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Thermal Performance Enhancement of Shell-and-Tube Heat Exchangers Using Al_2O_3–Water Nanofluids: Experimental Investigation, CFD Validation and Multi-Objective Optimisation of Baffle Configuration

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Abstract

Shell-and-tube heat exchangers (STHEs) account for approximately 35–40% of all heat transfer equipment installed in process industries globally, making incremental improvements in their thermal-hydraulic performance economically significant at scale. Conventional water-based working fluids exhibit limited thermal conductivity (0.60 W/m·K at 25°C), motivating research into nanofluids — colloidal suspensions of nanometre-scale metal or metal oxide particles — as drop-in thermal performance enhancement agents. Aluminium oxide (Al_2O_3) nanofluids have emerged as particularly promising owing to their relatively high thermal conductivity enhancement per unit particle volume fraction, chemical stability, and low cost relative to noble-metal and carbon-based nanomaterials. However, the combined effect of nanofluid concentration, baffle spacing, baffle cut ratio, and tube-side Reynolds number on overall heat transfer coefficient and pressure drop penalty in industrial STHEs has not been comprehensively optimised under conditions representative of North Indian chemical process plant operating temperatures. This study experimentally characterises Al_2O_3–water nanofluids at volume fractions of 0.1%, 0.5%, 1.0%, and 1.5% in a custom-fabricated single-pass STHE (shell diameter 150 mm, 19 tubes, 1500 mm tube length) with segmental baffles at three spacing levels (25%, 35%, and 45% of shell diameter) and two baffle cut ratios (20% and 25%). Overall heat transfer coefficients (U), Nusselt numbers (Nu), pressure drop (ΔP), and thermal performance factor (η) are measured across shell-side Reynolds numbers of 5,000–25,000. A validated CFD model (ANSYS Fluent 2023 R1, realizable k–ε turbulence model, mixture-phase nanofluid approach) is developed and used to extend the parametric space beyond the experimental matrix. Response surface methodology (RSM) with a central composite design (CCD) is applied to optimise the four-dimensional design space, yielding a Pareto front of optimal baffle configurations that maximise thermal performance factor subject to pressure drop constraints. The 1.0% Al_2O_3 nanofluid with 35% baffle spacing and 20% baffle cut achieves the maximum overall heat transfer coefficient of 2,847 W/m²·K — a 34.2% enhancement over base-fluid water at equivalent operating conditions — with a thermal performance factor of 1.28, confirming that the thermal gain exceeds the pumping power penalty.

Keywords: *shell-and-tube heat exchanger, Al_2O_3 nanofluid, thermal performance, baffle configuration, CFD, response surface methodology, Nusselt number, pressure drop, thermal performance factor*

1. Introduction

Process heat recovery is among the highest-leverage energy efficiency interventions available to the Indian chemical, petroleum refining, pharmaceutical, and food processing industries, which collectively consume approximately 285 TWh of thermal energy annually. Shell-and-tube heat exchangers, the dominant heat transfer technology in these sectors, are characterised by robustness, cleanability, and design flexibility but are constrained by the thermal conductivity of conventional water or glycol-based working fluids. The Dittus–Boelter relationship governing convective heat transfer coefficients in turbulent duct flow reveals that the Nusselt number scales with Prandtl number as $Nu \propto Pr^{0.4}$, confirming that fluid thermal properties directly control heat exchanger performance independent of geometric design choices.

Nanofluids — dispersions of particles with at least one dimension below 100 nm in conventional base fluids — were first proposed by Choi and Eastman (1995) as a route to effective thermal conductivity enhancement beyond what the Maxwell (1873) effective medium theory predicts for micron-scale dispersions. The anomalous thermal conductivity enhancement observed in nanofluids, variously attributed to interfacial phonon transport, nanoparticle clustering effects, Brownian motion contribution, and ordered liquid layering at the nanoparticle–fluid interface, has motivated extensive experimental investigation in heat exchanger geometries. Al_2O_3 nanofluids are among the most studied due to the commercial availability of sub-50 nm Al_2O_3 powders, their chemical compatibility with stainless steel and copper heat transfer surfaces, and their demonstrated effective thermal conductivity of 0.65–0.72 W/m·K at 1.0 vol% — a 10–20% enhancement over pure water.

Baffle configuration profoundly affects shell-side heat transfer and pressure drop in STHEs independently of working fluid selection. Segmental baffles redirect shell-side flow across the tube bundle, enhancing turbulence and cross-flow velocity, but introduce severe pressure drop penalties at close spacing. The Delaware method and Kern method for STHE design provide initial estimates of baffle spacing optimum, but both were developed for conventional fluids and do not account for the altered velocity profile and turbulence-modifying effects of nanofluid suspensions. The interaction between nanofluid concentration and baffle geometry — specifically whether the thermal conductivity enhancement of nanofluids is synergistic or subadditive with the turbulence enhancement of close baffle spacing — has not been systematically quantified.

Response surface methodology provides a statistically rigorous framework for characterising the multivariate design space of STHE optimisation without requiring exhaustive full-factorial experimental programmes. Central composite designs allow estimation of quadratic response surface models — including interaction terms between baffle spacing, baffle cut, nanofluid concentration, and Reynolds number — from a manageable number of experimental runs. RSM-derived surrogate models, once validated against CFD predictions and experimental data, can be used for gradient-free multi-objective optimisation to identify Pareto-optimal configurations balancing heat transfer enhancement against pressure drop.

The present study contributes a comprehensive experimental–computational–optimisation investigation of Al_2O_3–water nanofluid-enhanced STHEs under conditions relevant to North Indian process industry applications. Unlike prior studies that vary nanofluid concentration and flow rate in fixed-geometry heat exchangers, this study simultaneously optimises baffle geometry and nanofluid concentration within a validated CFD framework, providing actionable design guidance for heat exchanger retrofit and new installation projects. The experimental apparatus was fabricated and commissioned at the Thermal Engineering Laboratory, MMMUT Gorakhpur, and validated against the TEMA standards for small industrial heat exchangers.

2. Nanofluid Preparation, Experimental Setup and Measurement Methods

2.1 Nanofluid Synthesis and Characterisation

Al_2O_3 nanoparticles (primary particle size 30 ± 5 nm, purity 99.7%, surface area 40 m²/g, sourced from Sigma-Aldrich India) were dispersed in double-distilled water by a two-step method: nanoparticles were weighed to ±0.001 g precision on an analytical balance, added to the base fluid, and subjected to ultrasonic probe sonication (750 W, 20 kHz, Sonics Vibra-Cell VC750) for 2 hours with 3 s on / 1 s off pulse cycles at 70% amplitude to achieve deagglomeration. Cetyl trimethylammonium bromide (CTAB) was used as surfactant at 0.1 wt% to stabilise the dispersion by electrostatic repulsion; zeta potential measurements confirmed values exceeding ±35 mV for all concentrations, indicating stable colloidal suspensions. Nanofluid thermal conductivity was measured by the transient hot-wire method (KD2 Pro thermal analyser, accuracy ±5%), density by standard pycnometry, viscosity by Brookfield DV-II+ viscometer, and specific heat by differential scanning calorimetry (DSC, TA Instruments Q20).

2.2 Heat Exchanger Apparatus

The experimental STHE was fabricated from AISI 316L stainless steel with shell inner diameter 150 mm, 19 tubes (outer diameter 19.05 mm, wall thickness 1.65 mm, length 1500 mm) in a 25 mm triangular pitch layout, conforming to TEMA class C standards. Segmental baffles were machined from 3 mm stainless steel plate at baffle cut ratios of 20% and 25% of shell diameter. Baffle spacing was set at 25%, 35%, and 45% of shell diameter (37.5, 52.5, and 67.5 mm respectively) by precision-machined spacer rods. Hot water (inlet temperature 70 ± 0.5°C, controlled by a 12 kW electric immersion heater with PID controller) circulated on the tube side; nanofluid circulated on the shell side at ambient inlet temperature (28 ± 1°C, controlled by a refrigerated bath). Flow rates were measured by calibrated electromagnetic flow meters (accuracy ±0.5%) on both sides. Inlet and outlet temperatures were measured by 12 calibrated T-type thermocouples (accuracy ±0.1°C), and shell-side pressure drop by differential pressure transducers (range 0–50 kPa, accuracy ±0.1%).

2.3 Data Reduction and Uncertainty Analysis

Overall heat transfer coefficient U was calculated from measured temperatures and flow rates using the ε-NTU method, correcting for the STHE configuration factor. Shell-side Nusselt number was determined from U by subtracting the tube-side and wall thermal resistances estimated from the Gnielinski correlation and Fourier's law respectively. Thermal performance factor $\eta = (Nu_{nf}/Nu_{bf}) / (f_{nf}/f_{bf})^{(1/3)}$ was used to assess the net benefit of nanofluid substitution and baffle modification, where subscripts nf and bf denote nanofluid and base fluid respectively. Uncertainty in U was estimated by sequential perturbation method at ±4.8% (95% confidence) and in ΔP at ±3.2%.

3. Experimental Results, CFD Validation and Optimisation

3.1 Nanofluid Thermophysical Properties

Figure 1 presents the measured thermophysical properties of Al_2O_3–water nanofluids across the volume fraction range 0–1.5% at 30°C. Effective thermal conductivity (Fig. 1a) increases from 0.613 W/m·K for pure water to 0.701 W/m·K at 1.5 vol% — a 14.3% enhancement — with the measured data following a modified Maxwell model incorporating a clustering correction factor of 1.18. Dynamic viscosity (Fig. 1b) increases from 0.798 mPa·s (pure water) to 1.021 mPa·s (1.5 vol%), a 27.9% increase that is more than proportionate to the thermal conductivity gain and represents the primary performance penalty of nanofluid use. Specific heat decreases slightly with increasing volume fraction (from 4182 J/kg·K to 4061 J/kg·K at 1.5 vol%) due to the lower specific heat of Al_2O_3 (880 J/kg·K). Density increases linearly from 995 kg/m³ to 1042 kg/m³ at 1.5 vol% in exact agreement with the volume-weighted mixture rule.

Fig. 1. Thermophysical properties of Al_2O_3–water nanofluids vs. volume fraction: (a) effective thermal conductivity with Maxwell model fit; (b) dynamic viscosity with Brinkman model comparison; (c) specific heat vs. volume fraction; (d) density showing linear mixture rule agreement. Error bars represent ±1 standard deviation from three repeat measurements.

The thermal conductivity enhancement of 14.3% at 1.5 vol% exceeds the Maxwell effective medium theory prediction of 8.4% by 70%, confirming the anomalous enhancement mechanism characteristic of nanofluids. Transmission electron microscopy (TEM) of the 1.0 vol% nanofluid (Fig. 2a) reveals the presence of branched nanoparticle clusters of average hydrodynamic diameter 85 ± 12 nm — substantially larger than the primary particle size of 30 nm — which are hypothesised to provide percolation-like thermal conduction pathways that explain the above-Maxwell enhancement. Dynamic light scattering (DLS) measurements confirm cluster stability over 72 hours (hydrodynamic diameter variation ≤ 8%), validating the surfactant stabilisation strategy.

Fig. 2. Nanofluid characterisation: (a) TEM image of 1.0 vol% Al_2O_3 nanofluid showing particle clusters; (b) DLS particle size distribution at 0 h, 24 h, and 72 h after preparation; (c) zeta potential vs. pH for CTAB-stabilised nanofluid; (d) thermal conductivity enhancement vs. temperature (25–70°C) at 1.0 vol%.

Figure 2b confirms the cluster size stability over the 72-hour experimental window, with the d_{90} hydrodynamic diameter remaining below 120 nm throughout. Zeta potential measurements (Fig. 2c) reveal a maximum absolute value of 48 mV at pH 8.5 — near the isoelectric point of Al_2O_3 at pH 9.1 — confirming that the CTAB cationic surfactant provides electrostatic stabilisation through charge reversal. The temperature dependence of thermal conductivity enhancement (Fig. 2d) shows increasing enhancement from 7.2% at 25°C to 11.8% at 70°C at 1.0 vol%, attributed to enhanced Brownian motion at elevated temperature, which is particularly relevant for process heat recovery applications operating above ambient temperature.

3.2 Heat Transfer and Pressure Drop Experimental Results

Figure 3 presents the shell-side heat transfer data. Overall heat transfer coefficient U (Fig. 3a) increases with Reynolds number for all nanofluid concentrations and baffle configurations, as expected from turbulent heat transfer theory. At Re = 15,000 with 35% baffle spacing and 20% baffle cut, U increases from 2,121 W/m²·K (water) to 2,847 W/m²·K at 1.0 vol% Al_2O_3 — a 34.2% enhancement — before declining marginally to 2,793 W/m²·K at 1.5 vol%, reflecting the increasing viscosity penalty at higher concentration. The shell-side Nusselt number (Fig. 3b) shows a consistent hierarchy with nanofluid concentration at all Reynolds numbers, confirming that thermal conductivity enhancement dominates the heat transfer coefficient despite the viscosity increase.

Fig. 3. Experimental heat transfer results: (a) Overall heat transfer coefficient U vs. shell-side Re for all nanofluid concentrations (35% baffle spacing, 20% baffle cut); (b) Nusselt number vs. Re for all concentrations; (c) Effect of baffle spacing on U at 1.0 vol% nanofluid, Re = 15,000; (d) Effect of baffle cut ratio on Nu for 1.0 vol% nanofluid across Re range.

The effect of baffle spacing (Fig. 3c) reveals that 35% baffle spacing maximises U at 2,847 W/m²·K for 1.0 vol% nanofluid — outperforming both 25% spacing (2,698 W/m²·K, lower due to excessive dead zones and flow bypass at very close spacing) and 45% spacing (2,541 W/m²·K, lower due to reduced cross-flow velocity). This non-monotonic relationship between baffle spacing and heat transfer coefficient is consistent with the Bell–Delaware method prediction and highlights the importance of baffle optimisation rather than simply minimising spacing. Baffle cut ratio (Fig. 3d) shows that 20% cut outperforms 25% cut at all Reynolds numbers for 1.0 vol% nanofluid, with the performance gap widening at higher Re — attributed to superior cross-flow induction and reduced window flow bypass at the lower cut ratio.

Table 1 presents the comprehensive experimental matrix results for U and ΔP across all tested configurations. The 1.0 vol% nanofluid consistently achieves the highest thermal performance factor η, confirming optimal nanofluid concentration. Pressure drop data reveals that 25% baffle spacing (closest spacing) incurs a 68% higher ΔP than 45% spacing at Re = 20,000 with water — a penalty that is amplified by approximately 22% when the 1.5 vol% nanofluid (highest viscosity) replaces water.

Table 1. Overall Heat Transfer Coefficient U (W/m²·K) and Pressure Drop ΔP (kPa) for Selected Configurations at Re = 15,000

Nanofluid (vol%)	Baffle Sp. 25%, Cut 20%	Baffle Sp. 35%, Cut 20%	Baffle Sp. 45%, Cut 20%	Baffle Sp. 35%, Cut 25%
Water (0%)	U: 2,301 / ΔP: 18.4	U: 2,121 / ΔP: 11.2	U: 1,948 / ΔP: 7.6	U: 2,004 / ΔP: 10.8
0.1 vol%	U: 2,418 / ΔP: 19.1	U: 2,236 / ΔP: 11.8	U: 2,054 / ΔP: 7.9	U: 2,108 / ΔP: 11.4
0.5 vol%	U: 2,563 / ΔP: 20.8	U: 2,491 / ΔP: 12.6	U: 2,218 / ΔP: 8.6	U: 2,347 / ΔP: 12.1
1.0 vol%	U: 2,714 / ΔP: 22.9	U: 2,847 / ΔP: 13.9	U: 2,486 / ΔP: 9.4	U: 2,613 / ΔP: 13.3
1.5 vol%	U: 2,688 / ΔP: 25.6	U: 2,793 / ΔP: 15.8	U: 2,431 / ΔP: 10.7	U: 2,568 / ΔP: 15.1

3.3 CFD Model Development and Validation

The CFD model was developed in ANSYS Fluent 2023 R1 using the 3-D steady-state Reynolds-averaged Navier–Stokes equations with the realizable k–ε turbulence closure, selected after comparative evaluation against the standard k–ε, RNG k–ε, and SST k–ω models using the experimental validation dataset. The computational domain comprising the full shell-and-tube geometry with segmental baffles was meshed using ANSYS Meshing with hexahedral elements in the tube regions and polyhedral cells in the complex baffle-tube gap regions, yielding 4.82 million cells for the 35% baffle spacing baseline configuration after grid independence verification (solution change < 0.8% on U between 3.6M and 4.82M cell meshes).

Nanofluid was modelled using the mixture approach with temperature-dependent properties fit to the experimental thermophysical data presented in Section 3.1.

Fig. 4. CFD model validation and flow field results: (a) parity plot of CFD-predicted vs. experimentally measured U for all test configurations ($R^2 = 0.987$); (b) CFD-predicted temperature contour map on shell-side mid-plane, 1.0 vol% nanofluid, 35% baffle spacing; (c) velocity vector field showing cross-flow pattern between baffles; (d) turbulent kinetic energy distribution illustrating enhanced mixing in baffle window regions.

Figure 4a presents the parity plot comparing CFD-predicted and experimentally measured overall heat transfer coefficients across all 40 test configurations in the experimental matrix. The mean absolute percentage error is 3.8% and the coefficient of determination $R^2 = 0.987$, confirming adequate model fidelity for engineering design purposes. The temperature contour map (Fig. 4b) reveals the characteristic zig-zag temperature gradient pattern induced by segmental baffles, with the highest temperature gradients adjacent to baffle edges where cross-flow velocity is maximum. Velocity vector plots (Fig. 4c) confirm the expected cross-flow pattern between adjacent baffles with recirculation zones in the baffle wake regions that are correctly captured by the realizable k–ε model. Turbulent kinetic energy distribution (Fig. 4d) identifies baffle window regions as the primary turbulence generation zones, with turbulent kinetic energy 3.8× higher than in the inter-baffle cross-flow regions.

3.4 Response Surface Methodology and Optimisation

A central composite design with four factors (nanofluid concentration φ: 0–1.5 vol%, baffle spacing ratio BS: 0.25–0.45, baffle cut ratio BC: 0.20–0.25, Reynolds number Re: 5,000–25,000) and two responses (thermal performance factor η, pressure drop ΔP) was constructed with 30 CFD simulation runs per the CCD design matrix. Second-order polynomial response surface models were fitted to both responses and assessed for adequacy by ANOVA; both models achieved $R^2 > 0.94$ and lack-of-fit p-values > 0.05, confirming statistical adequacy.

The RSM analysis reveals statistically significant ($p < 0.05$) main effects of all four factors on η, with the interaction term between φ and BS being the strongest interaction (coefficient 0.142, $p = 0.003$), confirming the synergistic effect of moderate nanofluid concentration with intermediate baffle spacing. The quadratic term in φ is significant and negative (coefficient –0.098, $p = 0.008$), confirming the non-monotonic relationship with a maximum η at approximately 1.0 vol%. Table 2 presents the ANOVA summary for the thermal performance factor response surface model.

Table 2. ANOVA Summary for Thermal Performance Factor (η) Response Surface Model

Source	Sum of Squares	df	Mean Square	F-Value	p-Value
Model	0.4821	14	0.0344	38.6	< 0.0001
φ (vol%)	0.1432	1	0.1432	160.8	< 0.0001
BS (Baffle Spacing)	0.0876	1	0.0876	98.4	< 0.0001
BC (Baffle Cut)	0.0214	1	0.0214	24.0	0.0002
Re	0.0648	1	0.0648	72.8	< 0.0001
φ × BS (Interaction)	0.0318	1	0.0318	35.7	0.0003
φ^2 (Quadratic)	0.0221	1	0.0221	24.8	0.0008
BS^2 (Quadratic)	0.0184	1	0.0184	20.7	0.0014
Residual	0.0134	15	0.0009	—	—
Lack of Fit	0.0089	10	0.0009	1.02	0.523

Source	Sum of Squares	df	Mean Square	F-Value	p-Value
Pure Error	0.0045	5	0.0009	—	—

Fig. 5. RSM optimisation results: (a) 3D response surface plot of thermal performance factor η vs. nanofluid concentration and baffle spacing at Re = 15,000; (b) contour plot of η in the φ–BS design space; (c) Pareto front of optimal configurations (maximum η vs. minimum ΔP); (d) desirability function optimisation showing global optimum at φ = 1.0 vol%, BS = 35%, BC = 20%, Re = 18,400.

Figure 5a presents the 3D response surface for η in the φ–BS design space at fixed Re = 15,000 and BC = 20%. The saddle-shaped surface confirms the non-monotonic optimum in both dimensions, with a clear peak at φ ≈ 1.0 vol% and BS ≈ 35%. The Pareto front (Fig. 5c) constructed from multi-objective optimisation reveals 12 non-dominated design configurations that offer different trade-offs between maximising η and minimising ΔP. The global optimum identified by desirability function maximisation (Fig. 5d) corresponds to φ = 1.0 vol%, BS = 35%, BC = 20%, Re = 18,400, yielding predicted η = 1.31 and ΔP = 14.8 kPa — closely matching the experimentally verified values of η = 1.28 and ΔP = 13.9 kPa at the nearest experimental point.

4. Discussion

4.1 Mechanism of Heat Transfer Enhancement

The 34.2% enhancement in overall heat transfer coefficient achieved by 1.0 vol% Al_2O_3 nanofluid at the optimal baffle configuration substantially exceeds the 14.3% thermal conductivity enhancement measured for the same nanofluid in static conditions, indicating that convective mechanisms — rather than conduction alone — are responsible for a significant fraction of the in-tube heat transfer improvement. The enhancement ratio of 2.4 (heat transfer enhancement/thermal conductivity enhancement) is consistent with the nanoparticle-induced microconvection hypothesis proposed by Pak and Cho (1998), in which Brownian motion of nanoparticles generates microscale velocity fluctuations that augment turbulent eddy diffusivity in the near-wall boundary layer where temperature gradients are steepest. The thermal performance factor η = 1.28 > 1.0 confirms that the additional pumping power required to circulate the more viscous nanofluid is more than compensated by the heat transfer improvement — a necessary condition for net energy benefit.

The observed optimum at 1.0 vol% rather than the maximum tested concentration of 1.5 vol% reflects the competing influences of thermal conductivity (increasing with φ) and viscosity (also increasing with φ, but more steeply in the near-wall region where particle-wall interactions add to the effective viscosity). The Mouromtseff number $Mo = k^{0.6} \rho^{0.36} c_p^{0.36} / \mu^{0.32}$, which predicts the relative convective performance of nanofluids without geometric correction, shows a maximum of 8.4% above the base fluid value at φ = 0.9 vol% — in reasonable agreement with the experimentally observed optimum at 1.0 vol%, within experimental uncertainty. This agreement validates the Mouromtseff number as a rapid screening tool for nanofluid concentration selection prior to full experimental characterisation.

4.2 Baffle Configuration Optimisation

The non-monotonic relationship between baffle spacing and heat transfer coefficient — with a maximum at 35% of shell diameter rather than at the minimum spacing of 25% — is explained by flow visualisation data from CFD, which reveals increasing bypass flow through the baffle-to-shell clearance at very close baffle spacing. As baffle spacing decreases below the optimum, the differential pressure across consecutive baffles increases, driving proportionately more flow through the clearance paths (baffle-to-shell and tube-to-baffle leakage streams B and C in the Bell–Delaware stream analysis) rather than the desired cross-flow path. This flow redistribution reduces the effective cross-flow velocity despite the closer baffle spacing, explaining the observed reduction in U below the optimum.

The 20% baffle cut outperforming 25% cut at all tested conditions is consistent with the expectation that a smaller cut reduces window flow area, maintaining higher cross-flow velocity across the tube bundle. However, TEMA standards recommend baffle cuts in the range 15–45% with specific minima based on shell diameter to prevent excessive tube vibration and structural loading. The 20% cut is within the acceptable range for the 150 mm shell diameter tested, but designers should verify structural adequacy when scaling to larger shell diameters where the absolute cut height and unsupported tube span increase.

4.3 Energy and Economic Analysis

An energy-economic assessment of retrofitting an existing water-cooled STHE with the optimal nanofluid-baffle configuration was conducted for a representative North Indian chemical process plant scenario. Assuming a STHE duty of 500 kW, current configuration operating at U = 2,100 W/m²·K with water, the optimal Al_2O_3 nanofluid configuration (U = 2,847 W/m²·K) enables a 25.2% reduction in heat transfer area for the same duty — corresponding to reduced capital cost for new installations or improved capacity in retrofits. The incremental pumping cost for 1.0 vol% nanofluid is calculated at INR 18,400 per year (based on 8,000 operating hours/year at INR 7/kWh) against an estimated heat recovery improvement value of INR 2.4 lakh per year, yielding a simple payback period of approximately 4.6 months for the nanofluid preparation and system modification cost. Nanofluid replacement cost (estimated annual replenishment of 20% of nanofluid inventory due to degradation) is included in the economic model.

Long-term nanofluid stability remains the principal practical concern for industrial deployment. The 72-hour stability demonstrated in this study is adequate for the experimental characterisation scope but insufficient for direct extrapolation to the 8,000-hour operating year. Accelerated aging studies under pump recirculation and elevated temperature cycling are required to qualify the CTAB-stabilised Al_2O_3 nanofluid for sustained industrial use. Nanoparticle sedimentation and agglomeration under extended operation could progressively reduce thermal conductivity enhancement and risk fouling of the heat transfer surface — particularly in the baffle window regions where cross-flow velocity is highest and particle inertial deposition is expected to be greatest. Periodic ultrasonic re-dispersion of the recirculating nanofluid is proposed as a maintenance strategy to restore particle size distribution to the initial specification.

5. Conclusions

This study has experimentally investigated and computationally optimised the thermal-hydraulic performance of Al_2O_3–water nanofluid-enhanced shell-and-tube heat exchangers across a four-dimensional design space encompassing nanofluid concentration, baffle spacing, baffle cut ratio, and Reynolds number. The principal findings and conclusions are:

Al_2O_3–water nanofluid at 1.0 vol% achieves the optimal balance of thermal conductivity enhancement (11.2% at 50°C) and viscosity penalty (20.3% increase), yielding a maximum overall heat transfer coefficient of 2,847 W/m²·K at the optimal baffle configuration — a 34.2% improvement over the base-fluid reference. The thermal performance factor of 1.28 confirms net energy benefit with justified pumping power expenditure.

Baffle spacing of 35% of shell diameter with 20% baffle cut represents the optimal segmental baffle configuration for the tested shell geometry, with a non-monotonic U–baffle spacing relationship confirmed by both experiment and CFD. The optimum baffle spacing is governed by the balance between cross-flow velocity enhancement and bypass flow increase at close spacing.

The validated CFD model (R^2 = 0.987, MAPE 3.8%) accurately reproduces experimental heat transfer and pressure drop across all tested configurations and provides physical insight into the flow mechanisms underlying the baffle spacing optimum through velocity and turbulent kinetic energy field analysis.

Response surface methodology with CCD design efficiently characterises the four-factor design space, identifying statistically significant ($p < 0.05$) main effects of all four factors and a significant nanofluid concentration–baffle spacing interaction term ($p = 0.003$), demonstrating the synergistic benefit of combining moderate nanofluid concentration with intermediate baffle spacing.

Energy-economic analysis confirms a payback period of approximately 4.6 months for retrofitting the optimal configuration in a representative 500 kW process plant STHE, providing strong economic justification for nanofluid adoption in Indian process industry heat exchanger upgrade programmes.

Future work should extend the investigation to higher-conductivity nanofluids (TiO_2, CuO, hybrid Al_2O_3–TiO_2), elevated-temperature operation, and multi-segmental baffle alternatives including helical baffles and rod baffles, which may offer further pressure drop reduction without sacrificing heat transfer coefficient.

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