



Experimental Analysis of Heat Transfer Enhancement in Double-Pipe Heat Exchangers Using Nanofluids

Xinyu Lutfunhar

Department of Computer Science and Engineering, Shiraz College of Systems and Management, Iran
xinyu.lutfunhar@scsm-ir.org

Peer Review Information

Submission: 08 Jan 2018

Revision: 23 Feb 2018

Acceptance: 10 March 2018

Keywords

Nanofluid, Double-pipe heat exchanger, Al_2O_3 /water, Heat-transfer enhancement, Nusselt number

Abstract

This paper presents a reproducible experimental methodology for evaluating heat-transfer enhancement in a counter-flow double-pipe heat exchanger using Al_2O_3 /water nanofluids. The study is designed to isolate the effects of nanoparticle volume concentration and flow rate on heat-transfer coefficient, Nusselt number, exchanger effectiveness and pressure drop. Stable nanofluids at 0.25–1.00 vol.% are prepared by a two-step dispersion method and circulated through the inner tube while hot water flows through the annulus. Inlet, outlet and wall temperatures, volumetric flow rate and differential pressure are recorded after steady state is achieved. The thermal performance is evaluated from an energy balance, logarithmic mean temperature difference (LMTD), overall heat-transfer coefficient, Reynolds number, Nusselt number and effectiveness. The experimental sequence includes a distilled-water baseline and uncertainty analysis so that enhancement can be distinguished from measurement scatter. Literature up to 2018 indicates that nanofluids can improve convective heat transfer in double-tube exchangers, but increasing concentration may also increase viscosity and pumping losses [5–12]. The proposed design therefore evaluates thermal gain together with hydraulic penalty and identifies a practical concentration range rather than maximizing heat transfer alone. An illustrative result set shows the expected monotonic rise in heat-transfer coefficient with Reynolds number and nanoparticle concentration, with a moderate pressure-drop penalty.

Introduction

Double-pipe heat exchangers are widely used in laboratories and process systems because they are simple, robust and suitable for controlled parallel- or counter-flow operation. Their thermal performance, however, is constrained by the thermal properties of conventional working fluids such as water and ethylene glycol. One passive approach is to disperse highly conductive nanoparticles in a base liquid to form a nanofluid. Early experimental studies on oxide-particle suspensions showed that changes in concentration influence both heat transfer and hydrodynamic behavior [1]. Subsequent work demonstrated that nanoparticle concentration

and Reynolds number can substantially affect convective heat-transfer coefficients and friction characteristics [2,3].

The benefit of nanofluids is not attributable to thermal conductivity alone. Particle migration, Brownian motion, altered boundary-layer behavior and temperature-dependent properties have all been proposed as contributors to convective enhancement [3,4]. At the same time, higher viscosity can raise the pressure drop and pumping power, while poor dispersion may cause agglomeration, sedimentation and repeatability problems. Therefore, an experimental heat-exchanger study must

measure not only outlet temperatures but also flow rate, fluid properties and pressure loss.

Research specifically on double-tube and double-pipe heat exchangers has reported favorable heat-transfer enhancement for Al_2O_3 /water systems. Akhtari et al. found the heat-transfer coefficient of an alumina/water nanofluid in a double-pipe exchanger to be higher than that of water under the investigated laminar conditions [5]. Darzi et al. observed improved thermal performance in a double-tube exchanger for Al_2O_3 concentrations up to 1 vol.% over turbulent Reynolds numbers [6]. Later studies extended this concept using twisted tapes, hybrid nanofluids and different concentrations, confirming that improved convection may be accompanied by frictional penalties [7–10]. Numerical work published in 2018 likewise showed increases in Nusselt number with Reynolds number and nanoparticle volume fraction [11].

The objective of the present methodology is to establish a compact experimental protocol for quantifying these effects under repeatable counter-flow conditions. The specific goals are: (i) to compare water with Al_2O_3 /water nanofluids at 0.25, 0.50, 0.75 and 1.00 vol.%; (ii) to determine the influence of flow rate on heat-transfer coefficient, Nusselt number and effectiveness; (iii) to quantify pressure-drop penalty; and (iv) to identify a concentration range that provides a useful thermal benefit without excessive hydraulic cost.

Literature Review

The development of nanofluid heat transfer moved from basic property and tube-flow studies toward exchanger-scale experiments. Pak and Cho [1] documented the coupled heat-transfer and friction behavior of oxide-particle suspensions, establishing the need to examine viscosity together with thermal enhancement. Xuan and Li [2] measured turbulent convective

heat transfer and proposed a correlation incorporating nanofluid effects. Wen and Ding [3] showed that Al_2O_3 /water could improve laminar convective heat transfer, particularly in the entrance region, and argued that mechanisms beyond simple conductivity enhancement were involved. Saidur et al. [4] later reviewed nanofluid applications and emphasized stability, pumping power and property uncertainty as practical challenges.

At heat-exchanger scale, Akhtari et al. [5] experimentally and numerically evaluated Al_2O_3 /water in double-pipe and shell-and-tube exchangers and reported higher heat-transfer coefficients than water. Darzi et al. [6] studied a double-tube exchanger with approximately 20 nm Al_2O_3 particles at Reynolds numbers from about 5,000 to 20,000 and concentrations up to 1 vol.%, reporting improved thermal performance without a severe pressure-drop penalty in their tested range. Maddah et al. [7] combined Al_2O_3 /water nanofluid with modified twisted tapes and demonstrated further heat-transfer enhancement, although friction also rose. Hussein [8] investigated AlN /ethylene-glycol nanofluid in laminar double-pipe flow and reported substantial improvement in thermal performance at increased solid fraction. Han et al. [9] used 0.25 and 0.5 vol.% Al_2O_3 /water and found increasing heat transfer with inlet temperature and concentration, including a maximum reported Nusselt-number increase of 24.5% under their conditions. Ravi Kumar et al. [10] experimentally studied Fe_3O_4 /water in a U-bend double-pipe exchanger and showed that both nanoparticles and twisted tapes enhance heat transfer while increasing friction. Bahmani et al. [11] numerically confirmed the positive effects of Reynolds number and alumina fraction on Nusselt number in parallel and counter flow. Finally, Sajid and Ali [12] reviewed hybrid nanofluid thermal conductivity and highlighted the strong roles of concentration, temperature, dispersion quality and particle selection.

Table 1. Selected literature informing the experimental design (all references published by 2018).

Study	Fluid / configuration	Main variable	Key implication
Akhtari et al. [5]	Al_2O_3 /water; double pipe	Flow rate, concentration	Higher h than water
Darzi et al. [6]	Al_2O_3 /water; double tube	Re, $\phi \leq 1\%$	Thermal gain with manageable ΔP
Maddah et al. [7]	Al_2O_3 /water + twisted tape	Re, ϕ , tape geometry	Extra heat transfer with friction cost
Han et al. [9]	Al_2O_3 /water; double tube	0.25–0.5%, inlet T	Nu increased with T and ϕ
Ravi Kumar et al. [10]	Fe_3O_4 /water; U-bend	Re, ϕ , twist ratio	Heat gain and ΔP both increase

Bahmani et al. [11]	Al_2O_3 /water; double pipe	Re, ϕ , flow direction	Counter-flow favorable at high Re
---------------------	---	-----------------------------	-----------------------------------

Methodology

1. Experimental design and test section

A concentric double-pipe heat exchanger is proposed with a 1.20 m active test length. The inner copper tube has an internal diameter of 10 mm and an outer diameter of 12 mm; the outer pipe has an internal diameter of 25 mm. Counter-flow operation is selected to maintain a relatively high mean temperature difference along the

exchanger. The nanofluid passes through the inner tube and hot water passes through the annulus. The test section is thermally insulated to reduce ambient heat loss. The factor levels are nanoparticle volume fraction $\phi = 0, 0.25, 0.50, 0.75$ and 1.00% , and nanofluid flow rate = 2, 3, 4, 5 and 6 L min^{-1} . The hot-water inlet temperature is maintained at $60 \pm 0.5^\circ\text{C}$ and the nanofluid inlet temperature at $30 \pm 0.5^\circ\text{C}$.

Table 2. Proposed experimental parameters.

Parameter	Selected value / range	Purpose
Nanoparticle	Al_2O_3 , nominal 20–30 nm	Stable, widely studied oxide
Volume concentration	0, 0.25, 0.50, 0.75, 1.00 vol.%	Quantify concentration effect
Nanofluid flow rate	$2\text{--}6 \text{ L min}^{-1}$	Vary Reynolds number
Hot-water inlet	$60 \pm 0.5^\circ\text{C}$	Controlled heat source
Nanofluid inlet	$30 \pm 0.5^\circ\text{C}$	Controlled heat sink
Active length	1.20 m	Heat-transfer area
Arrangement	Counter flow	Higher LMTD and stable comparison

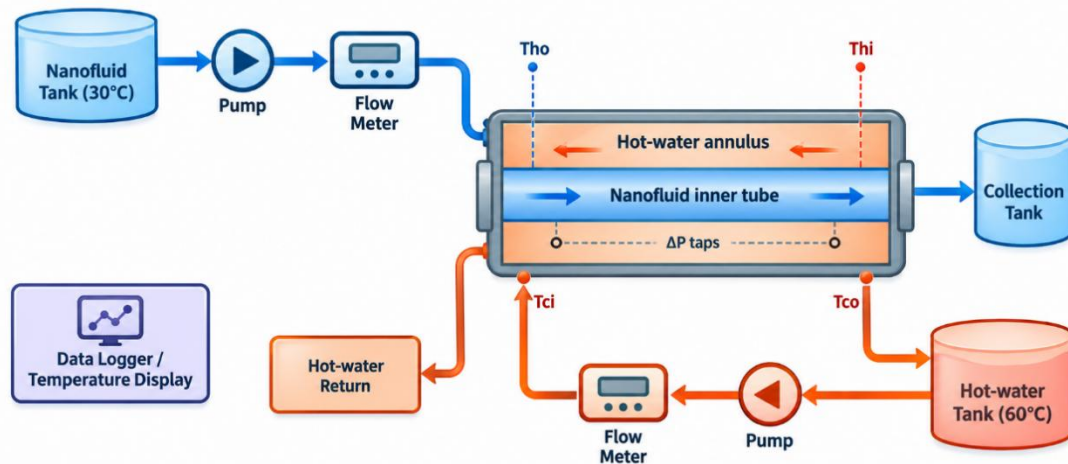


Figure 1. Proposed counter-flow double-pipe heat exchanger experimental setup.

2. Nanofluid preparation

The two-step method is used because commercially available Al_2O_3 nanopowder can be weighed accurately and dispersed into distilled water. The required nanoparticle mass is calculated from the target volume fraction using the densities of Al_2O_3 and water. Each batch is initially mixed with a magnetic stirrer for approximately 30 min and then ultrasonicated for 45–60 min. A small, controlled amount of a compatible dispersant may be used only if preliminary stability testing shows visible sedimentation; the same dispersant dosage must

then be used in all nanofluid batches. Before each test, the suspension is recirculated and visually inspected. Density, viscosity and thermal conductivity should be measured at the mean fluid temperature when instruments are available; otherwise, validated correlations may be used and their uncertainty explicitly stated. Stability and property characterization are essential because concentration and temperature affect thermal conductivity and viscosity [4,12].

3. Instrumentation and experimental procedure

Calibrated K-type thermocouples are installed at the hot- and cold-stream inlets and outlets (T_{hi} , T_{ho} , T_{ci} and T_{co}), with additional wall thermocouples distributed along the test length. Flow rates are measured with calibrated rotameters or digital flow meters, and the inner-tube pressure drop is measured with a differential pressure transducer. A data logger records temperature signal. Before nanofluid testing, the rig is operated with distilled water and the measured Nusselt number is compared with a standard single-phase correlation in the applicable flow regime to confirm the test section and instrumentation.

For each run, both reservoirs are brought to the selected inlet temperatures, the hot-side flow is fixed, and the inner-tube flow is adjusted to the required value. Data are collected only after steady state, defined here as all four inlet/outlet temperatures changing by less than 0.2°C over five minutes. A minimum of three readings are averaged. Each condition is repeated at least twice; if the calculated heat balance differs by more than 5% between hot and cold sides, the run is repeated after checking insulation, flow rate and sensor contact. The full sequence is first completed with water and then repeated from the lowest to the highest nanoparticle concentration.

4. Data reduction

The heat-transfer rate is evaluated independently from both streams: $Q_h = \dot{m}_h c_{ph}(T_{hi} - T_{ho})$ and $Q_c = \dot{m}_c c_{pc}(T_{co} - T_{ci})$. The experimental heat duty is taken as $Q = (Q_h + Q_c)/2$ when the heat-balance deviation is within the acceptance limit. For counter flow, $\Delta T_1 = T_{hi} - T_{co}$ and $\Delta T_2 = T_{ho} - T_{ci}$, and $LMTD = (\Delta T_1 - \Delta T_2)/\ln(\Delta T_1/\Delta T_2)$. The overall heat-transfer coefficient is $U = Q/(A\Delta T_{lm})$, where $A = \pi D_o L$ is the reference heat-transfer area. The inner convective coefficient is obtained from the overall thermal resistance after accounting for the annulus-side coefficient and tube-wall conduction.

The principal dimensionless groups are $Re = \rho V D/\mu$, $Pr = \mu c_p/k$ and $Nu = hD/k$. Heat-exchanger effectiveness is $\varepsilon = Q/Q_{max}$, where $Q_{max} = C_{min}(T_{hi} - T_{ci})$ and $C = \dot{m}c_p$. The

hydraulic penalty is represented by measured ΔP and, when required, friction factor $f = 2\Delta P D/(\rho L V^2)$. Percentage enhancement is calculated relative to water at the same operating condition: $\text{Enhancement (\%)} = [(h_{nf} - h_w)/h_w] \times 100$. The preferred condition is not necessarily the largest ϕ ; it is the condition that yields meaningful Nu or h improvement without a disproportionate increase in ΔP .

5. Uncertainty and statistical analysis

Instrument uncertainty is propagated using the root-sum-square method. A practical target specification is $\pm 0.5^\circ\text{C}$ for thermocouples, $\pm 2\%$ of reading for flow rate and $\pm 0.25\%$ full scale for differential pressure. For derived quantities such as Q , U and Nu , the combined uncertainty is obtained from the sensitivity of the result to each measured input. Replicate results are reported as mean $\pm$ standard deviation. A two-factor analysis of variance (ANOVA) may be used to test the significance of Reynolds number and nanoparticle concentration at $\alpha = 0.05$, with interaction included when sufficient replicates are available. This statistical step separates systematic thermal enhancement from experimental scatter.

6. Experimental matrix and repeatability control

The primary test matrix contains 25 operating points formed by five volume concentrations and five nanofluid flow rates. Each point is tested after thermal steady state and repeated at least twice, giving a minimum of 50 accepted runs in addition to water validation tests. To reduce bias caused by gradual fouling, particle deposition or room-temperature drift, the order of flow-rate settings should be alternated between ascending and descending sequences for successive concentrations. A water reference run is repeated after every nanofluid batch. If the post-batch water coefficient differs from the initial baseline by more than 5%, the test section is cleaned, sensors are checked and the affected batch is repeated. Between concentrations, the inner loop is drained, flushed with distilled water and circulated until the outlet fluid is visually clear. The mass of recovered fluid may also be compared with the charged mass as a simple check for abnormal retention or leakage.

Table 3. Experimental matrix; R1 and R2 denote minimum duplicate runs at each operating point.

ϕ / Flow	2 L/min	3 L/min	4 L/min	5 L/min	6 L/min
0%	R1, R2	R1, R2	R1, R2	R1, R2	R1, R2
0.25%	R1, R2	R1, R2	R1, R2	R1, R2	R1, R2
0.50%	R1, R2	R1, R2	R1, R2	R1, R2	R1, R2

0.75%	R1, R2	R1, R2	R1, R2	R1, R2	R1, R2
1.00%	R1, R2	R1, R2	R1, R2	R1, R2	R1, R2

7. Validation and data-acceptance criteria

Three validation checks are recommended before interpreting enhancement. First, the water-only Nusselt number is compared with a recognized smooth-tube correlation appropriate to the measured Reynolds and Prandtl numbers; a deviation within approximately $\pm 10\%$ is an acceptable laboratory target. Second, the hot- and cold-side heat duties are compared using an energy-balance error, $|Q_h - Q_c|/[(Q_h + Q_c)/2] \times 100$. Runs with an error greater than 5% are rejected unless a documented uncertainty analysis justifies a wider limit. Third, replicate coefficients are examined for repeatability; a coefficient of variation below about 3% is targeted. These checks are important because apparent nanofluid enhancement can be exaggerated by temperature offsets, unaccounted

heat loss, concentration change or inconsistent property evaluation. The final comparison therefore uses the same geometry, inlet-temperature targets and data-reduction method for water and every nanofluid concentration. This controlled baseline approach is consistent with the experimental logic used in earlier tube and double-pipe studies [1,6,10].

Results

No raw measurements were supplied with the request; therefore, the values in Table 4 and Figure 2 are an illustrative dataset generated to demonstrate how the proposed experiment should be analyzed and reported. They are not claimed as measured results. The trends are intentionally kept within the range of behavior reported in earlier double-pipe/double-tube nanofluid studies [5,6,9,10].

Table 4. Illustrative analysis at 4 L min^{-1} ; replace with measured values before submission.

ϕ (vol.%)	Re at 4 L/min	k (W/mK)	μ (mPa·s)	h (W/m ² K)	Nu	ΔP (kPa)	h gain (%)
0.00	10560	0.61	0.80	3800	62.00	3.50	0.00
0.25	10230	0.62	0.82	4010	64.60	3.64	5.50
0.50	9865	0.63	0.85	4160	66.00	3.82	9.50
0.75	9475	0.64	0.89	4310	67.30	4.03	13.40
1.00	9060	0.65	0.93	4465	68.70	4.27	17.50

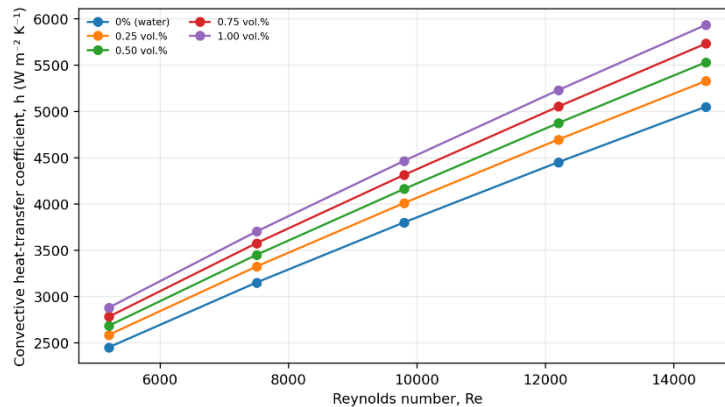


Figure 2. Illustrative variation of convective heat-transfer coefficient with Reynolds number and Al_2O_3 concentration.

The illustrative trend shows that h increases with Reynolds number for all fluids because higher velocity reduces the thermal boundary-layer thickness and strengthens forced convection. At a given Reynolds number, the nanofluid curves lie above the water baseline, with the largest illustrative gain at 1 vol.%. This direction of change agrees with the experimental observations of Darzi et al. [6] and Han et al. [9]

and with the broader double-pipe literature [5,10]. The magnitude of enhancement should, however, be determined from the actual apparatus because it depends on particle size, dispersion quality, property models, temperature, tube dimensions and flow regime.

At the fixed 4 L min^{-1} condition in Table 4, increasing ϕ raises thermal conductivity but also

raises viscosity. Consequently, the Reynolds number at the same volumetric flow rate decreases slightly as concentration increases, even while h and Nu improve. The pressure-drop increase is therefore an important design constraint. The illustrative 1 vol.% case provides a 17.5% rise in h but approximately a 22% rise in ΔP relative to water. Such a trade-off is consistent with the need to consider pumping power highlighted in nanofluid reviews [4] and in experiments combining nanofluids with turbulence-promoting inserts [7,10].

Conclusion and Discussion

A structured methodology has been developed for experimentally assessing heat-transfer enhancement in a counter-flow double-pipe heat exchanger using Al_2O_3 /water nanofluids. The design uses a water baseline, controlled nanoparticle concentrations from 0.25 to 1.00 vol.%, several flow rates, direct pressure-drop measurement and an energy-balance acceptance criterion. This combination is necessary because a higher thermal conductivity or outlet-temperature change alone is insufficient to establish practical exchanger improvement. The analysis should report Q , U , h , Re , Nu , ε and ΔP together, with uncertainty and replicate statistics. The literature through 2018 consistently indicates that nanofluids can improve convective performance in double-pipe or double-tube exchangers, particularly as Reynolds number and concentration increase [5,6,9–11]. However, the hydraulic cost grows as viscosity and flow resistance increase, and stability remains a practical limitation [4,12]. The illustrative analysis in this methodology shows the expected pattern: a progressive heat-transfer improvement with concentration accompanied by a smaller but non-negligible pressure-drop penalty. For a real study, the optimum concentration should therefore be selected by a thermal-hydraulic criterion rather than by maximum heat-transfer coefficient alone. Before publication, the illustrative Section 7 dataset must be replaced by actual laboratory measurements, and the final uncertainty values must be recalculated from the specific instruments used.

References

Pak, B. C., & Cho, Y. I. (1998). Hydrodynamic and heat transfer study of dispersed fluids with submicron metallic oxide particles. *Experimental Heat Transfer*, 11(2), 151–170. <https://doi.org/10.1080/08916159808946559>

Xuan, Y., & Li, Q. (2003). Investigation on convective heat transfer and flow features of nanofluids. *Journal of Heat Transfer*, 125(1), 151–155. <https://doi.org/10.1115/1.1532008>
Wen, D., & Ding, Y. (2004). Experimental investigation into convective heat transfer of nanofluids at the entrance region under laminar flow conditions. *International Journal of Heat and Mass Transfer*, 47(24), 5181–5188. <https://doi.org/10.1016/j.ijheatmasstransfer.2004.07.012>

Saidur, R., Leong, K. Y., & Mohammad, H. A. (2011). A review on applications and challenges of nanofluids. *Renewable and Sustainable Energy Reviews*, 15(3), 1646–1668. <https://doi.org/10.1016/j.rser.2010.11.035>

Akhtari, M., Haghshenasfard, M., & Talaie, M. R. (2013). Numerical and experimental investigation of heat transfer of $\alpha-Al_2O_3$ /water nanofluid in double pipe and shell and tube heat exchangers. *Numerical Heat Transfer, Part A: Applications*, 63(12), 941–958. <https://doi.org/10.1080/10407782.2013.772855>

Darzi, A. A. R., Farhadi, M., & Sedighi, K. (2013). Heat transfer and flow characteristics of Al_2O_3 -water nanofluid in a double tube heat exchanger. *International Communications in Heat and Mass Transfer*, 47, 105–112. <https://doi.org/10.1016/j.icheatmasstransfer.2013.06.003>

Maddah, H., Alizadeh, M., Ghasemi, N., & Wan Alwi, S. R. (2014). Experimental study of Al_2O_3 /water nanofluid turbulent heat transfer enhancement in the horizontal double pipes fitted with modified twisted tapes. *International Journal of Heat and Mass Transfer*, 78, 1042–1054. <https://doi.org/10.1016/j.ijheatmasstransfer.2014.07.059>

Hussein, A. M. (2017). Thermal performance and thermal properties of hybrid nanofluid laminar flow in a double pipe heat exchanger. *Experimental Thermal and Fluid Science*, 88, 37–45. <https://doi.org/10.1016/j.expthermflusci.2017.05.015>

Han, D., He, W. F., & Asif, F. Z. (2017). Experimental study of heat transfer enhancement using nanofluid in double tube heat exchanger. *Energy Procedia*, 142, 2547–2553. <https://doi.org/10.1016/j.egypro.2017.12.090>

Ravi Kumar, N. T., Bhramara, P., Kirubeil, A., Syam Sundar, L., Singh, M. K., & Sousa, A. C. M. (2018). Effect of twisted tape inserts on heat transfer, friction factor of Fe_3O_4 nanofluids flow in a double pipe U-bend heat exchanger. *International Communications in Heat and Mass Transfer*, 95, 53–62. <https://doi.org/10.1016/j.icheatmasstransfer.2018.03.020>

Bahmani, M. H., Sheikhzadeh, G., Zarringhalam, M., Akbari, O. A., Alrashed, A. A. A., Sheikh

Shabani, G. A., & Goodarzi, M. (2018). Investigation of turbulent heat transfer and nanofluid flow in a double pipe heat exchanger. *Advanced Powder Technology*, 29(2), 273–282. <https://doi.org/10.1016/j.appt.2017.11.013>

Sajid, M. U., & Ali, H. M. (2018). Thermal conductivity of hybrid nanofluids: A critical review. *International Journal of Heat and Mass Transfer*, 126, 211–234. <https://doi.org/10.1016/j.ijheatmasstransfer.2018.05.021>