HEFAT2010
7th International Conference on Heat Transfer, Fluid Mechanics and Thermodynamics
19-21 July 2010
Antalya, Turkey

EXPERIMENTAL ANALYSIS OF A TITANIUM PLATE HEAT EXCHANGER WITH ICE SLURRY AS COOLING MEDIUM

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ABSTRACT

The experimental analysis of a titanium plate heat exchanger using ice slurry as cooling medium has been carried out at our laboratory. The ice slurry was produced from an ethylene glycol 10 wt% aqueous solution. The pressure drop and heat transfer performance has been measured in an experimental test ring. In the paper, the characteristics of the plate heat exchanger and the experimental facility are described in detail. The heat transfer performance of the heat exchanger has been determined for different ice slurry flow rates and different ice fractions. The pressure drop measurements are presents as a function of mass flow rate and ice concentration. Moreover data of heat transfer rate, overall heat transfer coefficient and pressure drop operating with water and with ethylene glycol solutions are presented. The experimental results are discussed in the paper.

INTRODUCTION

Plate heat exchangers are used in many industrial, commercial and domestic applications. This type of heat exchangers provides significant advantages over other types, such as, high efficiency, compactness, flexibility and a competitive cost. The most widespread applications of plate heat exchangers are related to the heat exchange between single-phase fluids and as condensers or evaporators in refrigeration or heat pump systems.

Recently, plate heat exchangers are beginning to be used in indirect cooling systems with slurry ice as cooling medium. The slurry ice is a suspension of microcrystals of ice within a solution of water and an antifreeze substance. The high heat transport capability of the ice slurry and its low operating temperatures makes it an excellent alternative to conventional single-phase coolants in indirect refrigeration systems. Moreover, for moderate ice concentrations is not necessary to use special systems for pumping and distribution. Due to the small size of the channels in the plate heat exchangers, the presence of ice micro-crystals could become a problem. However, several experimental works show that this type of heat exchangers is being applied successfully in ice slurries systems with low or moderate ice fractions [1-5].

Generally, offset strip fins are used to enhance heat transfer in plate-and-fin heat exchangers. The thermal enhancement provided by offset strip fins is based on repeated growth and wake destruction of boundary layers [6]. However, it is worth pointing out that this type of fins also increases the pressure drop within the heat exchanger. Plate-and-fin heat exchangers with offset strip fins using air as heat transfer fluid on the fins side have been studied by many researchers [7].

The offset strip fins geometry is characterized by the fin height (h), transverse spacing (s), thickness (s) and length (L). Moreover, it is necessary to take into account that manufacturing irregularities, such as burred edges, bonding imperfections, and separating plate roughness, can influence the flow and heat transfer characteristics in the heat exchangers [8].

On the other hand, offset strip fins are not commonly applied in plate heat exchangers with liquids as working fluids. Therefore, experimental data about its pressure drop and heat transfer characteristics is scarce. Moreover, we did not find any data in open literature about the use of plate heat exchangers with offset strip fins to exchange heat with ice slurry. Nowadays, plate heat exchangers made of titanium with inner offset strip fins are being marketed by a Japanese company in the European market. Taking all this into account, the experimental analysis of a titanium plate heat exchanger using ice slurry as cooling medium has been carried out at our laboratory.

In the paper, the experimental setup and procedure are described, the reduction data processes are detailed, and the experimental results are presented and discussed.

NOMENCLATURE

A	[m^2]	Exchange surface
F	[-]	Correction factor
h	[kJ/kg]	Enthalpy
$\dot{m}$	[kg/h]	Mass flow rate
q	[kW]	Heat transfer rate
T	[K]	Temperature
U	[$W/m^2 \cdot K$]	Overall heat transfer coefficient
ΔT	[K]	Temperature difference

Special characters

$\varnothing$	[kg/kg]	Mass ice fraction
ρ	[kg/m^3]	Density

Subscripts

cf	Carrier fluid
c	Cold fluid
h	Hot fluid
i	Inlet
is	Ice slurry
ic	Ice
o	Outlet
$LMTD$	Logarithmic mean temperature difference

Table 1. Characteristics and geometry of plates and fins.

Model	TBHE-TiM-07-BH
Plate material	Titanium
Type	Brazed
Number of channels hot stream	4
Number of channels cold stream	3
Total heat transfer area (m^2)	0.128
Plate length (mm)	310
Plate width (mm)	90
Plate thickness (mm)	0.4
Weight (kg)	1,9
Inlet/Outlet ports	3/4"
Nominal flow rate (l/h)	300
OSF length (mm)	3
OSF height (mm)	2.7
OSF transverse spacing (mm)	3.9
OSF thickness (mm)	0.3

EXPERIMENTAL SETUP

Figure 1 shows two photographs of the plate heat exchanger used to carry out the experimental analysis. This heat exchanger is a prototype specifically manufactured to develop this work. The prototype is made of smooth titanium plates with offset strip fins in between. The characteristics and geometry of the plates and fins are given in Table 1. The geometrical configuration of the offset strip fins is detailed in Figure 2.

Figure 1 Photographs of the titanium plate heat exchanger.

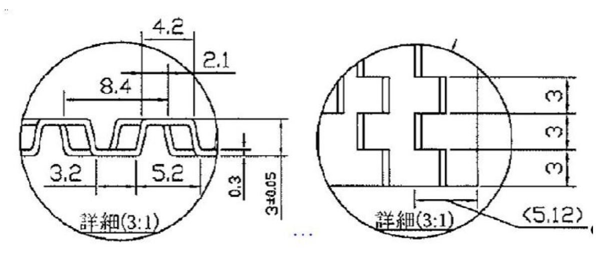

Figure 2 Geometrical configuration of the offset strip fins.

The experimental facility used to carry out this work was designed and built at our laboratory. Figure 3 shows the layout of the experimental setup.

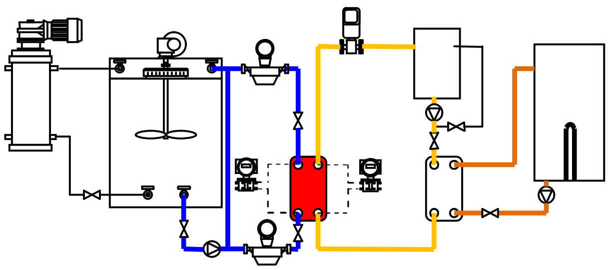

Figure 3 Schematic diagram of the experimental setup.

The main circuits of the experimental setup are the ice slurry loop and the hot single phase fluid loop. The ice slurry is pumped from a storage tank (500 litres capacity) through the tested heat exchanger. The single-phase fluid (water or glycol) is also pumped from a reservoir tank (20 litres capacity) through an auxiliary plate heat exchanger and the tested prototype, as shown in Figure 3. The auxiliary heat exchanger in the single-phase circuit allows the control of the fluid temperature at the prototype inlet by modifying the temperature of the hot water pumped from an electric heater, as can be seen in Figure 3. The electric heater is composed of a storage tank with three immersed electric resistances. One of the electric resistances is equipped with an adjustable input power. The overall heating power is 8 kW.

The ice slurry was generated by a standard ice slurry system equipped with a scraped surface type generator. The ice slurry produced was stored in an insulated tank with an inner capacity of 500 litres, as pointed out above. The ice slurry was pumped through the tested prototype by means of a centrifugal pump. The ice fraction in the storage tank was maintained constant in the desired value for each experiment by means of an on/off control strategy implemented in the ice generator control system. The ice slurry storage tank is equipped with a mixing device to keep the ice slurry homogeneous. The content of the tank was stirred continuously for the duration of the experiments. Ice slurry was produced from a 10 wt% ethylene glycol aqueous solution.

The experimental setup was equipped with a data acquisition system based on a PC. The mass flow rate and density of the ice slurry were measured at the tested heat exchanger inlet and outlet using two Coriolis flow meters (ELITE, Micro Motion F025S with a 2700 Micro Motion Transmiter) with an accuracy of ±0.1% of the measured value for the mass flow rate and ±0.5 kg/m^3 for the density. The measured values of the density were used to determine the ice fraction in the ice slurry at the inlet and outlet of the prototype.

The volumetric flow rate of the hot fluid circulating through the tested heat exchanger was measured by means of an electromagnetic flow meter with a standard uncertainty of ±0.25% of the actual flow rate.

The inlet and outlet temperatures of both the ice slurry and hot single-phase fluid streams were measured with type A Pt100 sensors at the heat exchanger ports. The temperature sensors were inserted in 50 mm long and 3 mm diameter stainless steel pockets. After calibration, the accuracy was within ±0.3 ºC for absolute temperatures and within ±0.1 ºC for temperature differences. The hot water temperature at the electric heater outlet was also measured by a calibrated type A Pt100 sensor inserted in a 50 mm long and 3 mm diameter stainless steel pocket.

The pressure drop on the ice slurry side was measured by a differential pressure transmitter (Siemens, Sitrans P series DS III) with an accuracy of ±0.075% of the measured value.

The heat transfer with the ambient was neglected due to the tested heat exchanger and its headers were properly insulated to reduce heat transfer to the surroundings.

EXPERIMENTAL PROCEDURE AND DATA REDUCTION

Experimental procedure

Previously to the analysis with ice slurry, several experiments using single-phase fluids were conducted to determine the pressure drop in the tested prototype and its heat transfer performance. Firstly, tests were carried out using water as cold and hot fluid. Then, new experiments were conducted circulating water through one side of the heat exchanger and an 30 wt% ethylene glycol aqueous solution through the other side. Later, new experiments using 10 wt% and 30 wt% ethylene glycol solutions were conducted. Finally, the experiments with ice slurry and with the 30 wt% ethylene glycol solution were carried out. In these experiments, the ice slurry was specifically generated for each experiment and all experiments were conducted immediately after its production with the aim of preventing any changes in properties due its time-dependant behaviour.

Experiments with single-phase fluids were conducted by varying each one of the flow rates keeping the other at the nominal value provided by the heat exchanger manufacturer. In these tests the inlet temperature of the cold fluid was kept constant at 10 ºC and the inlet temperature of the hot fluid were kept constant at 15, 20 and 25 ºC. Consequently, the temperature differences between the inlet temperatures of both fluids were kept constant at 5, 10 and 15 ºC.

The experiments with ice slurry were carried out considering the same mass flow rate as for the single-phase experiments. The inlet ice fraction was varied between 5 wt% and 25 wt%. The inlet temperature of the ethylene glycol solution was adjusted in each case on the way that the temperature difference between the inlet temperatures of hot and cold flows were kept constant and equal to 5, 10 and 15ºC, as in the single-phase experiments.

The experimental data were recorded for at least 15 minutes for each testing condition once all parameter were stable. The time interval between two measurements was 20 seconds. The values shown in this paper are the average values of the recorded data.

Data reduction

The heat transfer rate was determined from the energy balances on both fluid streams circulating through the heat exchanger, according to equation 1.

$$q = \dot{m} \cdot (h_o - h_i) \tag{1}$$

The specific enthalpy of water at the inlet and outlet of the heat exchanger was obtained from REFPROP 8.0 [9] as a function of the experimental temperature measurements. The thermo physical properties of the 10 wt% and 30 wt% ethylene glycol solutions used in the single-phase tests were obtained from ASRHAE Fundamentals [10]. To obtain the properties of the ice slurry is necessary to know the ice fraction. In this work the ice fraction at the inlet and outlet of the heat exchanger was obtained from density measurements using two Coriolis mass flow meters, according to equation 2.

$$\phi = \frac{\rho_{ic} \cdot (\rho_{cf} - \rho_{is})}{\rho_{is} \cdot (\rho_{cf} - \rho_{ic})} \tag{2}$$

Once the ice fraction is known, the specific enthalpy of the ice slurry was determined from equation 3.

$$h_{is} = \phi \cdot h_{ic} + (1 - \phi) \cdot h_{cf} \tag{3}$$

In equations 2 and 3, the ice properties were obtained according to Melinder [11] and the properties of the aqueous solutions according to Lugo [12]. The actual heat transfer rate in the experiments with single-phase fluids was calculated as the average value between the heat transfer rates determined from the energy balances on the hot and cold streams. In the experiments with ice slurry, the actual heat transfer rate has been determined from the energy balance on the glycol stream.

The overall thermal resistance was determined from equation 4, where F is equal to one for being a simple parallel counter flow configuration and the logarithmic mean temperature difference was obtained from equation 5.

$$q = A \cdot U \cdot F \cdot \Delta T_{LMTD} \qquad (4)$$

$$\Delta T_{LMTD} = \frac{(T_{ci} - T_{ho}) - (T_{co} - T_{hi})}{ln\left(\dfrac{T_{ci} - T_{ho}}{T_{co} - T_{hi}}\right)} \qquad (5)$$

RESULTS AND DISCUSSION

Single phase fluids

As mentioned above, the plate heat exchanger was initially tested considering heat transfer between single-phase fluids. Experiments were conducted with three different fluids combinations: water to water, water to 10 wt% ethylene glycol solution and 30 wt% ethylene glycol solution to 10% ethylene glycol solution. This section shows the main experimental results obtained for single-phase heat transfer experiments.

The heat transfer rate was determined by applying energy balances to the hot and cold streams. The values of the heat transfer rate obtained considering the energy balances on both streams and all tests with single-phase flows are compared in Figure 4. Results in Figure 4 show that, for all experimental data, the discrepancy between the heat transfer rate measured on the hot and cold streams are lower than ±3%. These results support the reliability of the experimental facility and the experimental measurements.

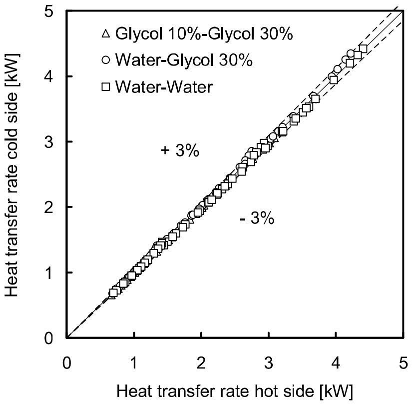

Figure 4 Comparison between heat transfer rates obtained from the energy balances on both streams.

Figure 5 shows the heat transfer rates obtained for the three combinations of the single-phase fluids considered in the analysis. For each fluids combination, three inlet temperature differences (5, 10 and 15 ºC) between both streams were included. The heat transfer rates were obtained by varying the cold fluid mass flow rate while keeping the cold fluid mass flow rate constant and equal to the nominal value indicated by the manufacturer.

Results in Figure 5 clearly show that the heat transfer rate increases when increasing the inlet temperature difference between both fluids. On the other hand, results in Figure 5 also show that the heat transfer rate increases with increasing the cold fluid mass flow rate for each one of the inlet temperature difference considered in the analysis. This figure also reveals the performance deterioration of the heat exchanger when replacing water with 10 wt% and 30 wt% ethylene glycol solutions as hot or cold fluids. The heat transfer rate deterioration after changing water in both circuits by the 10 wt% and 30 wt% ethylene glycol solutions is between 10-18%.

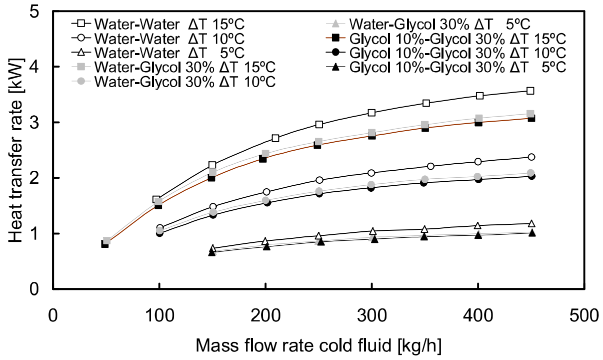

Figure 5 Heat transfer rate as function of cold fluid mass flow rate and inlet temperature difference for different combinations of single-phase fluids.

Figure 6 shows the overall heat transfer coefficients obtained from the heat transfer rates and the experimental conditions indicated in Figure 5.

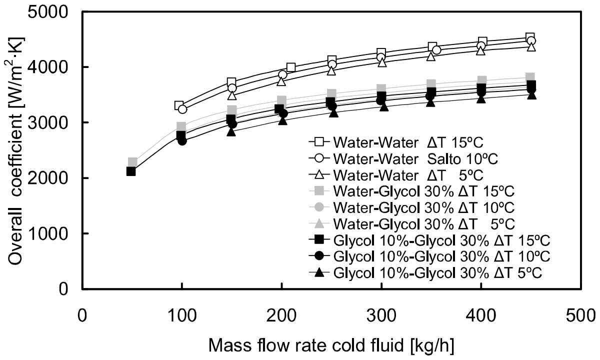

Figure 6 Experimental heat transfer coefficients as function of cold fluid mass flow rate for single-phase experiments.

Results in Figure 6 show that, for the same inlet temperature difference and mass flow rate, the overall heat transfer coefficients for the water-water flows are greater than for water-30 wt% ethylene glycol solution and for 10 wt%-30 wt% ethylene glycol solutions. The overall heat transfer coefficients for 10 wt%-30 wt% ethylene glycol solutions are between 20% and 25% lower than for water-water flows. On the other hand, results in Figure 6 also show that for any combination of fluids and for any inlet temperature difference, the overall heat transfer coefficient increases when the mass flow rate of the cold fluid increases.

Ice slurry

Figure 7 compares the heat transfer rates obtained from the energy balance on the ice slurry side and on the 30 wt% ethylene glycol solution side. Results in Figure 7 shows a deviation between ±10%. This deviation is greater than the deviation found with single-phase fluids, as can be seen in Figure 4, due to the uncertainties to determine ice fraction in ice slurry at the inlet and outlet of the heat exchanger. Taking into account the reasons cited above, the results shown hereafter are based on the energy balance on the single-phase fluid (30 wt% solution).

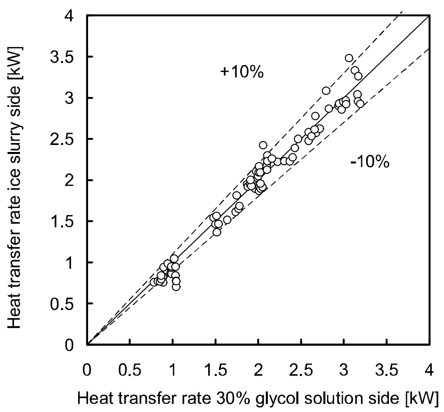

Figure 7 Comparison between the heat transfer rates obtained from the energy balance on the ice slurry and on the 30 wt% ethylene glycol solution streams.

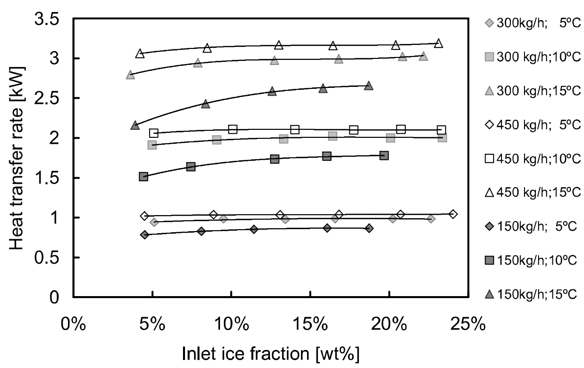

Figure 8 Heat transfer rate as a function of ice slurry mass flow rate, the inlet temperature differences between both fluids and the inlet ice fraction.

Figure 8 shows the heat transfer rates as a function of the inlet ice fraction obtained for ice slurry mass flow rates of 150 kg/h, 300 kg/h and 450 kg/h and inlet temperature difference between the 30 wt% solution and the ice slurry streams of 5 ºC, 10 ºC and 15 ºC. The solution mass flow rate (hot fluid) was kept constant and equal to 500 kg/h.

The experimental results shown in Figure 8 point out that the heat transfer rate clearly increases with increasing inlet temperature difference between the solution and the ice slurry. For the same inlet temperature difference, the heat transfer rate increases with increasing the ice slurry mass flow rate. However, results in Figure 8 reveal an uneven evolution of the heat transfer rate as a function of the inlet ice fraction. On the one hand, for high inlet temperature difference and low mass flow rates, the heat transfer rate clearly increases with increasing the ice fraction for ice fractions lower than 10%. This behavior is due to all the ice crystals melt before leaving the heat exchanger. On the other hand, for low inlet temperature differences (5 ºC), low mass flow rate (150 kg/h) and high inlet ice fraction, the heat transfer rate slightly increases or remains nearly constant when varying the inlet ice fraction. This behavior is justified by an incomplete melted of the ice crystals in the heat exchanger. Consequently, if the ice crystals are not melted completely, the influence of the inlet ice fraction on the heat transfer rate will be low.

Figure 9 compares the heat exchanger capacity operating with single-phase flows and with ice slurry. The heat transfer rate is presented as a function of the fluids combination and the inlet temperature difference between both fluids. The mass flow rates are the same for both streams and equal to the nominal flow rates indicated by the manufacturer. Results in Figure 9 show that, for the same inlet temperature difference, the heat transfer rates with water-water are higher than with all fluids combinations including ice slurry or an ethylene glycol solution. On the other hand, the heat transfer rates with ice slurry are higher than with water-30 wt% ethylene glycol and with 10 wt%-30 wt5 ethylene glycol.

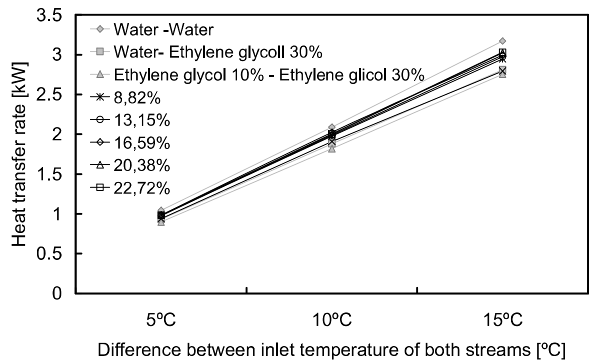

Figure 9 Heat transfer rate as a function of the fluids combination and inlet temperature difference.

Experimental measurements of the nearly isothermal pressure drop (without heat exchange) were carried out with water at 20 ºC, 10 wt% ethylene glycol aqueous solution at 20ºC and 0 ºC and ice slurry with mean inlet-outlet ice concentrations between 5 wt% and 25 wt% at ice fraction intervals of around 4 wt%. The experimental results are shown in Figure 10. For single phase fluids mass flow rate was varied between 100 kg/h and 500kg/h at intervals of 50 kg/h. With ice slurry, it was not possible to obtain stable measurements of pressure drop for mass flow rates lower than 300 kg/h. The unstable measurements can be attributed to the total or partial blockage of some channels in the heat exchanger due to the low velocities of circulation or the high ice fractions.

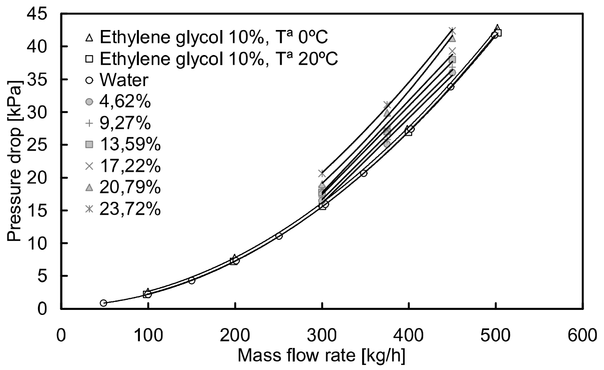

Figure 10 Variation of pressure drop with flow rate with water, glycol and ice slurry at different ice fractions.

The experimental measurements show that the ice slurry pressure drop increases with increasing the mass flow rate and the ice fraction. For the nominal operating mass flow rate provided by the manufacturer, the pressure drop increases around 30% when the ice fraction increases from 0 wt% to 24 wt%. These results agree with the data provided by other researchers such as Bellas [1].

CONCLUSION

Experimental research was carried out to investigate the thermal behavior of a titanium plate heat exchanger with offset strip fins. Experiments were conducted with several combinations of single-phase fluids and with ice slurry as cooling medium. Based upon the analysis of the experimental results, the following conclusions were drawn.

- The heat transfer rate with single-phase fluids increases when increasing the inlet temperature difference between both fluids and the cold fluid mass flow rate for each one of the inlet temperature difference considered in the analysis. The experimental results reveal the performance deterioration of the heat exchanger when replacing water with 10 wt% and 30 wt% ethylene glycol solutions as hot or cold fluids. The heat transfer rate deterioration after changing water in both circuits by the 10 wt% and 30 wt% ethylene glycol solutions is between 10-18%.

- Taking into account the same experimental conditions, the overall heat transfer coefficients for the water-water flows are greater than for water-30 wt% ethylene glycol solution and for 10 wt%-30 wt% ethylene glycol solutions. The overall heat transfer coefficients for 10 wt%-30 wt% ethylene glycol solutions are between 20% and 25% lower than for water-water flows. On the other hand, for all combination of fluids and all inlet temperature difference, the overall heat transfer coefficient increases with increasing the mass flow rate of the cold fluid.

- The heat transfer rate with ice slurry as cooling fluid increases with increasing inlet temperature difference between the solution and the ice slurry. For the same inlet temperature difference, the heat transfer rate increases with increasing the ice slurry mass flow rate. However, the experimental results reveal an uneven evolution of the heat transfer rate as a function of the inlet ice fraction. For high inlet temperature difference and low mass flow rates, the heat transfer rate clearly increases with increasing the ice fraction due to all the ice crystals melt before leaving the heat exchanger. However, for low inlet temperature differences, low mass flow rates and high inlet ice fraction, the heat transfer rate slightly increases or remains nearly constant when varying the inlet ice fraction due to an incomplete melted of the ice crystals in the heat exchanger. Therefore, if the ice crystals are not melted completely, the influence of the inlet ice fraction on the heat transfer rate will be low.

- The experimental results shown that, for the same inlet temperature difference, the heat transfer rates with water-water are higher than with all fluids combination including ice slurry or an ethylene glycol solution. On the other hand, the heat transfer rates with ice slurry are higher than with water-30 wt% ethylene glycol and with 10 wt%-30 wt% ethylene glycol.

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