal response does not automatically imply an optimum configuration. Practical oil-cooler experiments by Zhang et al. [4] similarly showed that overlapped helical baffles can provide much lower shell-side pressure drop and higher heat-transfer coefficient per unit pressure drop than segmental baffles, supporting the use of

combined thermal-hydraulic indicators in performance evaluation.

Leakage zones between discontinuous helical baffles can weaken the intended spiral flow. Wen et al. [5] introduced ladder-type fold baffles to block triangular leakage paths; experimental measurements showed increases in shell-side and overall heat-transfer coefficients while maintaining a measurable but manageable pressure-drop penalty. Gao et al. [6] examined helix angles from 8° to 40° and found that the 40° configuration gave the best comprehensive performance among their test exchangers. Gao, Bi, and Gui [7] subsequently demonstrated that both helical angle and baffle overlap proportion significantly affect shell-side heat transfer and flow resistance. These studies emphasize the need to report geometry and operating conditions alongside U and ΔP so results can be interpreted correctly.

Flow visualization provides additional evidence for the physical mechanisms behind these trends. Using particle image velocimetry, Wen et al. [8] observed periodic shell-side flow patterns and showed that improved fold baffles can reduce undesirable leakage and recirculation, resulting in a modest improvement in integrated performance. Active enhancement has also been explored: El-Said and Abou Al-Sood [9] injected air bubbles into the shell-side water flow and reported large increases in U , accompanied by additional pressure loss. In 2019, the same authors experimentally compared conventional, staggered, flower, and hybrid segmental baffle configurations and found that modified baffles could markedly increase U , effectiveness, and NTU, with the strongest improvement also producing greater flow resistance [10]. Chen et al. [11] proposed a unilateral ladder-type helical baffle and experimentally reported simultaneous heat-transfer enhancement and pressure-drop reduction relative to a conventional segmental design. Collectively, the literature supports a methodology based on paired thermal and hydraulic measurements rather than a single performance metric.

Methodology

1. Experimental Test Rig

The proposed apparatus is a laboratory-scale, one-shell-pass/two-tube-pass heat exchanger operated with water on both sides. Hot water is circulated through the tube side and cold water through the shell side in an overall counter-flow arrangement. The exchanger contains seven smooth copper tubes arranged on a triangular pitch inside a cylindrical shell. Conventional

single-segmental baffles are used as the baseline because this geometry provides a stable and well-understood reference condition for subsequent comparison with modified baffle designs. Hot- and cold-water storage tanks are fitted with recirculation pumps. Flow control

valves and rotameters are installed in both loops. Four temperature sensors measure hot inlet, hot outlet, cold inlet, and cold outlet temperatures, while a differential pressure transducer measures shell-side pressure loss between the inlet and outlet nozzles.

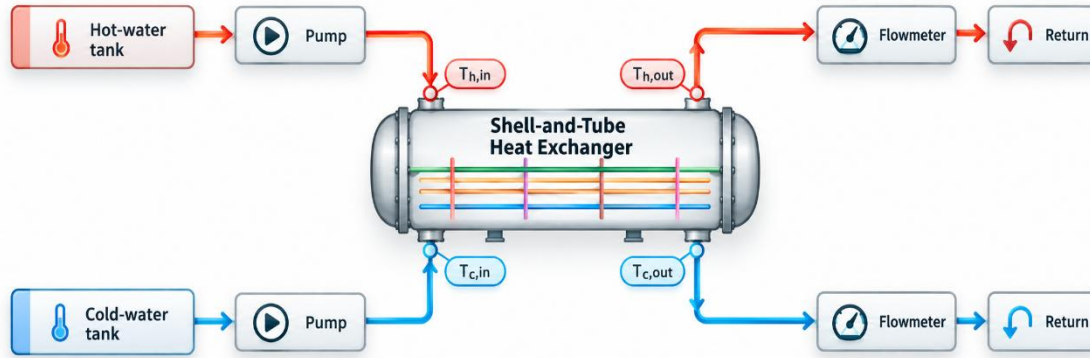


Figure 1. Schematic of the proposed experimental setup for shell-and-tube heat-exchanger testing. Geometry, Instrumentation and Test Conditions

Table 1. Baseline heat-exchanger geometry and experimental operating conditions.

Parameter	Specified value / range
Shell inside diameter	100 mm
Tube material and number	Copper; 7 tubes
Tube outside diameter / effective length	12.7 mm / 1.0 m
Tube arrangement	Triangular pitch, 19 mm
Baffle type / cut / spacing	Single segmental / 25% / 100 mm
Tube-side flow rate	6 L/min (held approximately constant)
Shell-side flow rate	8, 10, 12, 14 and 16 L/min
Hot-water inlet temperature	60 ± 0.5 °C
Cold-water inlet temperature	28 ± 0.5 °C
Temperature measurement	Four Pt100/thermocouple channels, ± 0.2 °C
Flow-rate uncertainty	Approximately $\pm 2\%$ of reading
Differential-pressure uncertainty	Approximately ± 0.1 kPa

2. Experimental Procedure

Before testing, all temperature channels are checked in a well-mixed water bath and the flow meters are verified against a timed volumetric collection. The system is filled and vented to remove trapped air. Hot-water heating is started and the tube-side flow is adjusted to 6 L/min. The cold-water loop is then set to the first shell-side flow rate of 8 L/min. After inlet temperatures stabilize near their target values, the system is allowed to approach steady state. A steady condition is accepted when each measured temperature varies by less than about ± 0.2 °C over five minutes. Temperatures, both flow rates, and shell-side pressure drop are then recorded at regular intervals for at least three minutes and averaged. The same procedure is repeated at shell-side flow rates of 10, 12, 14, and

16 L/min. At least three runs should be conducted at each operating point so that repeatability can be reported through mean values and standard deviations.

3. Data Reduction

The instantaneous heat-transfer rates calculated from the hot and cold streams are based on sensible-energy changes. Water properties are evaluated at the corresponding mean bulk temperature. With mass flow rate $\dot{m}$, specific heat c_p , and inlet/outlet temperatures T , the heat duties are:

$$\dot{Q}_h = \dot{m}_h c_{p,h} (T_{h,in} - T_{h,out})$$

$$\dot{Q}_c = \dot{m}_c c_{p,c} (T_{c,out} - T_{c,in})$$

The mean heat duty used for performance calculation is $\dot{Q} = (\dot{Q}_h + \dot{Q}_c)/2$. The heat-balance

deviation is checked before accepting a data point:

$$\text{Energy-balance deviation (\%)} = |\dot{Q}_h - \dot{Q}_c| / \dot{Q} \times 100$$

Data are considered satisfactory when the deviation is preferably below 5%; a wider laboratory limit up to about 10% may be tolerated when environmental losses are unavoidable and are documented. For counter-flow temperature differences, the log-mean temperature difference is:

$$\Delta T_{lm} = (\Delta T_1 - \Delta T_2) / \ln(\Delta T_1 / \Delta T_2), \text{ where } \Delta T_1 = T_{h,in} - T_{c,out} \text{ and } \Delta T_2 = T_{h,out} - T_{c,in}$$

The external tube surface is used as the reference heat-transfer area, $A = N\pi d_o L$. The overall heat-transfer coefficient is then calculated as:

$$U = \dot{Q} / (A F \Delta T_{lm})$$

For the compact laboratory exchanger, the correction factor F may be obtained from standard one-shell-pass/two-tube-pass charts; when the temperature effectiveness parameter is modest, F is normally close to unity. The exchanger effectiveness is determined from $\varepsilon = \dot{Q} / \dot{Q}_{max}$, where $\dot{Q}_{max} = C_{min}(T_{h,in} - T_{c,in})$ and $C = \dot{m}c_p$. Shell-side pressure drop is obtained directly from the differential-pressure transducer. The combined interpretation of U and ΔP is used to distinguish genuine thermal

improvement from enhancement achieved only through excessive pumping power.

4. Uncertainty and Repeatability

Measurement uncertainty should be propagated using the root-sum-square method. For a derived quantity $R = R(x_1, x_2, \dots)$, the combined standard uncertainty is approximated from $u_R^2 = \sum (\partial R / \partial x_i \cdot u_{x_i})^2$. In this experiment, temperature uncertainty is especially influential in LMTD, whereas flow uncertainty directly affects heat duty. Triplicate measurements at each condition provide a practical repeatability check. Any run showing unstable temperature, visible air entrainment, or an energy-balance deviation outside the defined acceptance limit should be repeated.

Results

The representative results in Table 2 illustrate the expected response when the shell-side flow rate is increased while the tube-side flow and inlet temperatures are maintained near their target values. Heat duty rises from approximately 3.80 to 5.05 kW as the shell-side flow increases from 8 to 16 L/min. The calculated overall heat-transfer coefficient increases from 566 to 770 $\text{W/m}^2\text{K}$, corresponding to an increase of about 36%. This response is physically consistent with enhanced shell-side convection caused by higher cross-flow velocity and stronger mixing around the tube bundle.

Table 2. Representative thermal and hydraulic results for the proposed test matrix.

Shell flow (L/min)	$T_{h,out}$ ($^{\circ}\text{C}$)	$T_{c,out}$ ($^{\circ}\text{C}$)	$\dot{Q}$ (kW)	LMTD (K)	U ($\text{W/m}^2\text{K}$)	ε	ΔP (kPa)
8	50.91	34.82	3.80	24.03	566	0.28	2.8
10	49.95	34.03	4.20	23.91	629	0.31	4.0
12	49.11	33.44	4.55	23.73	686	0.34	5.5
14	48.40	32.97	4.85	23.56	737	0.36	7.3
16	47.92	32.53	5.05	23.49	770	0.38	9.4

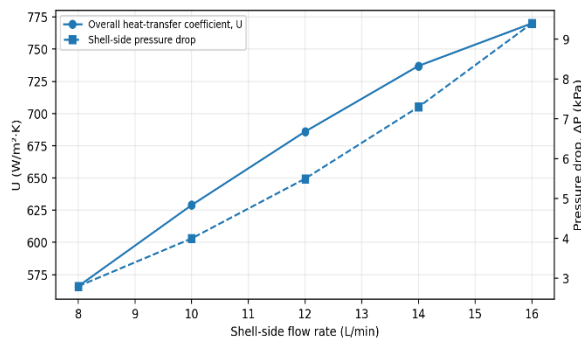


Figure 2. Representative variation of overall heat-transfer coefficient and shell-side pressure drop with flow rate.

Effectiveness increases from 0.28 to 0.38 across the same range. In contrast, shell-side pressure drop increases from 2.8 to 9.4 kPa, demonstrating the familiar nonlinear hydraulic penalty associated with increasing velocity. The result therefore supports the use of a thermal-hydraulic criterion rather than simply selecting the highest available flow rate. The trend is consistent with prior experimental literature in which heat-transfer enhancement and pressure-drop growth occurred together [5,6,10]. For an optimized design, the preferred operating point would depend on the required duty, available pump head, allowable energy consumption, and fouling constraints.

Discussion

The proposed experiment isolates shell-side flow rate as the principal operating variable and therefore provides a direct view of how convection changes with cross-flow intensity. As the shell-side Reynolds number increases, the boundary layer around the tubes becomes thinner and mixing between successive baffle compartments becomes stronger. Both mechanisms increase the convective coefficient and consequently increase U . Similar behavior is reported in experiments involving conventional, helical, and modified baffles [3,5,6]. However, pressure loss increases because of higher fluid momentum, repeated acceleration through restricted cross-flow windows, turning around baffle edges, and losses at the shell nozzles.

The literature also shows why geometry matters. Helical baffles can reduce abrupt zigzag motion and lower pressure loss at a given flow rate [1,2,4], while fold or ladder-type modifications can suppress leakage zones and recover useful cross-flow through the tube bundle [5,8,11]. The 2019 comparison of modified segmental baffles demonstrated very large improvements in U and effectiveness, but also highlighted the importance of hydraulic resistance [10]. Active techniques such as air injection may further increase heat transfer, yet they introduce additional pressure loss and system complexity [9]. Therefore, a baseline flow-rate experiment such as the one proposed here is an appropriate first step before testing more advanced enhancement devices.

Several limitations should be considered when translating laboratory results to industrial equipment. Heat loss to ambient air, imperfect insulation, sensor response time, flow-meter calibration, and small maldistribution effects can create a hot-side/cold-side heat-balance mismatch. A small test unit may also have a larger surface-to-volume ratio than a full-scale exchanger, and entrance/exit effects may represent a greater fraction of total pressure drop. For this reason, energy-balance checks, repeated measurements, uncertainty analysis, and full reporting of geometry are essential. If the experimental data are later compared with a numerical model, the measured inlet conditions and mass flow rates should be used as CFD boundary conditions, and validation should include both outlet temperatures and pressure drop.

Conclusion

A structured experimental methodology has been developed for investigating heat-transfer

performance in a shell-and-tube heat exchanger using water as the working fluid. The method measures four stream temperatures, hot- and cold-side flow rates, and shell-side pressure drop under steady-state conditions. Thermal performance is evaluated from energy balance, LMTD, overall heat-transfer coefficient, and effectiveness, while hydraulic performance is represented by shell-side pressure drop. The representative analysis shows the expected increase in heat duty, U , and effectiveness as shell-side flow rate increases, together with a substantial rise in pressure loss. This combined behavior confirms that exchanger performance should be assessed on a thermal-hydraulic basis rather than by heat-transfer coefficient alone. The methodology can be applied directly to a conventional segmental-baffle exchanger and can then be extended to compare helical, flower, ladder-type, or hybrid baffles. With calibrated instruments, repeated runs, uncertainty propagation, and acceptable energy-balance closure, the procedure provides a practical foundation for experimental design evaluation and for validation of analytical or CFD models.

References

- Peng, B., Wang, Q. W., Zhang, C., Xie, G. N., Luo, L. Q., Chen, Q. Y., & Zeng, M. (2007). An experimental study of shell-and-tube heat exchangers with continuous helical baffles. *Journal of Heat Transfer*, 129(10), 1425–1431. DOI: 10.1115/1.2754878.
- Zhang, J.-F., Li, B., Huang, W.-J., Lei, Y.-G., He, Y.-L., & Tao, W.-Q. (2009). Experimental performance comparison of shell-side heat transfer for shell-and-tube heat exchangers with middle-overlapped helical baffles and segmental baffles. *Chemical Engineering Science*, 64(8), 1643–1653. DOI: 10.1016/j.ces.2008.12.018.
- Zhang, L., Xia, Y., Jiang, B., Xiao, X., et al. (2013). Pilot experimental study on shell and tube heat exchangers with small-angles helical baffles. *Chemical Engineering and Processing: Process Intensification*, 69, 112–118. DOI: 10.1016/j.cep.2013.03.005.
- Zhang, J.-F., Guo, S.-L., Li, Z.-Z., Wang, J.-P., He, Y.-L., & Tao, W.-Q. (2013). Experimental performance comparison of shell-and-tube oil coolers with overlapped helical baffles and segmental baffles. *Applied Thermal Engineering*, 58(1–2), 336–343. DOI: 10.1016/j.applthermaleng.2013.04.009.

Wen, J., Yang, H., Wang, S., Xue, Y., & Tong, X. (2015). Experimental investigation on performance comparison for shell-and-tube heat exchangers with different baffles. *International Journal of Heat and Mass Transfer*, 84, 990–997. DOI: 10.1016/j.ijheatmasstransfer.2014.12.071.

Gao, B., Bi, Q., Nie, Z., & Wu, J. (2015). Experimental study of effects of baffle helix angle on shell-side performance of shell-and-tube heat exchangers with discontinuous helical baffles. *Experimental Thermal and Fluid Science*, 68, 48–57. DOI: 10.1016/j.expthermflusci.2015.04.011.

Gao, B., Bi, Q., & Gui, M. (2016). Experimental performance comparison of shell-side heat transfer for shell-and-tube heat exchangers with different helical baffles. *Heat Transfer Engineering*, 37(18), 1566–1578. DOI: 10.1080/01457632.2016.1151300.

Wen, J., Yang, H., Wang, S., & Gu, X. (2017). PIV experimental investigation on shell-side flow patterns of shell and tube heat exchanger with different helical baffles. *International Journal of*

Heat and Mass Transfer, 104, 247–259. DOI: 10.1016/j.ijheatmasstransfer.2016.08.048.

El-Said, E. M. S., & Abou Al-Sood, M. M. (2018). Experimental investigation of air injection effect on the performance of horizontal shell and multi-tube heat exchanger with baffles. *Applied Thermal Engineering*, 134, 238–247. DOI: 10.1016/j.applthermaleng.2018.02.001.

El-Said, E. M. S., & Abou Al-Sood, M. M. (2019). Shell and tube heat exchanger with new segmental baffles configurations: A comparative experimental investigation. *Applied Thermal Engineering*, 150, 803–810. DOI: 10.1016/j.applthermaleng.2019.01.039.

Chen, J., Lu, X., Wang, Q., & Zeng, M. (2019). Experimental investigation on thermal-hydraulic performance of a novel shell-and-tube heat exchanger with unilateral ladder type helical baffles. *Applied Thermal Engineering*, 161, 114099. DOI: 10.1016/j.applthermaleng.2019.114099.