

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 coefficients of a mixture of glycol and water, with 13.8% in weight, inside a U-plate heat exchanger. Tests were carried out with the ice slurry Reynolds numbers varying between 150 and 425. 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 increases with 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 m³/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 m³/h. For low volumetric flow rates (0.05 m³/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 m³/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 increasing 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.

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 remniscent liquid consists of water and additives (fp). The concentration of the additive in the liquid phase increases as the ice crystals grows. 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 and the total mass of the ice slurry. The carrying fluid is the remniscent fluid that contains the dispersed ice crystals. The concentration of the additive in the carrying fluid (C_{fp}) depends on the initial concentration of this additive for the solution without ice (C_0) and of the instantaneous ice fraction, as it is shown in Eq. (1). Equation (2) correlates the volume and the mass ice fraction.

$$C_{fp} = \frac{C_0}{1 - X_g} \quad (1)$$

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

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, Eq. (3) and (4). For the thermal conductivity (k_{pg}), Eucken proposes the Eq. (5), where k_{fp} and k_{pg} , were proposed by Kauffeld (1999). 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 between 3 and 2000, and ice fractions between 0 and 35%.

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

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

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

$$\mu_{pg} = \mu_{fp} (1 + 2.5 Cv_g + 10.05 Cv_g^2 + 0.00273 e^{16.6 Cv_g}) \quad (6)$$

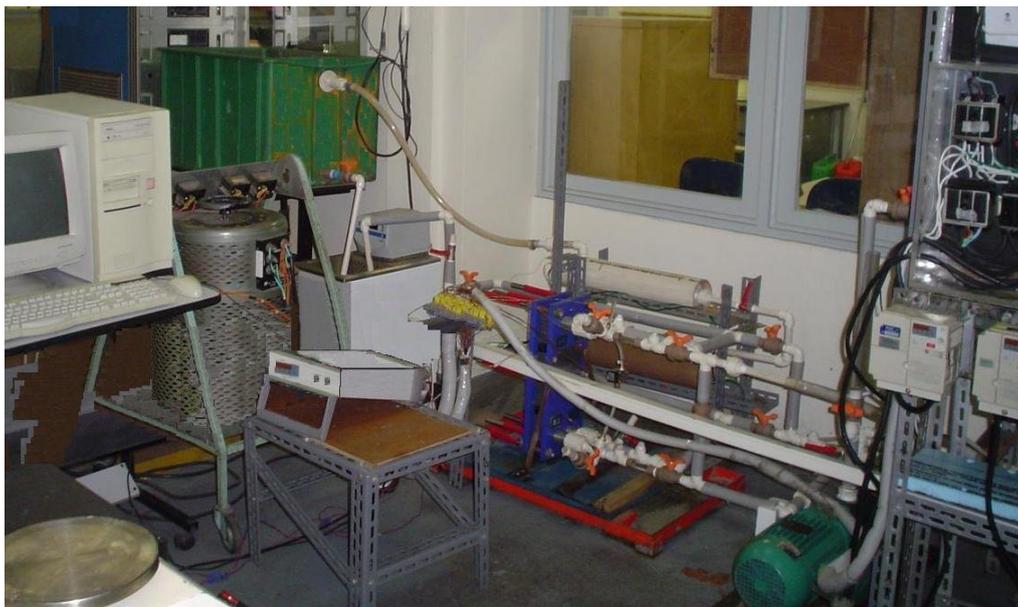


Figure 1. Test section, LRA –PUC-Rio.

2.2. Ice fraction

The calorimetric method was utilized to determine the ice fraction. The ice slurry sample is introduced inside the calorimeter. Equation (7) shows of the energy balance inside the calorimeter, the Eq. (8) make a differential analysis, taking into account the variation of the properties of the ice slurry, with the temperature (T) and the additive concentration of the carrier fluid (%C_m), during the samples warming-up process. The first term of the right hand side of the Eq. (8) represents the sensible heat of the carrier fluid. The second one is the sensible heat of the components of the calorimeter and the last one is responsible for the measurement of the latent heat that melts the ice and reduces the ice fraction. The value of the mean specific heat of the carrier fluid (C_{pm-fp}) can be assumed constant, for each differential time interval. The ice fraction is obtained by the integration of the Eq. (9) for each period of time.

$$\int P dt = (Q_{sen} + Q_{lat}) \quad (7)$$

$$\int_{t_1}^{t_2} P dt = \int_{T_1}^{T_2} m_{pg} \left(\frac{\%C_0}{\%C_m} \right) C_{p_{fp}(T, C_0, X_m)} dT + \int_{T_1}^{T_2} m_{eq-ag} C_{p_{ag}(T)} dT + m_{pg} L (X_2 - X_1) \quad (8)$$

$$X_{i+1-t} = X_i + \frac{1}{L} \left[\frac{\dot{Q}}{m_{pg}} (t_{i+1} - t_i) - \frac{\%C_0}{\%C_{i+1}} C_{p_{m-fp}} (T_i - T_{i+1}) - m_{eq-ag} C_{p_{m-ag}} (T_i - T_{i+1}) \right] \quad (9)$$

2.3. Data reduction

With the experimental data obtained in the laboratory, the overall heat transfer coefficient (U) can be obtained from the Eq. (10) and (11). The thermal load (Q_{ag}) is calculated through Eq. (12). The cooling capacity (Q_{pg}) that represents the heat absorbed by the ice slurry is calculated as shown in Eq. (13) for each test condition.

$$U = \frac{\dot{Q}_{ag}}{A F \Delta T_{LM}} \quad (10)$$

$$\Delta T_{LM} = \frac{(\Delta T_{e-tr} - \Delta T_{s-tr})}{Ln \left[\frac{\Delta T_{e-tr}}{\Delta T_{s-tr}} \right]} \quad (11)$$

$$\dot{Q}_{ag} = \dot{m}_{ag} C_{p_{m-ag}} \Delta T_{ag} \quad (12)$$

$$\dot{Q}_{pg} = \dot{m}_{pg} \left[C_{p_{m-fp}} (T_{fp-e} - T_{fp-s}) + L (X_{g-e} - X_{g-s}) \right] \quad (13)$$

In order to know the convective heat transfer for the ice slurry (h_{pg}), 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 (14) shows a correlation for the thermal resistances in an element of the heat exchanger, Fig. 2. In this case, the Nusselt number is well correlated as shown in Eq. (15). Equation (16) is obtained by substituting Eq. (15) in (13). 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.

$$\frac{1}{U} = \frac{1}{h_1} + \frac{1}{R_{F-1}} + \frac{e_{plac}}{k_{plac}} + \frac{1}{R_{F-2}} + \frac{1}{h_2} \quad (14)$$

$$Nu = a Re^b Pr^c \quad (15)$$

$$\frac{1}{U} - \frac{e_{plac}}{k_{plac}} = \frac{1}{a Re_{ag}^b Pr_{ag}^{0.3} k_{ag}} + \frac{1}{a Re_{sol}^b Pr_{sol}^{0.4} k_{sol}} \quad (16)$$

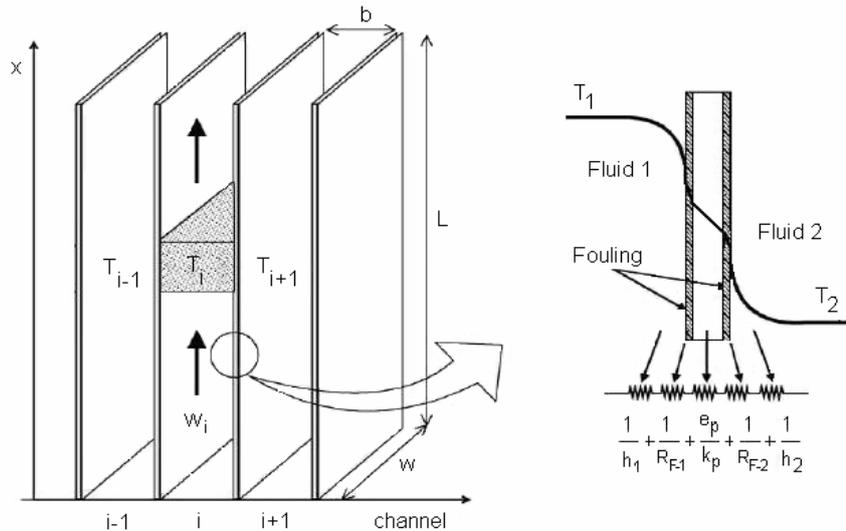


Figure 2. Resistances balance in a flow channel.

In a second phase, the tests are done with water (ag) and ice slurry (pg). 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 (Nu_{pg}), as shown in Eq. (18).

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

$$Nu_{pg} = a_2 Re^{b_2} Pr^{c_2} \quad (18)$$

The pressure drop can be calculated with a correlation used by Gut and Pinto (2003), which is based on Kaka (2002), and is presented on Eq. (19). The first term on the right hand side evaluates the pressure drop due to the friction inside the channels. 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. Being f the friction factor, L_H the large of the heat exchanger, N_p the number pass of the ice slurry, 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 D_H the hydraulic diameter.

$$\Delta p = \left[\frac{2 f L_H N_p G_C^2}{\rho_m D_H} \right] + 1.4 \left[N_p \frac{G_P^2}{2 \rho_m} \right] + \rho_m g L_H \quad (19)$$

3. 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, in Fig. 3 is shows the schematic diagram of the test section. The temperature inside of the primary reservoir is 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. Groups of electrical resistances 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 in the computer. Samples of the ice slurry are collected in both entrance and exit of the heat exchanger for their analysis in the calorimeter.

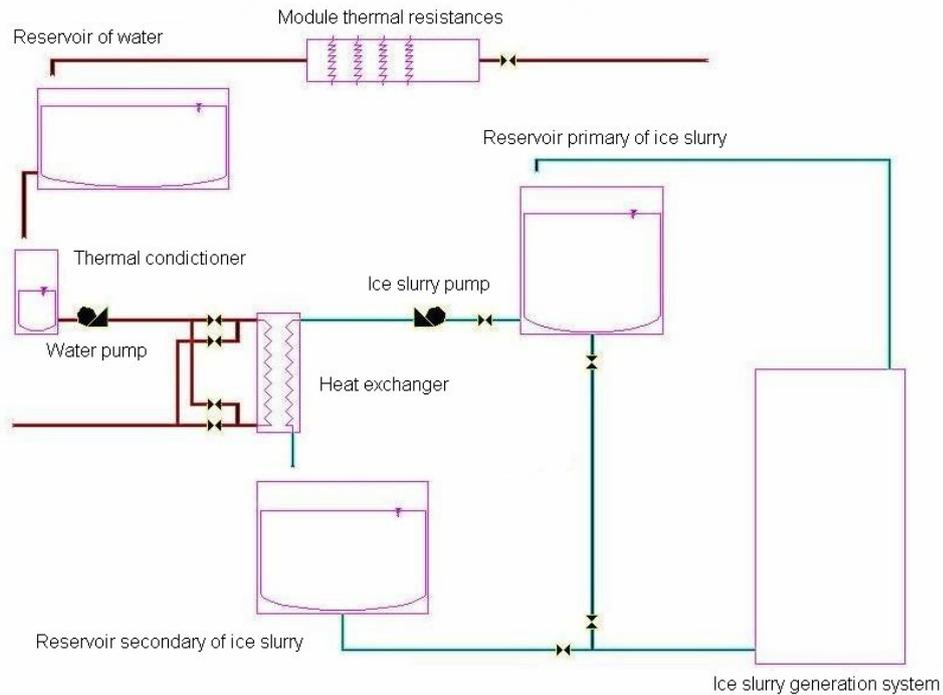


Figure 3. Schematic diagram of the test section.

4. Results and discussion

4.1. Cooling capacity

Figure 4, with the entrance conditions for the thermal load kept constant, shows the cooling capacity of the ice slurry (\dot{Q}_{pg}) increasing as its mass flow rate (\dot{m}_{pg}) increases for different initial ice fractions.

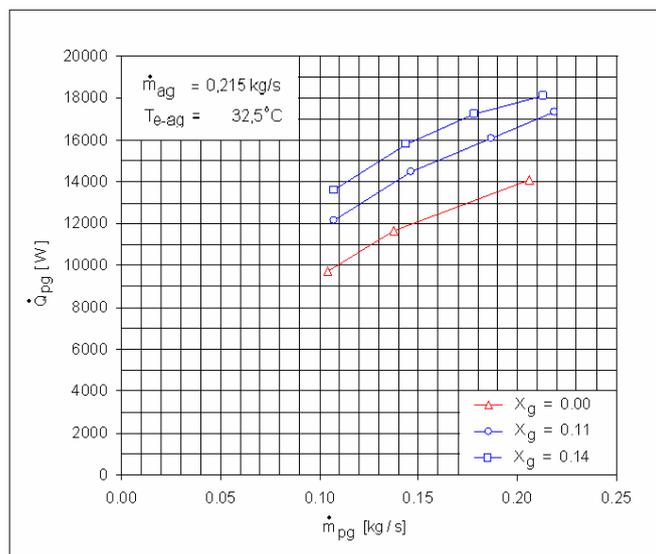


Figure 4. Cooling capacity vs. ice slurry mass flow rate.

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

Figure 5 presents two families of curves. In the first one Fig. 5.a, the water flow rate (\dot{m}_{ag}) is maintained constant with different temperatures imposed at the heat exchanger entrance ($T_{i,ag}$). The overall heat transfer coefficient decreases a little bit as the temperature is increased. The second Fig. 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 (Fig. 5.a and Fig. 5.b) 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.

Frei and Boyman (2004) found, in a similar experiment, using ice slurry from ethanol and a 9% concentration in weight, similar behavior for the overall heat transfer coefficient.

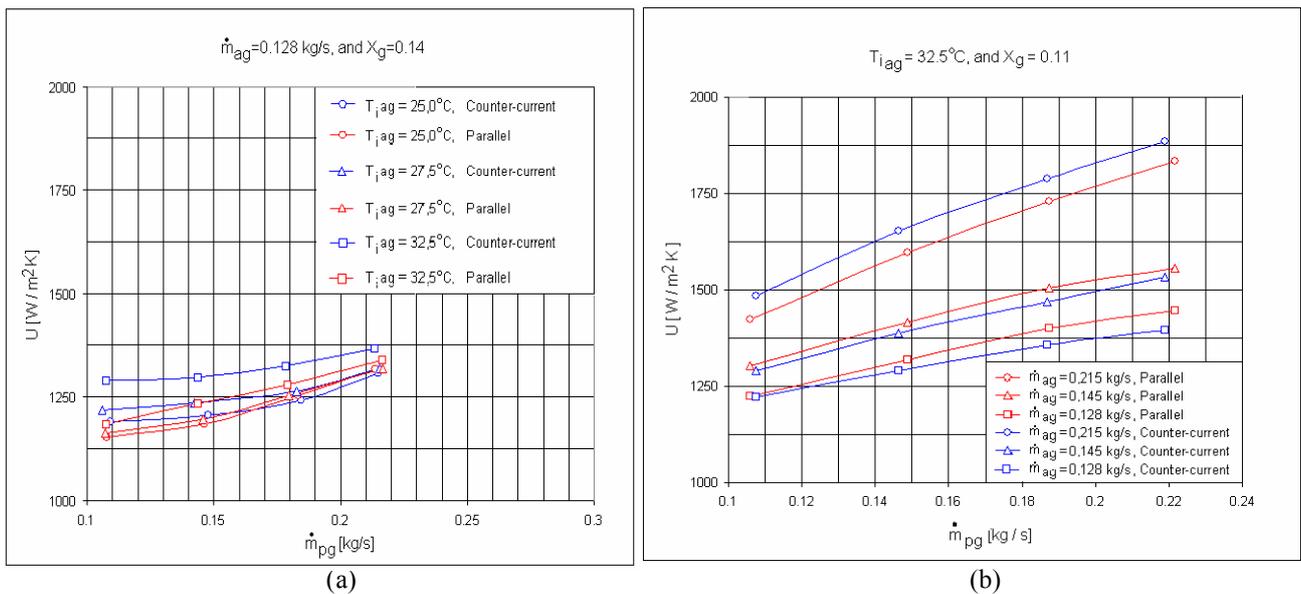


Figure 5. Overall heat transfer coefficient vs. mass flow of ice slurry.

4.3. Equivalent mass flow rates

When using the ice slurry instead of a solution without crystals, or water in similar conditions, it would be possible to reduce the mass flow rate to obtain the same cooling capacity. For instance, using an ice fraction of 0.11, keeping the same thermal load, the slurry mass flow rate could be reduced to approximately 87% of the water flow (\dot{m}_{ag-eq}). Under the same conditions, using an ice fraction of 0.14; the slurry mass flow rate could be reduced to approximately 75% of the water flow, Fig. 6.

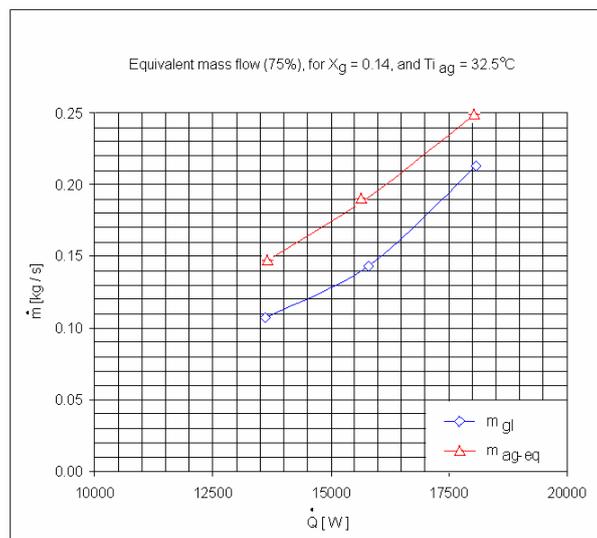


Figure 6. Equivalent mass flow rates for ice slurry.

4.4. Dimensionless correlations

Correlations for Nu, presented in Eq. (15), without phase change, were found imposing the exponents for the Prandtl number, were presented in Eq. (20) and (21). The adjustments had a mean error of the order of 4×10^{-9} .

$$Nu_{ag} = 0.159 Re^{0.742} Pr^{0.3} \quad (\text{Water side}), \quad 350 < Re < 705 \text{ and } 5.8 < Pr < 7.0. \quad (20)$$

$$Nu_{sol} = 0.159 Re^{0.742} Pr^{0.4} \quad (\text{Side without crystals}), \quad 296 < Re < 583 \text{ and } 5.9 < Pr < 7.9. \quad (21)$$

Similar expressions for Nu in different plate heat exchangers, without phase change, are presented in literature: Gut and Pinto (2004) for a mini flat plate heat exchanger Armfield Ft-43, Eq. (22). Holger (1992) displays Eq. (23), of Bassiouny (1985), in a treaty on heat exchangers.

$$Nu = 0.0169 Re^{0.897} Pr^{1/3} \quad (\text{with } R^2 = 0.98), \quad 100 < Re < 1000 \text{ and } 2.2 < Pr < 6.8 \quad (22)$$

$$Nu = 0.274 Re^{0.69} Pr^{0.4} \quad (\text{for plates of type H}), \quad 100 < Re < 10000 \text{ and } 2.0 < Pr < 40 \quad (23)$$

In Fig. 7.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).

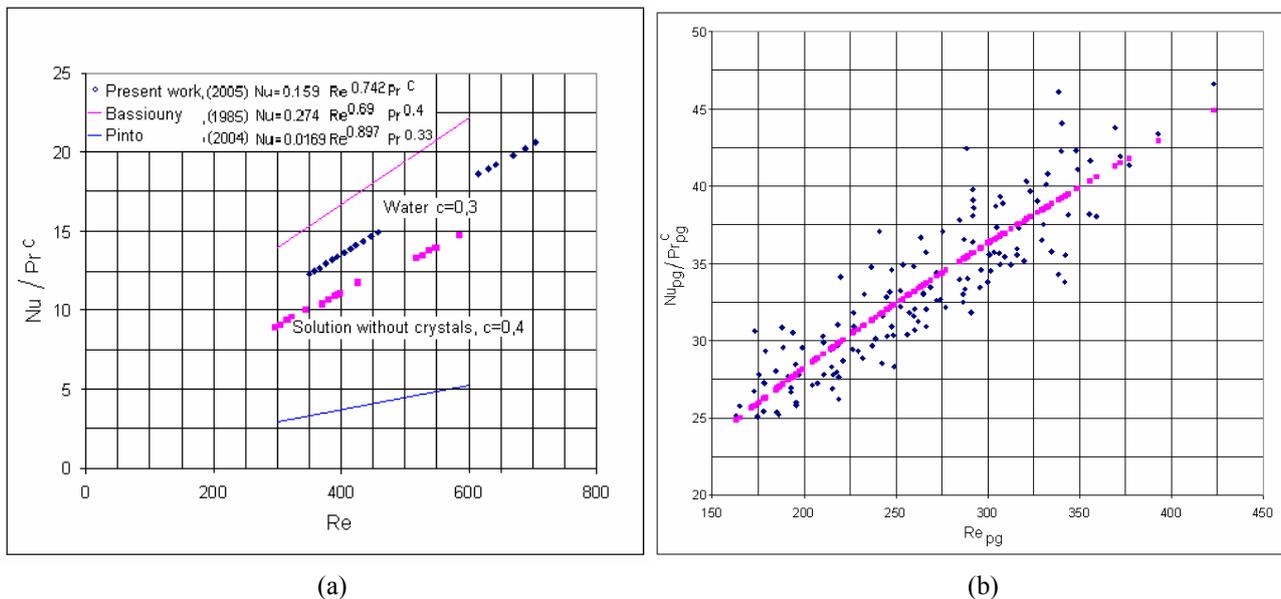


Figure 7. (a) Relations $Nu.Pr^c/Re$, for aqueous solution without Ice crystals, and (b) Ice slurry.

Figure 7.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 of 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. Adjusting all the ice slurry results for Nusselt by least square, Eq. (24), it is obtained an average quadratic deflection of $S=2.119$. 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 (26), with average quadratic deflections of $S=1.443$ and $S=1.347$ respectively.

$$Nu_{pg} = 1.057 Re^{0.62} Pr^{-0.115} \quad (24)$$

$$Nu_{pg} = 0.387 Re^{0.581} Pr^{0.4} \quad (\text{to } X_g=11\%) \quad (25)$$

$$Nu_{pg} = 0.784 Re^{0.445} Pr^{0.4} \quad (\text{to } X_g=14\%) \quad (26)$$

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. 8. Bellas et al. (2002) found similar results in the same range of the present work.

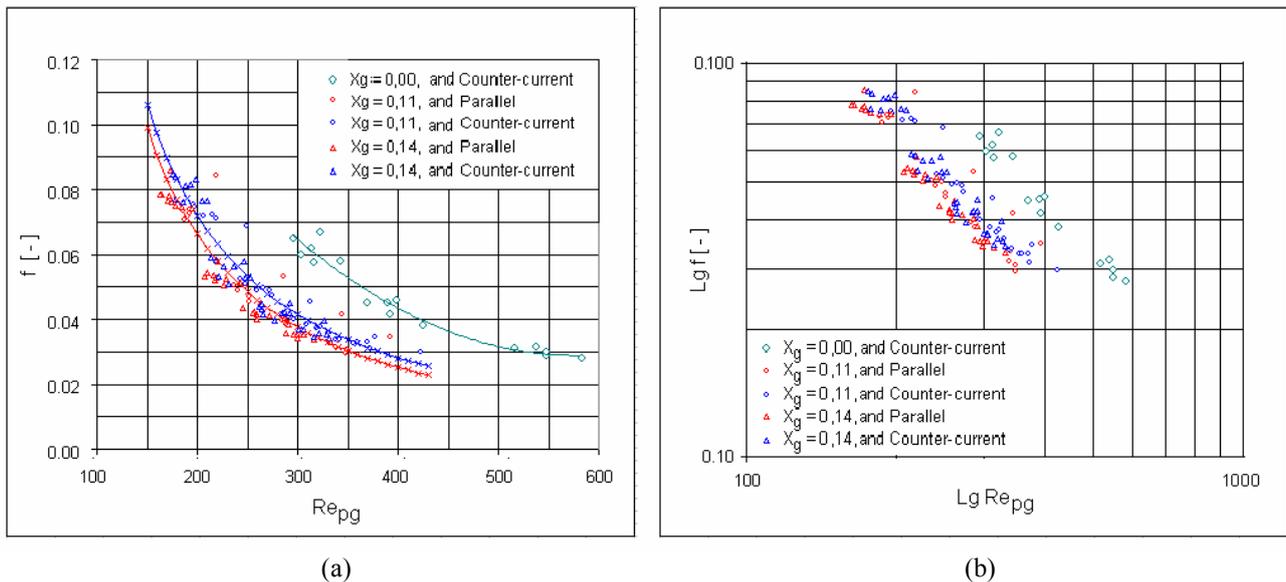


Figure 8. (a) Friction factor vs. Reynolds number, and (b) Logarithmic relation.

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

$$f_{pg} = 123.46 \text{ Re}^{-1.325} \quad (R^2=0.934) \quad (27)$$

For the ice slurry with ice fraction 0.11, Eq. (28).

$$f_{pg} = 94.14 \text{ Re}^{-1.355} \quad (X_g=0.11, R^2=0.972) \quad (28)$$

Finally, for the ice slurry with ice fraction 0.14, Eq. (29).

$$f_{pg} = 108.61 \text{ Re}^{-1.397} \quad (X_g=0.14, R^2=0.960) \quad (29)$$

From the present results, it is possible to conclude that the use of ice slurry has strong influence on both, the heat transfer and the pressure drop coefficients. Its use possibilities allow the mass flow rate reduction, keeping constant the cooling capacity.

Due to the strong dependency between viscosity and solid fraction, it is still 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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