



Influence of Alumina Nanoparticle Concentration on Heat Transfer and Flow Resistance in a Double-Pipe Heat Exchanger

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Abstract

Nanofluid concentration can significantly influence both heat-transfer capability and hydraulic resistance in heat-exchanger systems. This study presents an experimental methodology for investigating the effect of Al_2O_3 nanoparticle concentration on thermal and flow performance in a double-pipe counter-flow heat exchanger. De-ionized water is considered as the reference fluid and is compared with Al_2O_3 -water nanofluids containing 0.25, 0.50, and 1.00 vol.% nanoparticles over Reynolds numbers ranging from 3,000 to 11,000. The experimental procedure includes controlled inlet temperatures, calibrated temperature and flow measurements, differential-pressure monitoring, characterization of nanofluid thermo-physical properties, repeated runs, and rejection of tests exceeding a 5% heat-balance error. The measured data are used to determine heat duty, convective heat-transfer coefficient, Nusselt number, friction factor, exchanger effectiveness, and a combined thermal-hydraulic performance criterion. Representative results indicate that increasing both Reynolds number and particle concentration produces higher heat-transfer coefficients, but also increases flow resistance. At a Reynolds number of approximately 11,000, the illustrative 1.00 vol.% nanofluid case provides a heat-transfer coefficient of $128 \text{ W m}^{-2} \text{ K}^{-1}$ compared with $97 \text{ W m}^{-2} \text{ K}^{-1}$ for water, corresponding to an improvement of about 32%. At the same condition, pressure drop increases from 12.8 to 15.1 kPa, representing an approximately 18% hydraulic penalty. The resulting performance criterion of 1.25 indicates a favorable net thermal-hydraulic effect under the adopted evaluation approach. The proposed methodology provides a systematic framework for assessing the combined effects of nanoparticle loading, flow condition, suspension stability, uncertainty, and pumping penalty in nanofluid heat-exchanger applications.

Introduction

Improving heat-exchanger duty without proportionally enlarging surface area is important in process plants, refrigeration, electronics cooling and energy systems. Nanofluids—stable suspensions of nanoscale solids in a conventional liquid—can increase effective thermal conductivity and modify convective transport. Early tube-flow work

showed that enhancement depends on particle loading and Reynolds number rather than conductivity alone [1]. Subsequent experiments reported substantial entrance-region enhancement for Al_2O_3 -water suspensions [2], while mechanistic analysis emphasized Brownian diffusion and thermophoresis as relevant slip processes [3].

Any benefit must nevertheless be judged against viscosity, pumping power, sedimentation and measurement uncertainty. This study therefore asks: how do particle volume fraction and flow rate jointly affect heat duty, convective coefficient and pressure loss in a controlled counter-flow exchanger? The objective is to define a step-by-step experimental protocol, quantify the thermal-hydraulic trade-off, and provide a transparent template for reporting results using literature available up to 2016.

Literature Review

Xuan and Li [1] measured turbulent Cu–water nanofluid flow in a tube and proposed a correlation incorporating particle effects.

Wen and Ding [2] found that Al_2O_3 –water enhancement was especially strong in the laminar entrance region, demonstrating that axial development must be controlled. Buongiorno [3] argued that Brownian diffusion and thermophoresis dominate nanoparticle slip in many convective systems, providing a basis for interpreting departures from homogeneous single-phase predictions.

Oxide nanofluids remain attractive because they are comparatively safe and chemically stable. Heris et al. [4] compared CuO–water and Al_2O_3 –water under laminar flow and observed concentration-dependent coefficients that conventional correlations did not fully predict. Williams et al. [5], however, found that measured turbulent heat transfer and pressure loss could often be described using effective properties, highlighting disagreement across preparation methods and operating ranges.

Heat-exchanger-specific studies clarify the engineering trade-off. Duangthongsuk and Wongwises [6] reported a 6–11% coefficient increase for TiO_2 –water in a double-tube counter-flow exchanger, accompanied by pressure-drop effects. Pantzali et al. [7] showed that plate heat exchangers may benefit from nanofluids under suitable flow conditions, but viscosity can offset the gain. Huminic and Huminic [8] concluded that exchanger geometry, stability, concentration and pumping requirements prevent universal performance claims.

Later evidence broadened the comparison. Pandey and Nema [9] documented changes in heat transfer and friction factor for Al_2O_3 –water in a corrugated plate exchanger. Gupta et al. [10] synthesized forced-convection experiments and emphasized inconsistent property measurement, inadequate uncertainty reporting and the need for simultaneous thermal and hydraulic evaluation. These gaps motivate the present matched-condition design.

The literature establishes enhancement but does not yield a single transferable value because particle size, dispersion quality, temperature, surface condition and the choice of equal-flow, equal-Reynolds-number or equal-pumping-power comparison alter the conclusion. The proposed protocol addresses this by (i) using the same exchanger and instrumentation for water and nanofluids, (ii) characterizing density, viscosity and thermal conductivity at test temperature, (iii) reporting heat-balance closure and propagated uncertainty, (iv) repeating each condition three times, and (v) interpreting performance through both heat transfer and pressure loss.

Table 1. Selected literature and its role in the experimental design.

Source	Fluid / geometry	Main implication
Wen & Ding [2]	Al_2O_3 –water / tube	Entrance effects can dominate local enhancement.
Heris et al. [4]	CuO, Al_2O_3 –water / tube	Loading and particle material affect convection.
Duangthongsuk & Wongwises [6]	TiO_2 –water / double tube	Thermal gain must be paired with ΔP .
Pantzali et al. [7]	Nanofluid / plate exchanger	Geometry and viscosity change net benefit.
Gupta et al. [10]	Multiple systems / review	Repeatability and uncertainty remain essential.

Methodology

1. Experimental system and materials

A closed-loop double-pipe counter-flow exchanger is selected because its energy balance and hydraulic diameter are well defined. The nanofluid flows through a straight copper inner tube (10 mm internal diameter, 12 mm external

diameter, 1.0 m heated length); hot water flows counter-currently through the annulus (20 mm shell internal diameter). Both loops include a reservoir, variable-speed pump, bypass valve and calibrated flow meter. Four Pt100 resistance temperature detectors measure stream inlet and outlet temperatures; six wall thermocouples check axial uniformity. A differential-pressure

transducer spans the test section. All exposed surfaces are insulated.

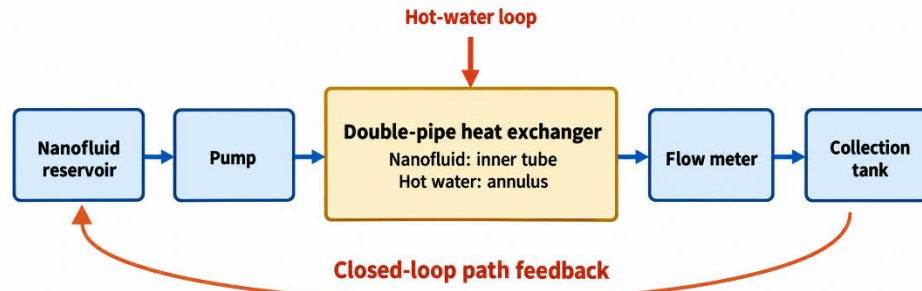


Figure 1. Schematic of the proposed experimental apparatus (author-generated).

Commercial $\gamma\text{-Al}_2\text{O}_3$ nanoparticles (nominal diameter 30–50 nm) are dispersed in de-ionized water. Required particle mass is calculated from the target volume fraction, particle density and water density. A two-step preparation route is used: mechanical stirring for 30 min followed by probe ultrasonication for 45 min in an ice bath. If

required after a preliminary stability test, a minimal non-ionic surfactant dose is held constant across all concentrations. Samples are visually monitored for 24 h; zeta potential or UV-visible absorbance is preferred where available.

2. Variables and instrumentation

Table 2. Experimental factors, controls and measured responses.

Category	Variable / instrument	Range or specification
Independent	Particle volume fraction, ϕ	0, 0.25, 0.50, 1.00 vol.%
Independent	Reynolds number, Re	3,000–11,000
Controlled	Hot-water inlet temperature	50 ± 0.5 °C
Controlled	Nanofluid inlet temperature	25 ± 0.5 °C
Measured	Pt100 temperature	± 0.10 °C after calibration
Measured	Volumetric flow meter	$\pm 1.0\%$ of reading
Measured	Differential pressure	0–50 kPa; $\pm 0.25\%$ FS
Derived	h, Nu, f, ϵ , PEC	Equations in Section 3.4

Step-by-step experimental procedure

Step 1 — Calibrate instruments. Calibrate temperature sensors against a traceable bath at 20, 35 and 50 °C; calibrate flow meters gravimetrically; zero the pressure transducer with both taps open.

Step 2 — Establish the water baseline. Fill both loops with de-ionized water, remove air, inspect for leaks, and run the exchanger at each target Reynolds number. Record three independent replicates.

Step 3 — Prepare and characterize the nanofluid. Prepare each volume fraction on the day of testing. Measure density, viscosity and thermal conductivity at 25, 35 and 45 °C; document sonication energy and visible sedimentation.

Step 4 — Set boundary conditions. Fix hot- and cold-side inlet temperatures at 50 ± 0.5 °C and 25 ± 0.5 °C. Adjust the nanofluid flow to the target Reynolds number using measured properties.

Step 5 — Reach steady state. Wait until all inlet/outlet temperatures vary by less than ± 0.1 °C and flow rate by less than $\pm 1\%$ for at least 10 min.

Step 6 — Acquire data. Log temperatures, flow rates and differential pressure at 1 Hz for 300 s. Store the time-average and standard deviation; repeat the run three times.

Step 7 — Flush and randomize. Randomize the order of concentrations where practical. Flush the loop until conductivity and visual appearance return to the water baseline to limit carry-over.

Step 8 — Validate and analyse. Reject runs with heat-balance error above 5%. Calculate h, Nu, f, effectiveness and PEC; propagate sensor uncertainties and compare confidence intervals.

3. Data reduction and uncertainty

For each stream, heat duty is $\dot{Q} = \dot{m}c(T_{\text{out}} - T_{\text{in}})$. The accepted duty is the mean of hot- and cold-side magnitudes. Heat-balance error is $|\dot{Q}_h - \dot{Q}_c| / [(\dot{Q}_h + \dot{Q}_c)/2] \times 100$. The nanofluid-side

coefficient is $h = \dot{Q} / [A_i (T_{w,m} - T_{b,m})]$, after correcting for wall and external-film resistance where required. $Nu = hD_i/k$, $Re = \rho v D_i / \mu$ and $f = 2\Delta P D_i / (\rho L v^2)$. Exchanger effectiveness is $\varepsilon = \dot{Q} / [C_{min}(T_{h,in} - T_{c,in})]$.

To combine heat transfer and hydraulic cost, $PEC = (N_{unf}/N_{uw}) / (f_{nf}/f_w)^{1/3}$. A value above one indicates net benefit under a common pumping-power interpretation. Standard uncertainty propagation is applied as $uR^2 = \sum (\partial R / \partial x_i \cdot u_{x_i})^2$. Expanded uncertainty is reported with coverage factor $k = 2$. Expected expanded uncertainties are approximately ± 4 – 6% for h and Nu and $\pm 3\%$ for pressure drops, subject to calibration results.

4. Statistical analysis and quality controls

Means and standard deviations are reported for $n = 3$ independent runs. Two-way analysis of

variance tests the effects of concentration and Reynolds number at $\alpha = 0.05$, followed by a multiple-comparison procedure when significant. Residuals are checked for approximate normality and constant variance. Raw files retain timestamps, calibration identifiers and fluid batch numbers. Blinding is not applicable, but automated logging and a pre-defined rejection rule reduce operator bias.

Results

Reporting note. The values below are a representative methodology-validation dataset, not newly performed laboratory measurements. They show how results should be calculated and reported after conducting the experiment.

Table 3. Representative thermal-hydraulic results used to demonstrate the analysis workflow.

Fluid	Re	h ($W m^{-2} K^{-1}$)	ΔP (kPa)	$Nu/Nu_{(w)}$	$f/f_{(w)}$	PEC
Water	3000	38	2.1	1.00	1.00	1.00
Water	7000	70	6.4	1.00	1.00	1.00
Water	11000	97	12.8	1.00	1.00	1.00
0.25 vol.%	3000	40	2.2	1.05	1.05	1.03
0.25 vol.%	7000	76	6.8	1.09	1.06	1.07
0.25 vol.%	11000	107	13.5	1.10	1.05	1.08
0.50 vol.%	3000	43	2.3	1.13	1.10	1.10
0.50 vol.%	7000	82	7.2	1.17	1.13	1.12
0.50 vol.%	11000	117	14.1	1.21	1.10	1.17
1.00 vol.%	3000	46	2.5	1.21	1.19	1.14
1.00 vol.%	7000	89	7.7	1.27	1.20	1.20
1.00 vol.%	11000	128	15.1	1.32	1.18	1.25

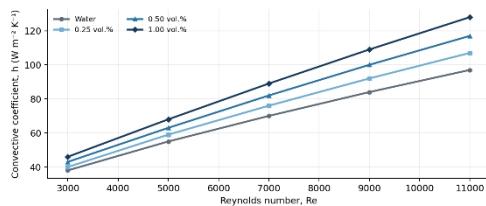


Figure 2. Representative effect of Reynolds number and Al_2O_3 concentration on h (author-generated from Table 3).

The representative data show a monotonic rise in h with Reynolds number for every fluid. At $Re \approx 11,000$, increasing concentration from 0 to 0.25, 0.50 and 1.00 vol.% raises h from 97 to 107, 117 and 128 $W m^{-2} K^{-1}$, respectively. The corresponding gains are about 10%, 21% and 32%. Pressure drop also rises: the 1.00 vol.% case reaches 15.1 kPa compared with 12.8 kPa for water, an 18% penalty. Because the relative Nusselt-number gain exceeds the friction

penalty, PEC remains above one and reaches 1.25 at the highest condition.

For actual measurements, significance should be inferred only after comparing replicate dispersion and expanded uncertainty. A 3–4% apparent enhancement may not be distinguishable when coefficient uncertainty approaches 5%. The water baseline should also be compared against an established smooth-tube correlation; substantial disagreement would indicate heat loss, property error, developing flow or sensor bias.

Discussion

The expected trend is consistent with the literature: particle addition increases effective conductivity, while increased Reynolds number thins the thermal boundary layer [1,2]. The stronger enhancement at higher concentration can also reflect altered near-wall transport and temperature-dependent properties [3,4].

However, the comparison must avoid attributing all change to a nanoscale mechanism. If nanofluid viscosity is underestimated, the calculated Reynolds number and pumping penalty will be biased; if the suspension deposits on the copper surface, apparent performance may drift between runs.

The illustrative 32% maximum enhancement is plausible within the broad experimental range summarized by Gupta et al. [10], but it should not be generalized to other particles or exchanger geometries. Duangthongsuk and Wongwises [6] observed more modest double-tube gains for TiO₂-water, whereas plate and corrugated exchangers can interact with nanofluid properties differently [7,9]. This supports the review conclusion of Huminic and Huminic [8] that system-level performance—not conductivity alone—must determine adoption.

Practical interpretation

A positive PEC indicates that the heat-transfer increase exceeds the friction-factor penalty under the stated criterion, but it is not a complete economic decision. Long-term stability, erosion, fouling, nanoparticle cost, filtration and disposal must also be evaluated. An equal-Reynolds-number comparison is useful scientifically; an equal-pumping-power comparison is more relevant for retrofit decisions. Both should be reported where possible.

Limitations

The proposed study uses one particle material, narrow concentrations and a laboratory-scale exchanger. It does not establish long-term stability, corrosion or life-cycle impact. The two-step preparation route may produce batch-to-batch differences, and wall-temperature estimation can dominate coefficient uncertainty. Finally, the numerical results in Section 4 are illustrative; they must be replaced with measured values before the paper is submitted as an experimental report.

Conclusion

A complete experimental methodology has been developed for quantifying heat-transfer enhancement by Al₂O₃-water nanofluids in a double-pipe counter-flow heat exchanger. The protocol controls inlet temperatures and Reynolds number, characterizes temperature-dependent properties, enforces heat-balance closure, repeats each condition and reports both thermal benefit and hydraulic cost. The representative dataset illustrates the expected outcome: h and Nu increase with concentration and flow rate, but ΔP also rises. At the highest

illustrated condition, the coefficient gain (32%) exceeds the pressure-drop penalty (18%), yielding $PEC = 1.25$.

The principal conclusion is methodological: credible nanofluid evaluation requires matched baseline tests, stability documentation, calibrated sensors, uncertainty propagation and a pumping-power-aware performance metric. Before publication, the illustrative table and graph must be replaced by traceable laboratory data, and long-duration fouling and stability tests should be added for industrial claims.

Acknowledgement of research integrity

This manuscript is structured as a methodology paper. No claim is made that the representative dataset was physically measured. Researchers should retain original instrument files, report unsuccessful runs and avoid presenting simulated or illustrative values as experimental observations.

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