

2010

A Numerical Study on the Application of Nanofluids in Refrigeration Systems

Juan Carlos Valdez Loaiza

Pontifical Catholic University of Rio de Janeiro

Frank Chaviano Pruzaesky

Pontifical Catholic University of Rio de Janeiro

Jose Alberto Reis Parise

Pontifical Catholic University of Rio de Janeiro

Follow this and additional works at: <http://docs.lib.purdue.edu/iracc>

Loaiza, Juan Carlos Valdez; Pruzaesky, Frank Chaviano; and Parise, Jose Alberto Reis, "A Numerical Study on the Application of Nanofluids in Refrigeration Systems" (2010). *International Refrigeration and Air Conditioning Conference*. Paper 1145.
<http://docs.lib.purdue.edu/iracc/1145>

This document has been made available through Purdue e-Pubs, a service of the Purdue University Libraries. Please contact epubs@purdue.edu for additional information.

Complete proceedings may be acquired in print and on CD-ROM directly from the Ray W. Herrick Laboratories at <https://engineering.purdue.edu/Herrick/Events/orderlit.html>

A Numerical Study on the Application of Nanofluids in Refrigeration Systems

Juan Carlos Valdez LOAIZA¹, Frank Chaviano PRUZAESKY², José Alberto Reis PARISE^{3*}

¹Pontifical Catholic University of Rio de Janeiro, Department of Mechanical Engineering,
Rio de Janeiro, RJ, Brazil
Phone: +5521-35271380, Fax: +5521-35271165 , E-mail: jc_valdez@aluno.puc-rio.br

²Pontifical Catholic University of Rio de Janeiro, Department of Mechanical Engineering,
Rio de Janeiro, RJ, Brazil
Phone: +5521-35271380, Fax: +5521-35271165 , E-mail: pruz@puc-rio.br

³Pontifical Catholic University of Rio de Janeiro, Department of Mechanical Engineering,
Rio de Janeiro, RJ, Brazil
Phone: +5521-35271380, Fax: +5521-35271165 , E-mail: parise@puc-rio.br

* Corresponding Author

ABSTRACT

The use of nanofluids as secondary coolants in vapor compression refrigeration systems was numerically studied. A simulation model for a liquid-to-water heat pump, with reciprocating compressor and double-tube condenser and evaporator was studied. The multi-zone method was employed in the modeling of the heat exchangers. The water-based nanofluid was supposed to flow through the inner circular section of the evaporator, while the refrigerant was left to the annular passage. A computational program was developed to solve the resulting non-linear system of algebraic equations. Different nanoparticles (Cu, Al₂O₃, CuO and TiO₂) were studied for different volume fraction and particle diameters. Simulation results have shown that, for a given refrigerating capacity, evaporator area and refrigerant-side pressure drop are reduced when: (i) the volume fraction of nanoparticles increase; (ii) the diameter of nanoparticles decrease. Also, nanofluid-side pressure drop and, consequently, pumping power, increase with nanoparticle volume fraction and decrease with nanoparticle size. Results from a typical case-study indicated an evaporator area reduction, with the use of nanofluids as secondary coolant, if compared to the conventional base-fluid (H₂O).

1. INTRODUCTION

Considerable attention has been recently given to nanofluids, nanoscale colloidal solutions, consisting of nanoparticles (with sizes of the order of 1 to 100 nm) dispersed in a base fluid (Choi, 1995; Cheng *et al.*, 2008). The idea of dispersing solid particles in a fluid to enhance its thermal properties is not new, but, with particle sizes of the order of 1 μ m to 1 mm, this effort has always been restricted by problems on deposition, abrasion, clogging and additional pressure drop, aggravated by the fact that high volume fractions of particles were required to attain appreciable results (Choi, 2008). Nanofluids, on the contrary, require relatively low concentrations to present enhanced thermal conductivity (Choi, 2008), to magnitudes that have astounded heat transfer researchers for the past decade - see, for example, Keblinski *et al.* (2008) and Jang and Choi (2004). Thermophysical properties and heat transfer mechanisms of nanofluids have been studied to a great extent in the last years, as the increasing number of related publications attest. A recent review, by Yu *et al.* (2008), shows a number of possible base fluids (water, organic fluids, lubricating oils, and others) and several nanoparticles (Al₂O₃, TiO₂, CuO, carbon nano tubes, Cu, to name but a few) and reports thermal conductivity enhancement ratios (nanofluid thermal conductivity divided by that of the base fluid, at same conditions) that range from 1 (no enhancement) to 3, with the majority of the cases not exceeding 2, the greater enhancement ratios being obtained with large volume fractions.

Understandingly, a major part of the literature on nanofluids is formed by papers focused on the study of thermophysical properties and heat transfer mechanisms. Only recently, a few publications have arisen on practical

applications of nanofluids, that include heat transfer equipment (Noie *et al.*, 2009; Paramatthanuwat *et al.*, 2009; Pantzali *et al.*, 2009a; Pantzali *et al.*, 2009b; Strandberg and Das, 2010; Farajollahi *et al.*, 2010; Shafahi *et al.*, 2010; Qu *et al.*, 2010), HVAC&R applications (Park and Jung, 2007; Weiting *et al.*, 2009; Lee *et al.*, 2010; Kulkarni *et al.*, 2009), internal combustion engines (Kole and Dey, 2010; Beck and Yuan, 2009) and industrial heat treatment (Prabhu and Fernades, 2008). One concludes that, in spite of the extensive number of studies on the enhancement of thermal conductivity and heat transfer mechanism of nanofluids, little has been done yet on the effect of such enhancements on the overall performance of a system.

This work presents a preliminary simulation effort on the performance of a vapor compression water-to-water heat pump working with nanofluid as the secondary fluid to the evaporator. Indirect refrigeration systems, operating with secondary coolants, have been regarded as an effective way of significantly reducing refrigerant charge (Wang *et al.*, 2010) and, consequently, reducing the direct environmental impact for a given refrigeration load. Water has been the natural choice for secondary fluids and, to cope with below zero temperatures, aqueous and non aqueous solutions have been used (Wang *et al.*, 2010).

2. MATHEMATICAL MODEL

2.1 System description

Figure 1a depicts a typical vapor compression refrigeration cycle operating with a secondary fluid. The thermal load is transferred to the secondary fluid that circulates in a closed circuit and rejects heat to the refrigerant, through the evaporator. Figure 1b shows the secondary loop, with a circulating pump and two heat exchangers, one of them the evaporator.

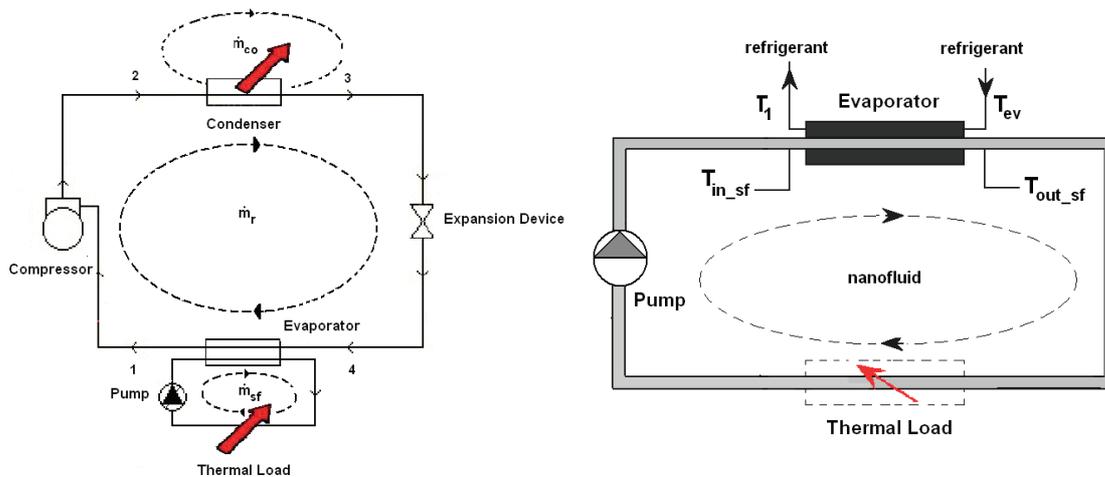


Figure 1: a) Vapor compression refrigeration system with secondary fluid; b) Secondary fluid circuit.

2.2 Compressor

The refrigerant mass flow rate is determined by:

$$\dot{m}_r = \eta_v v_1 \dot{V}_{cp} = \eta_v v_1 n_p \left(\frac{\pi D_p^2}{4} s_p \right) \frac{N}{60} \quad (1)$$

The volumetric efficiency takes into account the contribution of four factors, (Ciconkov and Ciconkov, 2007): the re-expansion of the gas in the clearance volume, η_c , the pressure drop in the suction side, η_p , heat transfer between gas and cylinder walls, η_q , and refrigerant leakage through the cylinder and piston clearances, as well as leakage through the suction and discharge valves, η_l :

$$\eta_v = \eta_l \eta_q \eta_p \eta_c = 0.97 \left(\frac{T_{ev}}{T_{cd}} \right) \left[1 - \left(\frac{\Delta P_{ev}}{P_{ev}} \right) \left(\frac{1+r}{\eta_c} \right) \right] \left\{ 1 - r \left[\left(\frac{P_{cd}}{P_{ev}} \right)^{\frac{1}{n}} - 1 \right] \right\} \quad (2)$$

The polytropic exponent is given by Ciconkov e Hilligweg (2004):

$$n = 1 + a \left(\frac{c_p}{c_v} - 1 \right) \quad (3)$$

where coefficient a is 0.5, 0.62, 0.75 or 0.88, depending on whether the suction pressure range is (<1.5 bar), (1.5 – 4 bar), (4 – 10 bar) or (10 – 30 bar), respectively, and specific heats are calculated at suction conditions. An empirical relation for the compressor isentropic efficiency, as a function of the pressure ratio, $\Pi = P_{cd}/P_{ev}$, is given by Ciconkov and Ciconkov (2007):

$$\eta_s = \begin{cases} -0.002515\Pi^4 + 0.03873\Pi^3 - 0.227968\Pi^2 + 0.577237\Pi + 0.275893; \Pi \geq 4 \\ -0.03\Pi + 0.892; \Pi < 4 \end{cases} \quad (4)$$

2.3 Condenser and Evaporator

Condenser and evaporator were of the counter flow double-tube type. In the condenser, refrigerant flows in the inner tube and cooling water, in the annular passage. The configuration is inverted for the evaporator, with the secondary fluid (nanofluid) flowing in the inner tube. The reason for this arrangement was that, at the time this work was carried out, there was no correlation available for flow of nanofluid in annular passages. In view of the local variation of the overall heat transfer coefficient along the heat exchangers, a multi-zone, or moving boundary, method (Martins Costa and Parise, 1993; Braun, 2004) was employed. The condenser was divided into three zones, namely, desuperheating, condensing and subcooling zones, and each one was treated as a separate heat exchanger. Analogously, boiling and superheating zones were provided for the evaporator. Equations (5) to (7), and (8) to (9), respectively, describe the energy balance for fluid and refrigerant, and the heat transfer between fluids, for each zone of the condenser and evaporator, .

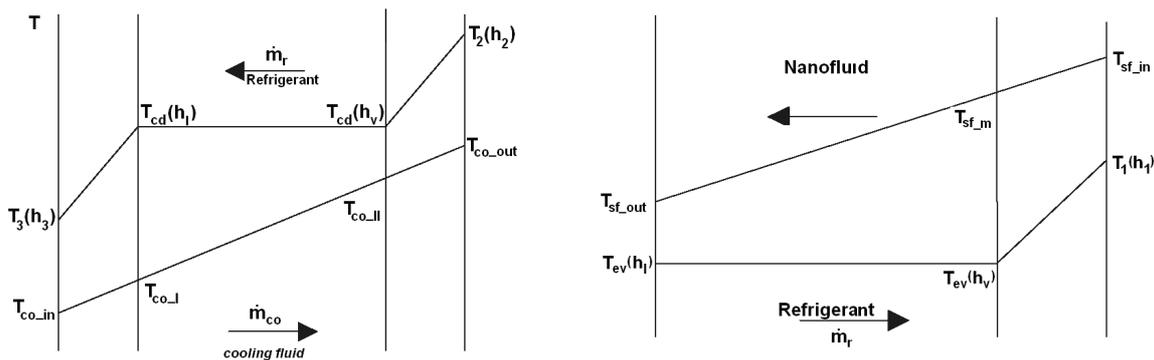


Figure 2: (a) Temperature distribution in the condenser; (b) Temperature distribution in the evaporator.

$$\dot{Q}_{ds} = \dot{m}_r (h_2 - h_{v,cd}) = \dot{m}_{co} c_{p,co} (T_{co,out} - T_{co,II}) = U_{ds} A_{ds} \overline{\Delta T}_{ds} \quad (5)$$

$$\dot{Q}_{cd} = \dot{m}_r (h_{v,cd} - h_{l,cd}) = \dot{m}_{co} c_{p,co} (T_{co,II} - T_{co,I}) = U_{cd} A_{cd} \overline{\Delta T}_{cd} \quad (6)$$

$$\dot{Q}_{sc} = \dot{m}_r (h_{l,cd} - h_3) = \dot{m}_{co} c_{p,co} (T_{co,I} - T_{co,in}) = U_{sc} A_{sc} \overline{\Delta T}_{sc} \quad (7)$$

$$\dot{Q}_{bo} = \dot{m}_r (h_{v,ev} - h_4) = \dot{m}_{sf} c_{p,sf} (T_{sf,m} - T_{sf,out}) = U_{bo} A_{bo} \overline{\Delta T}_{bo} \quad (8)$$

$$\dot{Q}_{sh} = \dot{m}_r (h_1 - h_{v,ev}) = \dot{m}_{sf} c_{p,sf} (T_{sf,in} - T_{sf,m}) = U_{sh} A_{sh} \overline{\Delta T}_{sh} \quad (9)$$

Temperatures in equations (5) to (9) follow the flow scheme depicted in Figures 2a and 2b. Zone overall heat transfer coefficients were calculated assuming a clean heat exchanger. Tube wall thermal resistance was taken into account. Refrigerant-side heat transfer coefficients for the condensing and evaporating zones were determined with correlations from Thome *et al.* (2003) and Gungor and Winterton (1986), respectively. A generalized correlation from Choi *et al.* (2001) was employed for the calculation of the two-phase pressure drop in both heat exchangers. The two-phase regions of both condenser and evaporator were discretized to take into account the local variation of the refrigerant condensing and boiling heat transfer coefficients. Heat transfer and pressure drop, in single-phase flows, for both refrigerant and condenser coolants, were treated with the usual Dittus-Boelter (1930) and Darcy-Weisbach correlations. The characterization of the nanofluid flow is detailed next.

2.4 Nanofluid

The nanofluid is treated as a homogeneous fluid. Based on experimental data of several authors, Velagapudi *et al.* (2008) proposed the following correlation for the thermal conductivity of nanofluids:

$$\frac{k_{nf}}{k_m} = c \text{Re}_m^{0.175} \phi_p^{0.05} \left(\frac{k_p}{k_m} \right)^{0.2324} \quad (10)$$

where the Reynolds number is given by:

$$\text{Re}_m = \left(\frac{1}{\nu_m} \right) \left(\frac{18k_p T}{\pi \rho_p d_p} \right)^{1/2} \quad (11)$$

Constant c depends on the particle-base fluid combination (Velagapudi *et al.*, 2008). For water-based nanofluids with nanoparticles of Al_2O_3 , CuO, Cu and TiO_2 , the value of c is 1, 1.298, 0.74 and 1.5, respectively. Concerning viscosity, specific correlations for each nanofluid were employed. For example, for water- Al_2O_3 nanofluid, the viscosity is given by Pak and Cho (1998), as follows:

$$\mu_{nf} = \mu_m (533.9\phi_p^2 + 39.11\phi_p + 1) \quad (12)$$

Finally, density and specific heat are determined based on mass and energy balances, respectively.

$$\rho_{nf} = \rho_p \phi_p + \rho_m (1 - \phi_p) \quad (13)$$

$$c_{p,nf} = \frac{\phi_p \rho_p c_{p,p} + (1 - \phi_p) \rho_f c_{p,f}}{\rho_{nf}} \quad (14)$$

Specific correlations for the Nusselt number of nanofluids have been available in the literature, as it is felt that the direct use of traditional correlations, such as from Dittus-Boelter (1930) or Gnielinski (1976), tend to underpredict the heat transfer (Yu *et al.*, 2008). Pressure drop on the nanofluid side was calculated in the same way as for any other fluid (Xuan and Roetzel, 2000).

2.5 Cycle calculations and method of solution

A thermostatic expansion valve was used, implying isenthalpic expansion. Input data for the simulation included the geometry of compressor and heat exchangers (inner and outer diameters), refrigerant type, nanofluid characteristics (base fluid and nanoparticle material, size and volume fraction), inlet temperatures of condenser coolant and secondary fluid (nanofluid), condensing and evaporating temperatures, evaporator outlet degree of superheating, condenser outlet degree of subcooling and refrigeration thermal load. The resulting system of equations was solved in the EES (Engineering Equation Solver) platform. Simulation main results include: (i) compressor - power consumption and discharge temperature; (ii) condenser – coolant mass flow rate, total heat transfer area and coolant and refrigerant pressure drops; (iii) evaporator – nanofluid mass flow rate, total heat transfer area and nanofluid and refrigerant pressure drops.

3. RESULTS

The simulation program was run for a small capacity system operating with four different water-based nanofluids, Cu, Al₂O₃, CuO and TiO₂. Volume fraction ranged from 1 % to 5% and particle size, from 10 to 50 nm. Figures 3 and 4 show the evaporator required heat transfer area and secondary fluid pressure drop as a function of volume fraction and particle size. It can be seen, from Figure 3, that greatest reductions in evaporator area were obtained with Cu+H₂O nanofluid, flowing with large volume fractions and lower particle diameters. This reduction in area would represent less refrigerant charge and, consequently, lower emissions of CO₂ equivalent (direct environmental impact). Figure 4 shows that pressure drop, as expected (enhanced viscosity), increases with particle size and volume fraction.

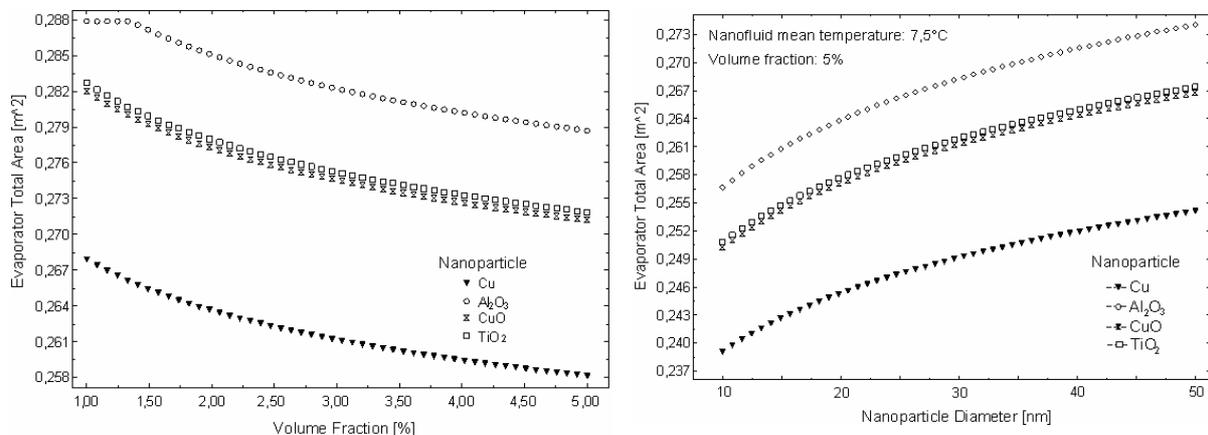


Figure 3: Variation of evaporator heat transfer area as function of: (a) volume fraction; and (b) nanoparticle size.

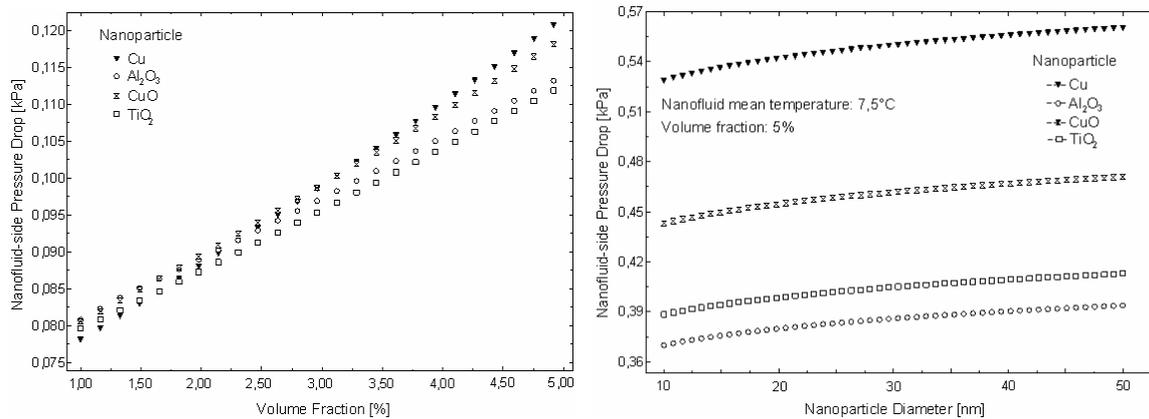


Figure 4: Variation of evaporator nanofluid-side pressure drop as function of: (a) volume fraction; and (b) nanoparticle size.

4. CONCLUSIONS

As far as the application of nanofluids is concerned, preliminary results here presented show that, for the same refrigerating capacity, evaporators with reduced area can be employed, as a result of the enhanced heat transfers characteristics of the secondary fluid. As far as the secondary loop pressure drop is concerned, there is a trade off between the enhanced viscosity of the nanofluid with the reduced evaporator area and, consequently, reduced channel length. Average temperature of the nanofluid may play an important role in this trade-off. It should be mentioned that, for the present study, no effort was made to optimize heat transfer enhancement effects due to the use of a nanofluid.

Practical issues, such as solution stability, still exist. And the discussion for a thermal conductivity model is still open. Without any doubt, there is a considerable potential for research and development in the application of nanofluids in refrigeration.

NOMENCLATURE

| | | | | |
|-------------|------------------------------------|-------------------|----|-------------------|
| a | coefficient in equation (3) | (-) | | Subscripts |
| A | heat transfer area | (m ²) | bo | boiling |
| c | constant in equation (10) | (-) | cd | condensing |
| c_p | specific heat at constant pressure | (J/kg·K) | ds | desuperheating |
| c_v | specific heat at constant volume | (J/kg·K) | ev | evaporating |
| D_p | piston diameter | (m) | m | base fluid |
| h | specific enthalpy | (J/kg) | nf | nanofluid |
| k | thermal conductivity | (W/m·K) | p | nanoparticle |
| \dot{m}_r | refrigerant mass flow rate | (kg/s) | sc | subcooling |
| N | compressor rotational speed | (rpm) | sh | superheating |
| n | polytropic exponent | (-) | r | refrigerant |
| n_p | number of cylinders of compressor | (-) | 1 | suction line |
| P | pressure | (Pa) | 2 | discharge line |
| \dot{Q} | heat transfer rate | (W) | 3 | liquid line |
| r | compressor clearance ratio | (-) | 4 | evaporator inlet |
| Re_m | modified Reynolds number | (-) | | |

| | | |
|----------------|-----------------------------------|----------------------|
| s_p | piston stroke | (m) |
| T | temperature | (K) |
| U | overall heat transfer coefficient | (W/K) |
| v | specific volume | (m ³ /kg) |
| \dot{V}_{cp} | compressor volumetric flow rate | (m ³ /s) |

Greek Symbols

| | | |
|-----------------------|---|----------------------|
| ΔP_{ev} | pressure drop in the suction side | (Pa) |
| $\overline{\Delta T}$ | logarithmic mean temperature diff. | (K) |
| η_c | vol. eff. due to clearance volume | (-) |
| η_l | vol. eff. due to wall-gas heat transfer | (-) |
| η_p | vol. eff. due to suction pressure drop | (-) |
| η_q | vol. eff. due to gas leakage | (-) |
| η_s | isentropic efficiency | (-) |
| η_v | compressor volumetric efficiency | (-) |
| μ | viscosity | (Ns/m) |
| Π | condenser-evaporator pressure ratio | (-) |
| ρ | density | (kg/m ³) |
| ϕ_p | volume fraction | (-) |

REFERENCES

- Beck, M. P., Yuan, Y., Warriar, P., Teja, A.S., 2009, The thermal conductivity of alumina nanofluids in water, ethylene glycol, and ethylene glycol + water mixtures, *Journal of Nanoparticle Research*, July, vol. 11, no. 5, pp. 1129–1136.
- Braun, J.E., 2004, Air-cooled condenser and direct-expansion evaporator modeling, USN/IIR Sponsored short course, Purdue Conferences, July 10 and 11.
- Cheng L., Bandarra Filho, E.P., 2008, Thome, J.R., Nanofluid Two-Phase Flow and Thermal Physics: A New Research Frontier of Nanotechnology and Its Challenges, *Journal of Nanoscience and Nanotechnology*, Vol.8, p. 1–18.
- Choi, J.Y., Kedzierski, M.A., Domanski, P.A., 2001, Generalized pressure drop correlation for evaporation and condensation in smooth and micro-fin tubes, *IIR Commission B1 Conference, Thermophysical Properties and Transfer Processes of New Refrigerants*, Paderborn, Germany; October 3–5.
- Choi, S.U.S., 1995, *Developments and applications of non-newtonian flows*, edited by D. D. Singer and H. P. Wang, American Society of Mechanical Engineering, New York, FED-Vol. 231/MD-vol. 66, p.99.
- Choi, S.U.S., 2008, Nanofluids: A new field of scientific research and innovative applications, editorial, *Heat Transfer engineering*, vol. 29, no. 5, pp. 429-431.
- Ciconkov, R., Ciconkov, V., 2007, Work Characteristics of Reciprocating Compressor for Refrigeration System Developed with Computer Simulation, paper ICR07-B2-1485, *Proceedings of the International Congress of Refrigeration*, Beijing.
- Ciconkov, R., Hilligweg A., 2003, Kolbenverdichter – Simulation des Leistungsverhaltens beim Einsatz in einem Verflüssigungssatz, *Ki Kälte und Klimatechnik, Ki Luft- und Kältetechnik*, No 3.
- Dittus, P.W., Boelter, L.M.K., 1930, Heat Transfer in Automobile Radiators of the tubular type, *Univ. Calif. Pub. Eng.*, pp. 443-461.
- Farajollahi, B., Etemad, S.G., Hojjat, M., 2010, Heat transfer of nanofluids in a shell and tube heat exchanger, *International Journal of Heat and Mass Transfer*, vol. 53, p. 12–17.
- Gnielinski, V., 1976, New equations for heat and mass transfer in turbulent pipe and channel flow., *Int. Chem. Eng.*, vol. 28, pp. 359–368.

- Gungor, K., Winterton, R., 1986, A general correlation for flow boiling in tubes and annuli, *International Journal of Heat Transfer*, 29 pp 351–358.
- Jang, S.P., Choi, S.U.S., 2004, Role of Brownian motion in the enhanced thermal conductivity of nanofluids, *Applied Physics Letters*, vol. 84, no. 21, pp. 4316–4318.
- Jiang, W., Ding, G., Peng, H., 2009, Measurement and model on thermal conductivities of carbon nanotube nanorefrigerants, *International Journal of Thermal Sciences*, vol. 48, p. 1108–1115.
- Kebllinski, P., Prasher, R., Eapen, J., 2008, Thermal conductance of nanofluids: is the controversy over?, *Journal of Nanoparticle Research*, vol. 10, pp. 1089–1097.
- Kole, M., Dey, T.K., 2010, Viscosity of alumina nanoparticles dispersed in car engine coolant, *Experimental Thermal and Fluid Science*, doi:10.1016/j.expthermflusci.2009.12.009.
- Kulkarni, D. P., Das, D. K., Vajjha, R. S., 2009, Application of nanofluids in heating buildings and reducing pollution, *Applied Energy*, vol. 86, p. 2566–2573.
- Lee, J. K., Koo, J., Hong, H., Kang, Y. T., 2010, The effects of nanoparticles on absorption heat and mass transfer performance in NH₃/H₂O binary nanofluids, *International Journal of Refrigeration*, vol. 33, p. 269–275.
- Martins Costa, M. L., Parise, J.A.R., 1993, A three-zone simulation model for a air-cooled condensers, *Heat Recovery Systems and CHP*, Volume 13, Issue 2, pp. 97–113.
- Noie, S.H., Zeinali Heris, S., Kahani, M., Nowee, S.M., 2009, Heat transfer enhancement using Al₂O₃/water nanofluid in a two-phase closed thermosyphon, *International Journal of Heat and Fluid Flow*, vol. 30, p. 700–705.
- Pak, B. C., Cho, Y. I., 1998, Hydrodynamic and Heat Transfer Study of Dispersed Fluids with Submicron Metallic Oxide Particles, *Experimental Heat Transfer*, Vol. 11, No. 2, pp. 151–170.
- Pantzali, M.N., Kanaris, A.G., Antoniadis, K.D., Mouza, A.A., Paras, S.V., 2009a, Effect of nanofluids on the performance of a miniature plate heat exchanger with modulated surface, *International Journal of Heat and Fluid Flow*, vol. 30, p. 691–699.
- Pantzali, M.N., Mouza, A.A., Paras, S.V., 2009b, Investigating the efficacy of nanofluids as coolants in plate heat exchangers (PHE), *Chemical Engineering Science*, vol. 64, p. 3290–3300.
- Paramatthanuwat, T., Boothaisong, S., Rittidech, S., Booddachan, K., 2009, Heat transfer characteristics of a two-phase closed thermosyphon using de-ionized water mixed with silver nano, *Heat Mass Transfer DOI* 10.1007/s00231-009-0565-y.
- Park, K., Jung, D., 2007, Boiling heat transfer enhancement with carbon nanotubes for refrigerants used in building air-conditioning, *Energy and Buildings*, vol. 39, p. 1061–1064.
- Prabhu, N., Fernades, P., 2008, Nanoquenchant for Industrial Heat Treatment, *Journal of Materials Engineering and Performance*, vol. 17, p. 101–103.
- Qu, J., Wu, H., Cheng, P., 2010, Thermal performance of an oscillating heat pipe with Al₂O₃–water nanofluids, *International Communications in Heat and Mass Transfer*, vol. 37, p. 111–115.
- Shafahi, M., Bianco, V., Vafai, K., Manca, O., 2010, An investigation of the thermal performance of cylindrical heat pipes using nanofluids, *International Journal of Heat and Mass Transfer*, vol. 53, p. 376–383.
- Strandberg, R., Das, D. K., 2010, Finned tube performance evaluation with nanofluids and conventional heat transfer fluids, *International Journal of Thermal Sciences*, vol. 49, p. 580–588.
- Thome, J., El Hajal, J., Cavallini, A., 2003, Condensation in horizontal tubes, part 2: two-phase flow pattern map, *International Journal of Heat and Mass Transfer*, 46, pp 3365–3387.
- Velagapudi, V., Konijeti, R.K., Aduru, C.S.K., 2008, Empirical Correlations to Predict Thermophysical Heat Transfer Characteristics of Nanofluids, *Journal of Thermal Science*, 12, pp 27–37.
- Wang, K., Eisele, Y.H., Hwang, Y., Radermacher, R., 2010, Review of secondary loop refrigeration systems, *International Journal of Refrigeration*, vol. 33, no. 2, pp. 212–234.

ACKNOWLEDGEMENT

Thanks are due to CAPES, CNPq and FAPERJ, Brazilian research funding agencies, for the financial support provided.