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Title: Heat exchanger performance modelling 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 should allow important energy savings in secondary refrigerant distribution loop. However, an accurate evaluation of these energy savings requires the knowledge of the thermal and rheological performance of the refrigerant studied.
Based on the experimental model developed by the authors for brine based ice slurry, a theoretical analysis of heat exchangers performance is presented in this work in order to calculate the potential energy savings associated to its use. The influence of ice concentration, mass flow rate, heat exchanger length and cooled fluid 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 us to find the most favourable operation conditions for ice slurry flow.
In order to assess the improvement obtained using ice slurry, results for ice slurry are compared to those obtained for carrier fluid at same inlet temperature.

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Departamento de Ingeniería Térmica y de Fluidos

Cartagena, 20th of July, 2011

Dear Editor,

Attached we are sending the paper titled:

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Authors: F. Illán and A. Viedma,

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Yours faithfully,

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Highlights

Assessment of the improvement obtained using ice slurry in heat exchangers.

Chilled carrier fluid compared to ice slurry.

The influence of ice concentration, mass flow rate, heat exchanger length and cooled fluid temperature over pumping power and heat transfer rate is studied.

The optimal ice concentration depends on specific operation conditions and the heat exchanger type.

As a general rule, the optimal ice concentration increases as the heat exchanger length increases.

Heat exchanger performance modelling using ice slurry as secondary refrigerant.

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Abstract.

Ice slurry is a biphasic secondary refrigerant that theoretically allows important energy savings in secondary refrigerant distribution loop compared to single phase refrigerants. However, an accurate evaluation of these energy savings requires the knowledge of the thermal and rheological performance of the refrigerant.

Based on the experimental model developed by the authors, a theoretical analysis of heat exchangers performance is presented in this work in order to calculate the potential energy savings associated to its use. The influence of ice concentration, mass flow rate, heat exchanger length and cooled fluid 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 us to find the most favourable operation conditions for ice slurry flow.

Results for ice slurry are compared to those obtained for carrier fluid at same inlet temperature to assess the improvement obtained.

Keywords: Ice slurry; Pressure drop; Heat transfer; Heat exchanger.

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Nomenclature

A	heat transfer area (m^2)
c_p	specific heat (J kg^{-1} K^{-1})
D	pipe diameter (m)
h	convective heat transfer coefficient (W m^{-2} K^{-1})
H_f	specific latent heat of fusion of ice (J kg^{-1})
HTR	heat transfer ratio $\left(HTR = \dot{q}_{is}/\dot{q}_{cf}\right)$
IR	improvement ratio $\left(IR = PR_{is}/PR_{cf}\right)$
k	thermal conductivity (W m^{-1} K^{-1})
L	tube length (m)
$\dot{m}$	mass flow rate (kg s^{-1})
P	pumping power (W)
PPR	pumping power ratio $\left(PPR = P_{is}/P_{cf}\right)$
PR	power ratio $(PR = \dot{q}/P)$
$\dot{q}$	heat transfer rate (W)
T	temperature (K)
ΔT_m	effective mean temperature difference (K)
U	overall heat transfer coefficient (W m^{-2} K^{-1})
v	flow velocity (m s^{-1})

Greek symbols

ϕ	ice mass fraction (-)
λ	Darcy friction factor (-)

Subscripts

c	corrugated tube
cf	carrier fluid
in	tube inner wall
is	ice slurry
out	tube outer wall
pg	propylene glycol
s	smooth tube

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 performance. The authors of this work have experimentally developed a thermal and rheological model for the ice slurry produced using 9 wt% sodium chloride brine as base solution (Illán and Viedma, 2009a; 2009b), which has been experimentally validated in a commercial corrugated tube heat exchanger (Illán and Viedma, 2009c). According to that model, the Darcy friction factor and the Nusselt number for ice slurry flowing through horizontal pipe can be obtained as a function of Reynolds number, ice content and ice particle-pipe diameter ratio: $\lambda = f(Re, \phi, d/D); Nu = f(Re, \phi, d/D)$. Based on this model, a theoretical analysis of smooth and corrugated tube heat exchangers performance 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 10 meters. Influence of ice content, mass flow rate and cooled fluid temperature have been analyzed for each tube length; the variation range of all these variables analyzed is given in Table 1.

According to Kozawa (Kozawa et al., 2005), one way of categorizing ice slurry thermal storage systems is to distinguish between stores with heterogeneous and homogeneous compositions of ice particles. When heterogeneous storage is employed, the ice slurry can be used in an indirect way. The chilled carrier fluid with no ice particles can be extracted from the bottom of the storage tank and used as conventional single phase secondary refrigerant, maintaining the high thermal storage capacity of ice slurry systems. On the other hand, when homogeneous storage is employed, ice slurry can be extracted from the storage tank and used in a direct way. In this case, some benefits of saving in pipe dimensions are expected.

A power ratio, defined as the ratio between heat transfer rate and pumping power $(PR = \dot{q}/P)$, has been used as comparison parameter. The ice slurry power ratio (PR_{is}) has been calculated for the case when ice slurry flows through the heat exchanger.

Similarly, carrier fluid power ratio $\left(PR_{cf}\right)$, has been calculated for the case when a heterogeneous storage is used and only carrier fluid flows through the heat exchanger.

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

Heat transfer ratio $\left(HTR = \dot{q}_{is}/\dot{q}_{cf}\right)$ and pumping power ratio $\left(PPR = P_{is}/P_{cf}\right)$ have also been calculated and plotted in order to obtain additional information about ice slurry performance.

2. - Procedure.

Figure 1 shows a sketch of the type of heat exchangers analyzed in this work. It is a tube-in-tube heat exchanger consisting of a single ice slurry tube (outer diameter 20 mm, inner diameter 18 mm) mounted inside an outer shell (outer diameter 31 mm, inner diameter 28 mm). The ice slurry stream flows inside the tube, cooling the product (10 wt% propylene glycol solution) which flows around it.

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, 2009a; 2009b). 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 the 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 have been calculated assuming that the ice particle mean diameter remain constant at 500 µm ($d/D = 36^{-1}$), whereas ice concentration has been varied between 5 wt% and 25 wt% and the outlet temperature for propylene glycol solution has been varied between 0 ºC and 50 ºC.

Since there are no existing recommendations for ice slurry flow through heat exchangers, in order to establish the tube-side velocity for ice slurry, general recommendations for water have been taken into account. According to Kuppan (Kuppan, 2000), the tube-side velocity for water and similar fluids must be maintained between 0.9 to 2.4 $m \cdot s^{-1}$; the lower velocity limit corresponds to limiting the fouling, and the upper velocity limit corresponds to limiting the rate of erosion. Other authors (Taborek, 1983) state that, based on overall cost optimization between pumping power cost and cost of fouling, the optimal tube-side flow velocity for cooling water is around 2 $m \cdot s^{-1}$. Since ice slurry can be used as a cleaning agent (Quarini, 2002), ice slurry mass flow has been initially fixed at 1500 $kg \cdot h^{-1}$ to obtain a flow velocity around 1.5 $m \cdot s^{-1}$ and then reduce pumping power cost without increasing the cost of fouling. In order to obtain similar values for the shell-side and tube-side heat transfer coefficients, mass flow for propylene glycol solution has been initially fixed at 1250 $kg \cdot h^{-1}$, leading to a flow velocity of around 1.1 $m \cdot s^{-1}$, which falls within the range of 0.6 to 1.5 $m \cdot s^{-1}$ recommended by Kuppan (Kuppan, 2000) for the shell-side velocity.

In a later analysis, the influence of the ice slurry Reynolds number and the shell-side heat transfer coefficient has also been studied by varying the ice slurry mass flow between 825 $kg \cdot h^{-1}$ and 2425 $kg \cdot h^{-1}$ (0.85-2.55 $m \cdot s^{-1}$ approx.) and the propylene glycol mass flow between 650 $kg \cdot h^{-1}$ and 1650 $kg \cdot h^{-1}$ (0.6-1.5 $m \cdot s^{-1}$ approx.).

Table 1 summarizes the variation range for all variables analyzed.

2.1.- Pumping power.

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

$$P = \lambda \frac{\dot{m} L v^2}{2D} \tag{1}$$

where $\dot{m}$ is the mass flow rate in kg·s^{-1}, L is the pipe length in m, v is the flow velocity in m·s^{-1} and D is the pipe diameter in m.

Due to the ice melting, the friction factor value varies along the tube length and its value depends on the heat transfer rate. Therefore the friction factor has been obtained simultaneously to the heat transfer rate, using the correlations proposed by the authors (Illán and Viedma, 2009a, 2009b).

2.2.- Heat transfer rate.

To obtain the heat transfer rate, the whole heat exchanger was divided into 1 mm length elements. In each element, the differences between inlet and outlet properties are nearly imperceptible and therefore the heat transfer rate can be obtained from the following equation system:

$$\dot{q} = AU\Delta T_m = A\left[\frac{1}{A_{in}h_{in}} + \frac{ln\left({D_{out}}/{D_{in}}\right)}{2\pi k L} + \frac{1}{A_{out}h_{out}}\right]^{-1}\left(\overline{T_{pg}} - \overline{T_{is}}\right) \tag{2}$$

$$\dot{q} = \dot{m}_{is}\left[(\phi_{inlet} - \phi_{outlet})H_f + (1-\bar{\phi})\overline{c_{p,cf}}\left(T_{is,outlet} - T_{is,inlet}\right)\right] \tag{3}$$

$$\dot{q} = \dot{m}_{pg}\overline{c_{p,pg}}\left(T_{pg,in} - T_{pg,out}\right) \tag{4}$$

where the only unknown variables are the heat transfer rate, $\dot{q}$, the ice slurry outlet temperature, $T_{is,out}$ and the propylene glycol inlet temperature, $T_{pg,in}$. In these expressions, the mean values for the ice slurry temperature, $\overline{T_{is}}$, the propylene glycol temperature, $\overline{T_{pg}}$, the ice concentration, $\bar{\phi}$, the propylene glycol specific heat, $\overline{c_{p,pg}}$, and the carrier fluid specific heat, $\overline{c_{p,cf}}$, have been obtained as the average between inlet and outlet values.

The heat transfer coefficient in the tube inner wall, h_{in}, has been obtained as a function of the ice slurry mean temperature using the correlations proposed by the authors (Illán and Viedma, 2009a, 2009b).

The heat transfer coefficient in the tube outer wall, h_{out}, has been obtained as a function of the propylene glycol mean temperature, using the correction factors given by Petukhov and Roizen (Petukhov and Roizen, 1964) for turbulent flow in concentric annular ducts. Assuming that the outer wall is insulated, the heat transfer at the inner wall in concentric annuli may be calculated as:

$$\frac{Nu_i}{Nu_{tube}} = 0.86\left(\frac{D_i}{D_o}\right)^{-0.16} \quad (5)$$

where D_i is the diameter of the inner wall of the duct and D_o is the diameter of the outer wall.

3. – Results and discussion.

Equations (1) to (5) have been used to obtain the improvement ratio for smooth (IR_s) and corrugated (IR_c) heat exchangers, as well as heat transfer and pumping power ratios. Results are shown below.

3.1.- Smooth tube heat exchanger results.

Fig.2 shows results for the ice slurry improvement ratio (a), the heat transfer ratio (b) and the pumping power ratio (c) for an ice slurry mass flow rate of 1500 $kg{\cdot}h^{-1}$, and a propylene glycol mass flow rate of 1250 $kg{\cdot}h^{-1}$ at an outlet temperature of 0 ºC. As can be seen in Fig. 2.a the improvement ratio increases as heat exchanger length increases, although a critical length exists, around 1.5 m., below which the improvement ratio is lower than 1. According to Fig.2.b, the heat transfer ratio is greater than 1 in almost all cases, except for heat exchangers shorter than 1 meter at very low ice concentrations. This mean that the direct use of ice slurry increases in almost all cases the heat transfer rate. On the other hand, as Fig.2.c shows, the pumping power ratio is always greater than 1, which means that the direct use of ice slurry raises in all cases the pumping power consumption. In those cases where the increase in heat transfer rate compensates for the increase in pumping power consumption, the improvement ratio will be greater than 1 and therefore the direct use of ice slurry improves the performance of the heat exchanger. On the other hand, in those cases where the increase in heat transfer rate does not compensate for the increase in pumping power consumption the indirect use of ice slurry will be recommended over its direct use.

Finally, the ice content has only a slight influence on the heat exchanger performance. As the ice content increases, the heat transfer ratio and the pumping power ratio increase. Due to these two opposite effects, the improvement ratio remains nearly independent of ice content for ice concentrations below 10 %, increasing slightly as ice concentration increases for ice content over 10 %.

Fig. 3 is similar to Fig.2 although in this case the values for improvement, heat transfer and pumping power ratios have been obtained for a propylene glycol outlet temperature of 50 ºC. In this case the ice concentration has a stronger influence over the heat exchanger performance although its influence depends on the heat exchanger’s length.

For low lengths the heat exchanger's performance improves as ice content increases whereas for medium lengths the tendency is reversed and finally, for high lengths, the influence is reversed again and beyond a length of 7.88 meters, the heat exchanger's performance clearly improves as ice content increases. Additionally, the critical length value varies in this case between 3.61 m. (for 5 % ice content) and 6.01 m. (for 25 % ice content). The values obtained for intermediate propylene glycol outlet temperatures vary progressively between the two extreme situations plotted in Fig. 2 and Fig. 3.

Fig.4 shows the evolution of all the parameters that influence the heat exchanger performance. Results correspond to an ice slurry mass flow rate of 1500 $kg \cdot h^{-1}$ and 25% of ice content and a propylene glycol mass flow rate of 1250 $kg \cdot h^{-1}$ and an outlet temperature of 50 ºC. Although as can be seen in Fig.4.a, the convective coefficient for ice slurry direct application is in many cases lower than that obtained using indirect application, Fig.4.c shows that the heat transfer rate for ice slurry direct application is always greater than that obtained using indirect application, mainly due to the greater temperature difference obtained thanks to the high thermal capacity of ice slurry. As a result, Fig.3.b shows how the heat transfer ratio initially increases slightly while the ice is melting and the convective coefficient decreases for direct application but, once the ice is completely melted (the vertical dotted line in Figs.3.a,b&c marks the end of the melting process), the increase in the convective coefficient together with the high temperature difference improves the heat transfer rate for direct application and therefore increases the slope for the heat transfer ratio. On another note, as Fig.4.b shows, due to its lower density, fluid velocity for direct application is initially greater but, as ice melts, ice slurry density increases and finally, when all ice is melted, density for direct application is greater than for indirect application and hence velocity for direct application is lower. Something similar occurs with the friction factor and as a result, initially the pumping power consumption is greater for direct application but, as the length increases, this tendency is reversed and finally, for high lengths, pumping power consumption is greater using indirect application. This effect can be seen in Fig.3.c which shows an increase in the pumping power ratio for low heat exchanger lengths and a decrease for high lengths. Nevertheless, when comparing Fig.3 and Fig.4 it is necessary to take into account that the ratios plotted in Fig.3 have been obtained for accumulated values of heat transfer and pumping power whereas the data plotted in Fig.4 are instantaneous values.

Fig. 5 shows the influence that ice slurry mass flow rate and propylene glycol mass flow rate have over the improvement ratio. Two extreme situations are plotted in Fig.5.a. In a minimum mass flow rate situation ice slurry and propylene glycol mass flow rates were fixed to 825 and 650 $kg \cdot h^{-1}$ respectively whereas in a maximum mass flow rate situation propylene glycol and ice slurry mass flow rates were fixed at 2425 and 1650 $kg \cdot h^{-1}$. As can be seen, in both cases the improvement obtained using ice slurry is lower than that obtained for the base case (1500 and 1250 $kg \cdot h^{-1}$). According to Fig.5.b, for a constant ice slurry mass flow rate of 1500 $kg \cdot h^{-1}$, the improvement ratio increases as the propylene glycol mass flow rate increases although the increase obtained is less important as mass flow rate increases. Fig.5.c shows, for a constant propylene glycol mass flow rate of 1250 $kg \cdot h^{-1}$, the influence of the ice slurry mass flow rate over the improvement ratio. According to this figure, in almost all cases, the improvement obtained by the direct use of ice slurry increases as ice slurry mass flow decreases. The exception is the ice slurry mass flow rate of 825 $kg \cdot h^{-1}$. In this case, due to the low velocity, the fluid flows under laminar flow conditions and pumping power increases due to the high increase in the friction factor.

Fig.6 helps to better understand the results plotted in Fig.5. According to Fig.6.a, for low propylene glycol mass flow rates, $\dot{m}_{pg}$, the overall heat transfer coefficient based on the inside tube area, U_{in}, is mainly dominated by the shell-side convective heat transfer coefficient, h_{pg}. As $\dot{m}_{pg}$ increases, the increase of h_{pg} is less important and simultaneously, the influence that a variation of h_{pg} has over U_{in} values become less important (the influence of h_{pg} is similar to the influence of h_{is}) but, in any case, an increase in $\dot{m}_{pg}$ improves the heat exchanger's performance. On the other hand, according to Fig.6.b, an increase in the ice slurry mass flow rate, $\dot{m}_{is}$, has very low influence over U_{in} because it is mainly dominated by h_{pg}. Although the friction factor, λ_{is}, decreases as $\dot{m}_{is}$ increases, in general terms this decrease in λ_{is} is not enough to compensate for the effect that the increase in flow velocity has over pumping power consumption. Therefore as $\dot{m}_{is}$ increases, the pumping power increases whereas the heat transfer rate remains nearly constant and so heat exchanger's performance worsens. Only at low $\dot{m}_{is}$ values (laminar flow) the heat exchanger's performance improves as $\dot{m}_{is}$ increases. In this case, thanks to the strong decrease in λ_{is} as $\dot{m}_{is}$ increases, the increase in the heat transfer rate is stronger than the increase in the pumping power consumption and therefore the improvement ratio increases.

As a conclusion, in general terms the improvement obtained using ice slurry is higher for the lower ice slurry mass flow rate, the higher ice concentration, the lower outlet temperature for the cooled fluid, the higher cooled fluid mass flow rate and the higher heat exchanger length. Nevertheless, as the outlet temperature for the cooled fluid 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.

Table 2 shows the absolute values of pressure drop and heat transfer rate obtained for direct and indirect application of ice slurry in cooling propylene glycol until 0 ºC, 25 ºC and 50 ºC. According to these values, 1500 $kg \cdot h^{-1}$ of ice slurry directly used in a 10 meter long heat exchanger allow a reduction in the temperature of 1250 $kg \cdot h^{-1}$ of propylene glycol from 6 to 0 ºC ($\dot{q} = 8.67\ kW$), from 65.3 to 25 ºC ($\dot{q} = 58.07\ kW$), or from 126 to 50 ºC ($\dot{q} = 111.84\ kW$), whereas if only the liquid phase is used as refrigerant fluid, the temperature of the propylene glycol flow only can be reduced from 4.5 to 0 ºC ($\dot{q} = 6.5\ kW$), from 56 to 25 ºC ($\dot{q} = 44.5\ kW$), or from 114 to 50 ºC ($\dot{q} = 94\ kW$).

3.2.- Corrugated tube heat exchanger results.

Results of the improvement ratio obtained in corrugated tube heat exchangers are shown in Fig. 7 and Fig. 8 for a propylene glycol outlet temperature of 0 ºC and 50 ºC respectively. According to Fig. 7.c the pumping power ratio increases as ice content increases whereas, according to Fig. 7.b, for heat exchanger length below 8 meters the heat transfer ratio increases as ice content decreases. Therefore, as Fig. 7.a shows, for heat exchanger lengths below 8 meters the improvement ratio clearly increases as the ice content decreases. Finally, whereas for 5 % ice content the improvement ratio remains over 1 for any heat exchanger length, for 25 % ice content the improvement ratio is below 1 for heat exchanger lengths below 2.4 meters. Fig. 8 shows results for a propylene glycol outlet temperature of 50 ºC. Results are qualitatively similar to those shown in Fig. 7, although the inflection point observed in Fig. 7 for 5 % ice content and 8 meters length heat exchanger appears in this case for all ice contents and therefore, for heat exchanger lengths over 6.16 meters, the improvement ratio increases as ice content increases.

Similarly to Table 2, Table 3 summarizes the absolute values of pressure drop and heat transfer rate obtained for direct and indirect application of ice slurry in a corrugated tube heat exchanger of length between 1 and 10 meters.

Fig. 9 shows the influence that ice slurry mass flow rate and propylene glycol mass flow rate have over the improvement ratio. In this case, the influence is similar to that observed in Fig. 5 for smooth tube and the same conclusions are valid for corrugated tube.

4. - Conclusions.

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

The optimal ice concentration depends on specific operation conditions and the heat exchanger type (smooth or corrugated tube). As a general rule, the optimal ice concentration increases as the heat exchanger length increases. Although in most cases the direct use of ice slurry improves the heat exchanger's performance, there are some cases, especially for low heat exchanger length, where the direct use of ice slurry is inadvisable. A careful analysis is strongly recommended before deciding about the use of ice slurry.

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Table 01

Tube type	Tube length (m)	Ice concentration (wt%)	Ice slurry mass flow rate ($kg \cdot s^{-1}$)	Propylene glycol outlet temperature (ºC)	Propylene glycol mass flow rate ($kg \cdot s^{-1}$)
Smooth	0 ÷ 10 (ΔL=1 mm)	5, 10, 15, 20 & 25	825, 1225, 1625, 2025 & 2425	0, 25 & 50	650, 900, 1150, 1400 & 1650
Corrugated	0 ÷ 10 (ΔL=1 mm)	5, 10, 15, 20 & 25	825, 1225, 1625, 2025 & 2425	0, 25 & 50	650, 900, 1150, 1400 & 1650

Table 1. Variation range of all variables analyzed.

Table 02

L (m)	$T_{pg,out} = 0\ °C$				$T_{pg,out} = 25\ °C$				$T_{pg,out} = 50\ °C$			
	$\dot{q}_{is}$ (kW)	Δp_{is} (kPa)	$\dot{q}_{cf}$ (kW)	Δp_{cf} (kPa)	$\dot{q}_{is}$ (kW)	Δp_{is} (kPa)	$\dot{q}_{cf}$ (kW)	Δp_{cf} (kPa)	$\dot{q}_{is}$ (kW)	Δp_{is} (kPa)	$\dot{q}_{cf}$ (kW)	Δp_{cf} (kPa)
1	0.64	2.20	0.60	2.13	3.89	2.19	3.62	2.11	7.75	2.19	7.27	2.09
2	1.32	4.39	1.22	4.25	8.11	4.37	7.40	4.18	16.08	4.35	15.04	4.10
3	2.04	6.59	1.84	6.36	12.69	6.54	11.36	6.21	25.01	6.50	23.32	6.04
4	2.82	8.78	2.47	8.47	17.66	8.70	15.50	8.20	34.60	8.62	32.10	7.90
5	3.64	10.97	3.11	10.57	23.05	10.84	19.83	10.15	45.28	10.68	41.35	9.70
6	4.51	13.15	3.77	12.66	28.88	12.97	24.36	12.06	56.92	12.64	51.06	11.44
7	5.45	15.34	4.43	14.74	35.23	15.09	29.09	13.93	69.43	14.50	61.19	13.13
8	6.45	17.52	5.11	16.82	42.31	17.17	34.02	15.76	82.81	16.27	71.72	14.77
9	7.52	19.70	5.80	18.89	49.95	19.17	39.15	17.56	97.01	17.95	82.67	16.37
10	8.67	21.87	6.50	20.95	58.07	21.11	44.50	19.33	111.84	19.58	94.03	17.93

Table 2. Absolute values of heat transfer rate and pressure drop for 25 % ice content ice slurry direct and indirect application in smooth tube heat exchangers.

Table 03

L (m)	$T_{pg,out} = 0\ ^oC$				$T_{pg,out} = 25\ ^oC$				$T_{pg,out} = 50\ ^oC$			
	$\dot{q}_{is}$ (kW)	Δp_{is} (kPa)	$\dot{q}_{cf}$ (kW)	Δp_{cf} (kPa)	$\dot{q}_{is}$ (kW)	Δp_{is} (kPa)	$\dot{q}_{cf}$ (kW)	Δp_{cf} (kPa)	$\dot{q}_{is}$ (kW)	Δp_{is} (kPa)	$\dot{q}_{cf}$ (kW)	Δp_{cf} (kPa)
1	0.98	4.12	1.00	4.08	5.44	4.11	5.56	4.05	10.57	4.09	10.87	4.02
2	2.06	8.23	2.02	8.15	11.68	8.18	11.43	8.03	22.81	8.13	22.54	7.90
3	3.24	12.34	3.06	12.21	18.82	12.22	17.60	11.93	36.91	12.10	34.97	11.65
4	4.55	16.44	4.12	16.25	27.00	16.23	24.10	15.77	53.07	16.05	48.10	15.29
5	5.99	20.53	5.21	20.28	36.43	20.18	30.91	19.53	70.55	19.81	61.86	18.83
6	7.59	24.62	6.33	24.29	47.19	24.16	38.05	23.22	89.23	23.42	76.22	22.29
7	9.36	28.69	7.47	28.29	58.75	28.02	45.51	26.86	108.97	26.90	91.16	25.68
8	11.32	32.76	8.63	32.27	71.04	31.76	53.29	30.44	129.58	30.29	106.66	29.01
9	13.52	36.81	9.82	36.24	84.00	35.38	61.37	33.97	150.99	33.60	122.67	32.28
10	15.97	40.86	11.04	40.19	97.57	38.91	69.73	37.45	173.12	36.82	139.16	35.50

Table 3. Absolute values of heat transfer rate and pressure drop for 25 % ice content ice slurry direct and indirect application in corrugated tube heat exchangers.

Figure 01

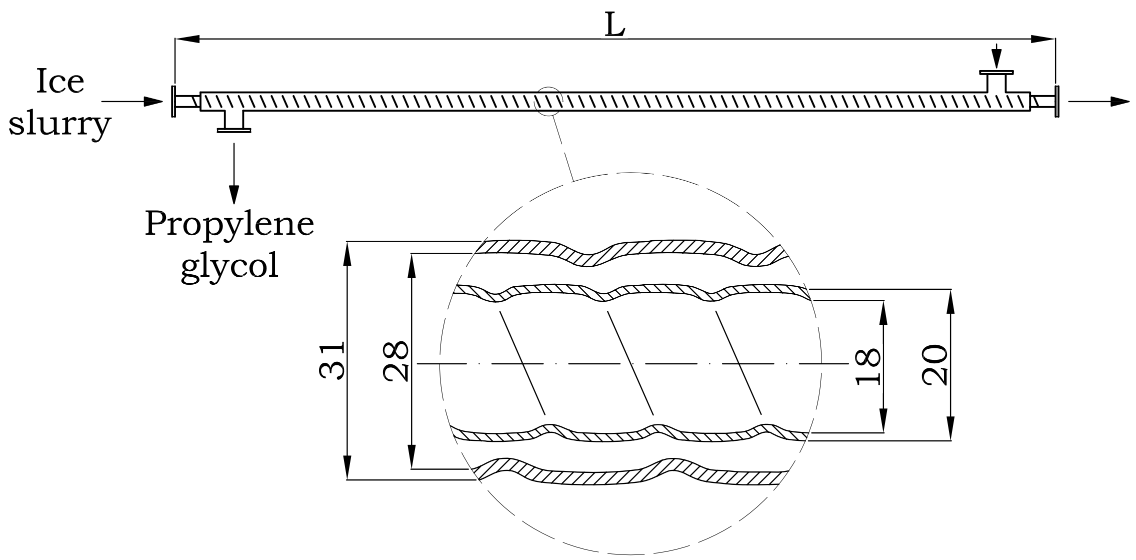
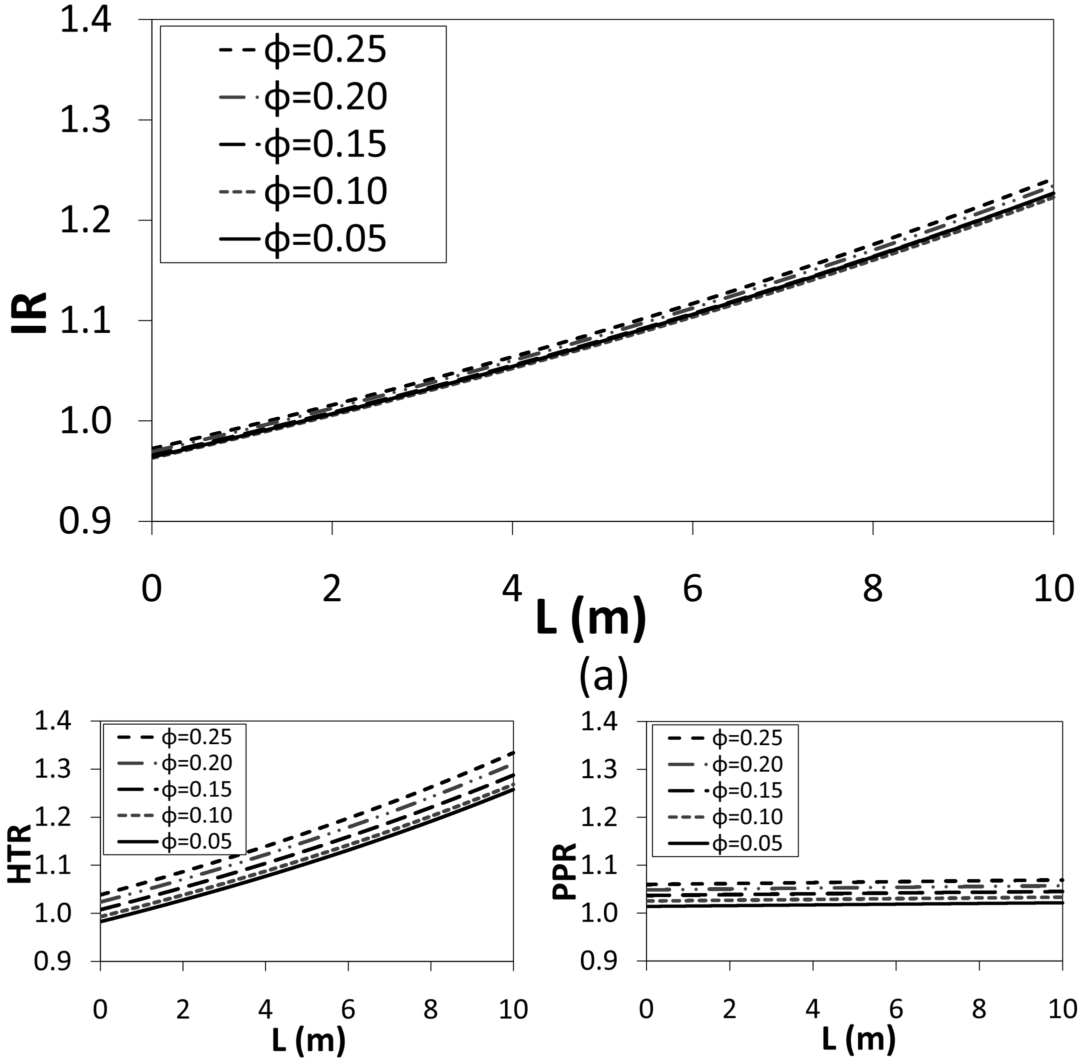


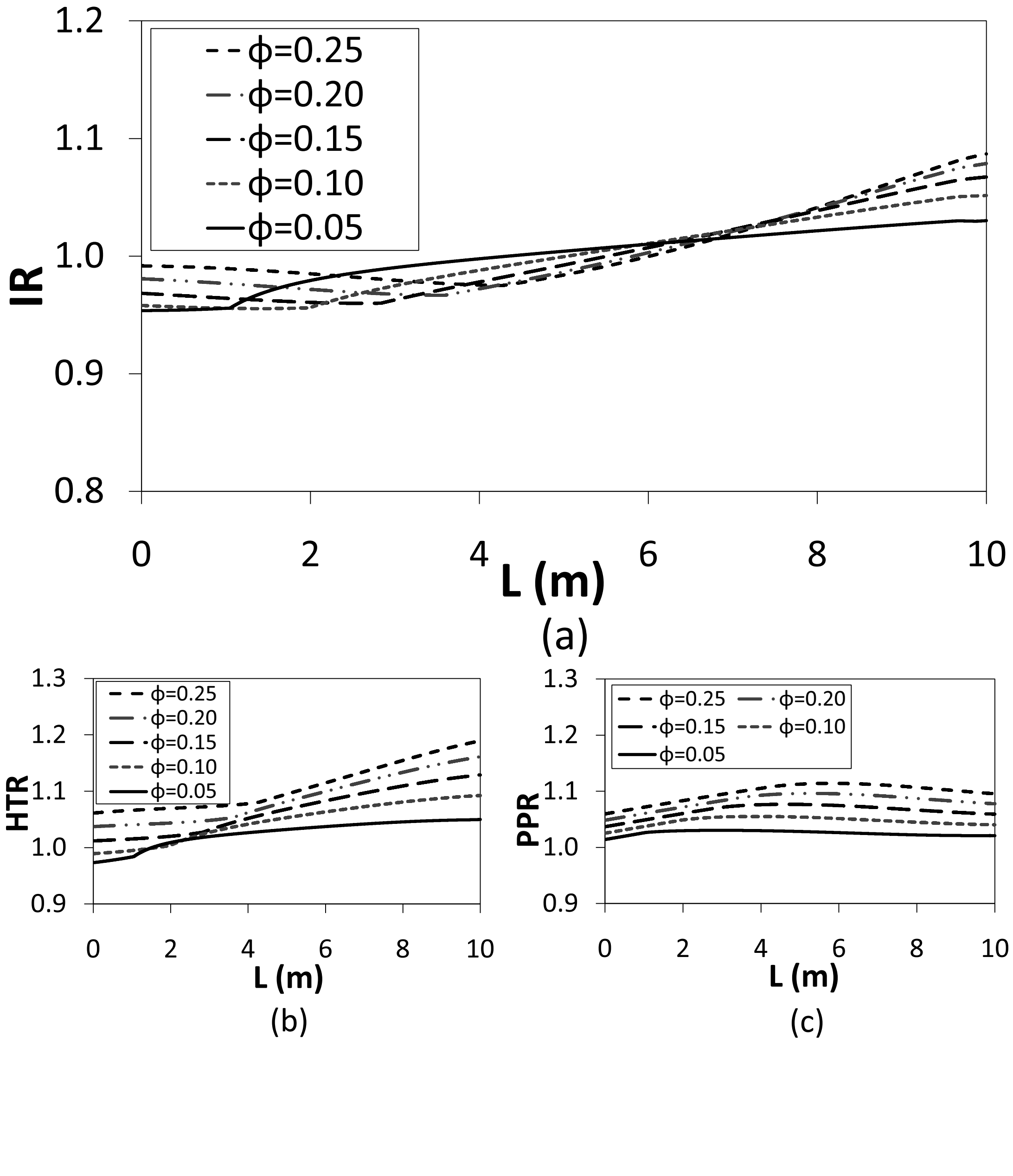
Figure 02



(a)

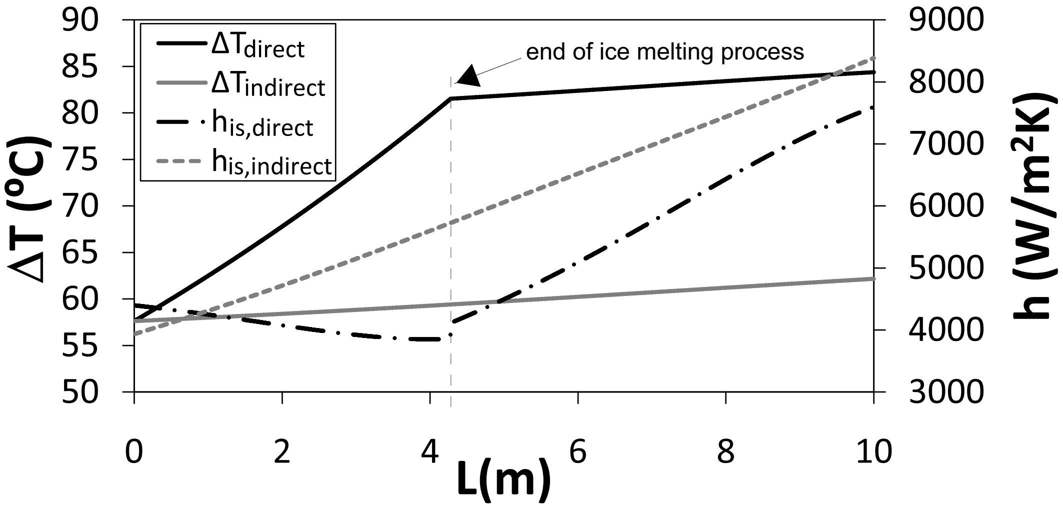
(b) (c)

Figure 03

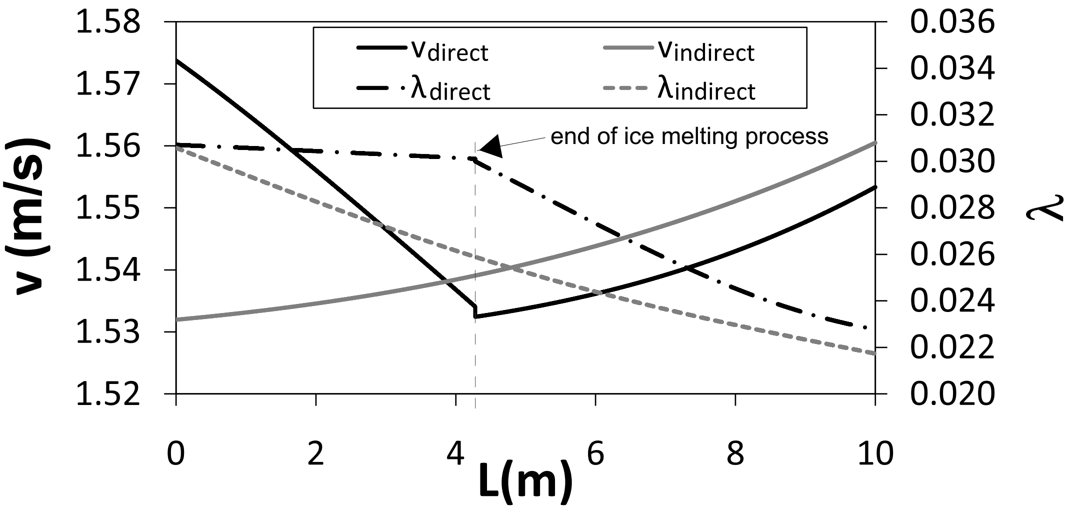


1.2
1.1
1.0
0.9
0.8
IR
0
2
4
6
8
10
L (m)
ϕ=0.25
ϕ=0.20
ϕ=0.15
ϕ=0.10
ϕ=0.05
(a)
1.3
1.2
1.1
1.0
0.9
HTR
L (m)
(b)
PPR
L (m)
(c)

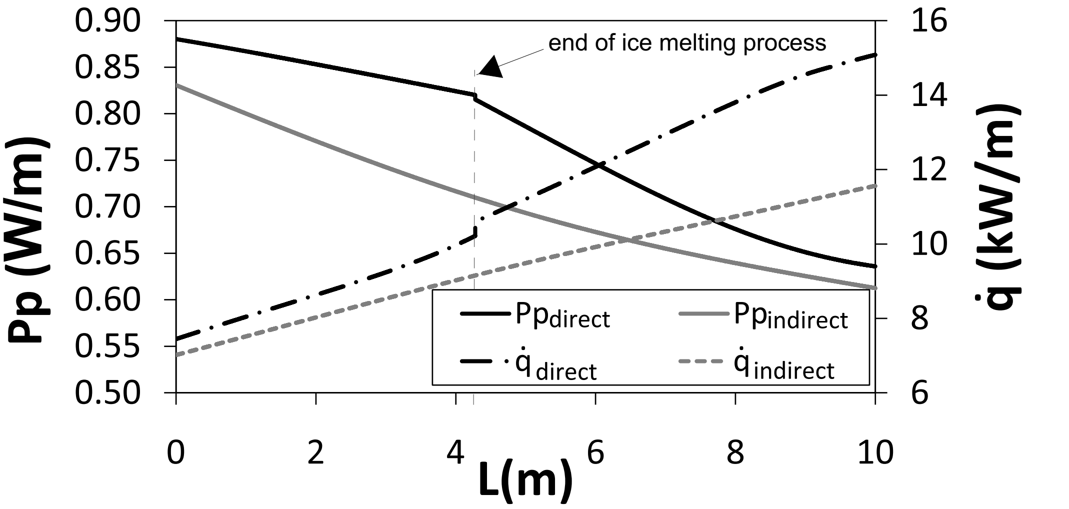
Figure 04



(a)

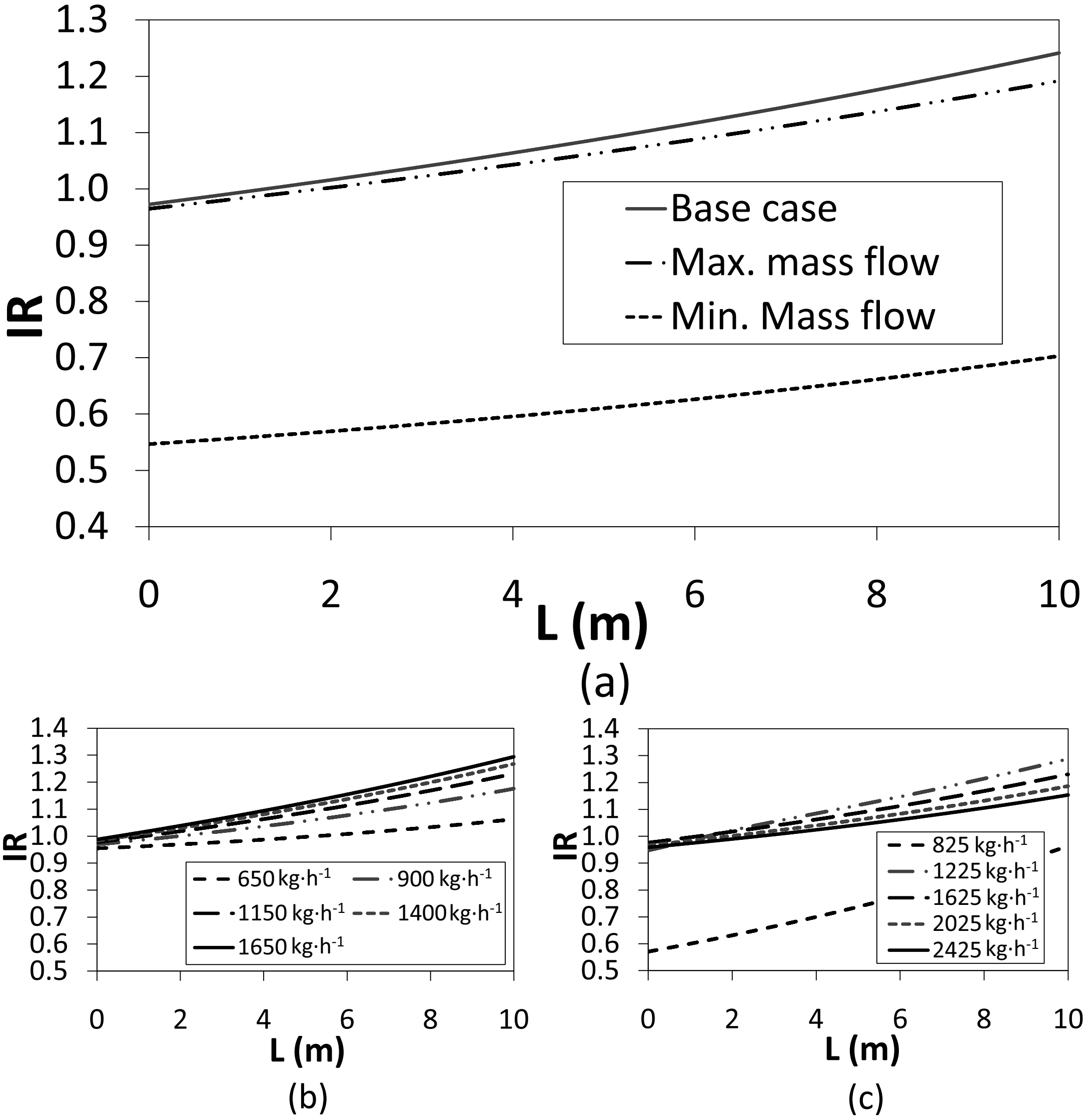


(b)



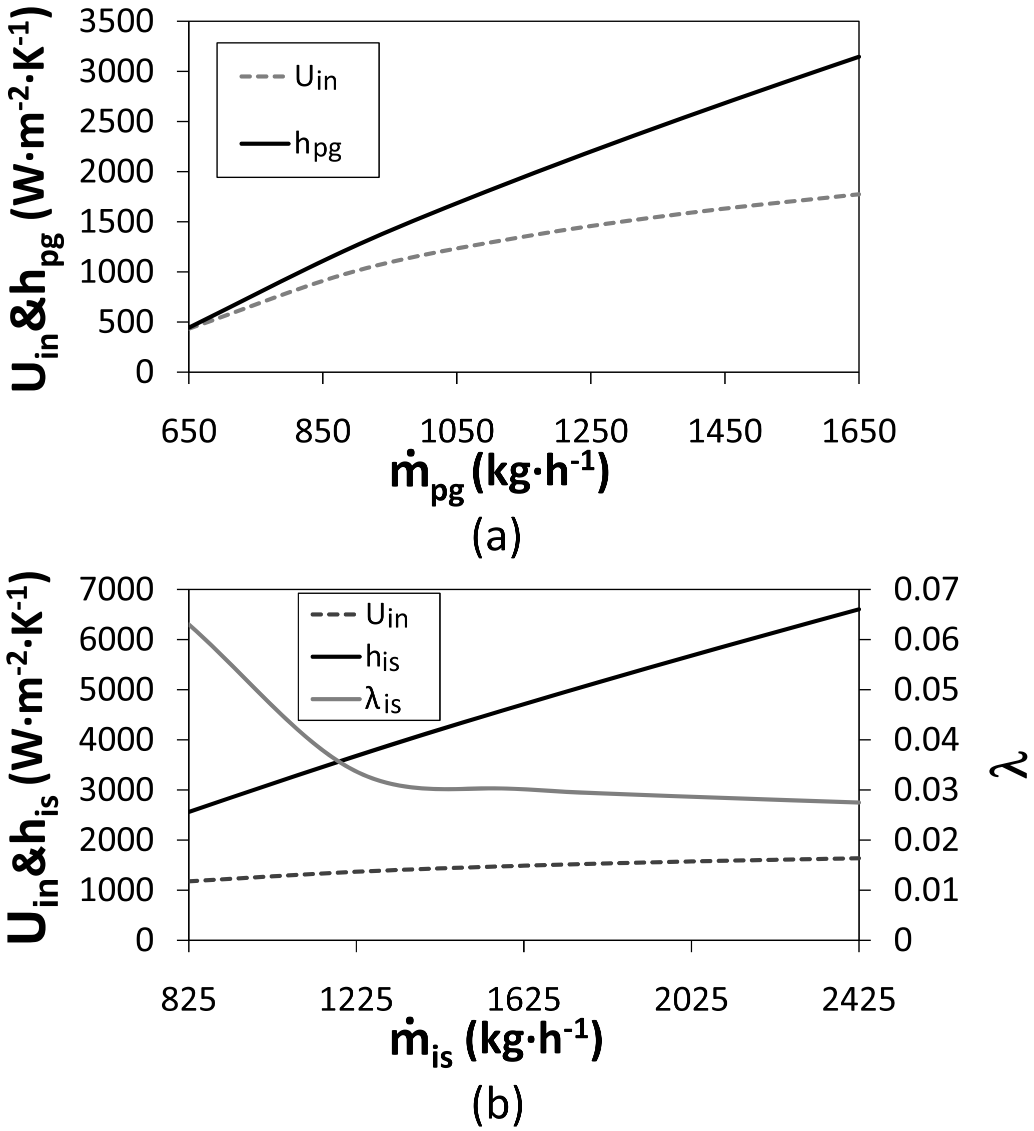
(c)

Figure 05



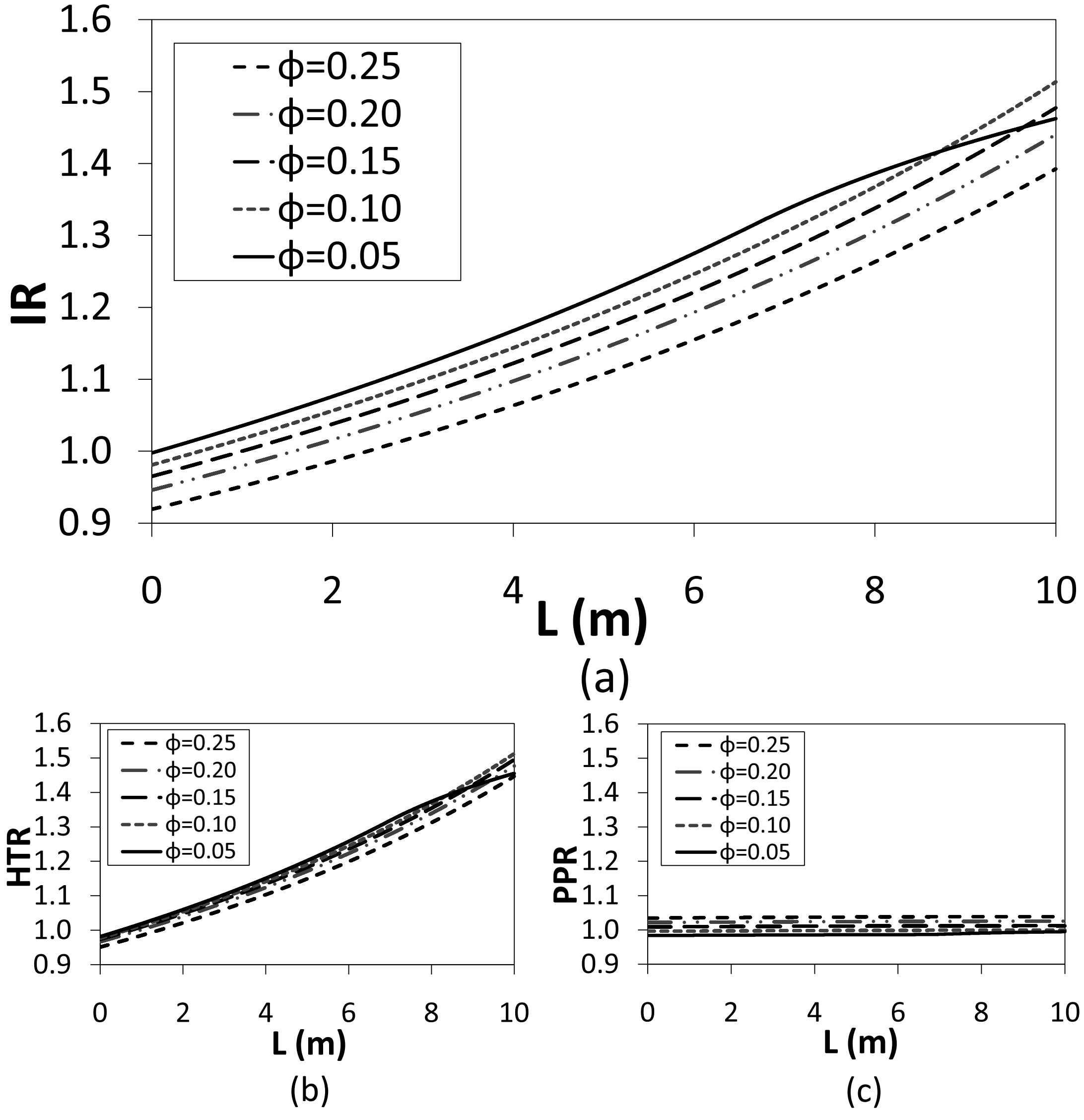
1.3
1.2
1.1
1.0
0.9
0.8
0.7
0.6
0.5
0.4
IR
0
2
4
6
8
10
L (m)
Base case
Max. mass flow
Min. Mass flow
(a)
1.4
IR
L (m)
650 kg·h-1
900 kg·h-1
1150 kg·h-1
1400 kg·h-1
1650 kg·h-1
(b)
825 kg·h-1
1225 kg·h-1
1625 kg·h-1
2025 kg·h-1
2425 kg·h-1
(c)

Figure 06



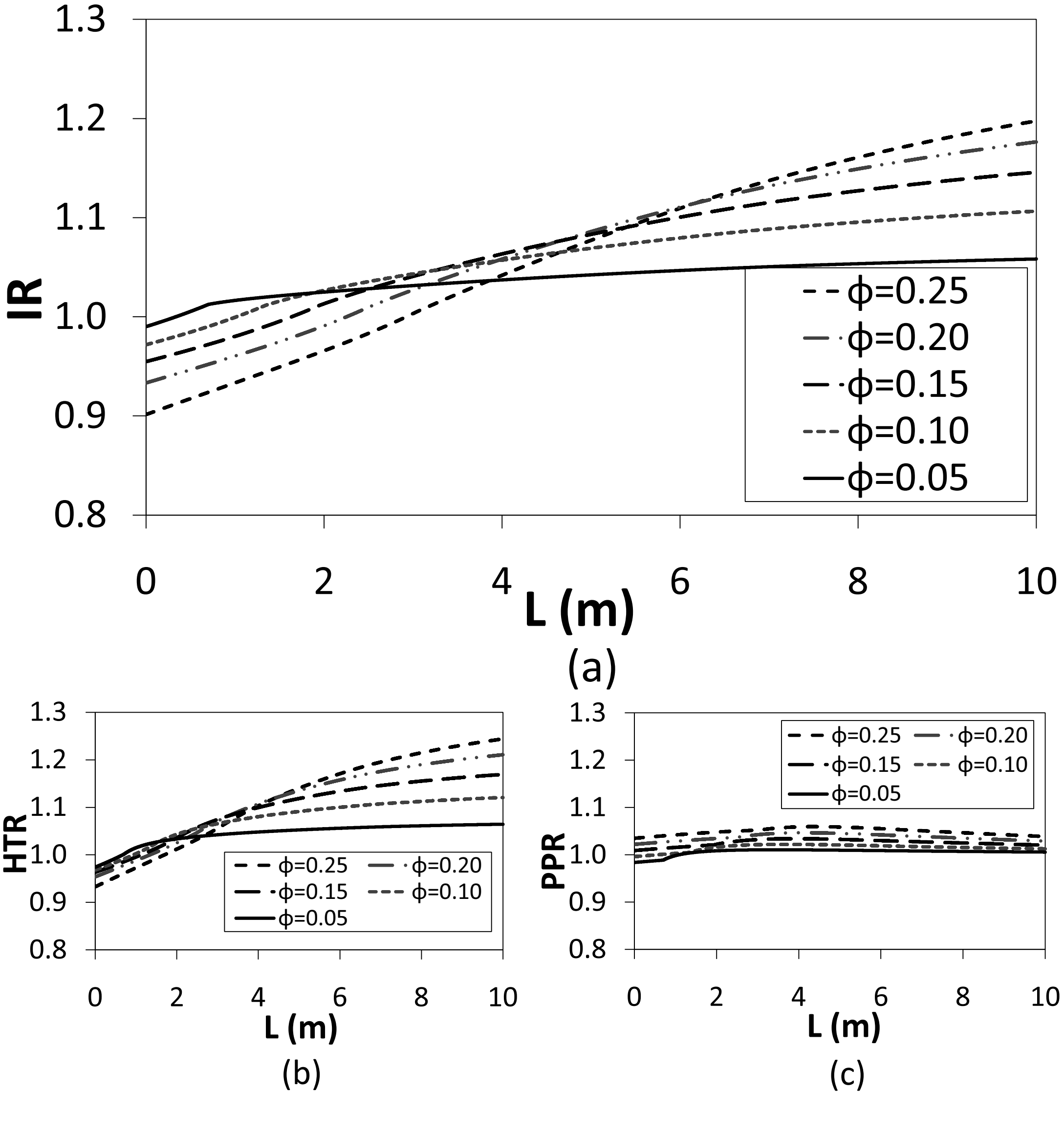
3500
3000
2500
2000
1500
1000
500
0
Uin
hpg
U_{in}&h_{pg} (W·m^{-2}·K^{-1})
650
850
1050
1250
1450
1650
$\dot{m}_{pg}$ (kg·h^{-1})
(a)
7000
6000
5000
4000
3000
2000
1000
0
0.07
0.06
0.05
0.04
0.03
0.02
0.01
0
Uin
his
λis
U_{in}&h_{is} (W·m^{-2}·K^{-1})
λ
825
1225
1625
2025
2425
$\dot{m}_{is}$ (kg·h^{-1})
(b)

Figure 07



1.6
1.5
1.4
1.3
1.2
1.1
1.0
0.9
IR
ϕ=0.25
ϕ=0.20
ϕ=0.15
ϕ=0.10
ϕ=0.05
0
2
4
6
8
10
L (m)
(a)
HTR
L (m)
(b)
PPR
L (m)
(c)

Figure 08



(b)
(c)

Figure 09

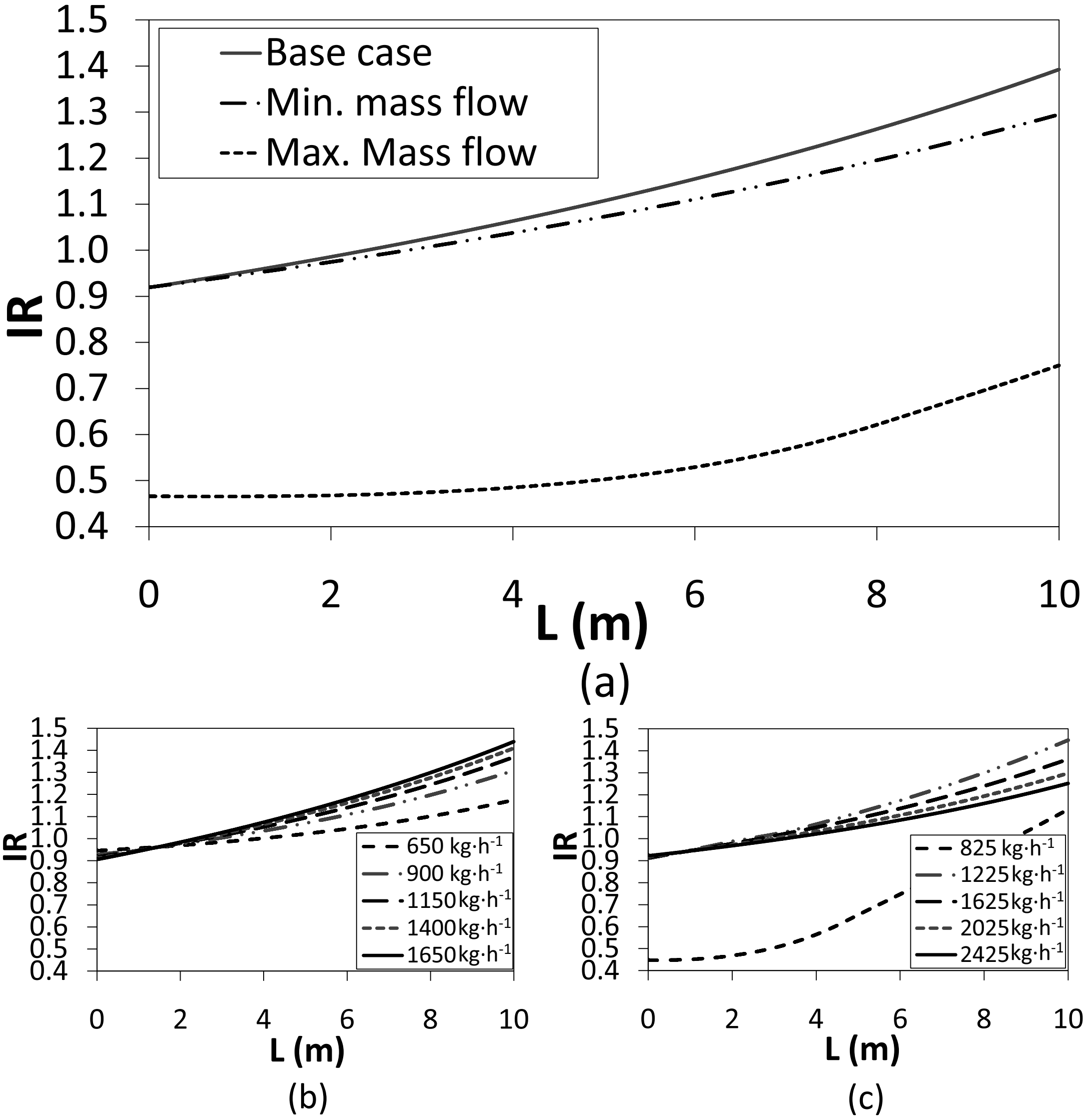

Base case
Min. mass flow
Max. Mass flow
IR
L (m)
(a)
650 kg·h⁻¹
900 kg·h⁻¹
1150kg·h⁻¹
1400kg·h⁻¹
1650kg·h⁻¹
(b)
825 kg·h⁻¹
1225kg·h⁻¹
1625kg·h⁻¹
2025kg·h⁻¹
2425kg·h⁻¹
(c)

Figure 1. Sketch of the heat exchanger analyzed.

Figure 2. Improvement ratio (a), heat transfer ratio (b) and pumping power ratio (c) for smooth tube heat exchangers with an ice slurry mass flow rate of 1500 $kg \cdot h^{-1}$ and a propylene glycol mass flow rate of 1250 $kg \cdot h^{-1}$ and outlet temperature of 0 ºC.

Figure 3. Improvement ratio (a), heat transfer ratio (b) and pumping power ratio (c) for smooth tube heat exchangers with an ice slurry mass flow rate of 1500 $kg \cdot h^{-1}$ and a propylene glycol mass flow rate of 1250 $kg \cdot h^{-1}$ and outlet temperature of 50 ºC.

Figure 4. Convective coefficient and temperature difference (a), flow velocity and Darcy friction factor (b) and pumping power and heat flow per unit length (c) for smooth tube heat exchangers with an ice slurry mass flow rate of 1500 $kg \cdot h^{-1}$ and 25 % ice content and a propylene glycol mass flow rate of 1250 $kg \cdot h^{-1}$ and outlet temperature of 50 ºC.

Figure 5. Improvement ratio for maximum, minimum and standard mass flow rate conditions (a). Influence of propylene glycol mass flow rate (b) and ice slurry mass flow rate (c) over heat exchanger improvement ratio for smooth tube heat exchangers with 25 % ice content and propylene glycol outlet temperature of 0ºC.

Figure 6. Influence of propylene glycol mass flow rate (a) and ice slurry mass flow rate (b) over the heat exchanger’s overall heat transfer coefficient for smooth tube heat exchangers with 25 % ice content and propylene glycol outlet temperature of 0ºC.

Figure 7. Improvement ratio (a), heat transfer ratio (b) and pumping power ratio (c) for corrugated tube heat exchangers with an ice slurry mass flow rate of 1500 $kg \cdot h^{-1}$ and a propylene glycol mass flow rate of 1250 $kg \cdot h^{-1}$ and outlet temperature of 0 ºC.

Figure 8. Improvement ratio (a), heat transfer ratio (b) and pumping power ratio (c) for corrugated tube heat exchangers with an ice slurry mass flow rate of 1500 $kg \cdot h^{-1}$ and a propylene glycol mass flow rate of 1250 $kg \cdot h^{-1}$ and outlet temperature of 50 ºC.

Figure 9. Improvement ratio for maximum, minimum and standard mass flow rate conditions (a). Influence of propylene glycol mass flow rate (b) and ice slurry mass flow rate (c) over heat exchanger improvement ratio for corrugated tube heat exchangers with 25 % ice content and propylene glycol outlet temperature of 0ºC.