EXPERIMENTAL DETERMINATION OF TRANSPORT COEFFICIENTS FOR ICE SLURRY IN PLATE HEAT EXCHANGER

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Abstract. *Ice slurry is an aqueous solution, with fine ice crystals This article presents an experimental procedure to measure the heat transfer, with phase change, and the pressure drop of a mixture of glycol and water, with 13.8% in weight, inside a U-plate heat exchanger with 16 plates. Tests were carried out with Reynolds numbers between 150 and 425 for the ice slurry. The thermal load was imposed using water as a second fluid under different mass flow rates and flow configurations. The results shown that the overall heat transfer coefficient increases up to 25% when the ice slurry flow is increased. By increasing the ice fraction the cooling capacity is improved while the overall heat transfer coefficient and the Nusselt number are reduced. The friction factor varies between 0.030 and 0.085. As it was expected, the friction factor decreases while the slurry mass flow rate is increased and it increases whith the ice fraction. Finally, correlations for the Nusselt number and the friction factor for the aqueous solution with and without ice crystals are shown as a function of Reynolds number.*

Keywords*: Heat Exchanger, pressure drop, ice slurry, secondary refrigerants, plate heat exchanger.*

1. Introduction

If fine ice crystals are dispersed, as a phase change material, in a water solution, it is created a kind of blend known as ice slurry, which combines a high specific thermal energy, through the phase change of the solids in suspension, with the capacity to be pumped. These properties are enough to create a relatively simple and interesting secondary coolant. For these reasons many studies have been oriented to determine their behavior in pipes, pumps and heat exchangers (Egolf, 2004).

Metz (1987) and Choi and Knodel (1992) studied the potential of the ice slurry for the internal flow in pipes. They showed that it is possible to reduce 35% of the mass flow rate using the ice slurry. The cooling capacity for ice fractions of 30% increases up to six times, when compared with conventional cold water systems.

Due to the described potential of these relatively simple mixtures, studies on the possibility of use of the ice slurry in other refrigeration system components began to be done. It is the case of the following studies on plate heat exchangers.

Grupta and Fraser (1990) conducted experimental studies for a mixture of ethylene glycol - water to 6%, with ice fractions reaching up to 20%, and volumetric flow rates between 0.18 and 2.16 m^3 /h. They reported an increase in the heat transfer coefficient with the mass flow rate and a decrease of the same coefficient while the ice fraction increases. The pressure drop stayed almost constant for ice fractions up to 20%.

Norgard (2001) studied a mixture of propylene glycol - water with concentration of 16% in weight, and mass flow rates between 0.05 and 0.3 m^3/h. For low volumetric flow rates (0.05 m^3/h) the results indicated an increase in the heat transfer coefficient and an increase also in the pressure drop as the ice fraction increases.

Bellas *et al* (2002) reported results for mixtures of propylene - water with concentration up to 5%, ice fractions up to 25%, and flow rates between 1.0 and 3.7 m^3/h. They observed an increase of 30% in the cooling capacity. The heat transfer coefficient increases with the ice slurry mass flow rate. On the other hand, the pressure drop presents an exponential increase.

Jiménez (2003) used a mixture of ethylene glycol - water with concentration up to 12%. He observed increases up to three times in the cooling capacity and overall heat transference coefficients when comparing the ice slurry flow with the liquid water flow.

Frei and Boyman (2003) reported results for the overall heat transfer coefficient and pressure drop as a function of the mass flow rate of ice slurry, for ice fractions up to 30%. They observed that the heat transfer coefficient decreases in the presence of ice crystals.

The behavior of the ice slurry depends of a great number of parameters (mixture composition, mass flow rate, ice fraction, size and space between crystals), however the influence of these parameters are not totally characterized. In the present work it is shown the effects of some parameters, which influences the behavior of the ice slurry when being used in plate heat exchangers.

2. Mathematical model

2.1. Thermophysical properties of ice slurries

In order to determine the thermophysical properties of the slurry, it is possible to suppose that the ice particles consist of pure water, and the reminiscent liquid consists of water and additives. The properties for the propylene glycol - water solution can be obtained from Melinder correlations (1997).

The ice fraction (X_g) is defined as the relationship between the mass of the ice (m_g) and the total mass of the ice slurry (m_{pg}). The carrying fluid (fp) is the reminiscent fluid that contains the dispersed ice crystals (g). The concentration of the additive in the carrying fluid (C_{fp}) depends on the initial concentration of this additive (C_0) for the solution without ice, and of the instantaneous mass ice fraction as it is shown in Eq. (1). Equation (2) correlates the volume (Cv_g) and the mass ice fraction.

$$C_{fp} = \frac{C_0}{1 - X_g} \tag{1}$$

$$Cv_g = \frac{\rho_g^{-1}.X_g}{\rho_g^{-1}.X_g + \rho_{fp}^{-1}.(1 - X_g)} \tag{2}$$

The mass density (ρ_{pg}) and specific heat at constant pressure of the ice slurry (Cp_{pg}) can be calculated by weighting the properties of the two phases, ice (g) and carrying fluid (fp). For the thermal conductivity (k_{pg}) was utilized the Eq. (5), proposed by Kauffeld (1999), with y as function of mass ice fraction. The viscosity (μ_{pg}) is determined from the Eq. (6), well-known for homogeneous Newtonian suspensions of Christensen and Kauffeld (1997) which is valid for Reynolds numbers (Re) between 3 and 2000, and ice fractions between 0 and 35%.

$$\rho_{pg}^{-1} = \rho_g^{-1}.X_g + \rho_{fp}^{-1}.(1 - X_g) \tag{3}$$

$$Cp_{pg} = Cp_g.X_g + Cp_{fp}.(1 - X_g) \tag{4}$$

$$k_{pg} = k_{fp}.\left[\frac{1 + 2.Cv_g.y}{1 - Cv_g.y}\right] \tag{5}$$

$$\mu_{pg} = \mu_{fp}.(1 + 2,5.Cv_g + 10,05.Cv_g^2 + 0,00273.e^{16,6.Cv_g}) \tag{6}$$

2.2. Ice fraction

The calorimetric method was utilized to determine the ice fraction of the sample. The ice slurry sample is introduced inside the calorimeter, after that a known amount of heat is generated by an electrical resistance, by measuring the initial and final temperature it is possible to calculate the initial ice fraction of the sample.

A differential energy balance inside the calorimeter is shows in the Eq. (7), P is the electrical power transfered, taking into account the variation of the properties of the ice slurry, with the temperature and the additive concentration of the carrier fluid, during the samples warming-up process. The first term of the right hand side of the Eq. (7) represents the sensible heat of the carrier fluid and the second term is the sensible heat of the components of the calorimeter, considering a mass equivalent of the components in water ($m_{eq\text{-}ag}$). The last term is responsible for the measurement of the latent heat that melts the ice and reduces the ice fraction, L is the latent heat of formation of ice.

Discreting the Eq. (7) in the time, the value of the mean specific heat of the carrier fluid ($Cp_{m\text{-}fp}$) can be assumed constant for each differential time interval, it is obtained the Eq. (8). The ice fraction is obtained by the integration of the Eq. (8) for each period of time of analysis of the ice slurry samples in the calorimeter.

$$\int_{t_1}^{t_2} P.dt = \int_{T_1}^{T_2} m_{pg}.\frac{C_0}{C_m}.Cp_{fp(T,C_0,X_m)}.dT + \int_{T_1}^{T_2} m_{eq-ag}.Cp_{ag(T)}.dT + m_{pg}.L.(X_2 - X_1) \tag{7}$$

$$X_{t(i+1)} = X_{t(i)} + \frac{1}{L}.\left[\frac{\dot{Q}.(t_{i+1} - t_i)}{m_{pg}} - \frac{C_0}{C_{t(i+1)}}.Cp_{m-fp}.\left[T_{t(i)} - T_{t(i+1)}\right] - m_{eq-ag}.Cp_{m-ag}.\left[T_{t(i)} - T_{t(i+1)}\right]\right] \tag{8}$$

2.3. Data reduction

With the experimental data obtained in the laboratory and using the definition of the logarithmic mean temperature difference (ΔT_{LM}), determined with the temperature differences in the input (e) and output (s) the heat exchanger, the overall heat transfer coefficient (U) can be obtained from the Eq. (9) and Eq. (10). Where ($\dot{Q}_{ag}$) is the thermal load, A is the area of transfer of heat exchanger, the correction factor of medium temperature is represented by F. The thermal load is calculated through Eq. (11), using the water mass flow rates and its temperatures. The cooling capacity that represents the heat absorbed by the ice slurry, is calculated as shown in Eq. (12), for each test condition.

$$U = \frac{\dot{Q}_{ag}}{A.F.\Delta T_{LM}} \tag{9}$$

$$\Delta T_{LM} = \frac{(\Delta T_{e-tr} - \Delta T_{s-tr})}{Ln\left[\dfrac{\Delta T_{e-tr}}{\Delta T_{s-tr}}\right]} \tag{10}$$

$$\dot{Q}_{ag} = \dot{m}_{ag}.Cp_{m-ag}.\Delta T_{ag} \tag{11}$$

$$\dot{Q}_{pg} = \dot{m}_{pg}.\left[Cp_{m-fp}(T_{fp-e} - T_{fp-s}) + L.(X_{g-e} - X_{g-s})\right] \tag{12}$$

In order to know the convection heat transfer (h_{pg}) for the ice slurry, it is necessary to know first the same coefficient for the water side (h_{ag}). This is done in a previous experiment, where the hot fluid is water and the cold one is a water solution without ice crystals (sol). In this case, it is possible to assume the same dimensionless heat transfer coefficient for both sides. Equation (13) shows a correlation for the thermal resistances in an element of the heat exchanger, Fig 1, considering the fouling effect (R_F) and thermal resistance by conductivity of the plate (k_{plac}) of heat exchanger, where e is the thickness of the plate. The convection heat transfer is correlated by Eq. (14) with of Nusselt number. In this case, the Nusselt number is well correlated as shown in Eq. (15). Exponent c of Prandtl number should be considered equal 0.3 for cooling process and 0.4 when it is warmed up.

$$\frac{1}{U} = \frac{1}{h_1} + \frac{1}{R_{F-1}} + \frac{e_{plac}}{k_{plac}} + \frac{1}{R_{F-2}} + \frac{1}{h_2} \tag{13}$$

$$h = \frac{Nu.k}{D_H} \tag{14}$$

$$Nu = a.\mathrm{Re}^b.\mathrm{Pr}^c \tag{15}$$

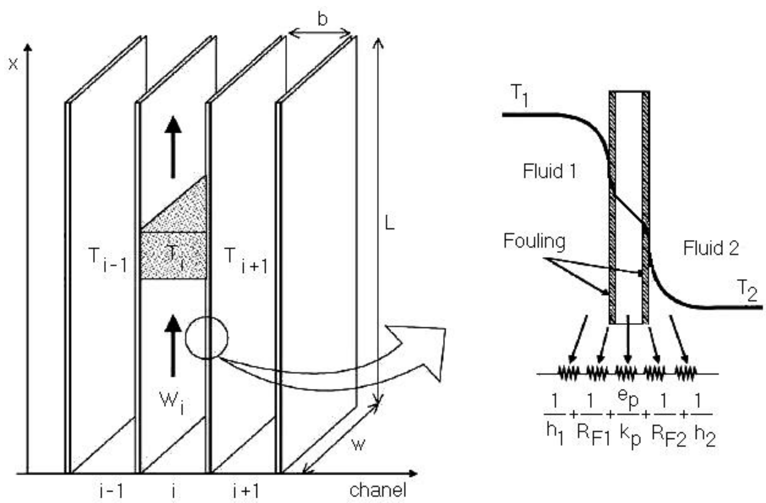


Figure 1. Resistances balance in a flow channel.

The Equation (16) is obtained by substituting Eq. (14) and Eq. (15) in Eq. (13), considering despresiveis of fouling efect, where D_H is the hydraulic diametrer. The values for the constants a and b, can be determined by using the previous equations and the knowledge of the mass flow rates, the temperatures in both fluids and dimensions of the heat exchanger.

$$\frac{1}{U}-\frac{e_{plac}}{k_{plac}}=\frac{1}{a.\mathrm{Re}_{ag}^{b}}\frac{D_{H-1}}{\mathrm{Pr}_{ag}^{0,3}.k_{ag}}+\frac{1}{a.\mathrm{Re}_{sol}^{b}}\frac{D_{H-2}}{\mathrm{Pr}_{sol}^{0,4}.k_{sol}} \tag{16}$$

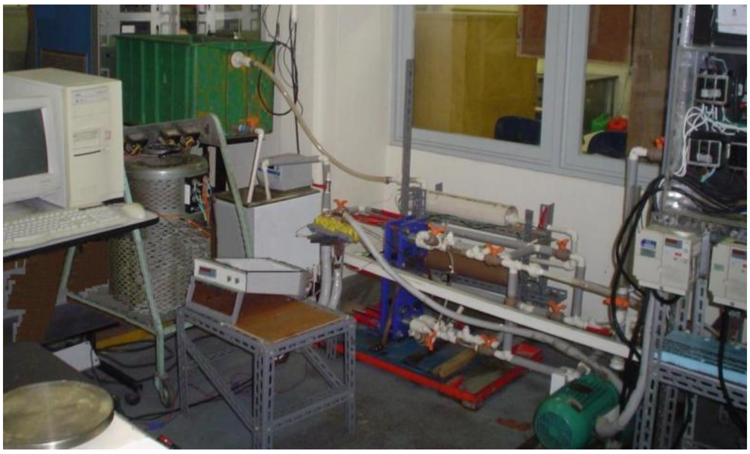


Figure 2. Test section, LRA –Puc-Rio

In a second phase, the testes are done with water and ice slurry. From the experimental data of this new phase, it is obtained the new overall heat transfer coefficient. By knowing the heat transference coefficient of the water side, calculated from the Nusselt number expression determined in the first phase, the ice slurry convective heat transfer coefficient is then determined by using the Eq. (17). Finally it is possible to derive a new correlation for the Nusselt number for the ice slurry, as shown in Eq. (18).

$$h_{pg}=\frac{1}{\frac{1}{U}-\frac{e_{plac}}{k_{plac}}-\frac{1}{h_{ag}}} \tag{17}$$

$$Nu_{pg}=a_2.\mathrm{Re}^{b_2}.\mathrm{Pr}^{c_2} \tag{18}$$

The pressure drop inside the plate heat exchanger can be calculated with a correlation used by Gut and Pinto (2003), which is based on Kakaç (2002), and is presented on Eq. (19). The first term in the right hand side evaluates the pressure drop due to the friction inside the channels, where f is the friction factor and N_P represents the number of pass. The second one represents the pressure variation caused by the change in the cross section area of the flow at the heat exchanger entrance and the last one accounts for the variation in the pressure due to the difference in elevation (L_H). Being G_C the mass flow by unit of area in the channels of the heat exchanger, G_P the mass flow by unit of area at the entrance of the heat exchanger and g the gravity specifies.

$$\Delta p=\left[\frac{2.f.L_H.N_P.{G_C}^2}{\rho_m.D_H}\right]+1.4\left[N_P\frac{{G_P}^2}{2.\rho_m}\right]+\rho_m.g.L_H \tag{19}$$

3. Experimental procedure

Figure 2 shows the test section of experimental apparatus composed essentially, Fig. 3, by plate heat exchanger, a ice slurry generation system, a data acquisition system, reservoirs of water and ice slurry, module thermal of resistances and thermal conditioner for the water.

Experimental procedure, in a first stage it deals with the refrigeration system and the primary reservoir, where ice slurry is generated in a continued manner. The temperatures inside the primary reservoir are continuously controlled to guarantee a pre-determined ice fraction condition. For the thermal load side it is necessary to pay attention to the fluid flow by using a needle valve, also monitoring the temperature inside the water tank. Groups of electrical resistances and the potentiometer are used to reach and maintain the desired condition. The thermal bath adjusts the water entrance temperature. The mass flow rate of the ice slurry can be adjusted between 0.10 and 0.23 kg/s. After obtaining the steady state condition, the data is acquired and stored in the computer. Samples of the ice slurry are collected in both entrance and exit of the heat exchanger for its analysis in the calorimeter. Different temperature conditions for the water entrance are imposed while its mass flow rate should be selected between 0.128 and 0.215 kg/s.

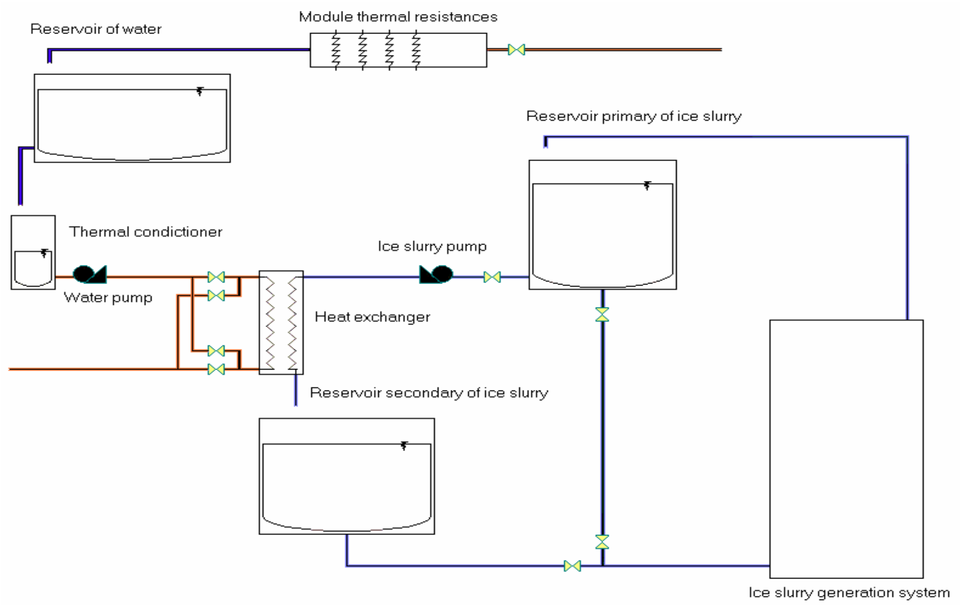


Figure 3. Schematic diagram of the test section

4. Results and discussion

4.1. Cooling capacity

With the entrance conditions for the thermal load kept constant, it is possible to see the cooling capacity of the ice slurry increasing as its mass flow rate increases. Figure 4 shows the cooling capacity for different initial ice fractions. A 21% increase is observed for the initial ice fraction of 0.11 while the flow rate is increased. For the 0.14 initial ice fractions, the increase is higher, 32%.

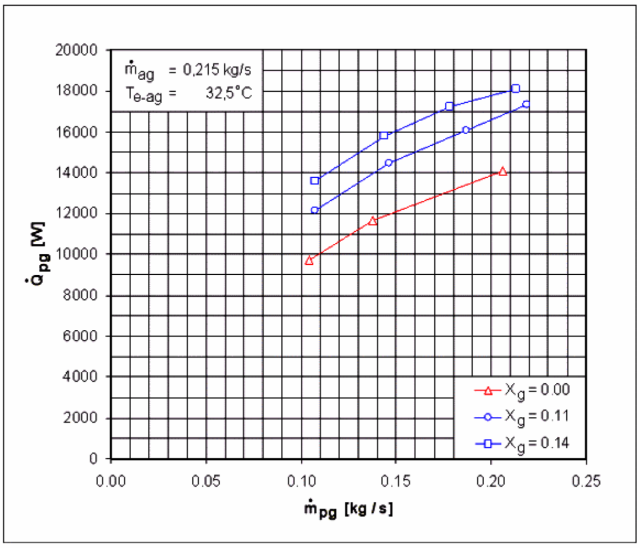


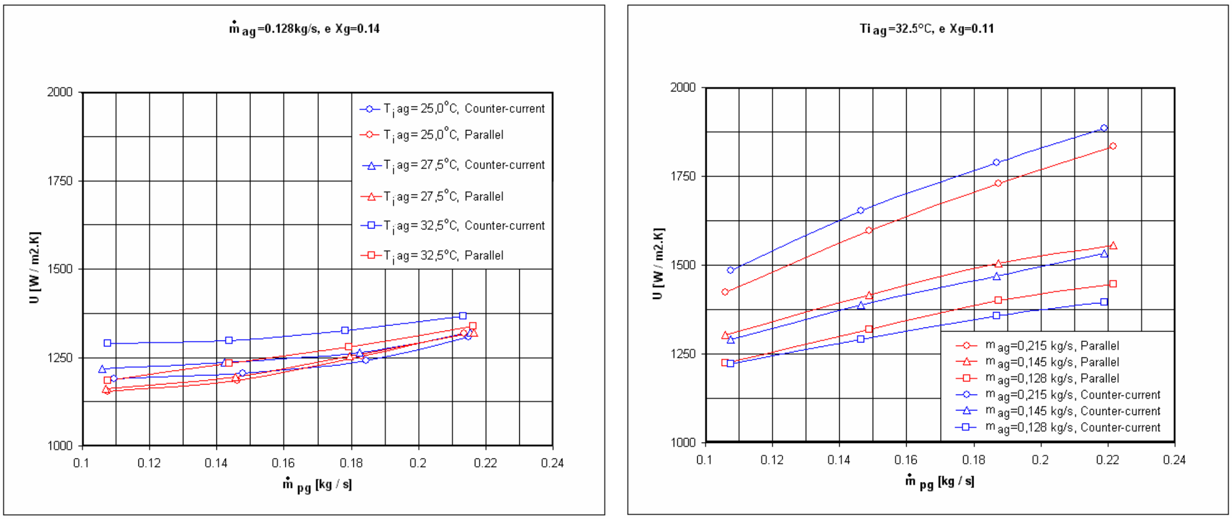
Figure 4. Cooling capacity vs. ice slurry mass flow rate

Bellas *et al.* (2002) found a similar behavior for the overall coefficient with respect to the volumetric flow rate. For ice fractions of 0.20 they observed a 30% increase in the cooling capacity. Jiménez (2003) showed similar results in the cooling capacity, for ice slurry made from ethylene.

4.2. Overall heat transfer coefficient

The Fig. 5(a) and Fig. 5(b) presents families of curves for overhall heat transfer coefficient. In the first one, Fig. 5(a), the water flow rate is maintained constant with different temperatures imposed at the heat exchanger entrance. The overall heat transfer coefficient decreases a little bit as the temperature is increased. Figure 5(b) shows the strong influence of the water flow rate on the overall coefficient that is reduced in approximately 23%, when the water flow drops from 0.215 to 0.128 kg/s, for an ice fraction equal to 0.11. In both figures it is possible to see that the overall heat transfer coefficient is greater in counter flow regime and greater slurry flow rates.

The overall heat transfer coefficient diminishes with the increase of the ice fraction. The observed results for the same water flow rate present 25% variations as it can be seen by comparing Fig. 5(a) and Fig. 5(b).



(a) (b)

Figure 5. Overall heat transfer coefficient vs. mass flow of ice slurry,

Frei and Boyman (2004) found, in a similar experiment, using ice slurry from ethanol and a 9% concentration in weight, similar behavior the overall heat transfer coefficient and the fall of the pressure in function of the mass flow, 0.11 to 0.22 kg/s, and the ice fraction of ice slurry.

4.3. Dimensionless correlations

Correlations for Nu, presented in Eq. (15), without phase change, were found imposing the exponents for the Prandtl number. The adjustments had a mean error of the order of 4×10^{-9}.

$$Nu_{ag} = 0{,}1588.\mathrm{Re}^{0{,}7417}.\mathrm{Pr}^{0{,}3} \quad \text{(Water side)} \qquad 350 < \mathrm{Re} < 705 \text{ e } 5{,}8 < \mathrm{Pr} < 7{,}0. \tag{20}$$

$$Nu_{sol} = 0{,}1588.\mathrm{Re}^{0{,}7417}.\mathrm{Pr}^{0{,}4} \quad \text{(Solution side without crystals)} \qquad 296 < \mathrm{Re} < 583 \text{ e } 5{,}9 < \mathrm{Pr} < 7{,}9. \tag{21}$$

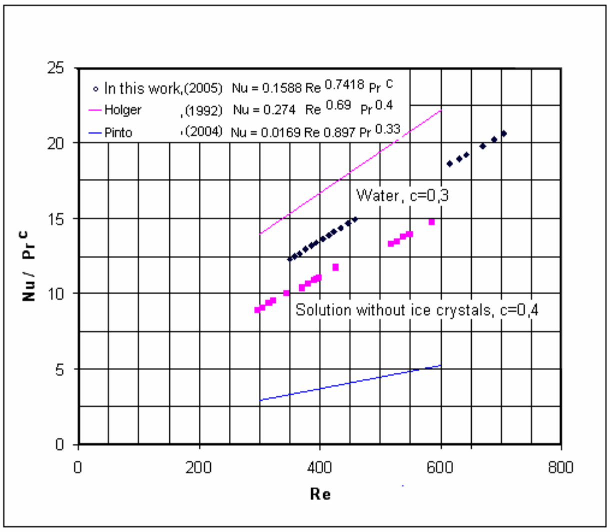
Similar expressions for Nu in different plate heat exchangers, without phase change, are presented in literature: Gut *et al* (2004) for a mini flat plate heat exchanger Armfield Ft-43, Eq. (22). Holger (1992) displays Eq. (23), in a treaty on heat exchangers.

$$Nu = 0{,}0169\,.\mathrm{Re}^{0{,}897}.\mathrm{Pr}^{1/3} \quad (\text{with } R^2 = 0{,}98), \qquad 100 < \mathrm{Re} < 1000 \text{ e } 2{,}2 < \mathrm{Pr} < 6{,}8 \tag{22}$$

$$Nu = 0{,}274.\mathrm{Re}^{0{,}69}.\mathrm{Pr}^{0{,}4} \quad \text{(for plates of type H)}, \qquad 100 < \mathrm{Re} < 10000 \text{ e } 2{,}0 < \mathrm{Pr} < 40 \tag{23}$$

In Fig. 6(a) these correlations are compared, for heat transfer without phase change. It is interesting to remember that the Eq. (22) is valid only for flat plate heat exchangers (without corrugation).

Figure 6(b) presents the results obtained for the Nusselt number, versus Reynolds number, for the ice slurry. A poor correlation factor of R^2=0.77 was found, which indicates that this expression even depends on another parameters, not considered in the present work. The Nusselt number experiments a strong decrease in the presence of ice crystals, due to its strong influence on the slurry viscosity.



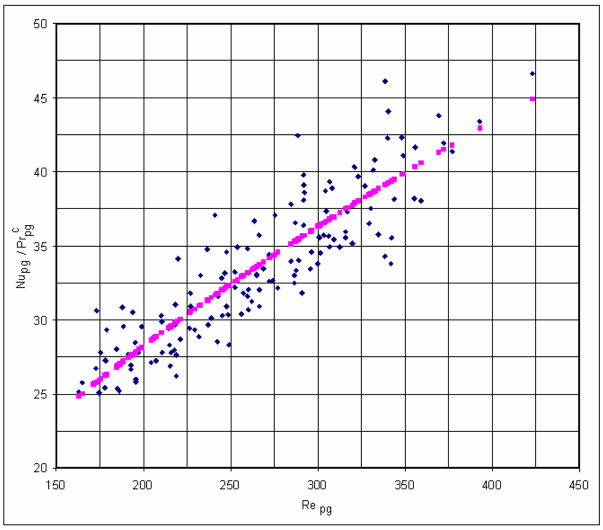


Figure 6. (a) Relations Nu.Pr^c/Re, for aqueous solution without ice crystals, and (b) for ice slurry

Adjusting all the ice slurry results for Nusselt by least square, Eq. (24), it is obtained an average quadratic deflection of S=2.1187. Doing the same with the results for ice fractions 0.11 and 0.14, imposing the Prandtl exponent, it is possible to derive the Eq. (25) and Eq. (26), with average quadratic deflections of S=1.443 and S=1.347 respectively.

$$Nu_{pg} = 1,0574.\mathrm{Re}^{0,62}.\mathrm{Pr}^{-0,115397} \tag{24}$$

$$Nu_{pg} = 0,387.\mathrm{Re}^{0,58132}.\mathrm{Pr}^{0,4} \qquad \text{(to } X_g\text{=11\%)} \tag{25}$$

$$Nu_{pg} = 0,784.\mathrm{Re}^{0,44521}.\mathrm{Pr}^{0,4} \qquad \text{(to } X_g\text{=14\%)} \tag{26}$$

The relation pressure drop versus mass flow rate is almost linear. The increase of the ice fraction at the heat exchanger entrance has a direct influence in the pressure drop inside the heat exchanger. The friction factor diminishes as the Reynolds number increases. It also decreases with the increasing of the ice fraction. For similar Reynolds numbers, the friction factor diminishes with the increase of the ice fraction, as it appears in Fig. 7, presenting a lineal behavior in a Lg-Lg graphic. Bellas *et al.* (2002) found similar results in the same range of the present work.

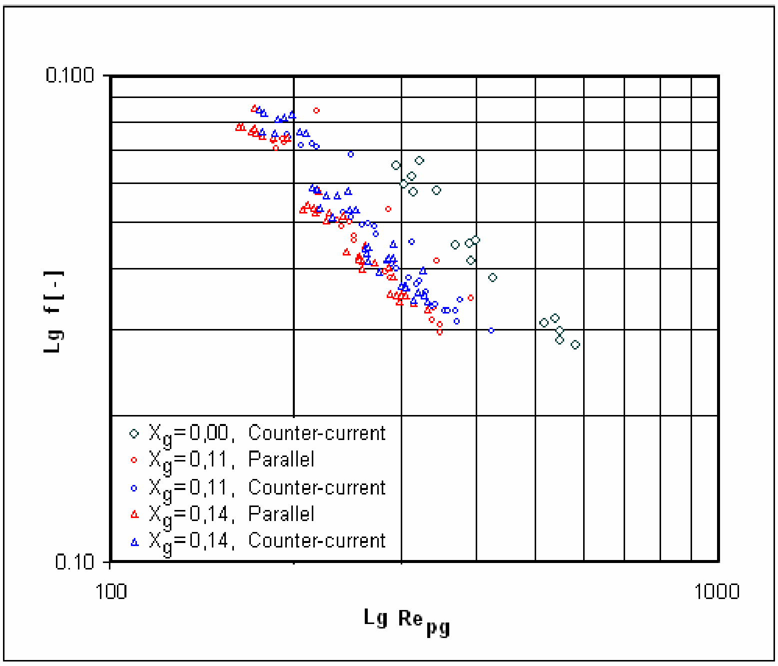


Figure 7. Lg(Friction factor) vs. Lg(Reynolds number)

Using the data acquired, by least square, the following equations were carried out. For the water solution without crystals, the Eq. (27) was derived,

$$f_{pg} = 123,4590.\,\mathrm{Re}^{-1,3249} \qquad (R^2=0,934) \tag{27}$$

For the ice slurry with ice fraction 0,11,

$$f_{pg} = 94,1383.\,\mathrm{Re}^{-1,35458} \qquad (X_g=0,11,\ R^2=0,972) \tag{28}$$

Finally, for the ice slurry with ice fraction 0,14,

$$f_{pg} = 108,6096.\,\mathrm{Re}^{-1,39715} \qquad (X_g=0,14,\ R^2=0,960) \tag{29}$$

From the present results, it is possible to conclude that the use of ice slurry has strong influence in both, the heat transfer and the pressure drop. Its use possibilities the mass flow rate reduction, keeping constant the cooling capacity. It is necessary to introduce new dimensionless parameters in the Nusselt and friction factor correlations, in order to account for the presence of the crystals in the fluid.

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