An Evaporative Style for Dynamic Ice-Slurry Producing with Super-Cooled Water

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ABSTRACT

Ice-slurry is an ideal form of ice product for ice-storage, which is an effective measure to save energy. The conventional producing method using super-cooled water suffers the troubles of instability and heavy dependence on electronic power. We propose a novel method to improve that: spraying water is super-cooled by evaporating in a chamber where the air is of low humidity and low wet bulb temperature; a refrigeration cycle and an inlaid liquid-dehumidifying unit are adopted to recover the air to run the cycle; the rejected heat from the condenser is used to drive the liquid dehumidifying unit. System and performance analysis were made. The results indicate that compared to the conventional super-cooled water system, this new system could avoid the ice block, and shoulder off the heavy dependence on the electric power while raise the efficiency by a maximum value of 98% under certain conditions.

1. INTRODUCTION

Ice has been widely used in many fields of today's life such as food reserving, air conditioning and industrial cooling. Particularly, ice-storage method is an effective and meaningful measure to save energy for refrigeration and air-condition systems, which have added a heavy load on the energy providing system of our modern society. Ice-slurry, the mixture of ice particles and some certain liquid, is considered as an ideal form for ice-storage because it features with good thermal properties and a flowing characteristic.

Producing ice-slurry with super-cooled water is a hotspot research in the ice-slurry producing field. As described by the work of Kozawa *et al* (Kozawa *et al*., 2005), the water is firstly being cooled to the super-cooled region and then sent to a unit called the super-cooling releaser where the super-cooled state is broken and the ice forms. However, this technology have to tolerate the troubles of instability in that the super-cooled water may change to ice before it enters the super-cooled releaser as expected, which will lead to the ice-block and suspend the whole process.

As we know, the water will evaporate while its vapor pressure is higher than that of the surrounding atmosphere. The saturation vapor pressure of water is corresponding to its temperature and it is around 600 Pa at the temperature of 0ºC. Therefore, while the vapor pressure of the atmosphere is below 600 Pa, the water will keep on evaporating until its saturation vapor pressure is equivalent to the vapor pressure of the atmosphere, and the temperature of water will correspondingly fall below 0ºC (temperature at the triple point of water). In other words, the water could be super-cooled this way and change to ice. And that's the principle for the evaporation-freezing method (Kim *et al*., 2001).

Actually, it is not necessary to vacuum for a low water vapor pressure environment. Since the water vapor pressure of the atmosphere is bound to its humidity, the low water vapor pressure could be got through decreasing the humidity. The water vapor pressure of 600Pa is related to the humidity of 4.5g/ (kg air). Isao Satoh et al proposed an evaporation-freezing system by utilizing the waste cold from the gasification process of liquefied natural gas (LNG) to remove the moisture from the air in order to decrease the humidity. Then the water vapor pressure is reduced and the water turns to ice (Satoh *et al*., 2001).

Summarizing all of these works, we propose a novel system: the whole system includes a liquid-dehumidifying unit, which could provide air of very low humidity (much less than 4.5g/ (kg air)), so does the vapor pressure. Therefore, in an evaporation-freezing manner, water could be super-cooled by spraying in this low vapor pressure air rather than be super-cooled within a tube in a conventional super-cooled water manner. In that way the ice-block could be avoided. Beyond the humidity, the bulb temperature of the air should be maintained, so a refrigeration cycle is adopted as well as a liquid-dehumidifying unit. Besides using the cooling capacity of this cycle, the rejected heat from the condenser is reutilized to fuel the regeneration process for the liquid-dehumidifying unit. Therefore, the heavy dependence on electric power could be alleviated and the performance of the whole system could be greatly uplifted.

2. SYSTEM DESCRIPTION

The whole system consists of different parts and they interact with each other closely, which makes it hard and obscure to understand. To overcome this problem, we propose the description of the system evolution here in order to make it clear. This evolution process could be divided into four stages.

2.1 Stage I

Shown in Figure 1, water is pumped from the water tank (the temperature could be 20ºC) to the "Evaporation Icing chamber" and then sprayed there; meanwhile, low humidity air is introduced to this chamber, which later produces an appropriate environment where the wet bulb temperature is maintained below -3ºC (the humidity is very low, but still, the air bulb temperature should be low enough to maintain a low wet bulb temperature).

The sprayed water droplets continue evaporating in this chamber due to the water vapor pressure difference between its saturation layer and the environment full of low humidity air, whose corresponding water vapor pressure is very low. As the evaporation process going, the temperature of the droplets falls quickly (Strub *et al*., 2003). As long as droplets temperature falls below that of the surround air, there is the heat flux from the environment to the droplets, which causes a temperature rising tendency. When this two process comes to be a balance, the droplets could be super-cooled and its temperature may approach -3ºC, which equals to the wet bulb temperature.

The humidity of the air leaving the chamber is enhanced as the moisture added during the evaporation process; while its temperature may have a tiny raise because the sensible heat transfer is overshadowed during evaporation process and the mass of the air is set much larger than the water.

Therefore, after this evaporation process, the sprayed water is competent for a traditional ice-slurry producing method with super-cooled water of around -2ºC. Following exactly the conventional super-cooled water manner, the super –cooled water droplets will release its super-cooled state and changes to ice particles in a super-cooled water releaser situating at the bottom of the chamber. Some water droplets have been turned while some have not. So, an ice-water separator is adopted to separate the water and the ice. Surplus water is sent back to the water tank for

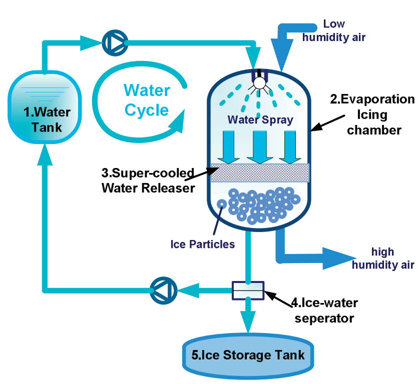

Figure 1: Stage I of the system

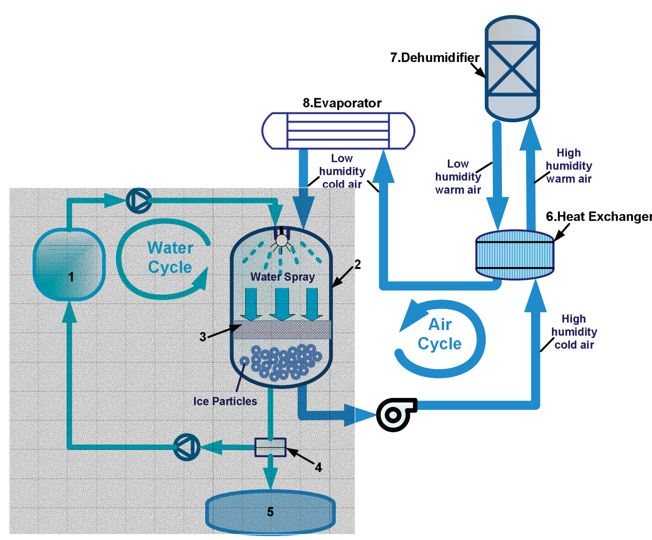

Figure 2: Stage II of the system

reusing, which forms the water cycle; ice could be stored in an ice storage tank and later made to be ice-slurry.

2.2 Stage II

As shown in Figure 2, Stage I is shadowed so that the new added parts could be highlighted. Mentioned before, the temperature and the humidity of the outlet air should be reduced in order to run a cycle. So, a dehumidifier, which adopts the liquid desiccant, is responsible for the dehumidification task; while an evaporator is assigned for the cooling load.

The air leaving the chamber should be dehumidified first and then sent to the evaporator for cooling. But actually, during the dehumidification process, what happens is not only the decrease of the humidity through mass transfer, but also the temperature of the air will increase if the air directly contacts with the liquid desiccant (the temperature of the outlet air may be much lower than that of the liquid desiccant without possible exceptional cooling measures, so there exists the heat transfer from the liquid desiccant to the air). If it goes that way, there must be a surplus cooling load brought in from the dehumidifier. To solve this problem, we suggest a heat exchanging before the air from the chamber entering the dehumidifier. A heat exchanger (Figure 2) takes the task of countercurrent heat exchanging between the air leaving the chamber and the air leaving the dehumidifier. In an ideal situation, after the heat exchanging, the temperature of the air leaving the chamber will be equal to the temperature of the air just leaves the dehumidifier while the temperature of the air leaving the dehumidifier will fall to that of the air just leaves the evaporation icing chamber. By this way, the surplus cooling load coming from the dehumidifier could be eliminated. Later, it only calls for the cooling load introduced by the evaporation process, a task left to the evaporator.

2.3 Stage III

Based on Stage II, Stage III is founded and the adding part is also highlighted in Figure 3 by shadowing Stage II. Naturally, the evaporator is one part of a refrigeration cycle, which is presented in this figure. It should be noticed that the energy input module stands for the uncertainty, as well as the flexibility of this refrigeration cycle: this cycle could be a vapor compression cycle driven by electric power, absorption cycle driven by solar energy or adsorption system driven by waster heat.

2.4 Stage IV

Grounded with Stage II and Stage III, the last stage, Stage IV is shown in Figure 4. As mentioned in Stage II, there's a dehumidifier using liquid desiccant responsible for reduce the humidity of the air.

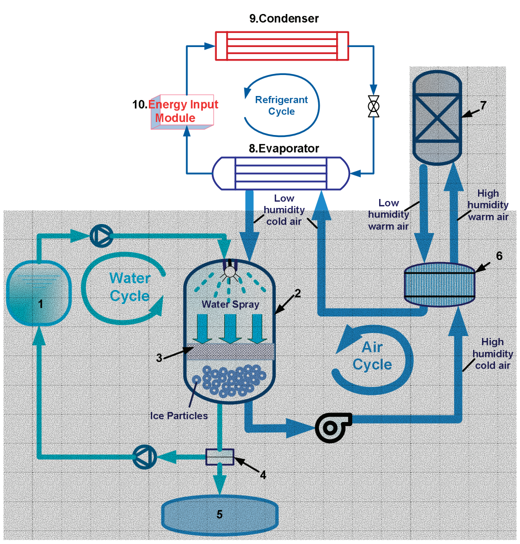

Figure 3: Stage III of the system

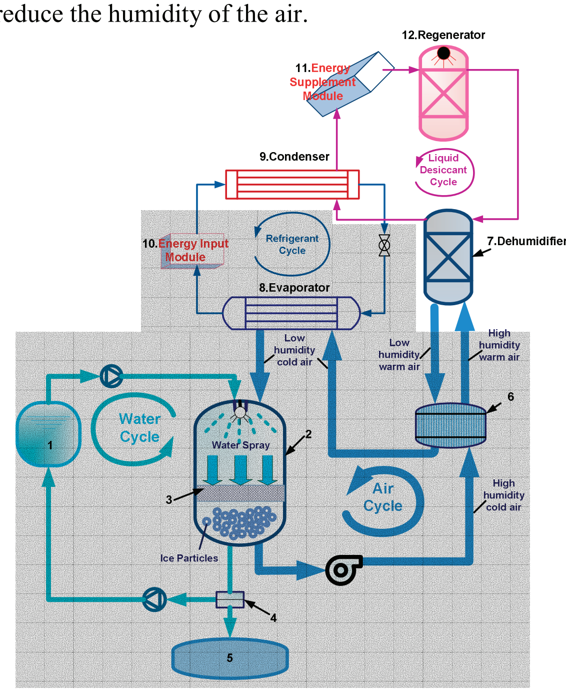

Figure 4: Stage IV of the system

When dehumidifying, the strong liquid desiccant solution absorbs the moisture from the air and becomes weak solution. So the desiccant should be regenerated and recovered to strong solution again. Therefore, a regenerator is added in this stage and highlighted in Figure 4. There must be heat source to run this cycle; we reutilized the rejected heat from the condenser to do this job. Moreover, considering the possible situation of insufficiency, an accessional energy supplement module is provided in this design. This module also stands for the uncertainty and flexibility in the Figure 4, as it could be solar energy, waste heat, geothermal heat or the others, as long as it could provided more energy for driving the dehumidifying cycle beyond the rejected heat gained from the condenser.

3. PERFORMANCE ANALYSIS

3.1 Conventional Super-cooled water System

For a conventional ice-slurry producing system with super-cooled water, the total cooling load for producing ice particles could be expressed as: (Δt means the super-cooled degree)

$$Q_0 = \dot{m}_{ice}(t_{inw} + \Delta t)C_w = \dot{m}_{sc}(t_{inw} + \Delta t)C_w \tag{1}$$

Suppose the driving energy for the refrigeration cycle is E with a certain COP (coefficient of performance). Since the cooling load is known, it has the form as:

$$E = \frac{Q_0}{COP} = \frac{\dot{m}_{sc}(t_{inw} + \Delta t)C_w}{COP} \tag{2}$$

We define CIPP (Coefficient of Ice Producing Performance) to evaluate the ice producing performance. The expression is:

$$CIPP = \frac{\dot{m}_{ice}}{E} = \frac{\dot{m}_{sc}}{E} \tag{3}$$

Substitute (2) to (3), we get:

$$CIPP = \frac{COP}{(t_{inw} + \Delta t)C_w} \tag{4}$$

3.1 Evaporative Style Super-cooled water System

Considering the enthalpy change in system evolution Stage I, we could obtain:

$$\Delta H = \dot{m}_{sc}h_{outw} + \dot{m}_{evp}h_{outevp} - (\dot{m}_{sc} + \dot{m}_{evp})h_{inw} \tag{5}$$

We use the absolute value of this enthalpy change and equation (5) could be further expressed as:

$$\Delta H = \dot{m}_{sc}C_w\left(T_{inw} - T_{outw}\right) + \dot{m}_{evp}\left(C_wT_{inw} - C_w{}'T_{outa}\right) \tag{6}$$

This enthalpy change consists of two parts: one is the change caused by the sensible heat transfer and the other is laid in the latent heat transfer because of the evaporation. The enthalpy change related to sensible heat transfer is:

$$\Delta H_s = \Delta H \text{-} l_w\dot{m}_{evp} \tag{7}$$

Integrating (6) and (7) together, we have:

$$\Delta H_s = Q_0{}' = \dot{m}_{sc}C_w\left(\Delta t + t_{inw}\right) + \dot{m}_{evp}\left[C_w\left(t_{inw} + 273.15\right) - C_w{}'\left(t_{outa} + 273.15\right) - l_w\right] \tag{8}$$

Suppose the driving energy for the refrigeration cycle is E' with COP' (coefficient of performance). By applying (8), it could be obtained that:

$$E' = \frac{Q_0'}{COP'} = \frac{\dot{m}_{sc} C_w (\Delta t + t_{inw}) + \dot{m}_{evp} \left[C_w (t_{inw} + 273.15) - C_w' (t_{outa} + 273.15) - l_w \right]}{COP'} \tag{9}$$

Moreover, the rejected heat   from the condenser of this refrigeration cycle could be expressed by integrating (8) and (9) together:

$$Q_a' = \left(\frac{COP'+1}{COP'} \right) \left\{ \dot{m}_{sc} C_w (\Delta t + t_{inw}) + \dot{m}_{evp} \left[C_w (t_{inw} + 273.15) - C_w' (t_{outa} + 273.15) - l_w \right] \right\} \tag{10}$$

During dehumidification process, this moisture mass will be transplanted into the liquid desiccant and reduce its concentration. Therefore, the liquid desiccant should be regenerated to recover its dehumidifying power by losing the same moisture mass to the atmosphere. Neglect sensible heat loss, the heating load demanded for regeneration could be expressed as: $Q_a'' = \dot{m}_{evp} l_w$.

We bear in our mind that in which condition the rejected heat is enough and capable for the needs of regeneration. When $Q_a' = Q_a''$ we get:

$$\dot{m}_{sc} C_w (\Delta t + t_{inw}) = \dot{m}_{evp} \left[\left(\frac{COP'}{COP'+1} + 1 \right) l_w + C_w' (t_{outa} + 273.15) - C_w (t_{inw} + 273.15) \right] \tag{11}$$

That is:

$$\frac{\dot{m}_{evp}}{\dot{m}_{sc}} = \frac{(\Delta t + t_{inw})}{\left[\left(\frac{COP'}{COP'+1} + 1 \right) 595.238 + (t_{outa} + 273.15) 0.438 - (t_{inw} + 273.15) \right]} \tag{12}$$

, when $\frac{\dot{m}_{evp}}{\dot{m}_{sc}}$ is smaller than the right part of equation (12), $Q_a' > Q_a''$: the rejected heat is enough, when $\frac{\dot{m}_{evp}}{\dot{m}_{sc}}$ is bigger than the right part, $Q_a' < Q_a''$: the rejected heat is not enough, and there should be other heat supplemented, and this heat could be sized as $\left(Q_a'' - Q_a' \right)$.

So there exist two situations: (1) rejected heat is enough and (2) rejected heat is insufficient:

In Situation (1):

$$E_1' = \frac{Q_0'}{COP'} = \frac{\dot{m}_{sc} C_w (\Delta t + t_{inw}) + \dot{m}_{evp} \left[C_w (t_{inw} + 273.15) - C_w' (t_{outa} + 273.15) - l_w \right]}{COP'} \tag{13}$$

$$CIPP_1' = \frac{\dot{m}_{sc}}{E_1'} = \frac{COP'}{C_w (\Delta t + t_{inw}) + \frac{\dot{m}_{evp}}{\dot{m}_{sc}} \left[C_w (t_{inw} + 273.15) - C_w' (t_{outa} + 273.15) - l_w \right]} \tag{14}$$

In Situation (2):

$$CIPP_2' = \frac{\dot{m}_{sc}}{E_2' + \Delta Q} = \frac{\dot{m}_{sc}}{E_2' + \left(Q_a'' - Q_a' \right)} \tag{15}$$

$$E_2' = \frac{Q_0'}{COP'} = \frac{\dot{m}_{sc} C_w (\Delta t + t_{inw}) + \dot{m}_{evp} \left[C_w (t_{inw} + 273.15) - C_w' (t_{outa} + 273.15) - l_w \right]}{COP'} \tag{16}$$

$$CIPP_2' = \frac{1}{-C_w (\Delta t + t_{inw}) + \left(\frac{\dot{m}_{evp}}{\dot{m}_{sc}} \right) \left[-C_w (t_{inw} + 273.15) + C_w' (t_{outa} + 273.15) + 2 l_w \right]} \tag{17}$$

3. RESULTS AND DISCUSSION

Performance analysis is made to compare the conventional method and this new method using the parameters listed in Table.1.

We define: $\Delta CIPP = \dfrac{CIPP_1'(or CIPP_2') - CIPP}{CIPP} \times 100\%$ to represent the performance change, subscript "1" and "2" stands for Situation (1) and Situation (2) discussed before; and we make $\Delta E = \dfrac{E_1'(or E_2') - E}{E} \times 100\%$ to represent the input energy (for the refrigeration cycle) change, subscript "1" and "2" have the same meanings.

For Fig.5, the X axis stands for $\dfrac{\dot{m}_{evp}}{\dot{m}_{sc}}$, while Y stands for both $\Delta CIPP$ and ΔE . The colorful points and the black points respectively stand for $\Delta CIPP$ and ΔE . With $\dfrac{\dot{m}_{evp}}{\dot{m}_{sc}}$ rising, $\Delta CIPP$ first increases and then falls. The highest performance enhancement is about 98%. It reaches this maximum value (this point is marked) when $\dfrac{\dot{m}_{evp}}{\dot{m}_{sc}}$ is equal to the right part expressed in Formula (12), which is exactly the boundary for judge between Situation (1) and Situation (2). $\Delta CIPP$ increases when $\dfrac{\dot{m}_{evp}}{\dot{m}_{sc}}$ is lower than that related to the maximum point, because system operates in Situation (1); $\Delta CIPP$ decreases when $\dfrac{\dot{m}_{evp}}{\dot{m}_{sc}}$ is bigger than that related to the maximum point, because system operates in Situation (2). In Situation (2), more additional energy (solar energy or waster heat) is needed to regenerate the liquid desiccant and its performance turns down.

ΔE increases with $\dfrac{\dot{m}_{evp}}{\dot{m}_{sc}}$ rising. The rising $\dfrac{\dot{m}_{evp}}{\dot{m}_{sc}}$ means there are more water evaporated to shoulder the cooling load, which leads to a decreasing demand of cooling load for the evaporator, and consequently reduces the energy injected to the refrigeration cycle.

In Fig.6, the colorful points and the black points respectively stand for $\Delta CIPP$ and ΔE ,related to different maximum performance enhancement points obtained under different COP'. When COP' is lower, both $\Delta CIPP$ and ΔE decrease; and the corresponding $\dfrac{\dot{m}_{evp}}{\dot{m}_{sc}}$, related to the maximum value of $\Delta CIPP$ under certain COP', increase with the COP' growing. That indicates that if an absorption or adsorption refrigeration cycle (they have a lower COP, but call for much less electric power and do not produce pollution) is adopted in our system, we could still gain a performance

Table 1: Parameters and their values

Parameters	Values
Δt	2ºC
t_{inw}	20～25ºC
t_{outa}	0ºC
COP	2.5
COP'	1~2.5
C_w	4.2 (kJkg^{-1}K^{-1})
C_w'	1.84 (kJkg^{-1}K^{-1})
l_w	2500 (kJkg^{-1}K^{-1})

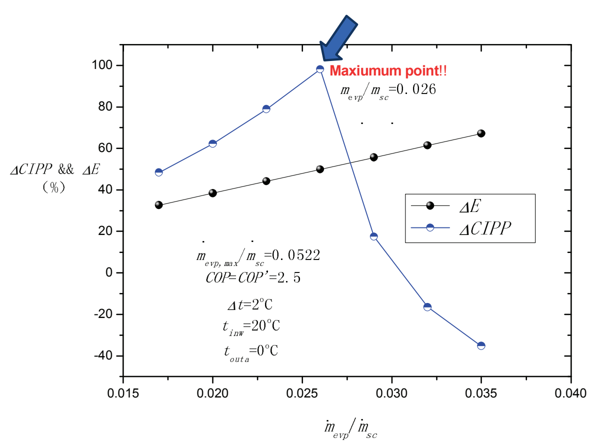

Figure 5: Performance Variation with $\dfrac{\dot{m}_{evp}}{\dot{m}_{sc}}$

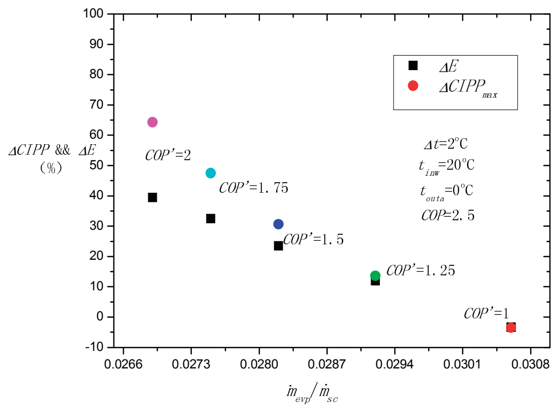

Figure 6: Maximum value varying with COP

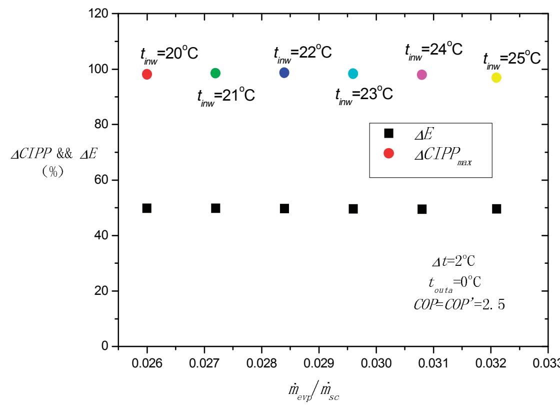

Figure 7: Maximum value varying with t_{inw}

enhancement (if COP' is above 1) under certain working conditions. In Fig.7, the colorful points and the black points respectively stand for $\Delta CIPP$ and ΔE , related to different maximum performance enhancement points obt ained with different initial temperature of the water entering the chamber. Both $\Delta CIPP$ and ΔE stay almost unc hanged. $\frac{\dot{m}_{evp}}{\dot{m}_{sc}}$, related to the maximum performance enhancement becomes larger with the rising temperature. T hat result provides a way for making the system operate with best performance by regulating the initial tempera ture of water.

It could be summed up: there will be a proper $\frac{\dot{m}_{evp}}{\dot{m}_{sc}}$ for reaching the maximum performance enhancement and its value is judged by some operation parameters; compared to a conventional super-cooled water system, performance of producing ice could be improved by making the system operate in Situation (1) in this new system; if system operates under Situation (2), it may have much lower performance but the heavy dependence on electric power could be shouldered off.

3. CONCLUSIONS

This evaporative style ice-slurry producing system is different from the conventional super-cooled water method in that it makes super-cooled water out of the tube, and consequently avoids the possible ice block.
Both the cooling capacity and rejected heat of the refrigeration cycle are utilized, which improve the performance of the whole system.
To reveal the best working condition and manipulate the process of evaporation super-cooling calls for more effort in the future, not only the experiments, but also the theoretical analysis.

NOMENCLATURE

C_w	specific heat of water	($kJkg^{-1}K^{-1}$)
C'_w	specific heat of water vapor	($kJkg^{-1}K^{-1}$)
CIPP, CIPP'	coefficient of ice production performance	
COP, COP'	coefficient of performance	
E, E'	input energy	(kJ)
h	specific enthalpy	($kJkg^{-1}K^{-1}$)
H	enthalpy	(kJ)
l	latent heat of evaporation	($kJkg^{-1}K^{-1}$)

Subscripts	
a	air
evp	evaporation
in	inlet
out	outlet
sc	super-cooled
w	water

m	mass flow rate	(kgs^{-1})
Q_0, Q'_0	cooling load	(kJ)
Q_a, Q'_a, Q''_a	rejected heat from condenser	(kJ)

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