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Juan Carlos Valdez Loaiza
Pontifical Catholic University of Rio de Janeiro

Frank Chaviano Pruzaesky
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Jose Alberto Reis Parise
Pontifical Catholic University of Rio de Janeiro

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A Numerical Study on the Application of Nanofluids in Refrigeration Systems

Juan Carlos Valdez LOAIZA[1], Frank Chaviano PRUZAESKY[2], José Alberto Reis PARISE[3]*

[1]Pontifical 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

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

[3]Pontifical 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, Al_2O_3, CuO and TiO_2) 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 (H_2O).

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 μm 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 (Al_2O_3, TiO_2, 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.

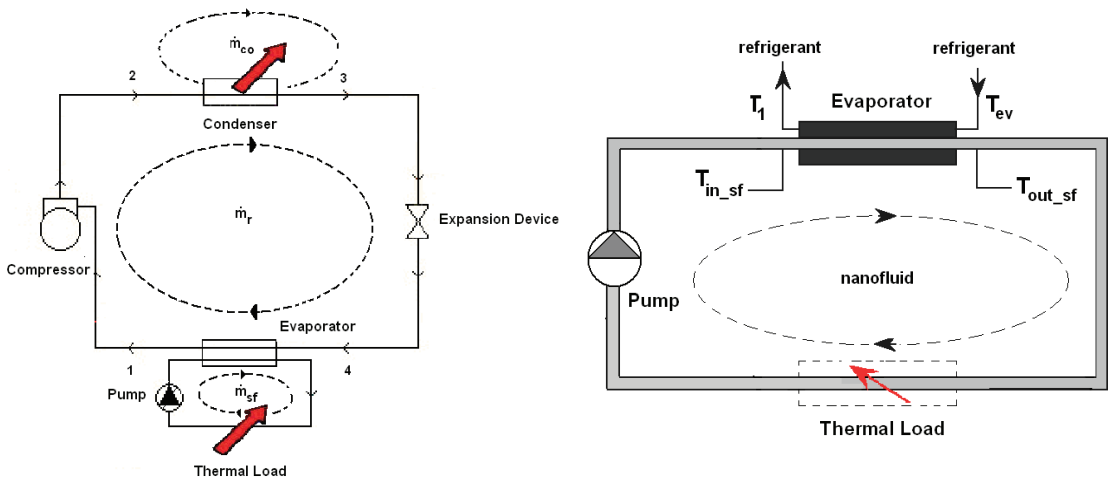

Figure 1: a) Vapor compression refrigeration system with secondary fluid; b) Secondary fluid circuit.

2.2 Compressor

The refrigerant mass flow rate is determined by:

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

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:

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

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

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

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):

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

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, .

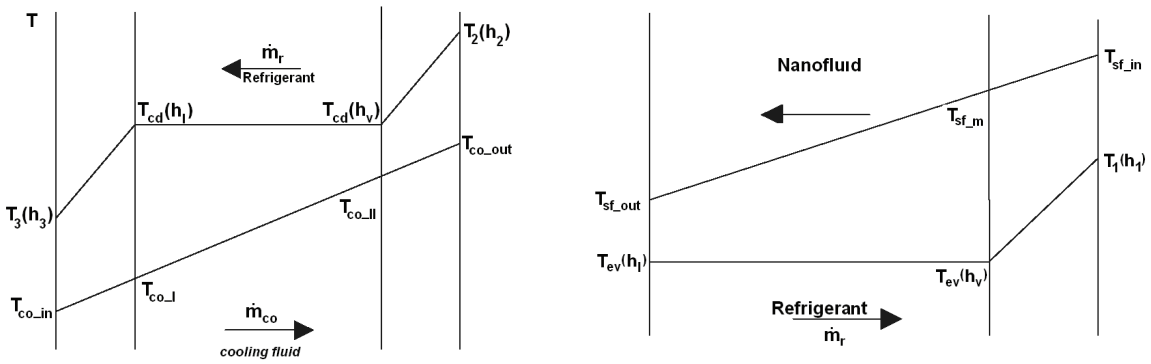

Figure 2: (a) Temperature distribution in the condenser; (b) Temperature distribution in the evaporator.

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

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

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

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

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

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:

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

where the Reynolds number is given by:

$$Re_m = \left(\frac{1}{\nu_m} \right) \left(\frac{18 k_p T}{\pi \rho_p d_p} \right)^{1/2} \tag{11}$$

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:

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

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

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

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

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, Al_2O_3, CuO and TiO_2. 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+H_2O nanofluid, flowing with large volume fractions and lower particle diameters. This reduction in area would represent less refrigerant charge and, consequently, lower emissions of CO_2 equivalent (direct environmental impact). Figure 4 shows that pressure drop, as expected (enhanced viscosity), increases with particle size and volume fraction.

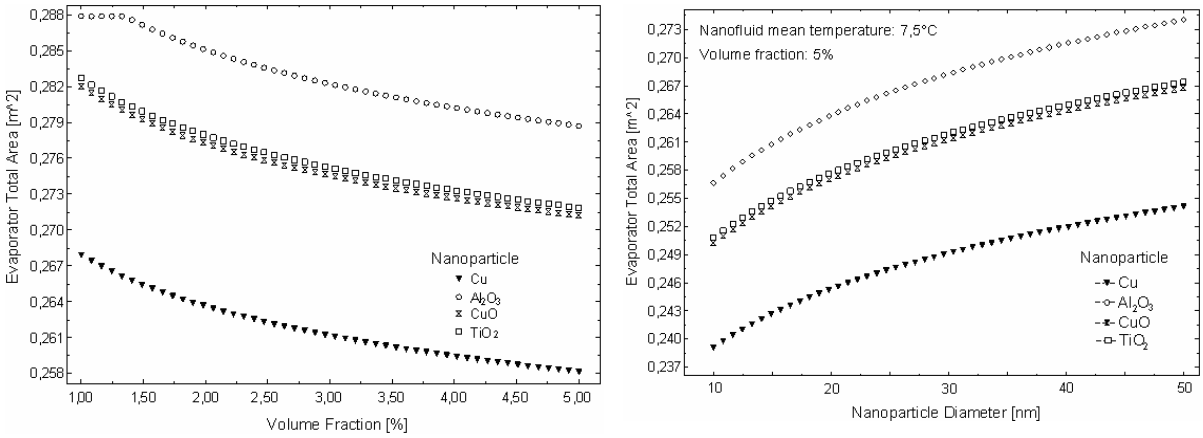

Figure 3: Variation of evaporator heat transfer area as function of: (a) volume fraction; and (b) nanoparticle size.

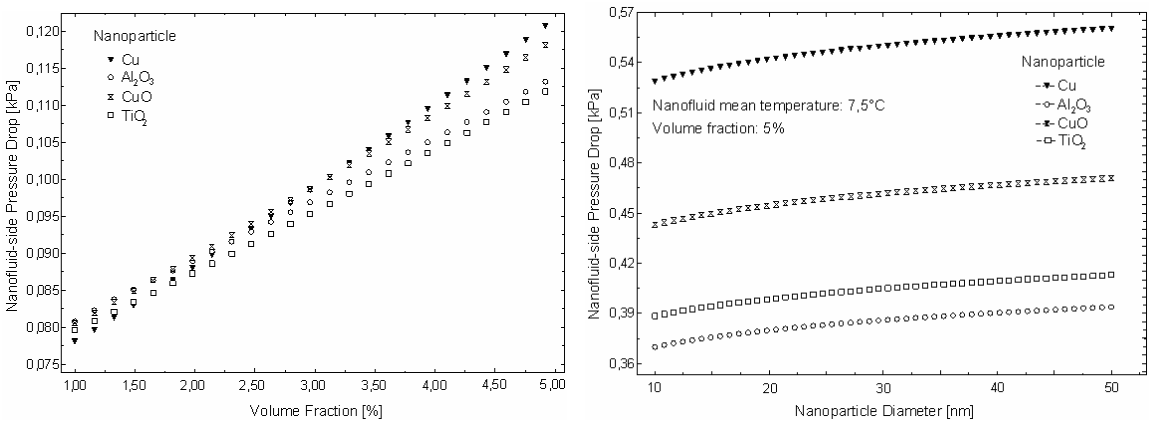

Figure 4: Variation of evaporator nanofluid-side pressure drop as function of: (a) volume fraction; and (b) nanoparticle size.

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	(-)

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