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Performance Enhancement and Life-Cycle Cost Savings of Supercooled Water Ice Slurry Generation Systems Using Heat Regeneration

Chong Zhang [1], Luwei Yang [1], Wenye Lin [2,3,*], Juan Wei [1] and Fengjun Guo [4]

[1] Technical Institute of Physics and Chemistry, Chinese Academy of Sciences, Beijing 100190, China; zhangchong@mail.ipc.ac.cn (C.Z.); lwyang2002@mail.ipc.ac.cn (L.Y.); weijuan@mail.ipc.ac.cn (J.W.)
[2] Fraunhofer Institute for Solar Energy Systems ISE, 79110 Freiburg, Germany
[3] Sustainable Buildings Research Centre (SBRC), University of Wollongong (UOW), Wollongong 2522, Australia
[4] National Engineering Research Centre for Agricultural Products Logistics, Jinan 250103, China; 20130751@sict.edu.cn
* Correspondence: wenye.lin@ise-extern.fraunhofer.de

Abstract: This paper presents the development and utilisation of a heat regeneration approach to enhancing the performance and reducing the life-cycle cost of supercooled water ice slurry generation systems. Two supercooled water systems with direct and indirect ice slurry generation were enhanced by the heat regeneration approach to avoid excessive cold loss and increase the supercooling degree, thereby improving system efficiency while reducing operational costs. Their respective performance and life-cycle costs were experimentally evaluated and compared to the ones without heat regeneration enhancement under different working conditions, as well as to a conventional scraped surface ice slurry generator used as a benchmark. It was found from the comparative investigation that the heat regeneration approach can effectively reduce the water temperature at the inlet of the supercooler, allowing a significant amount of cold loss to be saved for ice slurry generation. The effective utilisation rate of cold can be effectively improved by over 15% when using the heat regeneration approach, and the unit ice mass power consumption can be reduced by 12–20%. Due to the attractive energy-saving potential, the operational cost-effectiveness of the enhanced systems contributed to cut-down of life-cycle cost. It was found that the life-cycle costs of the enhanced direct and indirect ice slurry generation systems were 62.0% and 74.7% lower than that of the conventional scraped surface ice slurry generator, respectively.

Keywords: supercooled water; ice slurry; heat regeneration; performance enhancement; life-cycle cost analysis

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1. Introduction

Over the last several decades, a drastic increase in world energy consumption and the associated environmental deterioration and global warming have aroused wide and profound public concern [1]. To address these energy and environmental problems, various solutions from deepening the penetration of renewable energy (i.e., supply-side) to improving the energy efficiency of energy-using systems (i.e., user-side) have been developed and applied [2,3]. As an essential connection bridging these solutions between the supply and user sides, energy management has been attracting increasing attention for the rationalisation of energy utilisation [4,5], for whose fulfillment, efficient energy storage and transportation are indispensable [6,7].

In terms of energy management involving thermal energy, phase change materials (PCMs) have been widely recognised as efficient and effective media to fulfill thermal energy storage and transportation [8,9]. Ice, with the features of high heat of fusion and easy availability, has been applied in thermal energy storage of various sectors such as cold chain and air conditioning. In addition to its role in thermal energy storage media, an ice–water (or aqueous solution) mixture slurry can serve as a heat transfer fluid for thermal energy transportation. Compared to other PCM slurries (e.g., phase change emulsions), ice slurry was the first being industrialised and widely deployed thanks to its advantages of low cost, high efficiency, large thermal capacity, and good pumpability [10–12]. Recent research on ice slurry mainly focuses on its properties and performance characterisation (including heat transfer and flow), ice slurry generation, and practical applications. Among them, the generation of ice slurry is one of the primary key topics in any ice slurry systems. It is a hotspot in research regarding ice slurry, as the cost of current ice slurry generators accounts for a major proportion, which impedes the deeper application of this promising technology [13]. Currently, there are various technologies for ice slurry generation, such as supercooled water technology, scraped surface technology, vacuum technology, and refrigerant direct contact technology [14,15]. Scraped surface technology is one of the most popular methods, usually relying on the rotation of scraper blades within heat exchanger tubes to break the ice layer grown on the inner surface for ice slurry generation. This feature enables the easy control of the ice fraction in ice slurry; however, it tends to require custom evaporator design and lead to additional mechanical consumption, which typically causes additional energy consumption by around 10–20% compared to that of the supercooled method [13,16]. The supercooled water method utilises the supercooling feature of water, which can be cooled by a few degrees below the normal freezing point without crystallizing; then, by agitating the supercooled water to release the supercooled degree, ice crystals can be generated [17]. Compared to other technologies, supercooled water technology boasts of simplicity, low energy cost, and wide adaptability, and is suitable for large-scale air conditioning systems. More details about different ice slurry generation technologies can be found in Satio [10], Kauffeld et al. [13], and Kauffeld and Gund [18].

Recent research regarding supercooled water ice slurry generation technology usually focuses on avoiding ice blockage, development of control mechanisms, investigation of supercooling degree-releasing methods, and optimal system design. Ice blockage is likely to occur in the supercooling heat exchanger, which can lead to discontinuous ice slurry generation and a drop in system efficiency [19]. The occurrence of ice blockages is tightly associated with nucleation and thus affected by many factors, including but not limited to supercooling degree, water flow velocity, insoluble particles, and wall roughness. It has been reported that the supercooling degree is the most important factor, the increase of which contributes to system capacity and controllability while improving the probability of crystallization and decreasing the risk of blockage [20]. Tanino and Kozawa [21] experimentally studied the acceptable range of supercooling degree while considering the system efficiency. They summarised the optimal supercooling range from 1 to 2 °C, which can simultaneously avoid ice blockage and maintain decent system efficiency. As the maximum acceptable supercooling degree can be increased with a low wall roughness, surface modifications have been studied and utilised. For instance, Wang et al. [22] studied the water contact angle of stainless steel and nanocomposite fluorocarbon coatings. It was found that the water contact angle increased from 84 to 163° and the maximum supercooling degree in the coated supercooling heat exchanger reached around 1.7 °C compared to that of 0.9 °C in the uncoated supercooling heat exchanger. Furthermore, insoluble particles in water influenced the supercooling degree. Okawa [23] used the silver iodide as the foreign particles to study the nucleation characteristics under different foreign particle concentrations, and found that the crystallisation temperature depends on the total surface area of particles. In particular, the existence of tiny ice crystal seeds in the source water is one of the most common reasons for ice blockage in a supercooled water ice slurry generation system. In practice, preheating (usually by 0.5 °C) is of great necessity to remove ice crystals before beginning another supercooling cycle [14]; however, this preheating no doubt is a source of energy loss.

Releasing the supercooling degree is a necessary process for ice slurry generation when using supercooled water technology, and has also been extensively investigated.

Collison and stirring are among the main traditional methods for releasing the supercooling degree [17,22]. An alternative method is ultrasonic vibration; many studies have confirmed its advantages, including good uniformity and rapid response of crystallisation. For instance, Inada et al. [24] studied the performance of supercooling degree releasing using ultrasonic vibrations under 28 kHz, and found that the ice crystals could be generated under a lower supercooling degree compared to the other traditional methods. Zhang et al. [25] investigated and confirmed the relationship between cavitation bubbles induced by ultrasonic waves and the effect of supercooled state release. Hozumi et al. [26] experimentally compared ice slurry generation under the same frequency with different ultrasonic vibration powers, and the results showed that higher power contributed to the effect of supercooling degree release.

Due to the importance of establishment and releasing of supercooling degree in the ice slurry generation process using supercooled water technology, much effort has been assigned from the system aspect to stabilise the supercooling status while improving its efficiency. The indirect ice slurry generation system was first studied to enhance the stability of the cold source for ice slurry generation, and various indirect ice slurry generation systems have been developed and applied in practical air conditioning systems and cooling systems for different purposes [13,27,28]. To simplify the system and reduce the overall heat transfer temperature difference, direct ice slurry generation systems have been developed in recent years, and a number of solutions at the system level have been proposed to ensure its stability. For instance, Tanino and Kozawa [21] developed a direct ice slurry generation system with an ice concentrator that could produce ice slurry with a concentration of 5–10%, which enabled the enrichment of ice concentration based on ice slurry flow with low ice concentration, thereby avoiding ice blockage. Wang developed a direct ice slurry generation system with double supercooling heat exchangers for continuous ice slurry generation; these two supercooling heat exchangers can be switched when an ice blockage happened in one of them. However, these solutions introduce extra costs and may compensate for the benefits of direct ice slurry generation systems. Therefore, an in-depth comparison between direct and indirect ice slurry generation systems along with a comparison to other conventional ice slurry generation systems is of value.

From the above discussions, it can be seen that many questions remain to be addressed in supercooled water ice slurry generation, such as energy-saving in preheating process, quantitative performance assessment between direct and indirect ice slurry generation systems, and economic benefit enhancement compared to conventional ice slurry generation systems (e.g., scraped surface ice slurry generators). In this study, a regeneration cycle was proposed to effectively reduce the energy loss during the preheating of supercooled water ice slurry generation. Ice slurry generation systems with heat regeneration enhancement were developed, and their performance and life-cycle cost were experimentally investigated and assessed in comparison with a conventional scraped surface ice slurry generator. It is an innovative idea to apply heat regeneration for preheating the return water in ice slurry generation systems, thereby improving the system performance and economic benefits. The following parts of this paper are organised as follows: Section 2 details the research methodology in this study; specifically, Sections 2.1 and 2.2 introduce the principle of the heat regeneration enhancement and the experimental system, respectively. Section 2.3 states the experiment plan and procedure, while Section 2.4 presents the development of the key performance indicators. The setup of the economic analysis is detailed in Section 2.5. Section 3 presents and discusses the performance results of ice slurry generation systems with and without heat regeneration enhancement (Sections 3.1 and 3.2, respectively) and the results of the economic analysis. Finally, useful conclusions of this research are summarised in Section 4.

2. Research Methodology

The overall research methodology used in this study is presented in Figure 1. First, we established two flexible experimental platforms to allow switching between different ice slurry generation systems with and without heat regeneration enhancement. Based on these platforms, the working conditions and energy performance of ice slurry generation systems influenced by the ambient temperature and with/without a secondary coolant loop (i.e., corresponding to direct and indirect ice slurry generation systems) were initially evaluated through a series of preliminary tests. Then, the proposed heat regeneration approach was used with the ice slurry generation systems, the performance enhancement of which was further quantified and investigated in comparison to the results of preliminary tests on the systems without heat regeneration enhancement. Eventually, in order to achieve a clear perspective of the economic benefits of heat regeneration enhancement, the life-cycle cost of ice slurry generation systems with heat regeneration enhancement was assessed by comparison to a conventional scraped surface ice slurry generator used as a benchmark.

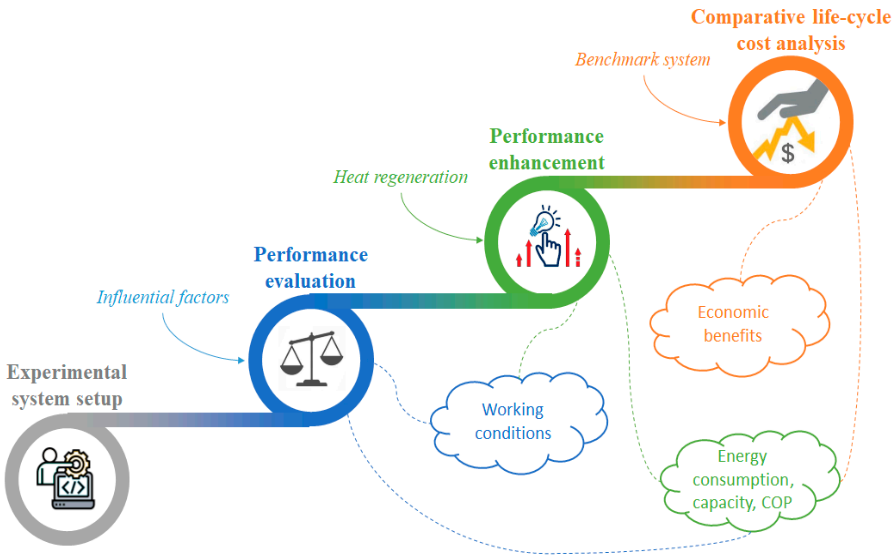
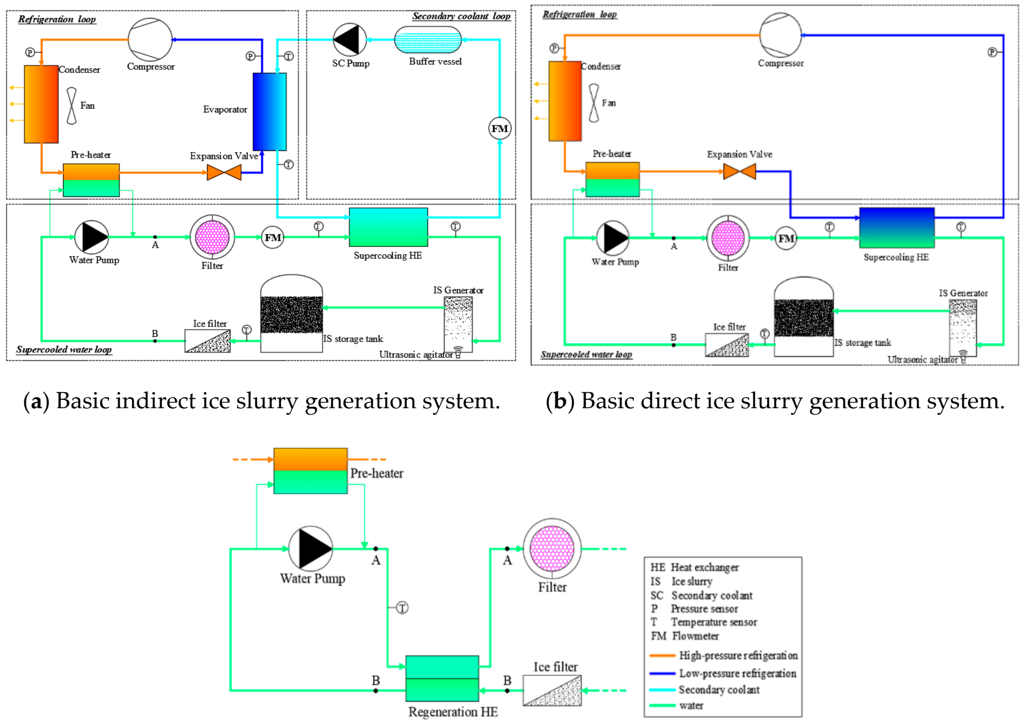


Figure 1. Sketch of the overall research methodology.

2.1. Heat Regeneration for Enhanced Ice Slurry Generation

The integration of heat regeneration for enhanced ice slurry generation is presented in Figure 2, with an indirect ice slurry generation system used as the baseline in this example. The basic indirect ice slurry generation system included three sections: a supercooled water loop for ice slurry generation, a secondary coolant loop for cold transportation, and a refrigeration loop for cold generation, as shown in Figure 2a. The supercooled water loop mainly consisted of a water pump, a supercooling heat exchanger (i.e., supercooler), an ultrasonic agitator, and an ice slurry storage tank. The water is supercooled when pumped through the supercooling heat exchanger, then stimulated by the ultrasonic agitator to generate ice slurry. The generated ice slurry is stored in the ice slurry storage tank, where the ice fraction can be improved by separating and cycling the carrier water. It is worth mentioning that in order to ensure the occurrence of supercooling within the supercooling heat exchanger, mechanical separation of the carrier water from the ice particles is usually not enough, and a pre-heating process is necessary in order to fully eliminate the ice crystal nucleates. Pre-heating was fulfilled by passing a portion of the carrier water through a preheater installed between the condenser and compressor, from which the exhaust heat from the refrigerant is used as the heat source. The pre-heating degree (typically 0.5 °C) is then controlled by mixing the heated bypassing carrier water flow with the main carrier water flow. Pre-heating can contribute to heat rejection, which improves the efficiency of the refrigerant loop. The direct ice slurry generation system included two sections (see Figure 2b): a refrigeration loop and a supercooled water loop. The supercooling heat exchange occurs directly between the refrigeration unit and the water for supercooled water generation. The other processes of the supercooled loop and refrigeration loop in the direct ice slurry generation system were similar to the indirect ice slurry generation system.



(**a**) Basic indirect ice slurry generation system. (**b**) Basic direct ice slurry generation system.

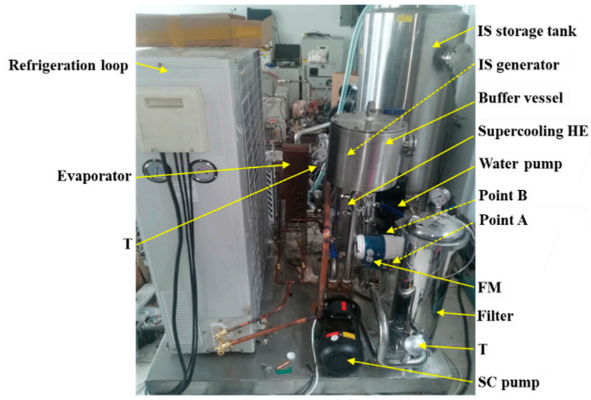
(**c**) Enhancement of heat regeneration

Figure 2. Integration of heat regeneration in the indirect and direct ice slurry generation systems. (A and B are the interface points for integrating the regenerator.)

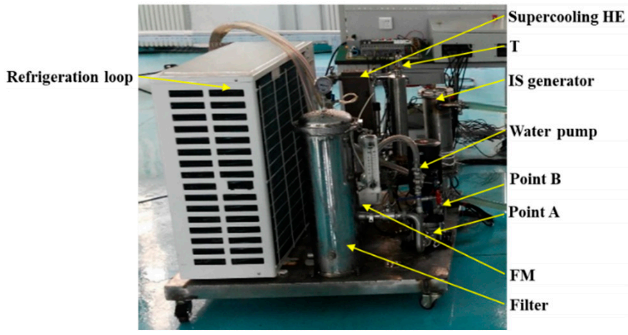
Heat regeneration was achieved by integrating a regeneration heat exchanger (i.e., regenerator) in the water loop of the basic ice slurry generation system at points A and B (see Figure 2a,b), coupling the carrier water flow before and after preheating and mixing, as highlighted in Figure 2c. The original carrier water from the storage tank and filter is first heated to around 0.25 °C when directed through the heating side of the regenerator, followed by further preheating to 0.5 °C using the preheater. The preheated water is then drawn through the cooling side of the regenerator to exchange heat with the original carrier water passing through the heating side. Compared to the basic ice slurry generation system with preheating only, with the assistance of heat regeneration enhancement the water temperature can be reduced to around 0.25 °C before entering the supercooler, thereby improving the efficiency of ice slurry generation while ensuring the full melting of ice particles given a sufficient preheating degree of 0.5 °C. The principle of the integration of heat regeneration in the direct ice slurry generation system was similar to that of the indirect ice slurry generation system, except that the secondary coolant loop was removed by directly coupling the refrigeration loop and the ice–supercooled water loop.

2.2. Establishment of the Experimental Platform

Two flexible platforms, for the direct and indirect ice slurry generation systems, respectively, were established to facilitate the experimental investigation of the ice slurry generation with heat regeneration enhancement, as illustrated in Figure 3. The only difference between the two platforms was whether a secondary coolant loop was integrated. Taking the indirect ice slurry generation platform as an example (see Figure 3a), the compressor adopted in the platform was a ZB21KQ compressor supplied by Emerson [29], and the condensers were custom-made finned tube heat exchangers supplied by Ruixue [30]. The evaporator, the supercooling exchanger, and the pre-heater were plate heat exchangers supplied by Leiman [31]. A connection interface was designed in the platform to allow the regenerator to be connected to the system by switching the inlet and the outlet piping of the water pump (see the sections between points B and A in the Figure 3a,b). The types and key parameters of these main components in the platforms are summarised in Table 1 for easier reference. In the indirect ice slurry generation platform, an ethylene glycol solution with a concentration of 50% was used as the secondary coolant in the indirect ice slurry generation system. Both systems were well insulated, with ambient heat loss to considered small enough to be neglected.



(**a**) Indirect ice slurry generation system.



(**b**) Direct ice slurry generation system.

Figure 3. The flexible experimental platforms with switch interface (IS: ice slurry; HE: heat exchanger; SC: secondary coolant; FM: flowmeter; T: temperature sensor).

Table 1. Summary of the main components in the experimental platforms. (DI: direct ice slurry generation system; IDI: indirect ice slurry generation).

Component	Part Number/Feature	Technical Parameter	Supplier
Compressor	ZB21KQ	Nominal cooling capacity: 6.2 kW	Emerson [29]
Water pump	CDLF2-2	Flow rate: 3 m^3/h	Nanfang Pump Industry [32]
Secondary coolant pump	JLF2	Flow rate: 3 m^3/h	Nanfang Pump Industry [32]
Condenser	Air-cooled finned tube heat exchanger	Heat transfer area: 30 m^2	Ruixue [30]
Supercooling heat exchanger of DI	Plate heat exchanger	Heat transfer area: 3.6 m^2	Leiman [31]
Evaporator of IDI	Plate heat exchanger	Heat transfer area: 2.4 m^2	Leiman [31]
Supercooling heat exchanger of IDI	Plate heat exchanger	Heat transfer area: 4.2 m^2	Leiman [31]
Regenerator	Plate heat exchanger	Heat transfer area: 1.8 m^2	Leiman [31]
Scrapped surface ice generator	ZJZB-5	Power consumption: 2.5 kW	Zhongjixinan [33]

Four PT100 temperature sensors with an accuracy of $\pm$0.01 °C were installed at the inlet of the supercooling heat exchanger, the outlet of the supercooling heat exchanger, the outlet of the water pump, and the ice storage tank in order to measure the corresponding water temperatures. Another two identical PT100 temperature sensors were installed to measure the secondary coolant temperatures at the inlet and outlet of the evaporator. Two electromagnetic flowmeters with an accuracy of 0.5% [34] were installed in the corresponding pipelines to measure the volumetric flow rates of the water and secondary coolant, respectively. Two strain-type pressure sensors with an accuracy of ±0.5% [34] were installed at the inlet and outlet of the compressor to measure refrigeration pressure. Total system power consumption was measured using a digital power meter with an accuracy of ±0.5% [35].

2.3. Experimental Plans

Three sets of experiments were designed and implemented to respectively quantify the performance of the control group, evaluate the influence of the heat regeneration enhancement on system performance, and assess the benefits of the system compared to the benchmark scraped surface ice slurry generator; the detailed experimental plans are summarised in Table 2. The first set of experiments were carried out based on direct and indirect ice slurry generation systems without heat regeneration enhancement, the performance of which was mapped under different ambient temperatures (from 15 to 35 °C in a step of 5 °C) by their placement in a climate chamber. The experiment lasted for one hour, during which the flow rate of supercooled water was fixed at 2.6 m^3/h and the pre-heating degree was set as 0.5 °C. It is worth mentioning that the starting time of a test was considered to be the moment when ice particles were first observed. Specifically, an amount of hot water was mixed into the ice storage tank after ice slurry generation until the water temperature in the tank reached 1–2 °C in order to ensure the full melting of the ice particles. Due to the fact that the ice particles were very small and quick to melt when exposed to room temperature, a heat balance method was used to estimate ice production rather than weighing the ice mass directly, as described by Equation (1),

$$m_{ice,tank} = \frac{c_{p,water} \times \left[m_{hot\ water} \times \left(T_{hot\ water} - T_{final,tank}\right) - m_{water,tank} \times \left(T_{final,tank} - T_{initial,tank}\right)\right]}{\gamma_{ice}} \quad (1)$$

where m is the mass, γ is the latent heat of fusion, T is temperature, c_p is the specific heat capacity, and the subscripts 'final' and 'initial' represent the conditions of the storage tank before and after hot water mixing.

Table 2. Summary of the experimental plans.

Test Set		System, Ambient Temperature, and Supercooled Water Flow Rate							
1	System	Direct and indirect ice slurry generation systems without heat regeneration enhancement							
	Ambient temperature (°C)	15	20		25		30		35
	Flow rate (m^3/h)	2.6							
2	System	Direct and indirect ice slurry generation systems with and without heat regeneration enhancement							
	Ambient temperature (°C)	25							
	Flow rate (m^3/h)	2.4	2.6	2.8	3.0	3.2	3.4	3.6	3.8
3	System	Direct and indirect ice slurry generation systems with heat regeneration enhancement, and the scraped surface ice slurry generator							
	Ambient temperature (°C)	25							
	Flow rate (m^3/h)	2.6							

The second set of tests involved the comparative performance evaluation of the direct and indirect ice slurry generation systems enhanced by heat regeneration. Facilitated by the heat regeneration approach, the flow rate of supercooled water was expected to affect the system performance. Different supercooled water flow rates (from 2.4 to 3.8 m^3/h in a step of 0.2 m^3/h) and their influence on ice slurry generation were therefore tested under a fixed ambient temperature of 25 °C. Eventually, in order to assess the economic benefit of the proposed systems in comparison with the conventional scraped surface ice slurry generator, the systems were experimentally tested and analysed under identical working conditions (i.e., an ambient temperature of 25 °C and a water flowrate of 2.6 m^3/h).

2.4. Key Performance Indicators

A number of key performance (KPIs) were developed in this study to assess the energy efficiency and effective utilisation of the different ice slurry generation systems. The coefficient of performance (*COP*) of the systems was calculated using Equation (2),

$$COP = \frac{\dot{Q}_c}{P} \tag{2}$$

where P is the system power and $\dot{Q}_c$ is the refrigeration capacity, which can be determined using Equation (3),

$$\dot{Q}_c = \dot{m}_{water} \times c_{p,water} \times (T_{in,water} - T_{out,water}) \tag{3}$$

where $\dot{m}$ is the water mass flow rate and the subscripts *in* and *out* represent the inlet and outlet of the supercooling heat exchanger. It is worthwhile to notice that due to supercooling, the cold energy was carried by the water in the form of sensible heat only.

Similar to COP, unit ice mass generation power consumption (w) can be used to quantify the system energy efficiency, as described in Equation (4),

$$w = \frac{W_{total}}{m_{ice}} \tag{4}$$

where W_{total} is the total energy consumption during the corresponding ice generation period.

Due to the fact that preheating in order to avoiding ice blockage results in excessive cold energy consumption to remove the sensible heat of water flow above 0 °C in the supercooler, not all of the cold energy is used for ice slurry generation. Accordingly, an effective utilisation rate of cold energy was defined as the ratio of supercooling degree to the difference in water temperature between the inlet and outlet of the supercooling heat exchanger (ambient heat loss was neglected), as described in Equation (5),

$$\eta = \frac{c_{p,water} \times \dot{m}_{water} \times (0\,^\circ\mathrm{C} - T_{out,water})}{c_{p,water} \times \dot{m}_{water} \times (T_{in,water} - T_{out,water})} = \frac{-T_{out,water}}{T_{in,water} - T_{out,water}} \tag{5}$$

where η is the effective utilisation rate.

The uncertainties for individual independent variables were determined using Equation (6), while the uncertainty propagation for refrigeration capacity, COP, and unit ice mass generation power consumption (i.e., dependent variables with the format as in Equation (7)) were calculated using Equation (8):

$$u = \sqrt{u_A^2 + u_B^2} = \sqrt{\frac{\sum_{j=1}^{K} (x_j - \bar{x}_j)^2}{K(K-1)} + (\frac{\Delta}{\sqrt{3}})^2} \tag{6}$$

$$z = f(x, y, \ldots) \tag{7}$$

$$\frac{u_z}{z} = \frac{1}{z} \cdot \sqrt{(\frac{\partial z}{\partial x} u_x)^2 + (\frac{\partial z}{\partial y} u_y)^2 + \ldots} \tag{8}$$

where u is the uncertainty, x and y are the value of direct measured variables, K is the number of measurement, Δ is the accuracy of the sensor/device, the subscripts A and B represent type A and type B evaluations of uncertainty, and z is the key performance indicator.

2.5. Setup of the Economic Analysis

The life-cycle cost of the ice slurry generation systems involves both investment cost and operational costs, which were determined using Equations (9)–(12) [36]. The life-cycle cost analysis approach was included in a comprehensive methodology for simulation, engineering economic analysis, and optimization of industrial thermal systems including power generation and cooling systems [37]. The ice generation systems investigated here are directly within this scope, and thus this method was considered applicable in this study.

$$C_{total} = C_{net,investment} + C_{operation} \tag{9}$$

$$C_{net,investment} = C_{investment} - C_{residual,0} = C_{investment} - b \cdot C_{investment} \cdot k^N \tag{10}$$

$$C_{operation} = \frac{k \cdot \left(1 - k^N\right)}{1 - k} \cdot C_{operation,0} \tag{11}$$

$$k = \frac{1 + r}{1 + i} \tag{12}$$

where C is the cost, within which the operational cost in the first year can be calculated using Equation (13), b is the ratio of residual value to investment in this study, r is the inflation rate in this study, i is the interest rate (i.e., discount rate) in this study, N is the life-cycle in number of years, and the subscript 0 indicates the first year.

$$C_{operation,0} = C_{energy,0} + C_{ma\ int\ enance,0} = W_{hour} \cdot e \cdot n + r_m \cdot C_{investment} \tag{13}$$

where W_{hour} is the energy consumption per hour, e is the electricity price per unit kWh, n is the number of ice slurry batches generated per year, and r_m is the ratio of maintenance cost to investment. The number of service hours within one year can be calculated using Equation (14),

$$n = \frac{M}{m_{ice,hour}} \tag{14}$$

where M is the ice slurry demand per year.

The key parameters used for the life-cycle cost analysis are summarised in Table 3.

Table 3. Summary of the key parameters used for life-cycle cost analysis.

Item	Ratio of Residual Value, b (%)	Inflation Rate, r (%)	Interest Rate, i (%)	Life-Cycle Period, N (year)	Ratio of Maintenance Cost, r_m (%)	Ice Demand per Year, M (t)
Value	10 [38]	2.5 [39]	4.0 [40]	10	1.25 [36]	50

The life-cycle cost analysis of the ice slurry generation systems was quantified in comparison with a conventional scraped surface ice slurry generator, shown in Figure 4. Due to its wide deployment, it was considered reasonable to select a scraped surface system as a benchmark for the comparison of different ice slurry generation systems. The scraped surface ice slurry generator was a commercial product supplied from Zhongjixinan [33]. It mainly consists of a compressor, a condenser, a cylindrical evaporator, and a number of scraper blades, and relies on the rotation of scraper blades to fragment the ice layer grown on the inner cylindrical surface of the evaporator to generate ice slurry. The scraped surface ice slurry generator had a capacity of 20–30 kg/h and a corresponding nominal power supply of 2.5–2.8 kW.

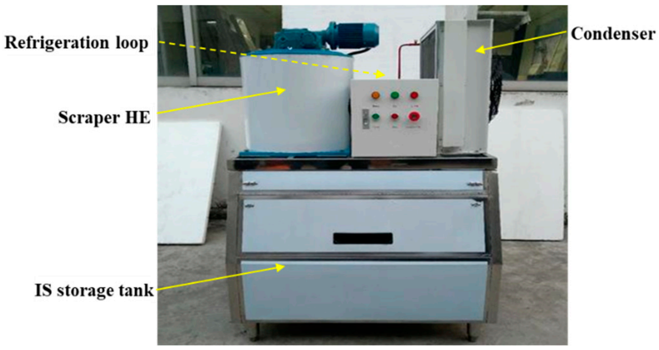


Figure 4. The scraped surface ice slurry generator (HE: heat exchanger; IS: ice slurry).

3. Results and Discussion

3.1. System Performance without Heat Regeneration Enhancement

Figure 5 presents the supercooling degrees of water at the outlet of the supercooler in both the direct and indirect ice slurry generation systems without heat regeneration under different ambient conditions. It can be seen that the supercooling degree in both systems shows a declining trend with increasing ambient temperature. A higher ambient temperature resulted in a higher refrigeration condensation temperature, which in consequence reduced the refrigeration capacity. It can be seen that the supercooling degree in the direct ice slurry generation system was always around 10% higher than in the indirect ice slurry generation system, ranging from 1.23 to 1.69 °C. This is due to avoidance of heat transfer in the secondary coolant loop, which contributes to a smaller overall heat transfer temperature difference on the cooling side in the direct ice slurry generation system. Correspondingly, the evaporation temperature in the direct ice slurry generation system was higher than in the indirect ice slurry generation system by around 1.4 °C, while the difference in condensation temperature between two systems was small. The relative uncertainties of the supercooling degree, evaporation temperature, and condensation temperature were all below 0.8% for both the direct and indirect systems (see Table 4), indicating the reliability of the above temperature analysis.

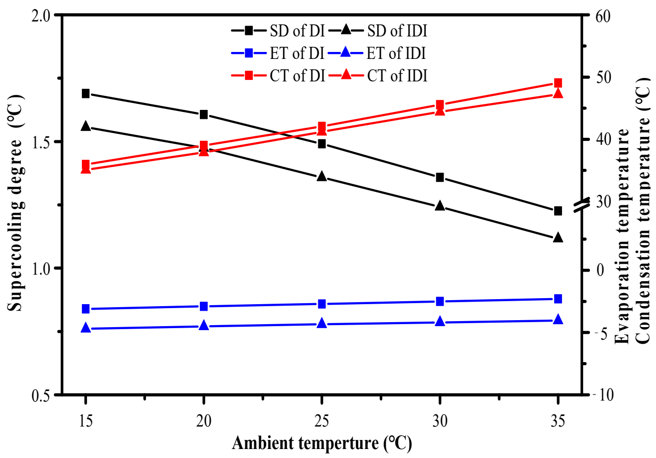


Figure 5. The variance of supercooling degree, evaporation, and condensation temperatures with the ambient temperature. (SD: supercooling degree; ET: evaporation temperature; CT: condensation temperature; DI: direct ice slurry generation system; IDI: indirect ice slurry generation system).

Table 4. Summary of temperature relative uncertainties.

Ambient Temperature (°C)	RU of SD in DI (%)	RU of SD in IDI (%)	RU of ET in DI (%)	RU of ET in IDI (%)	RU of CT in DI (%)	RU of CT in IDI (%)
15	0.37	0.44	0.39	0.26	0.24	0.19
20	0.38	0.45	0.56	0.71	0.07	0.15
25	0.40	0.45	0.78	0.28	0.18	0.13
30	0.45	0.61	0.49	0.29	0.13	0.12
35	0.53	0.58	0.71	0.41	0.10	0.10

(RU: relative uncertainty; SD: supercooling degree; ET: evaporation temperature; CT: condensation temperature; DI: direct ice slurry generation system; IDI: indirect ice slurry generation system).

Figure 6 presents the power consumption and COP of the direct and indirect ice slurry generation systems without heat regeneration. It can be seen from Figure 6 that energy consumption increased when increasing the ambient temperature. The main power consumption in both systems was due to the compressors, and the increase in their pressure ratios when improving the ambient temperature from 15 to 35 °C was the main reason for the ascending trend in compressor power consumption. The power consumption of indirect ice slurry generation was higher than the direct ice slurry generation system by almost 0.4 kWh, with the power draw of the secondary coolant pump accounting for this difference between the two systems. When increasing the ambient temperature, the condensation pressure increased sharply while the evaporation pressure varied only slightly, which tended to improve the pressure ratio of the compressor and resulted in a decrease in COP from from 2.80 to 1.90 and from 2.26 to 1.57, respectively, for the direct and indirect ice generation systems. Over the ambient temperature range studied, the direct ice slurry generation system outperformed the indirect ice slurry generation system, with a COP more than 10% higher. The relative uncertainties of refrigeration capacity and COP were below 1.1% and 2.9%, respectively.

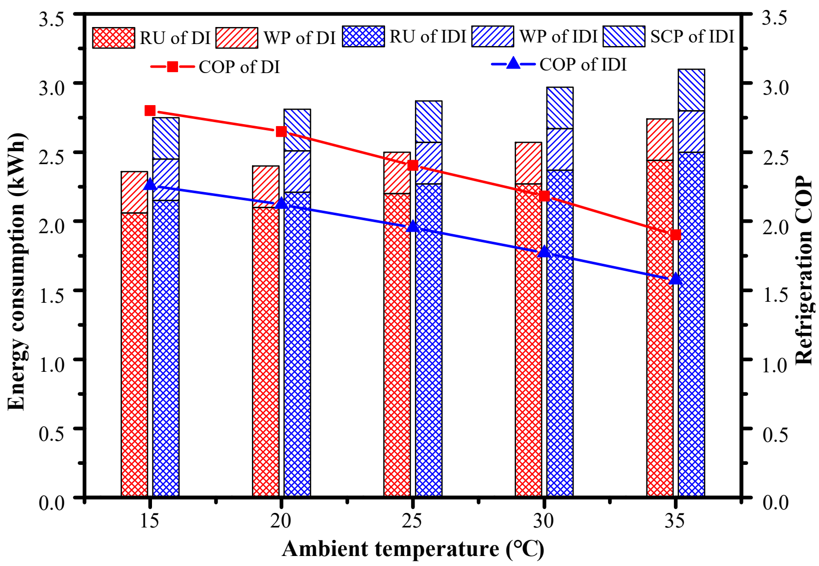


Figure 6. The variance in total energy consumption and COP with ambient temperature (RU: refrigeration unit; WP: water pump; SCP: secondary coolant pump; DI: direct ice slurry generation system; IDI: indirect ice slurry generation system).

Figure 7 illustrates the ice generation rate and the energy consumption per unit ice mass of the two systems. At an ambient temperature of 15 °C, the ice generation rate of the direct and indirect ice slurry generation systems reached 51.7 and 47.4 kg/h, respectively, which continued to decrease as the ambient temperature was increased from 15 to 35 °C. On the contrary, the variation in unit ice mass energy consumption experienced an increasing trend over the same ambient temperature range (i.e., 15–35 °C), from 0.046 to 0.075 kWh/kg for the direct ice slurry generation system. Due to this higher power consumption and lower hourly ice generation, the unit ice mass energy consumption of the indirect ice slurry generation system was around 20% higher than that of the direct ice slurry generation system. The relative uncertainties of unit ice mass energy consumption were all lower than 1.3%.

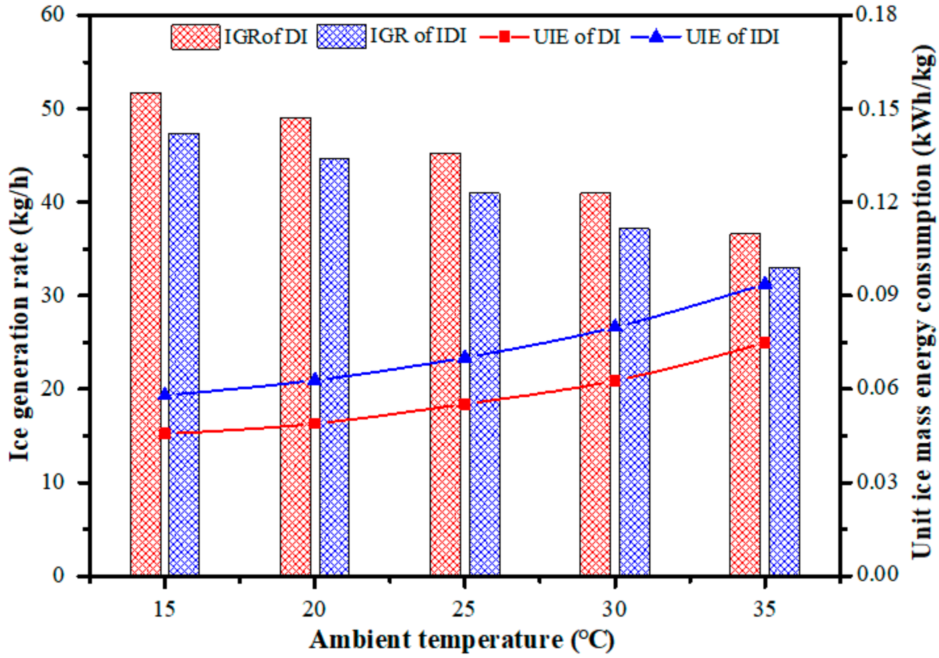


Figure 7. Ice generation rate and unit ice mass energy consumption variance with ambient temperature (IGR: ice generation rate; UIE: unit ice mass energy consumption; DI: direct ice slurry generation system; IDI: indirect ice slurry generation system).

It is worth mentioning that apart from its higher energy consumption, the indirect ice slurry generation system was more complicated and expensive compared to the direct ice slurry generation system. However, stable system control in the direct ice slurry generation system is harder to achieve due to the lack of the buffer provided by the secondary coolant loop.

3.2. Comparative Performance with Heat Regeneration Enhancement

Figure 8 presents the supercooling degrees in the direct and indirect ice slurry generation systems with and without heat regeneration enhancement under different water flow rates. It can be seen that the supercooling degree decreased with increasing water flowrate in all systems, with a reduction of 1.83 to 1.15 °C in the enhanced direct ice slurry generation system and from 1.63 to 0.92 °C in the system without heat regeneration enhancement. It is this regeneration circle that enables a higher supercooling degree of around 0.2 °C with the same flowrate. The supercooling degrees in the indirect ice slurry generation systems experienced similar trends, and were lower than in the direct system. The heat regeneration approach permitted a decrease in the water temperature at the inlet of the supercooler from 0.5 to 0.25 °C, contributing to a higher supercooling degree compared to the systems without the heat regenerator.

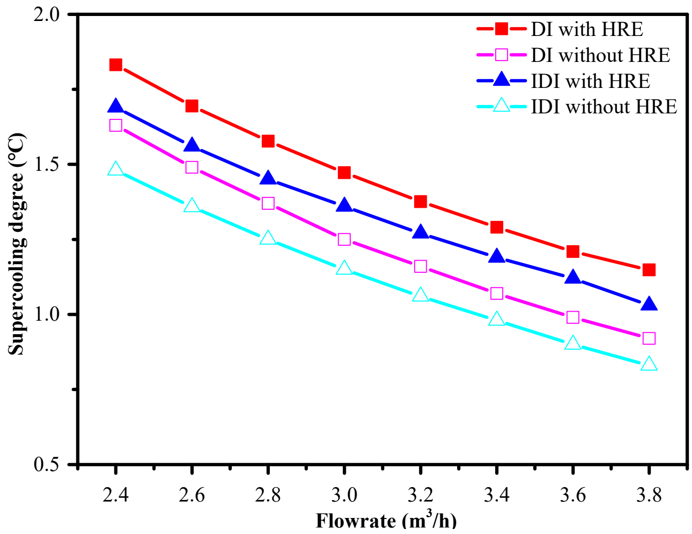


Figure 8. The variance of supercooling degree with water flowrate (HRE: heat regeneration enhancement; DI: direct ice slurry generation system; IDI: indirect ice slurry generation system).

Figure 9 illustrates the effective utilisation rate of the cold energy in the direct and indirect ice slurry generation systems with heat regeneration enhancement compared to ones without heat regeneration. In both systems with heat regeneration, the effective utilisation rates reached above 80%, while this value was only around 70% in the systems without heat regeneration. This indicates that using heat regeneration can improve the efficiency of ice slurry generation. The heat regeneration approach allowed for an increase in supercooling degree, avoiding the cold energy loss needed to bring the water to 0 °C. When increasing the water flow rate, the supercooling degrees in all systems were reduced as the preheating temperature needed to be maintained, leading to a decrease in the effective utilisation rate. However, the enhanced ice slurry generation systems were less sensitive to the influence of varying the water flow rate compared to the systems without heat regeneration enhancement. The effective utilisation rate of the direct ice slurry generation system without heat regeneration enhancement fell from 76.5 to 64.8% when the water flow rate was increased from 2.4 m^3/h to 3.8 m^3/h, a decrease of around 12%, while the enhanced system saw a decrease of only 6%. A similar phenomenon was observed for the indirect ice slurry generation systems with and without heat regeneration enhancement, with decreases in effective utilisation rate of around 12% and 6%. In general, using the heat regeneration approach successfully improved the effective utilisation rate by over 15%.

Figure 10 compares the ice generation rate and unit ice mass energy consumption of direct and indirect ice slurry generation systems with and without heat regeneration enhancement. The cooling capacities of the enhanced systems were slightly lower than those without heat regeneration enhancement because of the lower evaporation temperatures caused by their higher supercooling degrees under the same flow rate. However, this influence was offset by their higher effective utilisation rates (see Figure 9) leading to much higher final ice generation rates, as shown in Figure 10. The maximum ice generation rates of the enhanced direct and indirect ice slurry generation systems reached 52.0 and 47.8 kg/h, respectively, compared to rates of 46.0 and 41.4 kg/h for the corresponding systems without heat regeneration enhancement. The benefits from achieving this higher ice generation rate with similar energy consumption when using the heat regeneration enhancement effectively reduced the maximal unit ice mass power consumption from 0.056 to 0.049 kWh/kg for the direct and 0.069 to 0.060 kWh/kg for the indirect ice slurry generation system. Over the water flowrate range studied, the unit ice mass power consumption was reduced by 20–12% in the enhanced direct and 21–13% in the enhanced indirect ice slurry generation systems.

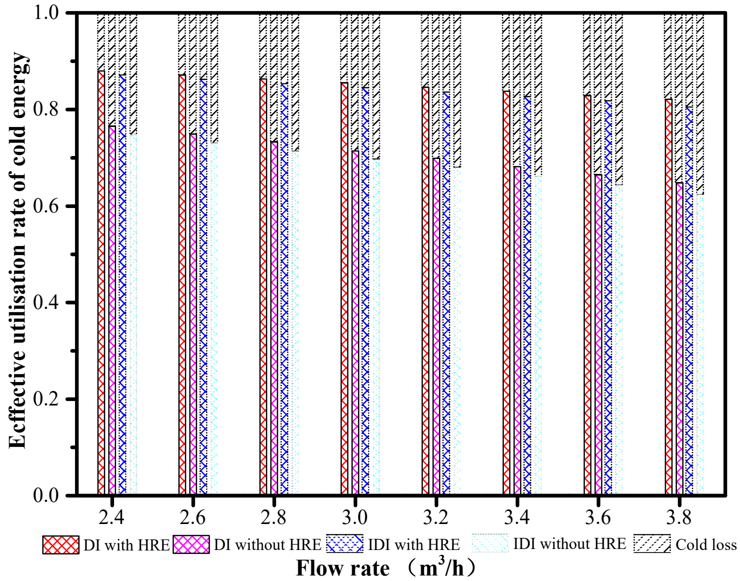


Figure 9. The variance in effective utilisation rate of cold energy with water flowrate (HRE: heat regeneration enhancement; DI: direct ice slurry generation system; IDI: indirect ice slurry generation system).

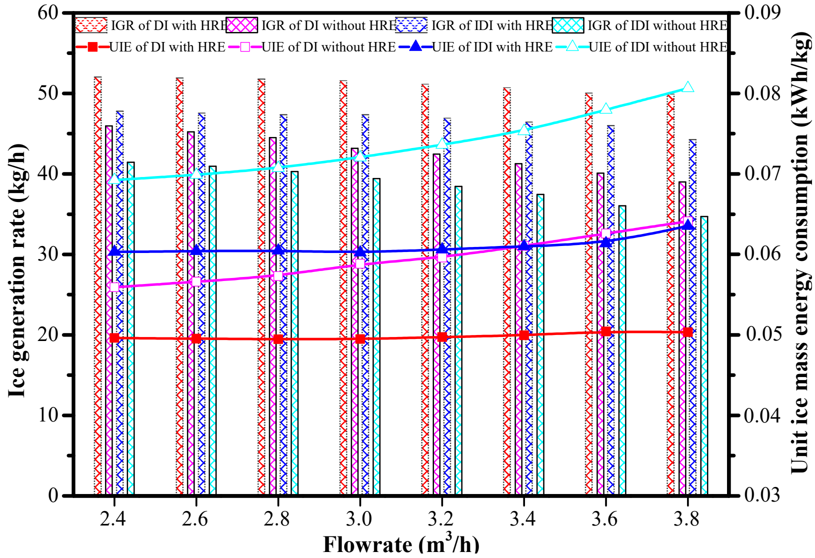


Figure 10. Ice generation rate and unit ice mass energy consumption variance with water flowrate (IGR: ice generation rate; UIE: unit ice mass energy consumption; HRE: heat regeneration enhancement; DI: direct ice slurry generation system; IDI: indirect ice slurry generation system).

3.3. Economic Benefit of the Proposed System

Table 5 compares the energy consumptions and ice generation rates of the ice slurry generation systems enhanced by heat regeneration and the scraped surface ice slurry generator used as a benchmark. By avoiding conductive heat transfer through the ice layer formed on the tube surface of the heat exchanger, the enhanced supercooled water ice slurry generation systems had higher evaporation temperatures, −3.0 and −4.3 °C, respectively, for the direct and indirect ice slurry generation systems, while that of the scraped surface ice slurry generator was −12.0 °C. This increase in evaporation temperature significantly contributed to the ice generate rate being increased around one-fold in the enhanced system (52.0 and 47.8 kg/h for the direct and indirect ice slurry generation systems, respectively) compared to the scraped surface ice slurry generator, which had a rate of 24.1 kg/h. Despite the slightly higher total energy consumption of the enhanced systems (2.55 and 2.87 kWh for the direct and indirect ice slurry generation systems compared to 2.53 kWh for the scraped surface ice slurry generator), the unit ice mass energy consumption of the enhanced systems was much lower than the benchmark (0.049 and 0.060 kWh/kg, respectively, compared to 0.105 kWh/kg), resulting in significant reductions of 53.3% for the enhanced direct and 42.9% for the enhanced indirect ice slurry generation systems. This benefit was attributed to better heat transfer within the supercooled water ice during slurry generation as well as to coolness recovery from heat regeneration.

Table 5. Performance comparison among the supercooled ice slurry generation systems with heat regeneration enhancement and the scraped surface ice slurry generator.

Methods	Evaporation Temperature (°C)	System Energy Consumption (kWh)	Hourly Ice Production (kg/h)	Unit Ice Mass Energy Consumption (kWh/kg)
Direct ice slurry generation system with heat regeneration enhancement	−3.0	2.55	52.0	0.049
Indirect ice slurry generation system with heat regeneration enhancement	−4.3	2.87	47.8	0.060
Scraped surface ice slurry generator	−12.0	2.53	24.1	0.105

Table 6 compares the life-cycle costs of the supercooled water ice slurry generation systems with heat regeneration enhancement and the scraped surface ice slurry generator. The total investment cost of the scraped surface ice slurry generator was USD5149 in this study, which was around 21.4% and 2.0% higher, respectively, than that of the enhanced direct and indirect ice slurry generation systems. Furthermore, the operating cost of scraped surface ice slurry generator was around twice that of the direct ice slurry generation system and 1.6 times that of the indirect ice slurry generation system, which was recognised to have extra energy consumption introduced by the heat transfer in the secondary coolant loop. Due to their cost-effectiveness in terms of both system operation and initial investment, it was found that the life-cycle costs of the enhanced direct and indirect ice slurry generation systems were USD7650 and USD9217, which was only 62.0% and 74.7% the cost of the scraped surface ice slurry generator. This indicates excellent cost saving potential, especially for the enhanced direct ice slurry generation system.

Table 6. Life-cycle cost analysis of the ice slurry generation systems.

System		Direct Ice Slurry Generation System with Heat Regeneration Enhancement	Indirect Ice Slurry Generation System with Heat Regeneration Enhancement	Scraped Surface Ice Slurry Generator
Investment (USD)	Refrigeration loop	1213	1183	-
	Water loop	1477	1507	-
	Coolant loop	-	687	
	Control system etc.	1552	1672	-
	Total investment, $C_{investment}$	4242	5049	5149
	Net investment, $C_{net,\ investment}$	3875	4612	4704
Operation cost (USD)	Net present energy cost over the life-cycle	3285	4022	7032
	Net present maintenance cost over the life-cycle	490	583	595
	Net present operation cost over the life-cycle, $C_{operation}$	3775	4605	7627
Life-cycle cost, C_{total} (USD)		7650	9217	12,331

4. Conclusions

Efficient ice slurry generation plays a primary role in energy savings and the cost-effective application of ice slurry for thermal energy storage and transportation. In this study, a heat regeneration approach was proposed to enhance the efficiency of supercooled water ice slurry generation; this was integrated into a direct and an indirect supercooled water ice slurry generation systems, and the corresponding system performances were experimentally investigated under different working conditions through a series of comparative studies. Based on our performance evaluation, the life-cycle costs of the enhanced ice slurry generation systems were further assessed via reference to a conventional scraped surface ice slurry generation system used as benchmark.

The experimental results showed that the supercooling degree of the enhanced systems was effectively increased, which improved the effective utilisation rate of cold energy from 76.5% to 88.0% in maximum for direct ice slurry generation. For both direct and indirect systems, the unit ice mass power consumption can be reduced by about 12–20%. When varying the ambient temperature from 15 °C to 35 °C, the COP of the direct and indirect ice slurry generation systems ranged from 2.80 to 1.90 and 2.26 to 1.57, respectively. Due to the existence of a secondary coolant loop, the efficiency of direct ice slurry generation system outperformed the indirect one, and its supercooling degree was 10% higher. As a result, the operational cost of the enhanced indirect ice slurry generation system reached USD 4605 compared USD 3775 for the enhanced direct ice slurry generation system, which were much lower than the scraped surface ice slurry generator (USD 7627). The total life-cycle costs of the direct and indirect ice slurry generation systems were USD 7650 and USD 9217, respectively only 62.0% and 74.7% that of the scraped surface ice slurry generator.

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Nomenclature

b	ratio of residual value to investment
C	cost, USD
c_p	specific heat capacity, kJ/(kg·K)
e	electricity price per kWh, USD/kWh
k	intermediate variable
K	number of measurement
i	interest rate, %
M	the ice slurry demand per year, ton
m	mass, kg
$\dot{m}$	water mass flow rate, kg/h
N	life-cycle period in year
n	number of the ice slurry generation batch per year
P	power, kW
$\dot{Q}_c$	capacity of the refrigeration unit, kW
r	inflation rate, %
r_m	ratio of maintenance cost to investment, %
T	temperature, °C
u	uncertainty
W	energy consumption, kWh

x, y	direct measured variables
z	response variable, i.e., key performance indicator
Δ	sensor/device accuracy
w	unit ice mass energy consumption, kWh/kg
γ	latent heat of fusion, kJ
η	effective utilisation rate, %
Subscripts	
0	the first year
A	type-A evaluation
B	type-B evaluation
j	index of measurement

References

1. Dincer, I. Energy and environmental impacts: Present and future perspectives. *Energy Sources* **1998**, *20*, 427–453. [CrossRef]
2. Abulfotuh, F. Energy efficiency and renewable technologies: The way to sustainable energy future. *Desalination* **2007**, *209*, 275–282. [CrossRef]
3. Ozbugday, F.C.; Erbas, B.C. How effective are energy efficiency and renewable energy in curbing CO_2 emissions in the long run? A heterogeneous panel data analysis. *Energy* **2015**, *82*, 734–745. [CrossRef]
4. Ammari, C.; Belatrache, D.; Touhami, B.; Makhloufi, S. Sizing, optimization, control and energy management of hybrid renewable energy system—A review. *Energy Built Environ.* 2021, *in press*. [CrossRef]
5. Olatomiwa, L.; Mekhilef, S.; Ismail, M.S.; Moghavvemi, M. Energy management strategies in hybrid renewable energy systems: A review. *Renew. Sustain. Energy Rev.* **2016**, *62*, 821–835. [CrossRef]
6. Yang, J.; Xu, X.; Peng, Y.; Zhang, J.; Song, P. Modeling and optimal energy management strategy for a catenary-battery-ultracapacitor based hybrid tramway. *Energy* **2019**, *183*, 1123–1135. [CrossRef]
7. Hemmati, R.; Saboori, H. Emergence of hybrid energy storage systems in renewable energy and transport applications—A review. *Renew. Sustain. Energy Rev.* **2016**, *65*, 11–23. [CrossRef]
8. Inaba, H. New challenge in advanced thermal energy transportation using functionally thermal fluids. *Int. J. Therm. Sci.* **2000**, *39*, 991–1003. [CrossRef]
9. Oro, E.; Gracia, A.; Castell, A.; Farid, M.M.; Cabeza, L.F. Review on phase change materials (PCMs) for cold thermal energy storage applications. *Appl. Energy* **2012**, *99*, 513–533. [CrossRef]
10. Saito, A. Recent advances in research on cold thermal energy storage. *Int. J. Refrig.* **2002**, *25*, 177–189. [CrossRef]
11. Wang, H.; Feng, R.; Duan, H.; Chen, A. Investigation into the ice generator with double supercooled heat exchangers. *Appl. Therm. Eng.* **2016**, *98*, 380–386. [CrossRef]
12. Zhang, C.; Yang, L.; Lin, W.; Wei, J.; Chen, J.; Ma, Z. Experimental investigation and life-cycle cost analysis of a cold storage enhanced vacuum cooling system using ice slurry. *Sustain. Energy Technol. Assess.* **2021**, *45*, 101074. [CrossRef]
13. Kauffeld, M.; Kawaji, M.; Egolf, P.W. *Handbook on Ice Slurries*; International Institute of Refrigeration: Paris, France, 2005.
14. Castaing, J.; David, T.; Bedecarrats, J.P.; Strub, F. Dynamic modelling and experimental study of an ice generator heat exchanger using supercooled water. *Energy Convers. Manag.* **2006**, *47*, 3644–3651. [CrossRef]
15. Wang, H.; He, G.; Feng, R. An effective method for preventing ice-blockage in dynamic generation system with supercooling water. *Int. J. Refrig.* **2014**, *46*, 114–122. [CrossRef]
16. Mouneer, T.A.; El-Morsi, M.S.; Nosier, M.A.; Mahmoud, N.A. Heat transfer performance of a newly developed ice slurry generator: A comparative study. *Ain Shams Eng. J.* **2010**, *1*, 147–157. [CrossRef]
17. Kozawa, Y.; Aizawa, N.; Tanino, M. Study on ice storing characteristics in dynamic-type ice storage system by using supercooled water. *Int. J. Refrig.* **2005**, *28*, 73–82. [CrossRef]
18. Kauffeld, M.; Gund, S. Ice slurry—History, current technologies and future developments. *Int. J. Refrig.* **2019**, *99*, 264–271. [CrossRef]
19. Yan, J.; Zhang, X.; Chen, Y.; Zhou, B. An improved ice producing system of assisting liquid desiccant evaporative supercooled water. *Energy Build.* **2013**, *62*, 530–538. [CrossRef]
20. Bedecarrats, P.J.; David, T.; Lasvignottes, J.C. Ice slurry production using supercooling phenomenon. *Int. J. Refrig.* **2010**, *33*, 196–204. [CrossRef]
21. Tanino, M.; Kozawa, Y. Ice-water two-phase flow behavior in ice heat storage systems. *Int. J. Refrig.* **2001**, *24*, 639–651. [CrossRef]
22. Wang, H.; He, G.; Tian, Q. Experimental study of the supercooling heat exchanger coated with fluorocarbon coating. *Energy Build.* **2012**, *55*, 526–532. [CrossRef]
23. Okawa, S.; Saito, A.; Suto, H. The experimental study on freezing of supercooled water using metallic surface. *Int. J. Refrig.* **2002**, *25*, 514–520. [CrossRef]
24. Inada, T.; Zhang, X.; Yabe, A.; Kozawa, Y. Active control of phase change from supercooled water to ice by ultrasonic vibration 1. Control of freezing temperature. *Int. J. Heat Mass Transf.* **2001**, *44*, 4523–4531. [CrossRef]
25. Zhang, X.; Inada, T.; Yabe, A.; Lu, S.; Kozawa, Y. Active control of phase change from supercooled water to ice by ultrasonic vibration 2. Generation of ice slurries and effect of bubble nuclei. *Int. J. Heat Mass Transf.* **2001**, *44*, 4533–4539. [CrossRef]

26. Hozumi, T.; Saito, A.; Okawa, S.; Matsui, T. Freezing phenomena of supercooled water under impacts of ultrasonic waves. *Int. J. Refrig.* **2002**, *25*, 948–953. [CrossRef]
27. Mito, D.; Tanino, M.; Kozawa, Y.; Okamura, A. *Application of a Dynamic Storage System to the Intermittent Cooling Process in the Food Industry*; Fourth IIR Workshop on Ice Slurries: Osaka, Japan, 2001; pp. 105–114.
28. Strand, R.K.; Pedersen, C.O.; Coleman, G.N. Development of direct and indirect ice-storage models for energy analysis. *ASHRAE Trans. Symp.* **1994**, *100*, 1230–1244.
29. Emerson. 2020. Available online: http://www.emerson.cn (accessed on 6 May 2020).
30. Ruixue. 2019. Available online: http://www.tjruixue.cn (accessed on 15 February 2019).
31. Leiman. 2020. Available online: http://www.wuxleiman.com (accessed on 15 September 2020).
32. Nanfang Pump Industry. 2020. Available online: http://www.cnppump.cn (accessed on 21 May 2020).
33. Zhongjixinan. 2018. Available online: http://www.zhongjixinan.cn (accessed on 30 March 2018).
34. Linuo. 2020. Available online: http://lenosensor.com (accessed on 6 July 2020).
35. Aitek. 2020. Available online: http://www.aitek.tw (accessed on 6 May 2020).
36. Shirazi, A.; Taylor, R.A.; Morrison, G.L.; White, S.D. A comprehensive, multi-objective optimization of solar-powered absorption chiller systems for air-conditioning applications. *Energy Convers. Manag.* **2017**, *132*, 281–306. [CrossRef]
37. Bejan, A.; Tsatsaronis, G.; Moran, M. *Thermal Design and Optimization*; Wiley-Interscience: New York, NY, USA, 1996.
38. Kong, R.; Deethayat, T.; Asanakham, A.; Kiatsititoat, T. Performance and economic evaluation of a photovoltaic/thermal (PV/T)-cascade heat pump for combined cooling, heat and power in tropical climate area. *J. Energy Storage* **2020**, *30*, 101507. [CrossRef]
39. Statista. 2020. Available online: https://www.statista.com (accessed on 19 February 2020).
40. Global Rates. 2020. Available online: https://www.global-rates.com (accessed on 19 February 2020).