ASSESSMENT OF IMPROVEMENT IN HEAT EXCHANGERS BEHAVIOUR USING ICE SLURRY AS SECONDARY REFRIGERANT

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ABSTRACT

Ice slurry is well known as a biphasic secondary refrigerant that presents several potential advantages compared to single phase secondary refrigerants. These potential advantages can be summarized in the ability to use the thermal storage and the high cooling capacity given by the latent heat. Theoretically, these features must allow important energy savings in secondary refrigerant distribution loop.

An accurate evaluation of these energy savings requires the knowledge of the thermal and rheological behavior of the refrigerant studied. Based on the experimental model developed by the authors for brine based ice slurry, a theoretical analysis of heat exchangers behavior is presented in this work in order to find out the potential energy savings associated to its use. The influence of ice concentration, mass flow rate, heat exchanger length and pipe outer wall temperature over pumping power and heat transfer rate is studied. The ratio between heat transfer rate and pumping power is used as the evaluation parameter, which allows to find the most favorable operation conditions for ice slurry flow.

In order to asses the improvement obtained using ice slurry, results for ice slurry are compared to those obtained for carrier fluid at same inlet temperature. Finally, a practical example is proposed where the behavior of a facility with several heat exchangers working in serial lay-out is analyzed for ice slurry and single phase flow.

1. INTRODUCTION

Ice slurry is considered as a very promising secondary refrigerant that, besides the reduction on the charge of primary refrigerant associated to any secondary refrigerant, allows a reduction in energy consumption compared to single phase secondary refrigeration systems as well as the possibility of thermal storage. This reduction in energy consumption has been extensively treated previously and is only outlined here. It is obtained in two different ways: firstly, the energy efficiency of an ice slurry plant is greater than that of a plant using a single phase secondary refrigerant (Rivet, 2007; Stamatiou, 2005); secondly, the energy consumption on the pumps used in the secondary refrigerant distribution system can be reduced compared to the energy consumption necessary to pump the traditional single phase secondary refrigerant (Kauffeld, 2005).

The attention of this paper is focused on the energy savings obtained in the secondary refrigerant distribution system. An accurate assessment of these savings requires the knowledge of ice slurry thermal and rheological behavior. The authors of this work have been experimentally developed a thermal and rheological model for the ice slurry produced using 9 wt% sodium chloride brine as base solution (Illán and Viedma, 2008a; 2008b), which has been experimentally validated in a commercial corrugated tube heat exchanger (Illán and Viedma, 2009). Based on this model, a theoretical analysis of smooth and corrugated tube heat exchangers behavior is developed in this work in order to analyze the potential energy savings associated to the use of ice slurry. Pumping power and heat transfer rate have been numerically obtained for tube lengths between 0 and 25 meters. Influence of ice content, mass flow rate and outer wall temperature has been analyzed for each tube length; variation range of all these variables analyzed is given in Table 1. The power ratio for ice slurry defined as the ratio between heat transfer rate and pumping power ($PR_{is} = \dot{Q}_{is} / P_{is}$) has been used as the evaluation parameter, which allows to find the most favorable operation conditions for ice slurry flow.

Table 1. Variation range of all variables analyzed.

Tube type	Tube length (m)	Ice concentration (wt%)	Mass flow rate (kg/s)	Outer wall temperature (ºC)
Smooth	0 ÷ 25 (ΔL=5 mm)	5, 10, 15, 20 & 25	375, 750, 1125, 1500, 1875, 2250, 2625 & 3000	0, 10, 20, 30, 40 & 50
Corrugated	0 ÷ 25 (ΔL=5 mm)	5, 10, 15, 20 & 25	375, 750, 1125, 1500, 1875, 2250, 2625 & 3000	0, 10, 20, 30, 40 & 50

A similar analysis has been made for the case when a heterogeneous storage is used and only carrier fluid flows through the heat exchanger. Carrier fluid power ratio ($PR_{cf} = \dot{Q}_{cf} / P_{cf}$) has been obtained and used to compare ice slurry and carrier fluid performance.

In order to asses the improvement obtained using ice slurry an improvement ratio has been defined as the ratio between ice slurry and carrier fluid power ratios ($IR = PR_{is} / PR_{cf}$). Values of improvement ratio higher than 1 will represent those situations where the use of ice slurry improves heat exchanger behavior; alternatively, values of improvement ratio lower than 1 imply that the use of ice slurry will be not recommended.

Finally, a practical example is proposed where the behavior of a facility with several heat exchangers working in serial lay-out is analyzed for ice slurry and single phase flow.

2. HEAT TRANSFER RATE AND PUMPING POWER IN HEAT EXCHANGERS

The model used in this work is based on the basic correlations obtained by the authors for Nusselt number and Darcy friction factor for ice slurry flow through horizontal smooth and corrugated pipes (Illán and Viedma, 2008a; 2008b). Pumping power and heat transfer rate can be obtained from these basic correlations as shown below.

According to the conclusions obtained in those works, the parameters which have more influence over heat transfer and pressure drop processes are the Reynolds number (Re), the ice concentration (ϕ) and the ice particle-pipe diameter ratio (*d/D*). All the results obtained in this work has been calculated assuming that the ice particle mean diameter remain constant at 500μm and for a constant tube inner diameter of 18 mm, whereas Reynolds number and ice concentration have been varied as shown in Table 1.

2.1 Pumping power

When the Darcy friction factor, λ, is known, pressure drop through a straight stretch of tube can be easily obtained by applying the Darcy-Weisbach equation. Therefore, the pumping power can be obtained as:

$$P = \lambda \frac{\dot{m} L v^2}{2D}, \qquad (1)$$

where $\dot{m}$ is the mass flow rate in kg/s, L is the pipe length in m, v is the flow velocity in m/s and D is the pipe diameter in m.

Due to the ice melting, friction factor value varies along the tube length and its value depends on heat transfer rate. Therefore friction factor has been obtained simultaneously to heat transfer rate, using the correlations proposed by the authors (Illán and Viedma, 2008a, 2008b).

2.2 Heat transfer rate
To obtain the heat transfer rate, the whole heat exchanger was divided in 5 mm length elements. In each element, differences between inlet and outlet properties are nearly imperceptibles and therefore heat transfer rate can be easily obtained from the following equation system:

$$\dot{Q} = AU\Delta T_m = A \times \left[\frac{1}{A_{in} h_{inner}} + \frac{\ln\left(D_{outer} / D_{inner} \right)}{2\pi k L} \right]^{-1} \times \left(T_w - \overline{T} \right), \tag{2}$$

$$\dot{Q} = \dot{m}\left[\left(\phi_{in} - \phi_{out} \right) H_f + \left(1 - \overline{\phi} \right) \overline{c}_p \left(T_{out} - T_{in} \right) \right], \tag{3}$$

where the only unknown variables are the heat transfer rate, $\dot{Q}$, and the flow outlet temperature, T_{out}. In these expressions, the mean values for the fluid temperature, $\overline{T}$, the ice concentration, $\overline{\phi}$, and the carrier fluid specific heat, $\overline{c}_p$, are obtained as the average between inlet and outlet values. Heat transfer coefficient in the tube inner wall, h_{inner}, can be obtained using the correlations proposed by the authors (Illán and Viedma, 2008a, 2008b).

3. RESULTS

Equations (1) to (3) have been used to obtain ice slurry (PR_{is}) and carrier fluid power ratio (PR_{cf}), as well as the improvement ratio for smooth (IR_s) and corrugated (IR_c) heat exchangers. Results are shown bellow.

3.1 Smooth tube heat exchanger results
The great amount of information obtained make difficult to include all the results in this paper and therefore only the extreme values for tube wall temperature and mass flow rate are plotted. Fig. 1 and Fig. 2 show results for the ice slurry power ratio and the improvement ratio for a pipe outer wall temperature of 0ºC and mass flow rates of 375 and 3000 kg/h. The values obtained for intermediate mass flow rates vary progressively between the two extreme situations plotted in Fig. 1 and Fig. 2.

Similarly, Fig. 3 and Fig. 4 show, for a pipe outer wall temperature of 50ºC, the same parameters plotted in Fig. 1 and Fig. 2 for 0ºC. The values obtained for intermediate pipe outer wall temperature vary progressively between the two extreme situations plotted in Fig. 1&2 and Fig. 3&4.

As it can be seen in these figures, in general terms the improvement obtained using ice slurry is higher for the lower pipe wall temperature, the lower mass flow rate, the higher ice concentration and the higher heat exchanger length. Nevertheless, as the pipe wall temperature increases, the influence of the length varies slightly and an initial decrease of the improvement ratio can be observed for low heat exchanger lengths until an inflection point is reached where *IR* value again increases with heat exchanger length.

Independent of operation conditions, it seems clear that the improvement obtained increasing heat exchanger length presents a horizontal asymptotic behavior. The value of heat exchanger length above which the improvement ratio remains nearly constant depends on operation conditions, although in any case, the inflection point is given by the moment at which the ice contained in the ice slurry flow has melted at all.

Figure 1. Ice slurry power ratio for smooth tube heat exchangers ($PR_{is,s}$) with an outer tube wall temperature of 0ºC and mass flow rates between 375 and 3000 kg/h.

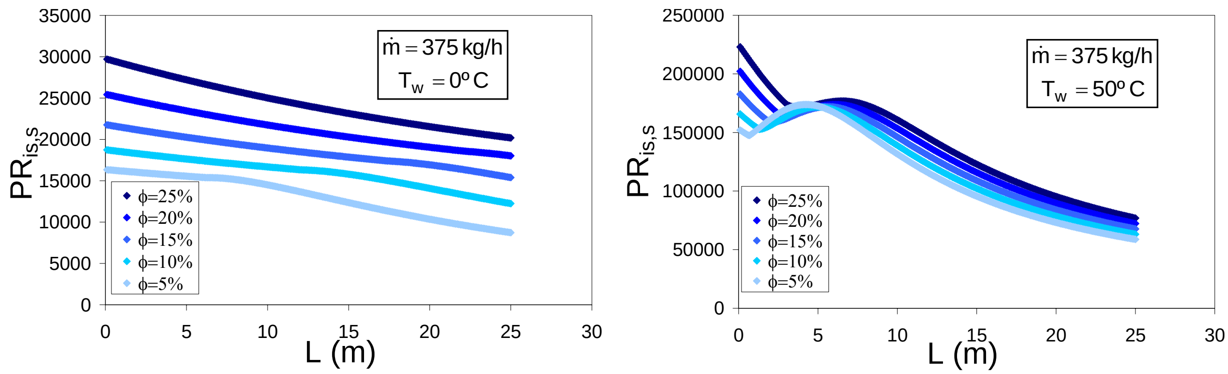
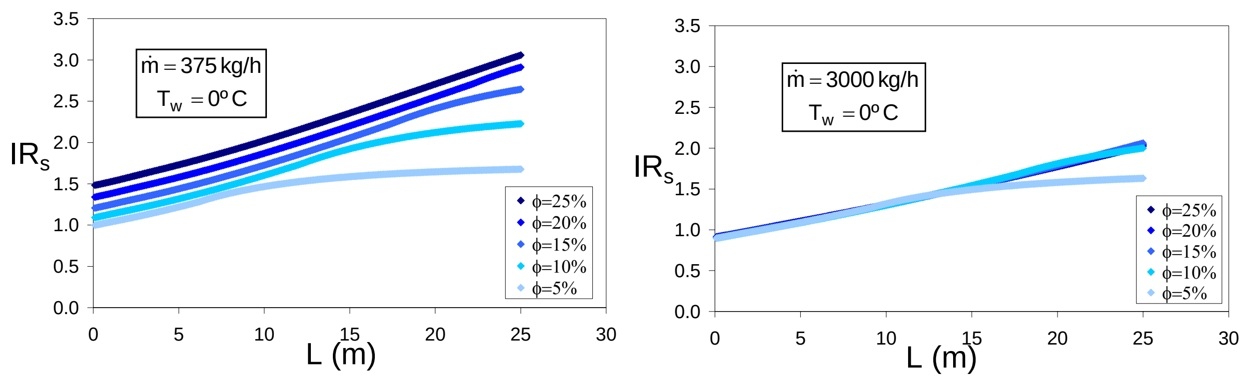


Figure 2. Improvement ratio for smooth tube heat exchangers (IR_s) with an outer tube wall temperature of 0ºC and mass flow rates between 375 and 3000 kg/h.



Finally, in Fig. 1 and Fig. 3 it can be clearly seen that the power ratio for ice slurry flow presents an opposite behavior, decreasing as pipe wall temperature, mass flow rate or heat exchanger length increases, although the influence of ice content is similar to that observed for the improvement ratio.

Therefore, an appropriate design of a heat exchanger using ice slurry as refrigerant must take into account not only the improvement obtained respect to the use of single phase refrigerant (*IR* value), but also the relationship between heat transfer rate and pumping power consumption (*PR* value) and the influence of all the other involved parameters like initial costs, space available, etc.

For example, from Fig. 3 and Fig. 4 it can be deduced that the optimum design for a pipe wall of 50ºC is a 6.5 meters length heat exchanger with a mass flow rate of 375 kg/h. If mass flow rate increases, *IR* decreases slightly as well as *PR* decreases strongly; if length increases, *IR* increases progressively until a nearly constant value is reached, but *PR* decreases strongly. Therefore, from an energy optimization point of view, the optimum design brings to a power rate of $177 \cdot 10^3$ (0.109 W of pumping power for 19.28 kW of heat transferred) and an improvement ratio of 1.09 (a 9% of improvement in power ratio with respect to that obtained for single phase flow). If only energy criteria are taken into account, the values of improvement ratio obtained for optimum design conditions are always relatively low. Nevertheless, in the practice of heat exchangers design other variables are involved that can lead to higher improvement when using ice slurry. A practical example with a commercial heat exchanger will be treated in section 4.

Figure 3. Ice slurry power ratio for smooth tube heat exchangers ($PR_{is,s}$) with an outer tube wall temperature of 50ºC and mass flow rates between 375 and 3000 kg/h.

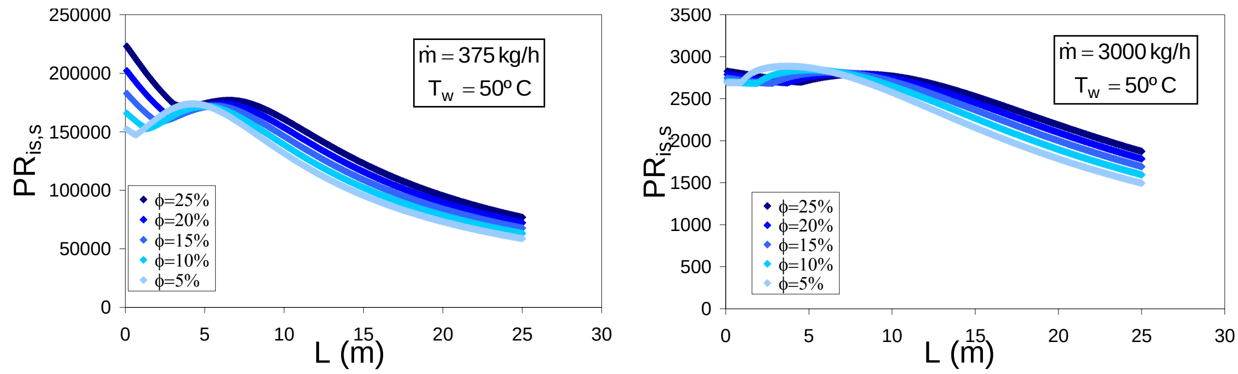
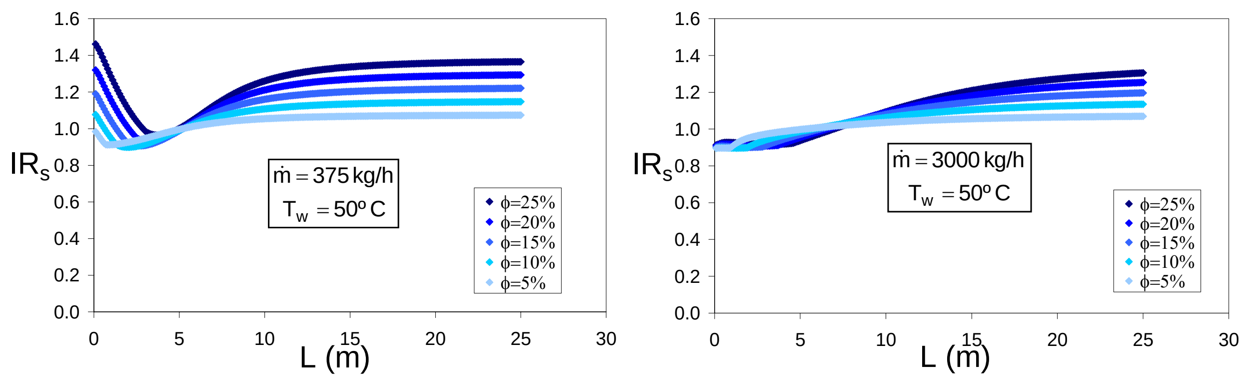


Figure 4. Improvement ratio for smooth tube heat exchangers (IR_s) with an outer tube wall temperature of 50ºC and mass flow rates between 375 and 3000 kg/h.



3.2 Corrugated tube heat exchanger results

Results of improvement ratio obtained in corrugated tube heat exchangers are shown in Fig. 5 and Fig. 6. For the same reason expounded for smooth tube heat exchanger, only the extreme values for tube wall temperature and mass flow rate are plotted in these figures.

Figure 5. Improvement ratio for corrugated tube heat exchangers (IR_c) with an outer tube wall temperature of 0ºC and mass flow rates between 375 and 3000 kg/h.

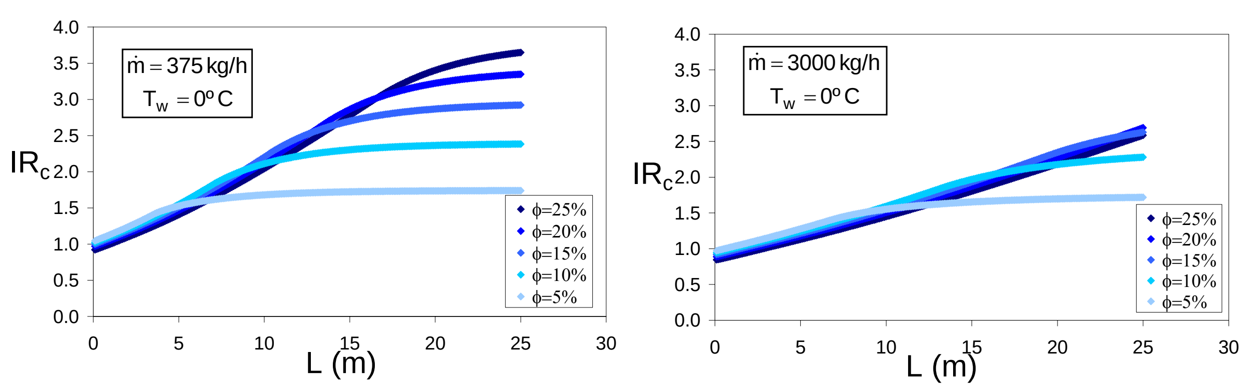
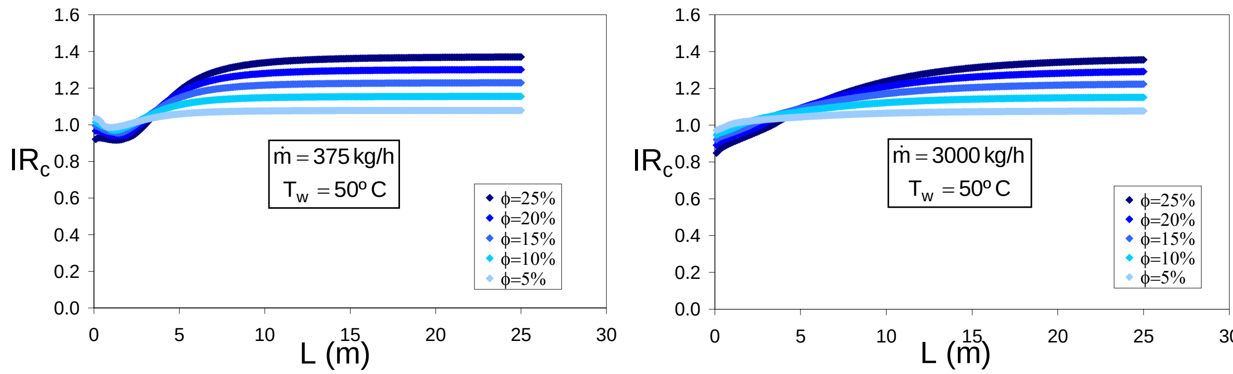


Figure 6. Improvement ratio for corrugated tube heat exchangers (IR_c) with an outer tube wall temperature of 50ºC and mass flow rates between 375 and 3000 kg/h.



The main conclusions obtained for smooth tube are also valid for corrugated tube, since the trends observed in both cases are qualitatively the same. Besides the quantitative differences, the only appreciable differences observed for corrugated tube are a greater influence of ice concentration as well as the fact that for the lower heat exchanger length, while the ice has not melted at all, the improvement obtained is higher for the lower ice concentration. Once the ice has melted at all, the behavior observed is quite similar to that presented in smooth tube.

4. ANALISYS OF A COMMERCIAL HEAT EXCHANGER FACILITY

A practical example of a refrigeration facility is analyzed in this section. The heat exchangers are K Series heat exchangers by the HRS-Spiratube Company. They are all welded stainless steel multitube heat exchanger with the inner tubes corrugated, designed for use in industrial applications.

Figure 7. Sketch of the heat exchanger (a) and the configurations (b) analyzed.

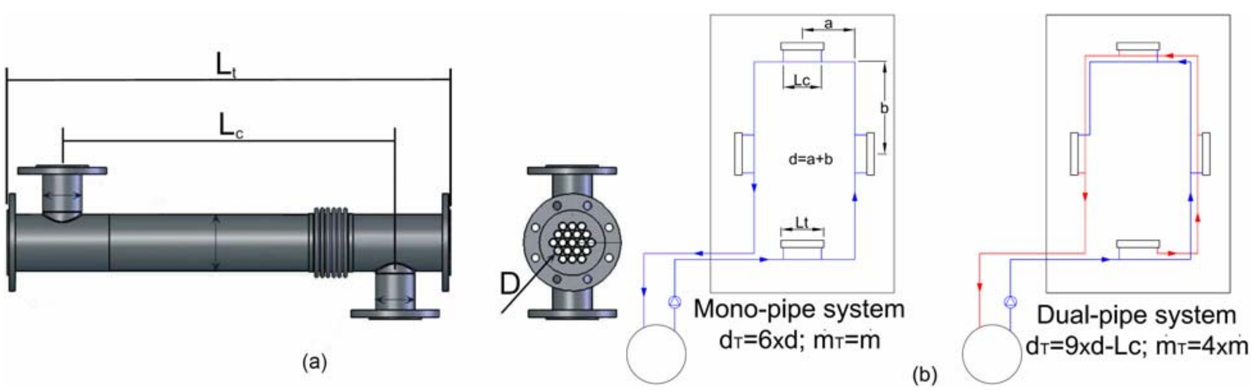


Fig. 7.a shows a sketch of this heat exchanger. A schematic representation of the configurations analyzed is shown in Fig. 7.b, where *d* is a scale factor which represents the distance between two consecutive heat exchangers. Features of the three different system configurations analyzed are summarized in Table 2.

When ice slurry is directly used as secondary refrigerant, multiple heat exchangers can be serially connected without introducing a significant reduction in the temperature difference in the downstream equipment. In these conditions, a mono pipe system can be used. If stirring devices are no used during the storage, ice crystals stratify and float to the surface of the liquid and only the carrier fluid is used as secondary refrigerant. In most cases is necessary to use a dual pipe system due to the increase in fluid temperature when passing through each heat exchanger and therefore the total piping length increases (d_t) as well as mass flow rate increases.

Table 2. Features of the three system configurations analyzed.

System size	**Heat exchanger reference**	**Number of tubes**	**D (mm)**	**L_t (mm)**	**L_c (mm)**	**d (m)**	**Mono-pipe d_t (m)**	**Dual-pipe d_t (m)**
Small (S)	K7-0.75.304/316LH	7	18	699	509	3	18	26.491
Medium (M)	K13-3.0.304/316LH	13	18	2939	2723	6	36	51.277
Large (L)	K55-6.0.304/316LH	55	18	5910	5600.9	12	72	102.399

Mass flow rate in each corrugated tube is fixed in 375 kg/h. If mass flow rate increases, the improvement obtained using ice slurry decreases (IR_c decreases), as well as the heat transferred by unit pumping power also decreases ($PR_{is,c}$ decreases). On the other hand, mass flow rates lower than 375 kg/h can lead to unsafe operation conditions with ice particle segregation. Corrugated tubes outer wall temperature has been fixed in 0, 20 and 50ºC, whereas ice concentration has been fixed in 5, 10, 15, 20 and 25%.

4.1 Results

For the small size configuration, the heat exchanger has an effective exchange length of only 0.509 m and, accordingly to the result presented in section 3, the higher improvement is obtained for the lower ice concentration (5%) and the lower pipe wall temperature (0ºC). In these conditions, a total heat transfer of 7.3 kW is obtained when using ice slurry as secondary refrigerant, whereas the pumping power is 0.11 kW (using a PVC DN 20 pipe and neglecting pressure drop in fittings). For these conditions and this heat exchanger model, the increase in fluid temperature when using carrier fluid as secondary refrigerant is relatively low and therefore a mono pipe system can be used even in this case. The total heat transferred using carrier fluid is 5.96 kW with a pumping power of 0.106 kW. Therefore, the improvement ratio obtained using ice slurry is around 1.16. For higher pipe wall temperatures, in most cases the use of ice slurry does not improve the behavior of the facility.

For the medium size configuration, due to the relatively high length of the heat exchanger used, the heat transferred in each heat exchanger is high and therefore is necessary to use a dual pipe system when using carrier fluid as secondary refrigerant. The higher improvement obtained in his model has been found for an intermediate ice concentration (15%) and for the lower pipe wall temperature (0ºC). For these conditions the total heat transferred is about 75 kW, with a pumping power of about 0.3 kW (using PVC DN 25 pipe), whereas the heat transfer rate is approximately constant in all the heat exchangers (varies from 19.7 kW in the first one to 17.9 kW in the last one). The heat transferred using carrier fluid in a dual pipe system is about 61.5 kW with a pumping power of about 0.77 kW (using PVC DN 40 pipe). Therefore, the improvement ratio obtained using ice slurry is around 3.16.

Finally, for the large configuration the higher improvement obtained has been found for the higher ice concentration (25%) and the lower pipe wall temperature (0ºC). In these conditions, the total heat transferred is about 603.6 kW, with a pumping power of about 0.8 kW (PVC DN 50). Due to its high length, in this model the heat transfer rate only remains approximately constant in the first three heat exchangers (varies from 182 kW in the first one to 162 kW in the third one, decreasing in the last one until 87 kW). Therefore, in this model and for a mass flow rate of 375 kg/h per pipe, is not interesting to use more than three serially connected heat exchangers; increasing unitary mass flow rate it will be possible to connect more than three heat exchangers maintaining the heat transfer rate approximately constant in all the heat exchangers. The heat transferred using carrier fluid in a dual pipe system is about 467 kW with a pumping power of about 3.52 kW (using PVC DN 70 pipe). Therefore, the improvement ratio obtained using ice slurry is around 5.66.

5. CONCLUSIONS

A theoretical analysis of heat exchangers behavior has been presented which allows to find out those situations where the use of ice slurry as secondary refrigerant, in substitution of single phase refrigerants, can lead to more important energy savings.

A practical example has been analyzed which clearly shows the improvement obtained using ice slurry in three different practical configurations.

The optimal ice concentration depends on specific operation conditions, increasing as increase heat exchanger length. Although in most cases the use of ice slurry improves the facility behavior, there some cases, especially for low heat exchanger length, where the use of ice slurry is inadvisable. A careful analysis is strongly recommended previously to decide about the use of ice slurry.

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NOMENCLATURE

A	heat transfer area (m^2)	c_p	specific heat ($J \cdot kg^{-1} \cdot K^{-1}$)
D	pipe diameter (m)	h	convective heat transfer coefficient ($W \cdot m^{-2} \cdot K^{-1}$)
H_f	specific latent heat of fusion of ice ($J\ kg^{-1}$)	k	thermal conductivity ($W \cdot m^{-1} \cdot K^{-1}$)
L	tube length (m)	$\dot{m}$	mass flow rate ($kg \cdot s^{-1}$)
P	pumping power (W)	Q	heat transfer rate (W)
T	temperature (K)	U	overall heat transfer coefficient ($W \cdot m^{-2} \cdot K^{-1}$)
v	flow velocity ($m \cdot s^{-1}$)	ϕ	ice concentration (-)
λ	Darcy friction factor (-)	ΔT_m	effective mean temperature difference (K)

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