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EXPERIMENTAL COMPARATIVE RESEARCH on ICE-SLURRY vs. COOLED WATER in COMFORT AIR-CONDITIONING SYSTEMS

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Abstract: *This paper reports experimental results of an ice-slurry system designed for comfort air conditioning. The cooling capacity of the system ranges from 3 to 7.5 kW. Ice-slurry is generated in 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 classical type heat exchanger namely a fan coil, made of copper, with the inner diameter of 9.0 mm, and aluminum fins, of 0.1 mm thickness. The experimental study aimed to comparatively analyze the operation of the system working with ice-slurry vs. cooled water, as the classical secondary cooling medium. The comparative study has been developed with regard to: overall heat transfer surface; working fluid mass flow rate; thermal performances. The conclusion of the paper is that ice-slurry represents a better option both from the energetic and indoor comfort points of view.*

Key words: *air-conditioning, ice-slurry.*

1. Introduction

This paper reports comparative experimental results on the thermal performances of the same air cooler fed with different cooling mediums: ice-slurry and cooled water. 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. Under these circumstances this study aims to quantify the ice-slurry over cooled water advantages, from the following points of view: required increase in water flow rate, for the same air temperature drop and the same heat transfer area; required increase in heat transfer area on cooled water operation, for the same temperature drop.

The experiments have been carried out for 3 different ice mass fractions, namely 10%, 15% and 20% and 2 different ice-slurry volumetric flow rates adjusted by a frequencies converter connected to the ice-slurry supply pump.

The increases in both water flow rates and air cooler heat transfer area have been calculated as percentages of the corresponding ice-slurry quantities and plotted vs. ice-slurry mass fraction.

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2. Experimental Stand

The comparative study on air cooling using as cooling medium ice-slurry and cooled water has been carried out on a laboratory experimental stand.

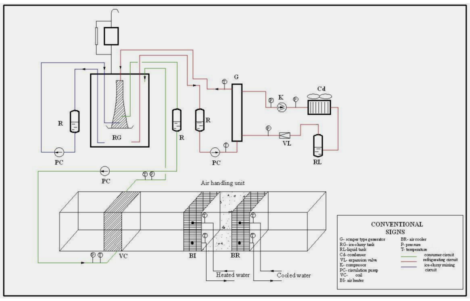

Fig. 1. *Layout of the experimental stand*

The layout of this stand is presented in Figure 1 and Photo 1.

As both of them show, the stand comprises 3 major sections:

1. ice-slurry generating system - of maximum 7.5 kW refrigeration capacity, working with R404A. The scraper-type generator feeds the ice-slurry storage tank, of 1m^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.

Photo 1. *Experimental stand*

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;

2. 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 of 3000m^3/h. The cooling/heating heat exchangers are fed with cooled/warm water from the existing water management system of the laboratory;

3. air cooler subjected to testing under conditions similar to an air conditioning system consumer. 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 0.1mm 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.

Figure 1 also shows the measuring devices and their locations. The following parameters have been measured throughout the experiments:

- on the air side: temperature and relative humidity at both the air cooler inlet and outlet;
- on the ice-slurry side: ice mass fraction, temperature at both the air cooler inlet and outlet and the flow rate;

- on the cooled water side: temperature at both the air cooler inlet and outlet and the flow rate.

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. The air velocity has been measured in a cross section at the air cooler outlet according to current standards requirements.

Measuring devices: for the air temperature have been used type K thermocouples, ± 0.25°C accuracy; for the cooled water and ice-slurry flow rates have been used Danfoss electronic flow meters; the air flow rate has been measured with an Ahlborn hot wire anemometer, of 3% accuracy.

3. Methodology

The same air cooler has been tested for ice-slurry and cooled water as a cooling medium. The criteria used for comparison was the same temperature drop on the air side.

The comparative experimental study has been carried out under the following conditions:

`- *ice-slurry*: two different volumetric flow rates set by different frequencies at the ice-slurry supply pump, resulting in different ice-slurry velocities (1.88 and 2.52 m/s, respectively); ice mass fraction of 10%, 15% and 20% at the air cooler inlet and approx.0% at the air cooler outlet; ice-slurry temperature at the air cooler inlet of approx. -1°C; air-cooler surface temperature: approx. 0°C;

- *air*: temperature at the air cooler inlet of approx. +28°C; relative humidity at the air cooler inlet of approx. 50%; volume flow rate of 2000m^3/h.

- *water*: temperature at the air cooler inlet of approx. 6°C; temperature increase of maximum 6°C (within the usual range); air-cooler surface temperature: approx. 9°C.

In order to achieve the objective of approx. 0% ice mass fraction at the air cooler outlet, for a given inlet air temperature and air flow rate, the ice-slurry volumetric flow rate has been modified. The values corresponding to 15 and 20 Hz turned out to the most suitable. Transparent tube sections next to the ice-slurry inlet and outlet made possible a visual assessment of the ice-slurry composition.

In order to achieve approx. the same temperature drop on the air side with cooled water as a cooling medium, the cooled water flow rate has been adjusted to meet this condition, given the same inlet water temperature (approx. 6°C) and admitting the usual maximum increase in water temperature (approx. 6°C).

Since the air-cooler surface temperature is different, depending on the cooling medium, the direction followed by the cooling process, starting from the same thermodynamic state, is also different; still, independent from the process direction, the air temperature drop has been kept at almost the same value, since this is the very criteria used for the proposed comparison.

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 ice-slurry to air and cooled water to air energy balances, set up to 10%.

4.Experimental Eesults

The measured parameters are shown in Tables 1 to 5.

Measured parameters with ice-slurry as a cooling medium Table1.

Measured parameters							
Ice-slurry				Air			
Frequency, [Hz]	Ice mass fraction, [%]	Volumetric flow rate, [m^3/s]	Air cooler surface temp., [°C]	Inlet temp., [°C]	Inlet relative humidity, [%]	Outlet temp., [°C]	Volumetric flow rate, [m^3/s]
15	10	0.000095	-0.1	28.0	49.0	25.3	0.55
20	10	0.000127	-0.2	28.2	49.6	24.9	0.55
15	15	0.000095	0.0	27.9	50.2	23.9	0.55
20	15	0.000127	-0.1	28.2	50.1	22.5	0.55
15	20	0.000095	0.1	27.8	49.8	22.1	0.55
20	20	0.000127	0.2	28.2	50.0	22.2	0.55

Measured parameters with cooled water as a cooling medium Table 2.

Measured parameters							
Cooled water				Air			
Inlet temp, [°C]	Outlet temp., [°C]	Volumetric flow rate, [m^3/s]	Air cooler surface temp., [°C]	Inlet temp., [°C]	Inlet relative humidity, [%]	Outlet temp., [°C]	Volumetric flow rate, [m^3/s]
6.1	11	0.000143	8.8	27.8	49.8	25.2	0.55
6.0	10.6	0.000189	9.1	28.0	49.5	24.8	0.55
6.2	11.5	0.000211	8.9	28.2	50.1	24.4	0.55
6.1	11.6	0.000285	9.0	28.1	50.0	22.6	0.55
6.0	11.5	0.000279	9.1	27.9	50.2	22.3	0.55
6.1	10.5	0.000379	8.9	28.1	50.1	21.9	0.55

Calculated quantities with ice-slurry as a cooling medium Table 3.

Calculated quantities				
Ice-slurry		Air		Energetic balance deviation, [-]
Mass flow rate, [kg/s]	Cooling capacity, [W]	Temperature drop, [°C]	Cooling capacity, [W]	
0.095445	3178.3	2.7	3224.1	-0.014
0.127717	4253.0	3.3	3934.7	0.075
0.094925	4741.5	4.0	4566.3	0.037
0.127021	6344.7	5.7	6612.6	-0.042
0.09441	6287.7	5.7	6441.5	-0.024
0.1126331	7501.4	6.0	7092.9	0.054

Calculated quantities with cooled water as a cooling medium Table 4.

Calculated quantities				
Cooled water		Air		Energetic balance deviation, [-]
Mass flow rate, [kg/s]	Cooling capacity,[W]	Temperature drop, [°C]	Cooling capacity,[W]	
0.14328	2940.3	2.6	2631.9	0.1049
0.189	3641.0	3.2	3618.8	0.0061
0.211	4683.4	3.8	4270.2	0.0882
0.285	6564.7	5.5	6040.2	0.0799
0.279	6426.5	5.6	6027.0	0.0622
0.3795	6993.1	6.1	6790.3	0.0290

Regarding the required increase in heat transfer area on cooled water operation, for the same temperature drop, calculations have been made, based on experimental results.

The air cooler surface area with cooled water as a cooling medium has been estimated based on the following equation, and the hypothesis that the overall heat transfer coefficient (k) may be considered as a quasi constant value; this hypothesis is acceptable taking into account that k is not primarily influenced by the cooling medium (ice-slurry or cooled water), but by the parameters on the air side. Not only that the air flow rate and the air temperatures were kept almost constant but the geometrical configuration of the tested air cooler also remained unchanged:

$$\frac{\dot{Q}_{is}}{\dot{Q}_{cw}} = \frac{k \cdot S_{is} \cdot \Delta T_m^{is}}{k \cdot S_{cw} \cdot \Delta t T_m^{cw}}$$

in which: $\dot{Q}_{cw}$ - cooling capacity with cooled water as a cooling medium, [W]; $\dot{Q}_{is}$ - cooling capacity with ice-slurry as a cooling medium, [W]; S_{is} / S_{cw} - heat transfer surface area required with ice-slurry / cooled water as a cooling medium, [m^2]; ΔT_m^{is} - logarithmic mean temperature difference with ice-slurry as a cooling medium, [°C]; ΔT_m^{cw} - logarithmic mean temperature difference with cooled water as a cooling medium, [°C].

Air cooler heat transfer surface are with cooled water as a cooling medium Table 5

Cooling capacity ice-slurry to air, $\dot{Q}_{is}$, [W]	Logarithmic mean temp. difference ice-slurry to air, ΔT_m^{is} [°C]	Cooling capacity cooled water to air, $\dot{Q}_{cw}$, [W]	Logarithmic mean temp. difference cooled water to air, ΔT_m^{cw}, [°C]	Heat transfer surface area, S_{cw}, [m^2]
3224.1	27.63	2631.9	17.69	9.53
3934.7	27.52	3618.8	17.82	10.61
4566.3	26.85	4270.2	17.05	11.00
6612.6	26.25	6040.2	15.87	11.29
6441.5	25.85	6027.0	15.70	11.50
7092.9	26.09	6790.3	16.12	11.57

Table 5 shows the calculated values of the air cooler heat transfer surface area with cooled water as a cooling medium.

5. Discussions

5.1. Regarding the Cooling Medium Flow Rate

The increase of the required cooled water flow rate for the same temperature drop on the air side, given the same heat transfer surface area, resulted based on the data shown in Tables 3 and 4. The percentage flow rate increase is illustrated by Figure 2, for both ice-slurry volumetric flow rates considered, as a function of the ice mass fraction.

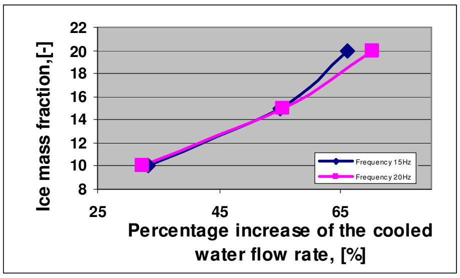

Fig. 2. *Water flow rate increase with respect to ice-slurry flow rate for the same air temperature drop*

Figure 2 shows an expected trend, namely: the increase in the ice mass fraction requires an increase of the cooled water flow rate, in order to compensate the ice latent heat of fusion. At the same time data plotted in Figure 2 show the non-dependence of the cooled water flow rate increase with the ice-slurry volumetric flow rate that is with the ice-slurry velocity over the coil cross section for all the ice fractions considered.

According to this experimental study the cooling effect on air of 10% ice fraction slurry requires an increase of about 30% in the cooled water flow rate, while the cooling effect of 15% ice fraction slurry requires an increase of about 50% in the cooled water flow rate, and 20% ice fraction require an increase of about 67% in the cooled water flow rate.

5.2. Regarding the Heat Transfer Cooling Surface Area

The increase of the required heat transfer cooling surface area for the same temperature drop on the air side resulted based on the data shown in Tables 3, 4 and 5. The percentage area increase is illustrated by Figure 3, for both ice-slurry volumetric flow rates considered, as a function of the ice mass fraction.

Figure 3 shows that as the ice mass fraction increases, the heat transfer surface area with cooled water needs to increase as well, in order to meet the condition of the same air temperature drop.

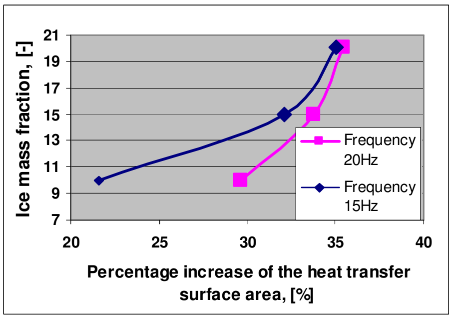

Fig. 3. *Heat transfer surface area increase, with respect to the ice-slurry surface area, for the same air temperature drop*

The explanation is mainly given by the logarithmic mean temperature difference, which takes significantly higher values under operating the air cooler with ice-slurry as a cooling medium, as compared to the cooled water option. The same Figure illustrates that the percentage increase of the air cooler surface area needs to take higher values as the volumetric flow rate increases. This trend is explained by the simultaneous increase of the ice quantity in the mixture.

According to this experimental study the same air cooling effect requires an increase of approx. 25% in the cooled water - air transfer surface area as compared to 10% ice-slurry - air transfer surface area; the increase goes up to approx. 33% if compared to 15% ice slurry as a cooling fluid and to approx. 35% if compared to 20% ice slurry.

5.3. Regarding Indoor Comfort Conditions

The direction followed by the cooled air as it washes the heat transfer surface area is different as a function of the surface temperature: approx. 0°C, if ice-slurry is used as a cooling medium and approx. 9°C, if cooled water is used as a cooling medium. For lower surface temperature, the air change in enthalpy is higher, for the same sensible temperature drop; as a consequence, the outlet air relative humidity is lower as compared to water-cooled air. Lower relative humidity is known as an indicator of indoor comfort. It is thus obvious the advantage of ice-slurry over cooled water as cooling medium. The above mentioned phenomenon is illustrated by the experimental data presented in Table 6.

Table 6

Frequency, [Hz]	Ice mass fraction, [%]	Cooling capacity (Ice-slurry – Air), [W]	Cooling capacity (Water – Air), [W]	Deviation, [-]
15	10	3224.1	2631.9	0.1837
20	10	3934.7	3618.8	0.0803
15	15	4566.3	4270.2	0.0648
20	15	6612.6	6040.2	0.0866
15	20	6441.5	6027.0	0.0644
20	20	7092.9	6790.3	0.0427

Experimental data in Table 6 show higher cooling capacities extracted from air by ice-slurry, in all the tested conditions. The explanation lies in the lower surface temperature provided by ice-slurry.

6. Conclusions

- *Regarding the cooling medium flow rate*

According to this experimental study, the cooling effect provided by 10 to 20% ice mass fraction slurry may be matched by an increase of approx. 30 to 67% in the cooled water flow rate.

- *Regarding the heat transfer cooling surface area*

This experimental study concludes that the heat transfer surface area needs to be enhanced by 25 to 35% if water is to be used as cooling medium for air with respect to the area needed in 10 to 20% ice fraction slurry as a cooling medium.

- *Regarding indoor comfort conditions*

As the cooling capacity of ice-slurry is higher, for the same air temperature drop, the outlet relative humidity of the air had lower values, which makes ice-slurry a better option regarding the indoor comfort. The laboratory size of the ice-slurry generator imposed R404A as primary agent used by the manufacturer, still as long as the refrigeration parameters are maintained (evaporating temperature mainly), the present results may be extended to the ammonia ice-slurry generating systems as well.As an overall conclusion, in spite of the higher energy consumption of the refrigerating system in ice-slurry case, it is the authors opinion that ice-slurry may be considered a valid option to cooled water in air-conditioning, based on the advantages brought by it, with respect to decreased required surface area, decreased tubes diameters, decreased flow rate and improved indoor comfort.

References

1. Horibe, A., Inaba, H., Haruki, N.: *Melting heat transfer of flowing ice slurry in a pipe*. In: S. Fukusako (Ed.), 4th Workshop on ice slurries, Osaka, Japan, pp.145-152, 2001.
2. Roy, S.K.,, Avanic, B.L.: *Turbulent heat transfer with phase change suspensions*.In: *Int. J.Heat Mass Transfer*, 44, p.2277-2285, 2001.
3. Stamatiou, E.; Kawaji, M.; Goldstein, V.: *Ice fraction measurements in ice slurry flow through a vertical rectangular channel heated from one side*. In: Proceedings of the Fifth IIR Workshop on Ice Slurries, Stockholm. Sweden, May 30-31, 2002.
4. Kawaji, M., Stamatiou, E., Hong, R., Goldstein, V.: *Ice slurry flow and heat transfer characteristics in vertical rectangular channels and simulation of mixing in a storage tank.* In: Proceedings of the 4th IIR Workshop on Ice Slurries, Osaka, Japan, 2001.
5. ASHRAE Final Reports 1166: *Behaviour of Ice Slurries in Thermal Storage Systems. Principal Investigator: Michael Kauffeld*. Danish Technological Institute; Conducted: April 2000-Oct. 2001, published in 2002.
6. Paul, L., Malter, et al.: *Generation and Utilization of Liquid Ice (binary ice)*. In: I.I.R. 20th International Conference on Refrigeration into the Third Millennium, Sept. 19-24, 1999, Sydney, Australia,1999.
7. Dumitrescu, R,, Ilie, A, Chiriac, F.: *Annual report 2007*. Grant of the National Council of University Scientific Research;
8. Dumitrescu, R., Ilie, A., Chiriac, F.: *Annual report 2008*. Grant of the National Council of University Scientific Research.