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Aspects concerning the use of binary ice for comfort air-conditioners

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Abstract

This paper refers to the use of ice-slurry, as an intermediary phase changing cooling fluid, in air-conditioning.

Ice-slurry is provided by a scraper-type generator that is part of a single stage compression system working with R404A, as primary refrigerant. A mixture of ice-slurry, water, and talin (10% mass concentration) represents the secondary cooling medium. This was used in a heat exchanger of classical type, namely an air cooler, with aluminum fins, of 0.1 mm thickness.

This paper reports experimental results of ice-slurry used as cooling fluid for air-conditioning consumers of various cooling capacity demands. Experiments have been carried out using two different ice mass fractions, namely approximately15% and 20% and two different air volumetric flow rates, namely Qair = 3000 m^3 / h and Qair = 2000 m^3 / h

The cooling capacity of the ice-slurry generator has been experimentally validated by simultaneously determining the thermal capacity for the evaporator (as a function power input to the compressor), for the air-cooler and air heaters.

***Key words**: ice-slurry, air-conditioning systems, heat exchanger.*

1. Introduction

The advantages of ice-slurry, as a viable alternative to the classical air conditioning solution, based on chiller prepared cooled water are well known and will not be insisted upon, [1].

This experimental study used a mixture of ice-slurry, water, and talin (10% mass concentration) and aimed to determine the cooling capacity of an ice-slurry fed air-cooler, under different working conditions, namely: different inlet ice mass fractions and different volumetric flow rate of ice slurry and cooled air.

2. Experimental Stand

The experimental study on air cooling using ice-slurry as cooling medium has been carried out on a laboratory stand. The layout of this stand is presented in Figure 1 and Photo1. As both of them show, the experimental stand consists of 3 major sections, [2]:

- ice-slurry generating system - of 7.5 kW refrigeration capacity, working with R404A, [3]. The scraper-type generator feeds the ice-slurry storage tank, of 1 m^3 volume. In order to prevent the ice particles agglomeration inside the tank, the binary system (water solution and ice particles) is continuously recirculated by a pump, inside the tank and through an exterior circuit. The ice mass fraction is controlled by an automation system that operates on the principle of electrical resistance measurement. The ice-slurry generating system is placed on a platform above the air cooler level;
- vertical closed air loop - made of insolating panels of 20mm thickness of sandwich type, exteriorly covered with thin aluminum sheet. This air loop contains inside its lower horizontal branch the studied air cooler or, in other words, the consumer of an air conditioning system. Inside the same lower horizontal branch there is the 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 consists of an air cooler, an air heater and a fan. The cooling/heating heat exchangers are fed with cooled/warm water from the existing water management system of the laboratory, [4];
- air cooler subjected to testing. This is a finned copper coil of 0.4m length and an inside diameter of 8 mm. The coil bank consists of 4 panels of 10 horizontal staggered tubes. The rectangular plate fins are made of aluminum of 0.1 mm thickness and have the overall dimensions of 0.65x0.25m. The inside surface area in contact with ice-slurry/cooled water is of 0.40192 m^2 and the overall exterior surface area in contact with the air is of 7.47 m^2. The air cooler is fed with ice-slurry by a circulating pump.

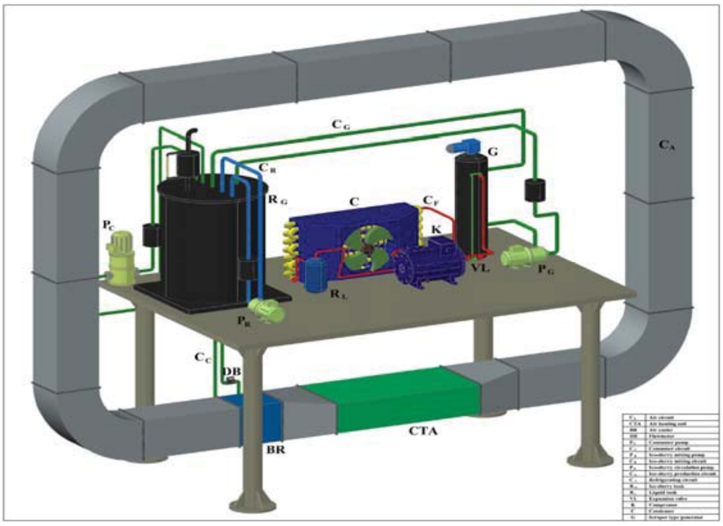

Fig. 1 Layout of the experimental stand

Legend: CTA – Air Handling Unit; G – ice-slurry Generator; C – condenser; K – compressor; P_C – ice-slurry Pump; R_G – ice-slurry tank; R_L – Liquid Refrigerant Receiver; VL – Expansion Valve; P_G - secondary cooling medium pump ; P_R – Ice-slurry Recirculation Pump; BR – Air Cooler; DB – flow meter; C_C – consumer circuit; C_G - ice slurry generation circuit; C_R – Recirculation circuit; C_A – air circuit

Photo1. Experimental stand showing the ice-slurry generation system on top, part of the air loop, containing the fan coil inside its lower branch and the connection tubes

3. Methodology

The experimental research started with determining the refrigeration capacity of the ice-slurry generator, under different consumer cooling loads. Experiments have been carried out for 2 different air flow rates at the consumer, namely: $Q_{air} = 3000\, m^3 / h$ and $Q_{air} = 2000\, m^3 / h$. To each of the above mentioned air flow rate value, corresponded 4 different ice-slurry mass flow rates, as follows: $Q_{is} = 0.350\, m^3 / h$; $Q_{is} = 0.330\, m^3 / h$; corresponding to $Q_{air} = 3000\, m^3 / h$, and; $Q_{is} = 0.350\, m^3 / h$; $Q_{is} = 0.330\, m^3 / h$, corresponding to $Q_{air} = 2000\, m^3 / h$, respectively. In addition, each of the above mentioned experiments was performed for 2 different values of the ice mass fraction: f = 15%; and 20% approximately, [5].

In order to experimentally validate the refrigeration capacity of the ice-slurry generator, Φ_{is}, the following quantities have been also measured: the refrigeration capacity of the evaporator, Φ_0 (based on measured compressor power consumption, P_k), the cooling capacity of the ice-slurry fed air-cooler, Φ_{air}, and the heating capacity of the air heater inside the air handling unit, Φ_w, [6].

The following balance equations have been used, [7], [8]:

$$\Phi_{is} \cong \Phi_0 \cong \Phi_{air} \cong \Phi_w, \text{ [W]} \qquad (1)$$

$$\Phi_o + P_k = \Phi_c, \text{ [W]} \qquad (2)$$

where: Φ_c - heat flow rate rejected by the condenser.

Calculations have been based on reliable measured data that is 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, along 10 consecutive readings, at 10 minutes apart. Data used in calculation represent the averaged values of the measured parameters. Another filter in considering data as reliable was represented by the deviation in energy balances set up to 12%.

3.1. Compressor Power Consumption has been measured by a power analyzer type Fluke 434. The values indicated by this device, under quasi-steady state experimental conditions, are shown in the Figure below. Of all the displayed data, just the power values, having kW as measuring unit, are relevant for this experiment and further calculations.

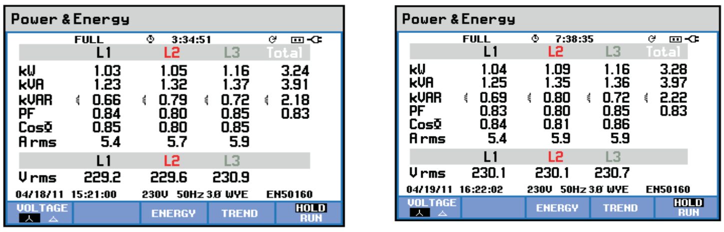

Fig. 2 Measured values of the compressor power consumption

As the above experimental data show, the measured compressor power consumption was 3.24 kW for 15% ice mass fraction and 3.28 kW for 20% ice mass fraction.
The uncertainty analysis related to this measurement indicated a mean value of 2%.

3.2. Refrigeration Capacity of the Ice-Slurry Generator
In order to experimentally determine the refrigeration capacity of the ice-slurry generator, Φ_{is}, the single-stage vapor compression refrigeration cycle has been drawn on the Pressure-Enthalpy Diagram, based on the following measured data, [9]: evaporating temperature and pressure; condensing temperature and pressure; compressor inlet superheated vapor temperature, (state point 1”); discharged superheated vapor temperature(state point 2); condenser outlet liquid temperature (state point 3), [10].

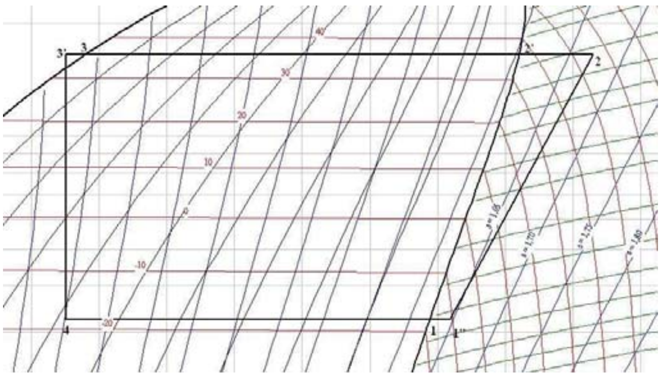

Fig. 3 Schematic of the experimental Single-Stage Vapor Compression Refrigeration cycle

Table 1 lists the measured and computed thermodynamic properties of R404A of the experimental refrigeration cycle.

Table 1. Measured and computed thermodynamic properties of the experimental refrigeration cycle

State point	1	1”	2	2’	3	3’	4
θ [°C]	-18	-10.99	55.14	36	36	32.76	-18
p [bar]	3.2	3.2	16	16	16	16	3.2
h[kJ/kg]	357.5	364	406	384	256	250	250

Based on the computed enthalpies in Table 1, the following quantities have been determined:

- the refrigerant mass flow rate, $\dot{m}_{R404A}$, as the ratio between the measured compressor power consumption, P_k, and the vapor enthalpy change between compressors suction and discharge, $(h_2 - h_{1"})$, [11], [12]:

$$\dot{m}_{R404A} = P_k / (h_2 - h_{1"}) = 0.078095 \text{ kg/s} \quad (3)$$

- the refrigeration capacity of the evaporator:

$$\Phi_0 = \dot{m}_{R404A} \cdot (h_1 - h_4) = 8.39 \text{ kW} \quad (4)$$

The ice-slurry refrigeration capacity was further obtained by the equation (1): $\Phi_{is} =$ 8.39 kW

3.3. Cooling Capacity of the Ice-Slurry Fed Air-Cooler

The air volumetric flow rate, $\dot{V}_{air}$, has been determined based on measured air velocity inside the air loop. The cross section of the loop selected for velocity measurements, placed at the air cooler outlet, complied with current standard requirements regarding stabilized air flow. The air flow rate has been measured with an Ahlborn hot wire anemometer, of 3% accuracy and was set to 3000 m^3/h and 2000 m^3/h.

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 averaged value. As air temperature measuring devices type K thermocouples, ± 0.25°C accuracy have been used.

The cooling capacity of the air-cooler, *on the air side*, was obtained from the following equation:

$$\Phi_{air} = \dot{m}_{air} \cdot (h_{inlet} - h_{outlet}) \quad (5)$$

where: h_{inlet}, h_{outlet} - air enthalpy at the inlet/outlet of the air-cooler, respectively.

The cooling capacity of the air-cooler, *on the ice-slurry side*, was obtained from the following equation:

$$\Phi_{is} = f \cdot \dot{m}_{is} \cdot r \quad (6)$$

where: - f - ice mass fraction; - r – fusion latent heat of ice; r = 333000 J/kg.

The ice-slurry volumetric flow rate, $\dot{V}_{is}$, has been measured using an ultrasound flow meter, [13]. The ice-slurry temperatures at both inlet and outlet of the air-cooler have also been measured and controlled. As a consequence it was possible to admit the hypothesis of just latent heat transfer between air and ice-slurry. Transparent tube sections next to the ice-slurry inlet and outlet made possible, to some extent, a visual assessment of the ice-slurry composition.

As mentioned above, the ice mass fraction was controlled by an automation system that operates on the principle of electrical resistance measurement, [14], [15]. The uncertainty analysis related to this measurement indicated a mean value of 4%.

3.4. Heating Capacity of the Air Heater

In order to experimentally determine the heating capacity of the air heater, *on the water side*, Φ_w, the water volumetric flow rate, $\dot{V}_w$, has been measured, along with the inlet and outlet water temperatures. The heating capacity of the air heater was obtained from the following equation:

$$\Phi_w = \dot{m}_w \cdot c_{p,w} \cdot \left(\theta_{inlet} - \theta_{outlet}\right) \qquad (7)$$

where: $c_{p,w}$ – specific heat of water at constant pressure.
The water flow rate has been measured with an electronic flow meter (Danfoss type).
The measured data have been stored and centralized by two data acquisition systems, namely: Almemo 2890-9 and Almemo 3290.

4. Experimental Results

Figures 4 and 5 show the increase of the cooling capacity of the consumer with the increase of the inlet ice slurry flow rate, for the same ice mass fraction under different air flow rates: 3000 m^3/h and 2000 m^3/h, respectively.

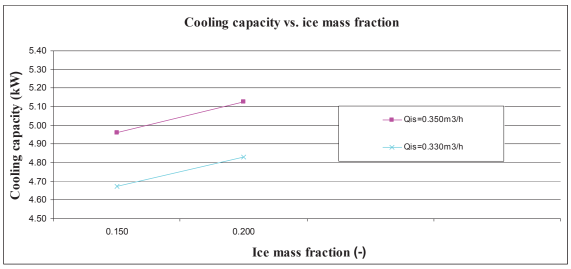

Figure 4. Cooling capacity of the air cooler as a function of the ice mass fraction, for 3000 m^3/h air flow rate

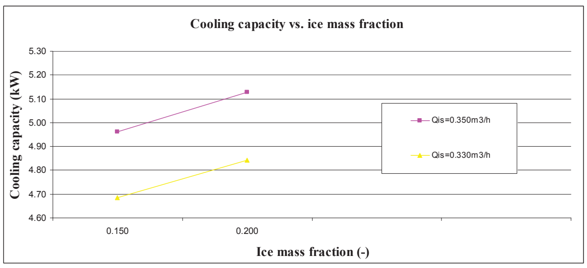

Figure 5. Cooling capacity of the air cooler as a function of the ice mass fraction for 2000 m^3/h air flow rate

Figures 6 and 7 show an approximate linear increase of the consumer cooling capacity with the increase of the air inlet temperature for both ice mass fractions under approximately the same ice slurry temperature. The data in Figure 6 correspond to 3000 m^3/h air flow rate and those in Figure 7 to 2000 m^3/h air flow rate.

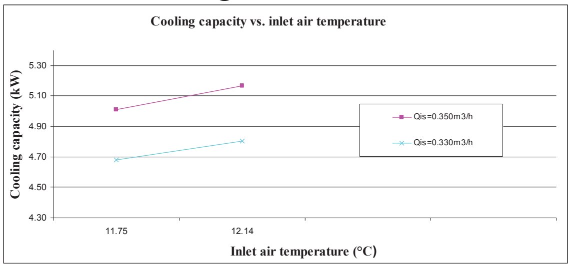

Figure 6. Cooling capacity of the air cooler as a function of the inlet air temperature for 3000 m^3/h air flow rate

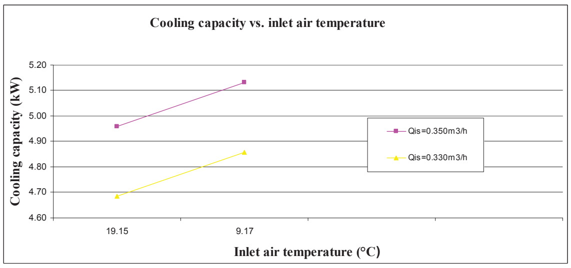

Figure 7. Cooling capacity of the air cooler as a function of the inlet air temperature for 2000 m^3/h air flow rate

Figures 8 and 9 show that for the same inlet ice slurry flow rate the heating capacity of the air heater increases with the inlet water temperature that is with the increase of the temperature difference between the mean water temperature and the mean air temperature. The trend is the same irrespective of the air flow rate. The data in Figure 8 correspond to 3000 m^3/h air flow rate and those in Figure 9 to 2000 m^3/h air flow rate. At the same time, for the same water inlet temperature, the heating capacity increases with the ice slurry flow rate.

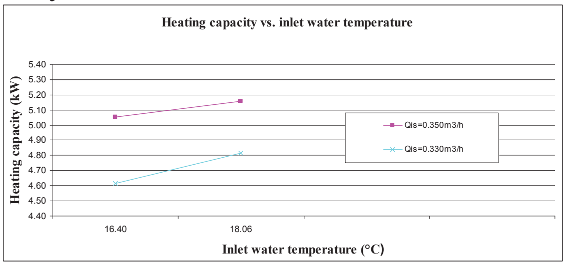

Figure 8. Heating capacity of the air heater depending on the inlet water temperature for 3000 m^3/h air flow rate

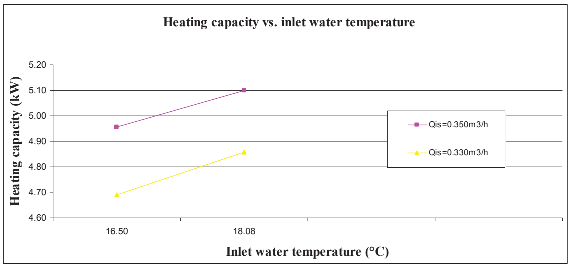

Figure 9. Heating capacity of the air heater depending on the inlet water temperature for 2000 m^3/h air flow rate

5. Conclusions

The experimental results showed that:

- for the same ice mass fraction the cooling capacity of the consumer increases with the inlet ice slurry flow rate, by 6.2% at 3000 m^3/h air flow rate and by 4.4% at 2000 m^3/h;
- the COP of the experimental system is approximately 2.7;
- the ice slurry generator refrigeration capacity was experimentally validated by simultaneously measured ice-slurry cooling capacity, air cooling capacity, water heating capacity and compressor power consumption;
- the cooling capacity of the air cooler has also been experimentally validated. Its averaged value is 5 kW and it depends on the ice slurry mass fraction: 5.05 kW for 15 % ice slurry mass fraction and 5.16 kW for 20 % ice slurry mass fraction and 3000 m^3/h air flow rate; 4.85 kW for 15% and 5.10 kW for 20 % ice slurry mass fraction and 2000 m^3/h air flow rate; the uncertainty analysis related to this measurement indicated a mean value of 8%;
- it was not possible to correlate the cooling capacity of the air cooler with the ice slurry generator refrigeration capacity because of the limited heat transfer surface of the air cooler.

7. References

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