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Theoretical and experimental study on ice-slurry use in comfort air-conditioning

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Abstract

This paper develops a theoretical and experimental study on convective heat transfer characteristics in a mini-channel type air-cooler, operating with ice-slurry. Although ice slurry is usually employed as a secondary fluid in air conditioning systems, this paper investigates its behaviour as a primary fluid. Furthermore, since previous investigations on ice slurry mostly referred to plate heat exchangers, little information is yet available in the literature regarding ice slurry use in mini-channel heat exchangers, this study employed several correlations recommended by the open literature for binary mediums, such as ice-slurry, in order to identify the best fitted correlation in terms of best agreement with the experimental results obtained for the given geometrical configuration of the heat exchanger, ice mass fraction, and specific operating conditions. Besides the theoretical study, the authors have also developed an experimental study and a comparative analysis of theoretical vs. experimental results with respect to the heat transfer coefficient on ice-slurry side as a function of ice mass fraction, of ice-slurry flow rate and air flow rate.

Keywords: ice-slurry, heat transfer, comfort air-conditioning.

1.Introduction

Ice slurry - a binary solution of water and ice -, with different possible ice-fraction represents an alternative solution to cooled water as a working fluid in technological and comfort air conditioning systems. This technical option has some indisputable advantages among which, the main one is its high heat transportation rate due to the latent heat of ice crystals, which leads to improved heat transfer performance. Ice slurry can provide the same cooling load with much lower volume flow rates as compared to cooled water, thus allowing smaller and more compact heat exchangers to be used, such as plate heat exchangers and mini-channel heat exchangers, characterized by a large heat transfer area to volume ratio, reduced weight and capital cost.

Besides advantages regarding cooling capacity, in moderate concentrations ice slurry offers significant savings in pumping costs as well as in pipe and equipment size. It can be easily pumped, transported and stored without needing to change pumping equipment, distribution network or accumulation tanks. In addition, ice-slurry is one of the environmentally friendly working fluid.

Due to all the previous considerations ice slurry continues to be a subject of intense research. Since in our knowledge, there are no available correlations in literature to describe the behavior with ice slurry in mini channel heat exchangers this paper attempts to identify the best correlation developed for compact heat exchangers to fit the experimental results.

2.Experimental stand

The experimental facility presented in Figure 1 and Photo 1 consisted of 3 major sections:

1. ice-slurry generating system of 7.5 kW maximum refrigerating capacity, working with R404A. Ice slurry was produced by circulating the carrier fluid (10% wt ethylene glycol water solution) through the ice slurry scraped-surface generator. Ice-slurry is stored into an insulated tank with a capacity of 1m^3. In order to prevent the ice particles agglomeration inside the tank and to keep the ice slurry homogeneous, the binary system is continuously stirred during the experiments by a mixing device. Ice slurry is pumped from the tank, by means of a centrifugal pump to the heat exchanger, placed inside the vertical closed air loop. The ice mass fraction is controlled at both inlet and outlet of the heat exchanger, by means of two Coriolis flow meters. The ice-slurry generating system is placed on a platform above the air cooler level;

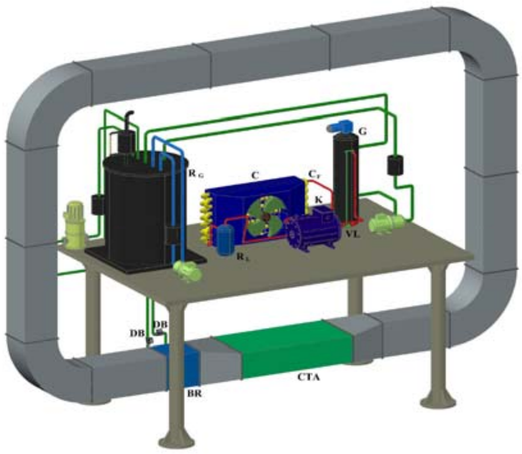

Figure 1.Experimental facility

Legend: CTA – Air Handling Unit; G – ice-slurry Generator; C – Condenser; K – Compressor; P– ice-slurry Pump; R_G – ice-slurry tank; R_L – Liquid Refrigerant Receiver; VL – Expansion Valve; BR – Air Cooler; DB – flow meter.

Photo 1.Experimental facility

2. vertical closed air loop - made of insolating sandwich type panels of 20mm thickness, exteriorly covered with a thin aluminum sheet. This air loop contains inside its lower horizontal branch the tested air cooler or, in other words, the consumer of an air conditioning system. Inside the same lower horizontal branch there is an air

handling unit whose role is to maintain at approximately constant values the air parameters at the air cooler inlet. In order to do this, the air handling unit comprises its own air cooler, air heater and a variable speed fan. The cooling/heating heat exchangers are fed with cooled/warm water from the existing water management system of the laboratory;

3. tested air cooler shown in Figure 2. This compact mini-channel type heat exchanger, specially designed and built for this project, is defined by the following geometric configuration: 25 parallel aluminum tubes with 6 inner orifices each, of 1.58 mm hydraulic diamet er, aluminum plate fins of 0.1mm thickness and the following overall dimensions: length of 0.5m; width of 0.018 m; height of 0.287 m. The total exterior heat transfer area is 1.1498 m^2 and the side area of the tubes is 0.45 m^2. Number of tubes connected in parallel: 8, 8 and 9. Number of flow paths: 3.

The experimental setup was equipped with a data acquisition system.

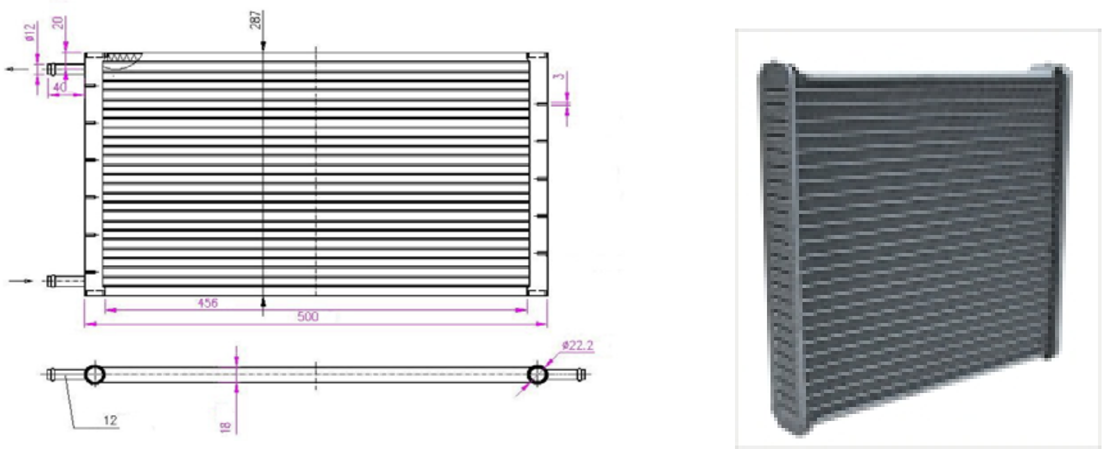

Figure 2. Compact mini-channel heat exchanger subjected to testing

3. Methodology

The heat transfer rate was experimentally determined from the energy balances on the air side and on the ice slurry side, as an arithmetic average, as stated Eq. (1):

$$\overline{\dot{Q}}_{\exp} = \left(\dot{Q}^{air}_{\exp} + \dot{Q}^{is}_{\exp}\right)/2 \quad (1)$$

where:

$\dot{Q}^{air}_{\exp}$ was calculated as a function of the specific enthalpies of the air, using the relation:

$$\dot{Q}^{air}_{\exp} = \left(h^{in}_{air} - h^{out}_{air}\right)\cdot \rho \cdot \dot{V} \quad (2)$$

where:

h^{in}_{air} and h^{out}_{air} are the specific enthalpies of the air at the inlet and outlet of the heat exchanger, respectively [J/kg]; ρ is the air density, evaluated using the average of the inlet and outlet air temperature [kg/m^3]; $\dot{V}$ is the air volumetric flow rate [m^3/s];

$\dot{Q}^{is}_{\exp}$ was calculated based on the following relation:

$$\dot{Q}^{is}_{\exp} = \dot{m}_{is} \cdot (h^{in}_{is} - h^{out}_{is}) \quad (3)$$

where:
h_{is}^{in} and h_{is}^{out} are the specific enthalpies of the ice slurry at the inlet and outlet of the heat exchanger, respectively [J/kg]; h_{is}^{in} is given by the relation: $h_{is}^{in} = f^{in} \cdot h_s + (1 - f^{in}) \cdot h_{cf}$ and h_{is}^{out} is given by the relation: $h_{is}^{out} = f^{out} \cdot h_s + (1 - f^{out}) \cdot h_{cf}$; $h_s = c_{ps} \cdot t - r_s$; the ice and the carrier fluid enthalpies (h_s and h_{cf} , respectively) were obtained according to Melinder (1997) and Lugo et al., 2002, respectively; $\dot{m}_{is}$ is the ice slurry mass flow rate [kg/s].
The average experimental heat flux density, $\bar{\dot{q}}_{\exp}$, was obtained by dividing the average experimental heat flux, $\bar{\dot{Q}}_{\exp}$, to the total exterior heat transfer area, S_T , - side area of the tubes, S_t , ratio:

$$\bar{\dot{q}}_{\exp} = \frac{\bar{\dot{Q}}_{\exp}}{\left(\frac{S_T}{S_t}\right)} \tag{4}$$

The average inner tube surface temperature was numerically calculated by solving the steady-state one - dimensional heat conduction equation, given the experimental heat flux density, $\bar{\dot{q}}_{\exp}$, and the measured outer tube wall temperature, as:

$$T_{iw} = T_{ow} - \bar{\dot{q}}_{\exp} \cdot \frac{\delta}{\lambda} \tag{5}$$

where:
T_{iw} and T_{ow} are the inner tube wall and the outer tube wall temperatures, respectively [ºC]; δ is the tube wall thickness [m]; λ is the tube thermal conductivity [W/(m·K)].
For a constant wall heat flux density, $\bar{\dot{q}}_{exp}$, the ice slurry experimental convective heat transfer coefficient, $\alpha_{\exp}$, was obtained according to Eq.6:

$$\alpha_{\exp} = \frac{\bar{\dot{q}}_{\exp}}{\left(T_{iw} - T_{is}\right)} \tag{6}$$

The following parameters have been measured throughout the experiments:
- on the air side: air cooler outlet cross section velocity, w_{air} [m/s]; inlet and outlet temperature t_{air}^{in} , t_{air}^{out} [ºC]; relative humidity, φ_{air} [%];
- on the ice-slurry side: density, ρ_{is} [kg/m^3]; mass flow rate, $\dot{m}_{is}$ [kg/s]*;* temperature at both the air cooler inlet and outlet t_{is}^{in} , t_{is}^{out} [ºC];
- outer tube wall temperature, T_{ow}, [ºC].

Two cross sectional grids have been used in order to measure the air temperature at the air cooler inlet and outlet, each of them with 5 measuring points. The temperature has been thus calculated as an average value, according to current standards requirements.
The following parameters have been calculated based on the measured ones:
- air volumetric flow rate, $\dot{V}$, based on air velocity measurements;

- inlet and outlet ice mass fractions, *f*, based on ice slurry density, ρ_{is}, measured using a Coriolis effect mass flow meter, as: $f = \dfrac{\rho_{cf} - \rho_{is}}{\rho_{cf} - \rho_{s}}$; the ice and the carrier fluid densities (ρ_s and ρ_{cf}, respectively) were obtained according to Melinder (1997) and Lugo et al., 2002, respectively;
- ice slurry specific enthalpy, h_{is}, and ice specific enthalpy, h_s;
- inner tube wall temperature, T_{iw}.

Calculations have been based on reliable measured data corresponding to a quasi-steady state operating regime. It was considered as such the regime characterized by maximum ±5% variation of the measured parameters, recorded during at least 20 minutes at 20 seconds intervals. Data used in calculation represent the average values of the measured parameters.

Measuring devices:

Device	Type	Accuracy
Hot wire anemometer (air velocity)	FV A935-TH5	± 2%
Mass flow rate and density (ice slurry)	Coriolis ProMass 63	±0,1%
Temperature sensors (air and outer tube surface)	T-type thermocouples	± 0,25°C
Data acquisition unit	Almemo 5590	500kB memory

In order to calculate the convection heat transfer coefficient on the ice-slurry side the correlations have been selected from the literature based on the following criteria: to be applicable to ice slurry, to be applicable to compact heat exchangers and to have the validity range satisfied, as much as possible, by the present experiment.

The following correlations have been used:

- Stamatiou, E.; Kawaji, M. (2005) for laminar flow:

$$Nu_{cf,x} = 4.0 \cdot Gz^{0.486} \cdot \bar{f}^{\,0.30} \cdot \left(\frac{\eta_{cf}}{\eta_w}\right)^{0.24} \tag{7}$$

where:

Gz - Graetz number; *x*- axial distance from the inlet of the heat exchanger. The validity range of the correlation above is: laminar flow defined as: $2100 < \mathrm{Re}_{cfl} < 4000$; $1\ \% < f < 25)$ %.

- Since in our knowledge, there are no consensual available data in literature to indicate the range of Reynolds number defining the ice slurry flow in mini-channel heat exchangers, the authors have decided to also take into consideration for this study the correlation by Stamatiou, E.; Kawaji, M. (2005) for turbulent flow:

$$\overline{Nu}_{is} = \frac{\overline{\alpha}_{is} \cdot D_h}{\lambda_{cf}} = \overline{Nu}_{gn} \cdot (1 + 1.85 \cdot 10^5 \cdot \bar{f}^{\,0.72} \cdot (\mathrm{Re}_{cf})^{-1.3} \cdot \left(\frac{\eta_{cf}}{\eta_w}\right)^{2.47} \tag{8}$$

where:

- $\overline{Nu}_{gn}$ is the single-phase Nusselt number predicted by Gnielinski (1976 and 1983) given by Eq. 9:

$$\overline{Nu}_{gn} = \frac{(f_{fr}/2)\cdot(\mathrm{Re}_{cf}-1000)\cdot \mathrm{Pr}_{cf}}{1+12{,}7\cdot(f_{fr}/2)^{0.5}\cdot(\mathrm{Pr}_{cf}^{2/3}-1)} \tag{9}$$

- f_{fr} is the friction factor defined as: $f_{fr} = [1.82\cdot\log(\mathrm{Re}_{cf}) - 1.64]^{-2}$ (10)
- Re_{cf} is the Reynolds number evaluated based on the ice slurry velocity (w_{is}), the hydraulic diameter of the mini-channels (D_h), and the carrier fluid kinematic viscosity, (ν_{cf}), defined as:

$$\mathrm{Re}_{cf} = \frac{w_{is}\cdot D_h}{\nu_{cf}} \tag{11}$$

The validity range of the correlation above is: turbulent flow, defined as: $3300 < \mathrm{Re}_{cf} < 11000$; $0\ \% < f < 25)\%$.

- Norgaard E. et al., (2001)

$$\alpha_{is} = \frac{\lambda_{is}\cdot\varphi}{D_h}\cdot 0.31\cdot(\mathrm{Re}_{is})^{0.61}\cdot(\mathrm{Pr}_{is})^{y} \tag{12}$$

where:

φ - correction factor, [-]; $\varphi = \left(\frac{\eta_{is}}{\eta_{cf}}\right)^{\frac{0,3}{(\mathrm{Re}_{is}+6)^{0,125}}}$ and $y = e^{\frac{6,4}{3\cdot(\mathrm{Pr}_{is}+30)}}$ (13)

The validity range of the correlation above is: laminar or turbulent flow; glycol concentration of 30 wt %; ice mass fraction between 10% and 30%; equivalent diameter of 2 mm; heat flux between 10 and 20 kW/m^2.

4.Experimental results

The ice slurry temperature was set at approximately - 1°C and slightly varied around this value throughout the experiments, as data in Table 1 show. The air flow rate was controlled and maintained at an average value of 2200 m^3/h through the experiments. The average inlet ice mass fraction was set at 8% by varying the ice slurry mass flow rate. The average outlet ice mass fraction was 5%.

The main measured and calculated values, of those mentioned above, are summarized in Tables 1 and 2.

Table 1.

Measured values of ice-slurry and air

Ice-slurry			Air			
Inlet/outlet temperature, [°C]	Mass flow rate, [kg/s]	Outer tube wall temperature, [°C]	Inlet temperature, [°C]	Inlet relative humidity, [%]	Outlet temperature, [°C]	Velocity, [m/s]
-1.0	0.397	-0.5	28.2	49.0	22.9	6.93
-1.2	0.383	-0.7	27.9	50.2	23.1	6.30
-1.4	0.404	-1.0	28.1	50.1	23.0	6.25
-1.3	0.390	-0.8	28.0	49.8	22.9	6.30
-1.1	0.383	-0.6	28.1	50.0	23.2	6.51

Table 2.

Calculated values of ice-slurry and air

Ice-slurry			Air		Deviation, [-]
Average ice mass fraction [%]	Specific enthalpy change, [J/kg]	Heat transfer rate, [W]	Volumetric flow rate, [m^3/s]	Heat transfer rate, [W]	
6.5	9.900	3940	0.66	4514	0,13
7.0	9.900	3804	0.6	4104	0,07
6.75	8.300	3336	0.595	4070	0,18
7.0	9.900	3865	0.6	4104	0,06
7.0	9.900	3802	0.62	4241	0,10

As can be seen in Table 2, the heat transfer rate calculated by applying the energy balance in the air side is on average with 10% greater than that obtained by applying the energy balance in the ice slurry side.

As Table 2 shows the average deviation between the energy balances applied in both sides of the heat exchanger (ice slurry and air) varied between 6% and 18%, throughout the series of experiments. These values lead to an overall deviation of 11%, computed as an arithmetic average. This value is considered acceptable by the authors taking into account the well-known experimental difficulties in terms of measurements accuracy associated with the air flow.

The measured and averaged quantities based on which the experimental ice slurry convective heat transfer coefficient was computed, according to Eq.6 are given in Table 3.

As data in Table 3 show the experimental α value ranges between 3309 and 3623 W/ (m^2 K), which leads to an averaged value of 3467 W/ (m^2 K).

Table 3.

The experimental ice slurry convective heat transfer coefficient

Temperature,[°C]		Average heat transfer rate, $\overline{\dot{Q}}$,[W]	Average heat flux density, $\overline{q}$, [W/m^2]	Convective heat transfer coefficient, α_{exp} , [W/ (m^2 K)]
Ice slurry	Difference between the inner tube wall and ice slurry, (T_{iw} - T_{is})			
-1.0	0.50	4227	1654	3309
-1.2	0.45	3954	1547	3439
-1.4	0.40	3703	1449	3623
-1.3	0.45	3985	1559	3465
-1.1	0.45	4021	1574	3497

Figure 3 illustrates the variation of the local ice slurry α values predicted by Eq. 7 (Stamatiou, E.; Kawaji, M. (2005) for laminar flow) with the axial distance from the heat exchanger inlet. It shows that the local ice slurry α values constantly decrease with the axial distance from the inlet of the mini-channel heat exchanger, in all tests carried out.

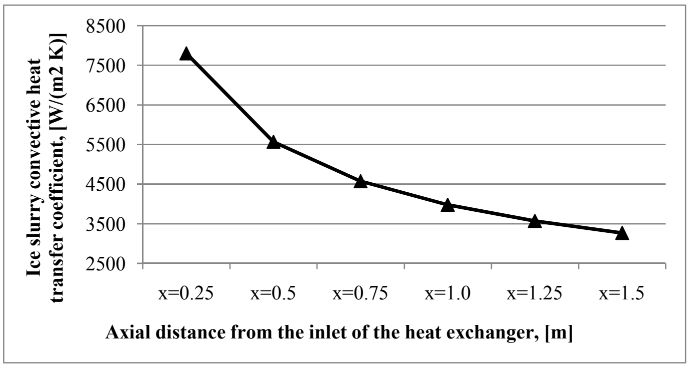

Figure 3. Convective heat transfer coefficient variation with the axial distance from the heat exchanger inlet

Figure 4 compares the experimental ice slurry convective heat transfer coefficient results with those predicted by Eq. 7 (Stamatiou, E.; Kawaji, M. (2005) for laminar flow), by Eq. 8 (Stamatiou, E.; Kawaji, M. (2005) for turbulent flow), and by Eq.12 (Norgaard E. et al., (2001).

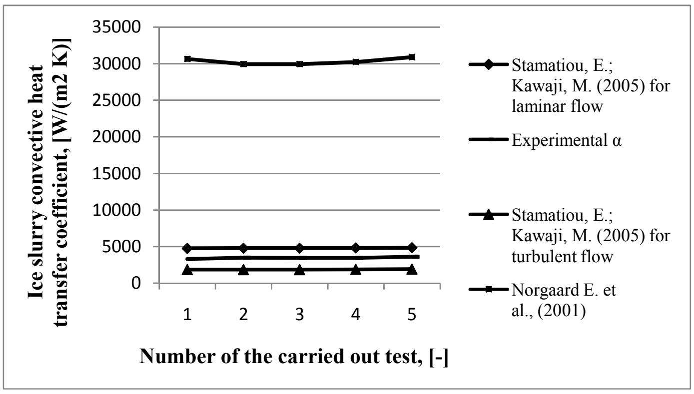

Figure 4. Experimental and predicted ice slurry convective heat transfer coefficient obtained in each carried out test

In Figure 4 the average theoretical value from Eq. 7 (Stamatiou, E.; Kawaji, M. (2005) for laminar flow) is plotted. Based on the local theoretical α values, the average α value has been computed, as an average corresponding to the total axial distance from the inlet of the heat exchanger. This average value ranges from 4612 to 4860 W/(m^2 K). Laminar or slightly turbulent the flow may be assumed since the 25 tubes were connected in three flow paths (of eight, eight and nine tubes, in parallel, respectively).

5. Conclusions

This theoretical and experimental study was performed on a compact mini-channel type air cooler and aimed to identify the best fitted correlation, of those available at present, in the authors' knowledge, to evaluate the ice slurry convective heat transfer coefficient. Three correlations, whose validity range generally satisfies the present experimental conditions have been considered: Stamatiou, E.; Kawaji, M. (2005) for laminar flow, Stamatiou, E.; Kawaji, M. (2005) for turbulent flow and Norgaard E. et

al., (2001). The study assumed laminar or slightly turbulent ice slurry flow and concluded that, for the specific geometrical configuration of the compact mini-channel type air cooler and the specific operating conditions, the best agreement between the experimental and theoretical results was obtained using the correlation proposed by Stamatiou, E.; Kawaji, M. (2005) for laminar flow, with an average deviation of 28%. The deviations have even lower values for axial distances higher than 1.25m and range between -9% and +7%.
The other two correlations showed unacceptable high deviations from the experimental results.

Nomenclature

$\bar{\dot{q}}$ - average heat flux density, W/m^2; $\bar{\dot{Q}}$ - average heat flux, W; h - specific enthalpy, J/kg; $\dot{m}$ - mass flow rate, kg/s; $\dot{V}$ - volumetric flow rate, m^3/s; f - ice mass fraction %; c_p - specific heat at constant pressure, J/(kg K); r - specific latent heat of fusion, J/kg; S - area, m^2; T - temperature, K; Gz- Graetz number; $Gz = \frac{Re_{cf} \cdot Pr_{cf} \cdot D_h}{x}$; Re - Reynolds number; Pr - Prandtl number; Nu - Nusselt number; D_h - hydraulic diameter, m; x - axial distance from the inlet of the heat exchanger, m; w - velocity, m/s;
Greek symbols: α - convective heat transfer coefficient, W/(m^2 K); ρ - density, kg/m^3; δ - tube wall thickness, m; λ - thermal conductivity, W/(m K); η - dynamic viscosity, Pa s; ν - kinematic viscosity, m^2/s;
Subscripts: is - ice slurry; *exp* – experimental; *air* - on the air side; *in* – inlet; *out* - outlet; *s* - solid (ice); *cf* - carrier fluid (10% wt ethylene glycol water solution); *T* - total; *t* – tubes; *iw* - inner wall; *ow* - outer wall; *w* – wall.

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