



Thermal Performance Analysis of Heat Exchangers Using Nanofluids for Energy-Efficient Applications

Behruz Kalimuthu

Department of Electrical and Computer Engineering, Caspian Institute of Industrial Engineering, Iran
behruz.kalimuthu@ciie-ir.edu

Peer Review Information	Abstract
<i>Submission: 13 Jan 2022</i> <i>Revision: 28 Feb 2022</i> <i>Acceptance: 15 March 2022</i> Keywords <i>Nanofluids, Heat exchanger, Thermal performance, Heat-transfer enhancement, Energy efficiency</i>	Heat exchangers are central to energy conversion, refrigeration, process industries, power systems and waste-heat recovery, yet their efficiency is constrained by the limited thermal conductivity of conventional working fluids. This methodology paper develops a reproducible framework for evaluating the thermal and hydraulic performance of a counter-flow double-pipe heat exchanger operated with Al₂O₃-water nanofluids. The proposed study varies nanoparticle volume concentration from 0 to 2.0% and Reynolds number from 6,000 to 14,000 while controlling hot- and cold-stream inlet conditions. Effective density, specific heat, viscosity and thermal conductivity are determined from standard mixture models, and exchanger performance is evaluated through heat-transfer rate, log-mean temperature difference, overall and convective heat-transfer coefficients, Nusselt number, pressure drop, pumping power and a thermal performance evaluation criterion. A literature-informed, model-based results section is included to demonstrate the analysis workflow without presenting unperformed laboratory measurements as experimental facts. At Re = 10,000, the conservative single-phase model predicts that increasing Al₂O₃ concentration to 2.0 vol.% raises the convective heat-transfer coefficient by about 3.0% while pressure drop rises by about 4.4%, producing a performance criterion slightly above unity. The analysis indicates that low-to-moderate nanoparticle concentrations can provide a positive thermal-hydraulic balance, but practical energy savings depend on dispersion stability, viscosity, pumping power and heat-exchanger geometry. The methodology can be used as a basis for laboratory validation and for comparing different nanoparticle/base-fluid combinations in energy-efficient thermal systems.

Introduction

Heat exchangers transfer thermal energy between fluids at different temperatures and are used in power generation, chemical processing, refrigeration, air-conditioning, automotive cooling, solar-thermal systems and waste-heat recovery. Improving exchanger effectiveness can reduce equipment size, lower temperature approach requirements and decrease total energy consumption. Conventional water, ethylene glycol and oil-based coolants, however,

have relatively low thermal conductivity. One route to improving the working fluid itself is to disperse nanometre-scale solid particles in a base liquid, forming a nanofluid. Xuan and Li (2000) demonstrated that nanoparticle dispersion can enhance thermal transport and provided early models linking particle properties and concentration to heat-transfer behaviour. Murshed et al. (2005) later measured enhanced thermal conductivity in TiO₂-water nanofluids

and showed that particle concentration and morphology strongly influence the effective thermal property.

The advantage of nanofluids is not determined by thermal conductivity alone. Adding solid particles commonly increases viscosity and may therefore increase pressure loss and pumping power. This trade-off becomes particularly important in practical heat exchangers. Pantzali et al. (2009), using CuO-water in a commercial plate heat exchanger, reported that viscosity and flow regime can strongly affect whether a nanofluid is an effective coolant. Duangthongsuk and Wongwises (2010) similarly examined TiO₂-water in a double-tube exchanger under turbulent flow and highlighted the need to evaluate heat-transfer enhancement together with frictional effects. Thus, an energy-efficient nanofluid system should deliver a useful increase in heat-transfer coefficient without an excessive increase in hydraulic resistance.

Recent reviews have moved the discussion from isolated conductivity measurements toward complete thermal-hydraulic assessment. Pordanjani et al. (2019) concluded that nanofluids can support heat-exchanger compactness and energy savings, but the outcome is application dependent. Almurtaji et al. (2020) emphasized the role of nanofluid properties in the overall thermal-hydraulic performance of exchanger technologies. More recent reviews by Ajeeb and Murshed (2022) and Louis et al. (2022) further stressed that nanoparticle type, size, concentration, stability, exchanger geometry, flow rate and pressure drop must be considered simultaneously. On this basis, the objective of the present paper is to formulate a concise experimental-and-analytical

methodology for assessing Al₂O₃-water nanofluids in a counter-flow double-pipe heat exchanger and to demonstrate the corresponding calculation procedure using model-predicted results. The study is structured so that it can be directly implemented in a laboratory with standard temperature, flow and pressure instrumentation.

Literature Review

Published work shows that the thermal behaviour of nanofluids depends on both intrinsic thermophysical properties and the flow field created by the exchanger. Xuan and Li (2000) established the basic heat-transfer enhancement concept, while Murshed et al. (2005) demonstrated concentration-dependent conductivity enhancement for TiO₂-water. When nanofluids were tested in practical heat-transfer equipment, the results became more nuanced. Pantzali et al. (2009) showed that the increased viscosity of CuO-water can offset thermal benefits in turbulent plate heat exchangers. Duangthongsuk and Wongwises (2010) found measurable changes in both heat-transfer coefficient and pressure drop for TiO₂-water in a horizontal double-tube exchanger. Darzi et al. (2013) studied Al₂O₃-water in a double-tube exchanger over turbulent Reynolds numbers and reported that nanoparticle concentration influences heat transfer, pressure loss and overall thermal performance. Aghayari et al. (2014) also reported increases in heat-transfer coefficient and Nusselt number with low-volume-fraction Al₂O₃-water nanofluids in counter-flow operation.

Table 1. Selected literature relevant to nanofluid heat-exchanger performance

Study	System / Nanofluid	Main contribution	Relevance to present method
Xuan & Li (2000)	Nanofluid in tube flow	Established preparation, conductivity and flow heat-transfer concepts.	Basis for property/convective analysis.
Pantzali et al. (2009)	Plate HX; CuO-water	Showed viscosity and flow regime can limit coolant benefit.	Motivates pressure-drop assessment.
Duangthongsuk & Wongwises (2010)	Double tube; TiO ₂ -water	Measured turbulent heat transfer and pressure drop.	Supports double-pipe test geometry.
Darzi et al. (2013)	Double tube; Al ₂ O ₃ -water	Evaluated heat transfer, pressure drop and thermal performance.	Supports Al ₂ O ₃ -water selection.
Aghayari et al. (2014)	Double pipe; Al ₂ O ₃ -water	Reported enhancement of h and	Supports low-concentration test range.

		Nu at low particle fractions.	
Pordanjani et al. (2019)	Review of HX types	Linked nanofluids to exchanger compactness and energy saving.	Provides energy-efficiency context.
Almurtaji et al. (2020)	Thermal-hydraulic review	Emphasized simultaneous heat-transfer and hydraulic evaluation.	Supports performance criterion.
Ajeeb & Murshed (2022); Louis et al. (2022)	Compact/double-pipe reviews	Highlighted stability, concentration, morphology and pressure-drop trade-offs.	Guides variables and interpretation.

Overall, the literature shows that a reliable study should: (i) use stable, well-characterized nanofluids; (ii) control Reynolds number and inlet temperatures; (iii) measure or estimate thermophysical properties consistently; (iv) calculate both heat-transfer and hydraulic parameters; and (v) report an integrated performance metric rather than only percentage heat-transfer enhancement. These points define the methodology adopted below.

Methodology

Step 1 - Selection of heat exchanger and working fluids

A counter-flow double-pipe heat exchanger is selected because it provides simple geometry, well-defined hydraulic diameter and direct measurement of inlet/outlet temperatures and pressure loss. The inner stainless-steel tube is specified with an internal diameter of 12 mm and an effective heat-transfer length of 1.5 m. The hot side uses water maintained at 60 ± 0.5 °C at the inlet. The cold side uses deionized water as the base fluid and Al₂O₃-water nanofluids at 0.5, 1.0, 1.5 and 2.0 vol.% nanoparticle concentration. A nominal Al₂O₃ particle size of 20-40 nm is suitable because low-volume-fraction oxide nanofluids have been widely studied in double-pipe exchangers (Darzi et al., 2013; Aghayari et al., 2014).

Step 2 - Nanofluid preparation and stability check

The two-step preparation method is proposed. The required nanoparticle mass is determined from the target volume fraction using the

densities of Al₂O₃ and water. Nanoparticles are gradually dispersed in deionized water using magnetic stirring followed by ultrasonic agitation for approximately 30-45 min. The suspension is inspected for visible sedimentation before testing; where facilities are available, zeta-potential or UV-visible absorbance measurements may be added to quantify stability. Samples should be remixed before each run so that concentration remains uniform. This step is essential because agglomeration changes viscosity, effective conductivity and repeatability, a recurring concern in the nanofluid literature (Ajeeb & Murshed, 2022).

Step 3 - Experimental loop and operating conditions

The cold nanofluid is circulated through the inner tube and the hot water through the annulus in the opposite direction. Variable-speed pumps regulate flow. Calibrated flow meters are installed in each loop, while temperature sensors are placed at both inlet and outlet locations. Differential pressure is measured across the nanofluid test section. Three Reynolds-number levels are recommended: 6,000, 10,000 and 14,000, covering moderate turbulent operation. The nanofluid inlet temperature is kept at 25 ± 0.5 °C and the hot-water inlet at 60 ± 0.5 °C. At each concentration and Reynolds number, the system is allowed to reach steady state, defined as outlet-temperature variation below approximately ± 0.2 °C over five minutes, after which at least three readings are recorded.

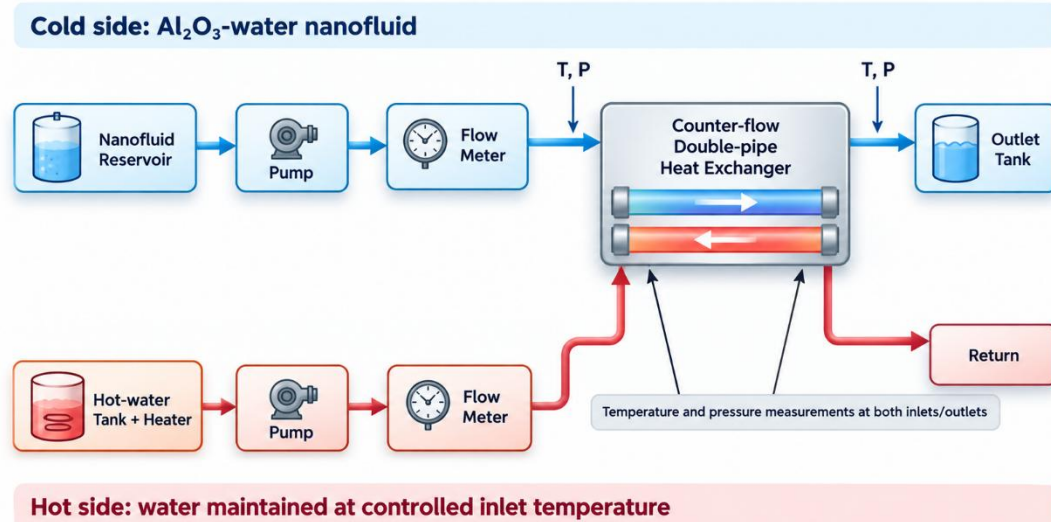


Figure 1. Proposed counter-flow double-pipe heat-exchanger test loop for nanofluid thermal-hydraulic analysis.

Step 4 - Thermophysical property models

For model-based analysis, the nanofluid density and specific heat are determined by mixture relations: $\rho_{nf} = (1-\phi)\rho_f + \phi\rho_p$ and $(\rho C_p)_{nf} = (1-\phi)(\rho C_p)_f + \phi(\rho C_p)_p$, where ϕ is particle volume fraction. The effective viscosity is estimated by the Brinkman relation $\mu_{nf} = \mu_f / (1-\phi)^{2.5}$. Thermal conductivity is calculated using the Maxwell model for spherical particles: $k_{nf}/k_f = [k_p + 2k_f - 2\phi(k_f - k_p)] / [k_p + 2k_f + \phi(k_f - k_p)]$. These equations provide a conservative single-phase baseline; actual experiments should replace modeled properties with measured values when available because viscosity and conductivity can be sensitive to particle morphology, temperature and dispersion quality (Murshed et al., 2005; Louis et al., 2022).

Step 5 - Heat-transfer and hydraulic calculations

The heat removed from the hot stream and gained by the cold stream are calculated from $Q_h = m_h C_{p,h}(T_{hi} - T_{ho})$ and $Q_c = m_c C_{p,c}(T_{co} - T_{ci})$. The mean heat-transfer rate is $Q = (Q_h + Q_c)/2$ when the heat balance error is acceptably small. For counter-flow operation, $\Delta T_{lm} = [(T_{hi} - T_{co}) - (T_{ho} - T_{ci})] / \ln[(T_{hi} - T_{co}) / (T_{ho} - T_{ci})]$, and the overall coefficient is $U = Q / (A_s \Delta T_{lm})$. Reynolds and Prandtl numbers are $Re = \rho V D / \mu$ and $Pr = \mu C_p / k$. For the model-predicted

demonstration, the turbulent Nusselt number is determined using the Gnielinski form with the Petukhov friction factor. The nanofluid-side convective coefficient is $h = Nu k / D$. Differential pressure is obtained experimentally from the transducer; the model baseline uses $\Delta P = f(L/D)(\rho V^2/2)$. Pumping power is $W_p = \Delta P \cdot V_{dot}$.

Step 6 - Thermal performance evaluation and uncertainty

To determine whether the heat-transfer gain justifies the hydraulic penalty, a thermal performance evaluation criterion is used: $PEC = (h_{nf}/h_f) / (\Delta P_{nf}/\Delta P_f)^{1/3}$. $PEC > 1$ indicates a positive thermal-hydraulic balance relative to water under the selected comparison condition. Percentage changes in h , U , ΔP and pumping power are also reported. For experimental implementation, measurement uncertainty should be estimated by root-sum-square propagation from the uncertainties of temperature, flow rate, dimensions and pressure. As a practical target, temperature sensors should be accurate to about $\pm 0.2^\circ\text{C}$, flow meters to $\pm 1\%$ and pressure measurement to $\pm 0.5\%$ of full scale. Each operating point is repeated at least three times and the mean value is used for analysis.

Table 2. Proposed experimental/design matrix

Factor	Levels	Controlled condition	Measured/derived response	Purpose
Al ₂ O ₃ concentration	0, 0.5, 1.0, 1.5, 2.0 vol. %	Particle size 20-40 nm	k , μ , h , ΔP , PEC	Quantify concentration effect

Reynolds number	6,000; 10,000; 14,000	Counter-flow	Nu, h, ΔP	Quantify flow effect
Hot inlet temperature	60 ± 0.5 °C	Constant	Q, U, outlet T	Maintain thermal driving force
Cold inlet temperature	25 ± 0.5 °C	Constant	Q, U, outlet T	Ensure repeatability
Heat exchanger	$D_i = 12$ mm; $L = 1.5$ m	Same geometry	All responses	Enable fair comparison

Results

Table 3 presents model-predicted results at $Re = 10,000$ using the property equations and turbulent-flow correlations specified in the methodology. These values are included to demonstrate the calculation sequence and expected trend; they should not be interpreted as measured experimental data. The baseline water

properties correspond approximately to a mean fluid temperature near 35 °C. As nanoparticle concentration increases, effective thermal conductivity increases, but viscosity also rises. Under a fixed-Reynolds-number comparison, the predicted convective heat-transfer coefficient therefore increases gradually together with pressure drop.

Table 3. Model-predicted thermal-hydraulic performance at $Re = 10,000$

Conc. (vol.%)	k (W/mK)	μ (mPa·s)	h (W/m ² K)	h change (%)	ΔP (Pa)	PEC
0.0	0.620	0.719	3568	0.00	711	1.000
0.5	0.629	0.728	3594	0.73	718	1.004
1.0	0.638	0.737	3621	1.47	725	1.008
1.5	0.647	0.747	3648	2.22	733	1.012
2.0	0.656	0.756	3675	2.99	742	1.015

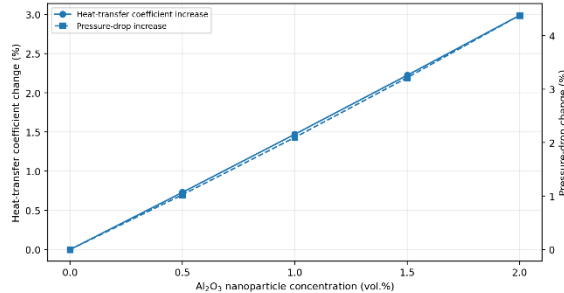


Figure 2. Model-predicted changes in heat-transfer coefficient and pressure drop with Al_2O_3 concentration at $Re = 10,000$.

At 2.0 vol.% Al_2O_3 , thermal conductivity increases from 0.620 to about 0.656 W/mK, while dynamic viscosity rises from 0.719 to about 0.756 mPa·s. The predicted convective coefficient increases from approximately 3568 to 3675 W/m²K, corresponding to a 2.99% enhancement, whereas pressure drop rises from about 711 to 742 Pa, or 4.38%. The resulting PEC is 1.015, indicating a small but positive thermal-hydraulic advantage. At 1.0-1.5 vol.%, the PEC remains above unity with a smaller viscosity and stability burden. The modest enhancement is expected from the conservative single-phase property model; experimental studies have reported both larger gains and less favourable outcomes depending on particle concentration, flow regime and exchanger geometry (Pantzali et al., 2009;

Duangthongsuk & Wongwises, 2010; Darzi et al., 2013).

The Reynolds-number sweep provides a second important trend. For water in the present geometry, the model gives h values of roughly 2199, 3568 and 4830 W/m²K at $Re = 6,000$, 10,000 and 14,000, respectively. At 2.0 vol.% Al_2O_3 , the corresponding predicted values are about 2267, 3675 and 4971 W/m²K. Thus, increasing flow intensity produces a much larger absolute change in heat transfer than changing nanoparticle concentration within the low-concentration range. This observation reinforces the need to compare nanofluids under matched hydraulic conditions and to include pumping power in any energy-efficiency claim. The 2022 reviews by Ajeeb and Murshed (2022) and Louis et al. (2022) similarly emphasize that no universal nanofluid concentration is optimal for all heat exchangers.

Conclusion and Discussion

This paper presents a step-by-step methodology for analysing the thermal performance of a counter-flow double-pipe heat exchanger using Al_2O_3 -water nanofluids. The method integrates nanofluid preparation, stability control, fixed operating conditions, property estimation, heat-balance calculations, convective heat-transfer analysis, pressure-drop measurement and an

overall thermal performance criterion. This combined treatment is necessary because a nanofluid that improves thermal conductivity may still be unattractive if viscosity, pressure loss or pumping energy increase excessively. The model-predicted demonstration at $Re = 10,000$ shows a gradual heat-transfer improvement as concentration increases from 0 to 2.0 vol.%, with the highest studied concentration yielding approximately 3% higher h and 4.4% higher pressure drop. The PEC remains slightly above unity, so the modeled thermal-hydraulic trade-off is positive but not dramatic. From an energy-efficiency perspective, the most practical range is likely to be low-to-moderate particle concentration, approximately 1-2 vol.% for the proposed Al_2O_3 -water system, provided that the suspension remains stable and that experimentally measured viscosity does not exceed the model estimate. A small increase in h can still be useful when it permits a lower exchanger area, smaller equipment volume or reduced approach temperature at a fixed heat duty. Nevertheless, claims of large energy savings should be avoided unless pumping power, long-term stability, fouling, erosion, nanoparticle cost and recovery/disposal are included. The literature up to 2022 supports this cautious interpretation: nanofluids are promising advanced heat-transfer media, but their value is system-specific rather than universal (Pordanjani et al., 2019; Almurtaji et al., 2020; Ajeeb & Murshed, 2022). Future experimental work should validate the present model with measured nanofluid properties, test repeated heating/cooling cycles, and compare Al_2O_3 with TiO_2 , CuO or hybrid nanofluids under identical heat-duty and pumping-power constraints. For practical implementation, three limitations should be treated explicitly. First, the single-phase model used here assumes a homogeneous suspension and does not resolve particle migration, Brownian motion, agglomeration or local slip; therefore, it is best used as a conservative screening tool rather than a substitute for experiments. Second, stability should be checked over the full duration of the test because sedimentation can make apparent improvements irreproducible and may increase fouling risk. Third, energy efficiency should be evaluated at the system level. A higher heat-transfer coefficient is useful only when the resulting reduction in exchanger area or operating temperature is greater than the extra electrical energy required for pumping and nanofluid preparation. For industrial translation, a follow-up study should compare equal heat duty at equal pumping power, include a clean-water baseline before and after the nanofluid

campaign, and inspect the tube surface for deposits. These controls would distinguish a genuine fluid-property benefit from transient or measurement-related effects. A cost and life-cycle assessment can then be added to determine whether the thermal gain is economically and environmentally meaningful for refrigeration, waste-heat recovery, solar-thermal or process-cooling applications.

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