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Slurry-HP

A feasibility study on the use of a super-cooling ice slurry heat pump for solar heating applications

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Abstract

The present project provides a technical and economic feasibility study of solar ice-slurry systems for the provision of SH and DHW in buildings. The competitiveness of such systems with respect to state-of-the art solar-ice systems (ice-on-hx) as well as respect to ground source heat pumps (GSHP), both in terms of electricity consumption and heat generation costs, is analysed by means of system simulations and lifetime cost calculations.

The analyses show that, considering all available slurry technologies and their current technological readiness level, the supercooling method is the most promising solution for solar-ice applications. Systems based on the supercooling approach show a high potential for cost reduction, while having a high energetic efficiency, especially for ice storage volumes of at least 1 m^3 per MWh of yearly heating demand. For example, with a storage volume of 1.3 m^3/MWh and uncovered selective collectors with an area of 1 m^2/MWh, heat generation costs around 28 Rp./kWh can be achieved, i.e. 1.5 Rp./kWh below the costs of a GSHP system with an increase of the seasonal performance factor of 30 % (SPF_{SHP+} =5.2). Comparing with ice-on-hx solutions, the ice slurry solution is likely to have a higher SPF_{SHP+} together with lower investment and heat generation costs, if the ice storage volume is larger than that of the ice-on-hx case.

For single-family houses, the ice slurry concept can be a valid option, depending on the available space for installing an ice storage. Nevertheless, it is envisaged that the main potential use of the proposed method will be in multi-family or tertiary buildings, where space restrictions are less constringent.

Finally, in order to fully exploit the benefits of the supercooling ice slurry method, some technological barriers still need to be overcome. The most important challenge is to achieve a stable and sufficiently high supercooling degree, without crystallisation of the ice particles before entering the ice releaser or storage vessel.

Zusammenfassung

Mit der vorliegenden Studie wurden die technische Machbarkeit und die Wirtschaftlichkeit von solaren Eisbrei-Wärmepumpen-Heizungen untersucht, die zur Bereitstellung von Raumwärme und Warmwasser in Gebäuden eingesetzt werden sollen. Der Elektrizitätsbezug und die Wärmegestehungskosten wurden mit aktuellen Solar-Eis-Systemen (mit Eis-Anwachsen auf den Wärmetauschern: ice-on-hx) und mit aktuellen Erdsonden-Wärmepumpen unter Verwendung von Systemsimulationen und Kostenberechnungen über die Lebensdauer verglichen.

Der Vergleich verschiedener Eisbrei-Technologien bezüglich ihrer Eigenarten und ihrer technologischen Reife zeigt, dass die Extraktion von Latentwärme aus Wasser mittels Unterkühlung (supercooling) die grösste Aussicht auf eine erfolgreiche Umsetzung in Solar-Eis-Heizungen hat. Eisbrei-Systeme, die mit Unterkühlung von Wasser arbeiten, haben ein hohes Kosteneinsparpotential im Vergleich zu den anderen untersuchten Heizungsarten einerseits und andererseits eine hohe Effizienz bezüglich des Elektrizitätsverbrauchs. Dies gilt insbesondere für Eisspeichervolumina grösser 1 m^3 pro MWh jährlichem Wärmebedarf. Beispielsweise resultieren mit einem Eisspeichervolumen von 1.3 m^3/MWh und unverglasten selektiven Kollektoren mit einer Fläche von 1 m^2/MWh Wärmegestehungskosten von rund 28 Rp./kWh, was rund 1.5 Rp./kWh unter den Gestehungskosten der simulierten Erdsondenheizung ist, bei einer gleichzeitigen Steigerung der Systemjahresarbeitszahl um 30 % (SPF_{SHP+} =5.2). Der Vergleich mit gängigen ice-on-hx-Heizungen zeigt, dass die Eisbrei-Technologie zu einer Steigerung des SPF_{SHP+} und zu geringeren Investitions- und Wärmegestehungskosten führen kann, wenn das Eisspeichervolumen grösser gewählt wird im Vergleich zum ice-on-coil-System.

Falls für die Installation eines Eisspeichers genügend Platz vorhanden ist, kann die Eisbrei-Technologie in Einfamilienhäusern gut eingesetzt werden. Das grösste Potential sehen die Autoren jedoch in Mehrfamilienhäusern und Gebäuden aus dem tertiären Sektor, in denen der Platz für die Installation eines grossen Eisspeichers mutmasslich leichter zur Verfügung gestellt werden kann.

Damit die Eisbrei-Erzeugung mittels Unterkühlung erfolgreich eingesetzt werden kann, müssen noch einige technologische Hürden genommen werden. Die grösste Herausforderung ist die Gewährleis-

tung einer stabilen und ausreichend starken Unterkühlung, die verhindert, dass die Eispartikel vor dem Erreichen des Eiserzeugers oder des Eisspeichers entstehen.

Résumé

Dans le présent projet, une étude techno-économique de faisabilité à propos de systèmes solaires à coulis de glace pour le chauffage de bâtiments et la production d'eau chaude sanitaire (ECS) est proposée. La compétitivité de tels systèmes est analysée au moyen de simulations et de calculs de coûts d'utilisation sur la totalité de la durée de vie et comparée à celle de systèmes conventionnels à stockage de glace solaire (formation de glace sur les échangeurs) ainsi que de pompes géothermiques.

Considérant l'ensemble des technologies de coulis de glace ainsi que leur maturité technologique actuelle, l'analyse montre que la méthode avec sur-refroidissement est la plus prometteuse en ce qui concerne les applications de type glace solaire. Les systèmes basés sur cette méthode du sous-refroidissement montrent un fort potentiel de réduction des coûts tout en garantissant un rendement énergétique élevé et ceci plus spécialement pour les systèmes ayant des volumes de stockage sous forme de glace supérieurs à 1 m^3 par MWh de besoins de chauffage annuels. Par exemple, avec un volume de stockage de 1.3 m^3/MWh et des capteurs sélectifs non vitrés ayant une surface de 1 m^2/MWh, des coûts de production d'environ 28 Rp./kWh peuvent être atteints, ce qui correspond à 1.5 Rp./kWh de moins qu'un système de pompe géothermique et le facteur de performance annuel augmenté de 30 % (SPF_{SHP+} =5.2). En comparaison des systèmes avec formation de glace sur les échangeurs, les systèmes à coulis de glace montrent vraisemblablement un SPF_{SHP+} plus élevé ainsi que des coûts d'investissement et de génération de chaleur plus faibles à condition que le volume du stockage de glace soit plus grand de celui d'un système à stockage de glace conventionnel.

Selon l'espace disponible pour le stockage de glace, le concept à coulis de glace peut être une alternative intéressante pour les maisons individuelles. Néanmoins il est prévu que le système proposé soit potentiellement le plus utilisé dans l'habitat collectif et les bâtiments du tertiaire, où les restrictions en termes d'espace sont moins drastiques.

Enfin il reste nécessaire de surmonter certains verrous technologiques pour exploiter pleinement les bénéfices de la méthode du coulis de glace sur-refroidit, le challenge le plus important étant d'atteindre un sur-refroidissement stable et suffisamment élevé sans qu'une cristallisation des particules de glace ne se produise avant l'entrée du générateur de glace ou du réservoir de stockage.

Contents

1. Motivation

The increase of system efficiency by raising the share of renewable energy in the building sector is necessary to reach the targets of the Energy Strategy of Switzerland by 2050 (Swiss Federal Council, 2013) and will have a high relevance also at European level. A recent example for a heating system with a high share of renewable energy is the combination of solar thermal and heat pump systems with ice storages, the so-called solar-ice systems. The interest in solar-ice systems is growing in central Europe, where climatic conditions are appropriate for this technology. For this reason, solar-ice systems have been established as an alternative to ground source heat pump (GSHP) systems with advantages such as (Carbonell et al., 2016b, 2015):

- usually not restricted to regulations of water and soil.
- no need to regenerate the ground in regions densely populated with boreholes.
- the ice storage is accessible, allowing for solving leakages or replace heat exchangers.
- the ice storage can be installed in the cellar when no ground space is available.
- it is a flexible system able to adapt to building size restrictions, i.e. the same system performance can be reached with different combinations of collector area and ice storage volume.
- higher performance compared to GSHP can be achieved if direct solar heat is used extensively.

However, solar-ice systems have some disadvantages respect to GSHP such as:

- higher number of hydraulic components.
- added complexity of the control.
- higher installation cost if the same performance is desired.
- degradation of performance when ice grows.

Therefore, research is still needed to bring robust solar-ice systems to the market with comparable cost with respect to GSHP and higher efficiency. To achieve the above mentioned goals, a solar-ice system based on a supercooling method is proposed in this project. A summary of the advantages of the supercooling method with respect to state-of-the-art ice-on-heat-exchanger systems are:

- The always free-of-ice heat exchanger will lead to a higher heat pump performance. The evaporation temperatures of the refrigerant during ice making will be higher compared to state-of-the-art ice-on-heat-exchanger systems, where the evaporation temperature is decreasing with increasing thickness of the ice layer.
- Avoidance of the in-tank heat exchanger where ice is produced with the respective cost reduction. Heat exchangers are only necessary to bring solar heat into the storage.
- Possibility to avoid the brine loop between the heat pump and the storage vessel.
- No risk of cracking the storage casing due to the expansion of water when freezing and no need to have a structure that keeps the ice immersed, i.e. very cheap vessels can be used to store slurries.
- Slurry production and storing takes place in two different devices. This provides flexibility for the storage concept, i.e. flexible, cheap and modular storage designs are possible (cylindrical, cubic or almost any other shape).
- Higher melting rates due to the large contact area between ice particles and water.

2. Review on ice slurry technologies

Ice slurry is a mixture of small ice crystals or particles and a carrier fluid. As carrier fluid water or a mixture of water and antifreeze can be used. Ice crystals are in the size of 0.1 to 1 mm; when they are above 1 mm they are usually referred to as ice particles (Kauffeld et al., 2010). Usually, the carrier fluid is either water or a mixture of water and a freezing point depressant, e.g. sodium chloride, ethanol, ethylene glycol and propylene glycol.

Three cristallization steps are involved in all ice slurry generators (Kauffeld et al., 2005):

(a) **Supersaturation**: is a condition where the solution is not in equilibrium, i.e. there is a chemical potential difference between the solid and the crystalline phase. The method used to cause supersaturation in ice generators is to supercool the solution or to change the pressure bringing the solution to the triple point, where water partially evaporates and creates a chemical potential difference that induces cristallization. The degree of supersaturation conditions the rates of the subsequent steps of nucleation and growth.

(b) **Nucleation**: when the solution is supersaturated, molecules can form stable clusters from initial nuclei. The type of nucleation can be homogeneous or heterogeneous. In homogeneous nucleation the ice particles aggregate inside the aqueous solution by statistical fluctuations of cluster of molecules. In heterogeneous nucleation the ice particles aggregate on foreign bodies, e.g. surface of the heat exchangers, solid particles, or impurities. In homogeneous nucleation, crystallization occurs at lower temperature and ice particles generated are usually smaller compared to heterogeneous nucleation.

(c) **Growth**: when nucleation has started ice crystals are able to grow.

Besides these three processes, ice slurries can undergo other steps depending on the technology (Stamatiou et al., 2005): i) **attrition** is the breaking-down process of ice crystals submitted to stress, ii) **agglomeration** is the process where crystals collide and adhere forming larger crystals and in iii) **ostwald ripening** small crystals tend to dissolve and deposit in larger crystals by the different solubility between small and large crystals.

2.1. Ice slurry technologies

The key aspects of any solar ice storage system for heating applications are reliability, energy efficiency and cost effectiveness. The main idea beyond all ice slurry generators is to avoid the loss of energetic efficiency by ice thickness growth on the surface of the heat exchanger, i.e. by avoiding the growth of ice on the surface or by actively removing it. There are several methods to produce slurries and each type of generator has its own advantages and disadvantages that may fit better to specific applications. Therefore, a single best technology for all applications does not exist.

A classification of most of the commercially available ice makers can be based on the type of nucleation, i.e. heterogeneous and homogeneous (Egolf and Kauffeld, 2005):

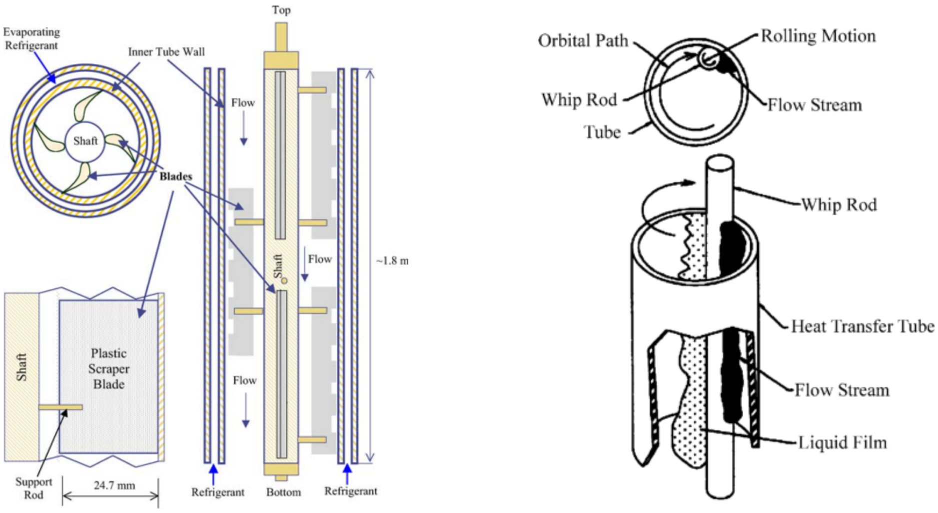
2.1.1 Heterogeneous nucleation

When latent heat of water is extracted by bringing the water into contact to a heat exchanger, the process of icing is called heterogenous nucleation. The heterogenous ice slurry generation methods can be classified as:

(a) **Scraper type** : this is the most well known and commercially available technology, where ice is removed by a mechanical device and accumulated in the suspension. The most widespread systems use mechanical scrapers, brushes, helical screws or rotating rods in a special evaporator, usually a co-axial cylinder or circular shell and tube (Stamatiou et al., 2005). These systems are widely accepted and have been used for more than 30 years. However, these heat exchangers require high investments and maintenance cost.

    The heat exchanger area is usually small and an aqueous solution with an antifreeze that facilitates the scraping effect and reduces agglomeration is used. Therefore, low refrigerant temperatures are needed to compensate the low area and the low freezing point of the aqueous solution leading to a reduction of the COP of the chiller.

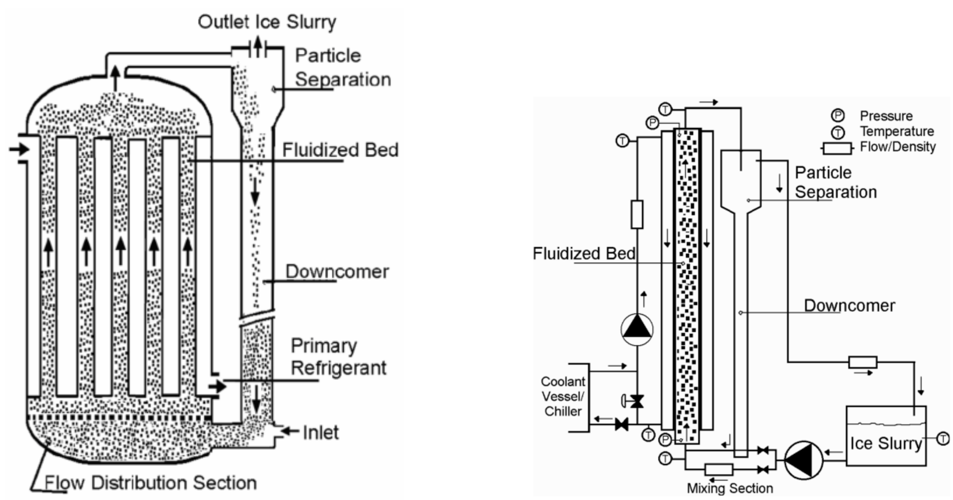
    The only publication on solar-ice slurry concepts for solar heating applications found is related to a research project conducted at CanmetENERGY, Natural Resources of Canada (Tamasauskas et al., 2015, 2012). The ice slurry method used is based on a scraping technology and, thus, the efficiency reported in the publications is relatively low. Seasonal performances in the order of an air source heat pump were achieved. However, with significant increase on system cost. Therefore, the scraper method does not seem to be a promising technology for this application.



(a) (b)

Figure 2.1: Schematic of ice slurry generators with moving parts, showing a) a scraper type, and b) an orbital rod. Schemes from Stamatiou et al. (2005)

(b) **Fluidized bed ice slurry generator:** this concept uses shell and tube or coaxial tube heat exchangers. The zone where the aqueous solution is circulating is filled with small particles of stainless steel called bed, which move together with the fluid and it is thus *fluidized*. This movement provokes that the solid particles collide on the walls where ice is being built, thereby preventing its cristallization. These heat exchangers need a vertical arrangement and, in order to reduce



(a) (b)

Figure 2.2: Schematics of fluidized beds with (a) an ice slurry generator, and (b) an experimental system set-up (Meewisse, 2004).

the amount of refrigerant, usually a falling film method is used for the refrigerant. The main disadvantage is that there is a maximum temperature difference between the refrigerant and the fluid below which ice can be removed safely. If the temperature difference is higher the impact of particles is not sufficient to remove ice from the walls. Moreover, high fluid velocities are needed to ensure enough momentum of the particles for proper ice removal. Currently only research activities are known for this technology (Meewisse, 2004, Pronk et al., 2005, 2003).

(c) **Prevention of ice formation by a secondary fluid**. Those methods are based on the injection of a fluid, either compressed air or supercooled water, through small holes in the cold walls where ice is formed. Zhang et al. (2008) used a bubbling device and coated walls with PTFE (poly-tetra-fluoro-ethylene) to suppress ice adhesion of an aqueous ethylene solution on the cold walls. The air bubbles were created by blowing compressed air from bottom to top and were expected to increase turbulence, enhance heat transfer and prevent ice adhesion. In the experiments optimum flow rates

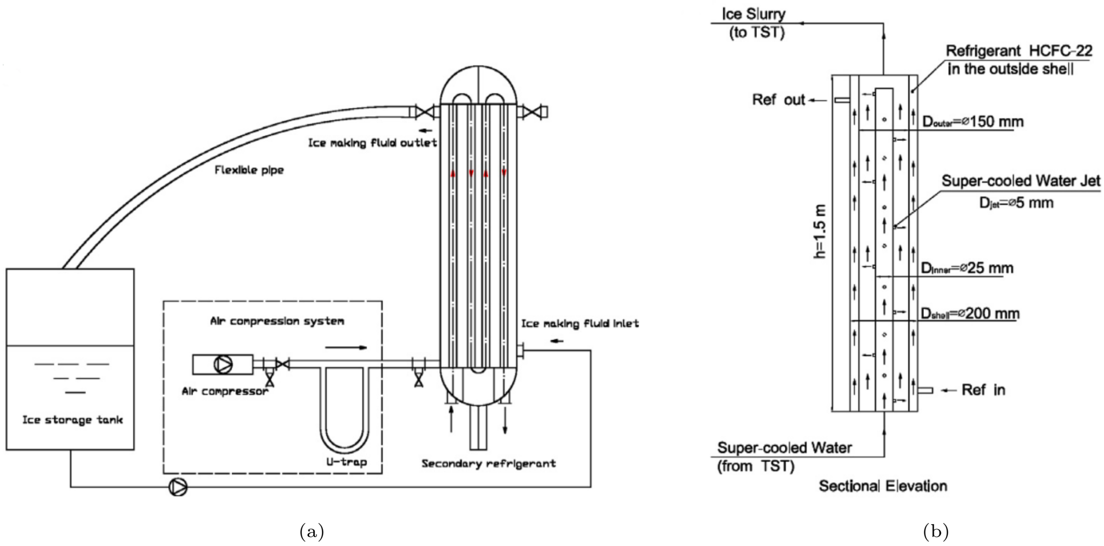


Figure 2.3: Schematics of ice slurry generators based on (a) a bubbling device (Zhang et al., 2008) and (b) supercooled water jets Mouneer et al. (2010).

of the compressed air and refrigerant were investigated. The temperature of the refrigerant was also analyzed. The optimum temperature of the refrigerant resulted in a higher degree of supercooling and ice formation. If the temperature of the refrigerant is too low, the temperature difference is high, but the wall temperature is very low and ice adheres to the walls. If the refrigerant temperature is high, ice does not adhered on the surface, but the temperature difference is small and thus the heat transfer rate is small, too.

Ice formation can also be prevented by using supercooled water jets (Mouneer et al., 2010). The idea is to spray the cold walls, where ice particles will grow, with several supercooled water jets. In the experiments fifteen water jets with diameters of 5 mm each were used. This method was estimated to reduce the cost by 40-60 % and the energy consumption by 10-20% compared to the scraper type. However, this seems to be at an early development stage.

2.1.2 Homogeneous nucleation

Homogeneous refers to the process of the nucleation inside the water or the aqueous solution. Ice slurry technologies using homogeneous nucleation can be divided as:

(a) **Direct contact.** The refrigerant is injected with a two phase flow state (liquid-gas) at very low temperature directly into a pressurized vessel containing water. When getting in contact with the water at 0 °C, the injected refrigerant expands and evaporates at constant pressure and temperature. This technology is expected to have a high efficiency since the evaporator heat exchanger is eliminated. High volumetric heat transfer coefficients were obtained by Kiatsiriroat et al. (2003) using R12 and R22 and by Thongwik et al. (2008) using CO_2. Water can also be used as a refrigerant if the storage is under vacuum. However, this technology presents several difficulties such as ensuring that i) the refrigerant and water do not mix ii) the refrigerant is not trapped in the ice slurry iii) water should not enter the cooling circuit because it can freeze and block the refrigerant circuit and iv) ice is not formed in the nozzle used to inject the refrigerant into the storage (Wijeysundera et al., 2004). Although, this method could be of interest, it is still on a research level with no real applications yet. A robust running device does not seem to be available even at the laboratory level.

(b) **Supercooling.** The supercooling method is a technology in which pure water or an aqueous solution with a low percentage of a freezing point depressant is supercooled by few degrees in a heat exchanger called supercooler. The ice particles are formed after the fluid has left the supercooler by a specific device, e.g. by ultrasonic vibrations. From the authors perspective this is the method that could fit better for the solar-ice applications and it is the one selected in the present feasibility study. The arguments for this decision and details about the method are provided in section 3.

Other methods for supercooling exist, e.g. Li et al. (2012) proposed a method of **evaporative supercooling**. Instead of supercooling water inside a heat exchanger, the idea is to evaporate water by bringing it to a supercooled state in an atmosphere of low humidity. However, this method needs a process to remove humidity in the circulating air.

2.2. Preventing agglomeration of slurry in the storage

Agglomeration of the ice slurry particles will happen in the ice storage and also in pipes. A method for preventing agglomeration and growth of ice particles is to use suitable additives. Inaba et al. (2005) reviewed anti-agglomeration of ice particles and suppression of ice growth by using anti-freeze protein and surfactants. A surfactant is a substance which tends to reduce the surface tension of a liquid in which it is dissolved. Grandum et al. (1999) showed that the addition of anti-freeze protein into an ice slurry is an effective way to to prevent agglomeration using very low concentrations. However, anti-freeze proteins are currently very expensive for practical applications. Therefore, substitutes were being searched for. Both substitutes of anti-freeze proteins and surfactant additives show promising results (Inaba et al., 2005) and are able to disperse ice particles without a relevant decrease of the freezing temperature. Matsumoto et al. (2006) showed that water oil emulsions were suitable for ice storages and prevented ice adhesion to the walls thanks to the inherent structure. However, in Matsumoto et al. (2010) it was found that ice propagation was difficult and slow. In order to promote propagation they used electric charging with DC voltage. Hong et al. (2004) found that an aqueous solution with ethylene glycol and a silane coupling agent as surfactant was able to suppress ice adhesion under specific concentration levels.

Theoretically, in solar-ice applications, agglomeration will not be an important issue because slurries will not be pumped but only stored. An exception is discussed in section 4.3, where the possibility to pump slurries to the thermal collectors is considered.

3. Ice slurry production by supercooling

This ice slurry production by supercooling allows to exchange heat through a surface which is always free of ice without the need of any special mechanical device or foreign body in the fluid flow. A liquid is considered to be in a supercooling state when it remains liquid below the freezing point. All substances, when being cooled down to the freezing point, undergo some supercooling, which is defined as the temperature difference between the supercooled liquid (T_s) and the solidification temperature (T_{fr}) of the heat transfer fluid.

$$\Delta T_s = |T_s - T_{fr}| \tag{1}$$

In the case of ice slurries, the heat transfer fluid is either water or an aqueous solution. The supercooling effect can be reduced by nucleation agents, but it can not be completely avoided. An example of a typical temeprature evolution over time of a substance being cooled down below its freezing point is shown in Fig. 3.1. The substance remains in the liquid state until the crystallization starts. After this, the latent heat of fusion is released, the temperature increases until the melting point, and solid particles are formed. The relation between the supercooling degree and the ice formation can be expressed as (Ernst and Kaufeld, 2016):

$$\dot{m}_{ice} = \dot{m}_s \frac{c_p}{L_f} \Delta T_s \simeq 1.26e^{-2} \Delta T_s \qquad [kg/s] \tag{2}$$

where $\dot{m}_{ice}$ and $\dot{m}_s$ are the mass flow rates of ice and water respectively; c_p is the specific heat capacity of water and L_f the heat of fusion.

The supercooling degree depends on: i) sample size of the particles formed ii) velocity of the heat transfer fluid iii) evaporation temperature of refrigerant, and thus pressure and iv) characteristic surface of the heat exchanger. From all of them, the most influencing parameter of the super-cooling degree is the

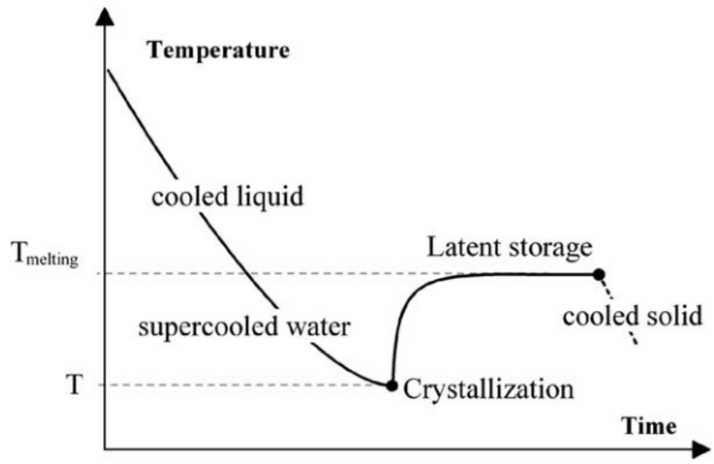


Figure 3.1: Supercooling phenomena: temperature versus time (from Castaing-Lasvignottes et al. (2006)).

sample size. Water can reach a supercooling degree of 14 K in volumes of a few m^3 and up to 35 K with micro-capsules of few μm^3 (Bédécarrats et al., 2010).

The COP of the heat pump increases with the temperature of aqueous solution and therefore the ideal supercooler will have a high freezing temperature, i.e. pure water as a carried fluid without any freezing point depressant.

Although the supercooling method is used commercially in few systems in Japan, it can suffer from unstable operation. One of the challenges is to avoid or reduce the formation of ice on the surface of the heat exchanger. A second challenge is to provoke ice formation when desired by promoting ice nucleation. The unwanted ice growth in the coldest part of the heat exchangers may cause the blockage of the heat exchangers and although the process is self-sealing, the heat pump operation is stopped and an extra energy and supply system for melting the ice formed is necessary. The process of crystallization has an stochastic character, i.e. samples that are apparently identical will not crystallize at the same temperature and time during the cooling process (Bédécarrats et al., 2010). The probability of nucleation depends on the critical size particle and thus on the type of nucleation and temperature, i.e. supercooling degree. Therefore, a safe margin needs to be considered in order to completely avoid nucleation.

Castaing-Lasvignottes et al. (2006) investigated numerically and experimentally a supercooler heat exchanger where stable operation was achieved with 1 K of supercooling degree. The coaxial coil heat exchanger had internal diameters of 11 mm and 17.5 mm for the inner and outer tube respectively. Wall thicknesses of 0.7 mm were used for both tubes whose length was 5 m. Tap water without any additives was circulated in the annular zone. In their experiments, supercooling of 2 K was found to be unstable. The heat power with this low supercooling degree of around 1.3 kW was not enough to completely evaporate the refrigerant and a second evaporator was used for this purpose. Mass flow rates used in the experiments were in the order of 540 - 650 kg/h. Tap water and heat exchanger without any special treatment were used. The same exchanger was used in following experiments by Bédécarrats et al. (2010). On this new experiments, a maximum supercooling degree was found to be 2 K, but blocking was very frequent. For supercooling degrees below 1.2 K there was no blocking but ice production was very slow. Between 1.3 K and 1.9 K there were regular but acceptable blockages (lower than 2 per hour). It was found that the supercooling degree increases by lowering the flow rate. However, higher flow rates produce more ice, thus an optimum exists. For the same supercooling degree, higher flow rates lead to higher number of blockages because turbulence promotes cristallisation. The largest supercooling degree corresponding for a stable operation was approximately of 1.8 K for a flow rate of 0.12 kg/s, but only of 0.9 K for a flow rate of 0.18 kg/s. An optimal operation was found for 0.14 kg/s and a supercooling degree of 1.6 K. These experiments showed that supercooling of water is not easy and without any treatment of the surface not robust. Therefore, a surface treatement seems to be necessary.

3.1. Surface of the heat exchanger

A method to reduce the probability that ice sticks on the surface of the heat exchanger and thus to increase the supercooling degree is to reduce the roughness of the heat exchanger surface or to coat it with an hydrophobic material.

3.1.1 Roughness of the heat exchanger

Faucheux et al. (2006) analyzed the influence of aluminum roughness effect on the supercooling degree of an aqueous solution with different concentrations of ethanol. The conclusion was that the supercooling degree increases with the decrease of the roughness. The supercooling degree was found to follow the equation $\Delta T_{sc} = 7.15 r_a^{-0.196}$, where r_a is the roughness in μm. Roughness between 0.63 and 13.33 μm were analyzed leading to 8.5 and 4.2 K supercooling degrees. The suspected relationship between supercooling degree and antifreeze concentration was not experimentally confirmed. These investigation showed very high supercooling degrees. However, experiments were performed using static conditions, and for cases where the aqueous solution is circulating, the supercooling degree rarely exceeds 3 K.

Analyses of the roughness of a coaxial copper tube in circulating conditions was provided by Ernst and Kaufeld (2016). Tap water was supercooled in a coaxial tube heat exchanger with an inner diameter of 8 mm and a total area of 0.24 m^2. Using a mass flow rate of 265 kg/h the maximum degree of supercooling was 3 K. Roughness was varied between 0.4 μm and 2.1 μm and the degree of supercooling was 2.3 K and 3 K respectively. The supercooling degree was found to follow a linear relationship:

$$\Delta T_{sc} = 3.1 - 0.7 r_a \qquad r_a [0.4 - 2.1] \mu m \tag{3}$$

The supercooling degree was higher than the ones reported by Bédécarrats et al. (2010) and Castaing-Lasvignottes et al. (2006). Though the frequency of ice blockage on the maximum supercooling degree that was achieved was not listed in the paper, the authors reported by personal communication that the supercooling degree was stable over several hours.

3.1.2 Coating heat exchanger surfaces

A common approach to avoid or delay the nucleation is to use hydrophobically coated surfaces.

Saito and Okawa (1994) investigated the effect of the heat transfer surface characteristics on the freezing of supercooled pure water. It was found that the supercooling degree is highly dependent on the characteristics of the surface.

Wang et al. (2012) found that fluorocarbon coating was able to increase the maximum supercooling degree from 0.9 K to 1.7 K for tap water, and from 1 K to 2.35 K for pure water. It was also observed that the maximum supercooling state lasted 6 minutes for the uncoated surface and 9 minutes for the coated one when tap water was used. Pure water has a higher supercooling degree compared to tap water because it has less impurities that could initiate nucleation. The maximum supercooling degree was obtained with velocities of 2 m/s. Values below 1.5 m/s were not able to produce any slurry and above 2.5 m/s the heat exchanger was frequently blocked by the growth of ice. The heat exchanger was a co-axial tube with 12 mm internal diameter, 1 mm thickness of the inner tube, and 22 mm internal diameter and 1.5 mm thickness of the outer tube. The heat exchanger was coated with a polymer which contains a fluoric binder and an organic solvent.

Tsuchida et al. (2002) investigated the adhesion of ice particles from a water oil emulsion in a vessel with a stirring device. It was found that a stainless steel vessel coated with 20 μm of a PFA resin (tetra-fluoro-ethylene-perfluoro-alkylvinil-ether-copolymer) was effective for preventing ice adhesion. The use of a vessel made of PTFE (poly-tetra-fluoro-ethylene) was also found to prevent ice adhesion.

3.2. Release methods

The initiation of freezing of the supercooled water has to be ensured after it leaves the supercooler. The locations for the initiation can be inside the pipe in direction to the ice storage or in the ice storage itself. It is possible to end the state of supercooling actively and thus, in a controlled way by a device.

Without this active controlling, supercooled water could be pumped into the ice storage and again back to the supercooler. A high risk that nucleation starts spontaneously inside the supercooler causing a blockage would be a consequence. A second effect is that the whole water content in the system could be supercooled, which lowers the source temperature of the evaporator, and therefore the COP of the heat pump.

After ice is produced first and after it starts to accumulate in the ice storage a further active controlling of icing is not necessary. When ice is present in the ice storage, supercooled water that enters the ice storage and gets into contact with ice particles freezes spontaneously.

The frequency of the need for using a releasing device depends on the system design. When the ice storage is used as seasonal storage, icing has to be initiated once at the beginning of winter. In heating systems with small ice storages that are heated above 0 °C several times during winter, freezing has to be initiated more often.

In the following, a short literature review is given on release methods that are able to end a supercooling state of water.

Okawa and Saito (1998) investigated methods to provoke nucleation using external factors. Several methods that are influencing the initiation of freezing of supercooled water were described. According to their review, the following methods applied to the bulk fluid (purified water) have been analysed in the past:

- Methods that do initiate freezing: turbulence, cavitation, applying direct current, electric spark discharge.
- Methods with no effect or even stabilization of supercooling: vibrating, applying an electric field, impulses against a wall next to the fluid.

In their publication, Okawa and Saito analyze the effect of supercooling for pure, non-stirred water. Supercooling rates in the range of 12 to 5 Kelvin were analyzed. Two successful methods for releasing ice were found i) rubbing of glass surfaces against each other with frequencies around 100 Hz and ii) applying an electric charge (direct current) with stainless steel electrodes at distances between 0.5 mm and 1 mm. The lower the degree of supercooling the higher the voltage that is needed to initialize freezing. The results show that for a supercooling of 4 K (which is the lowest value analyzed) only a low probability of less than 20 % for releasing the latent heat is reached with the analyzed voltages up to 130 V DC. The resulting current was not stated. As pure water is used in the experiments the effect in tap water remains unclear.

Hozumi et al. (2002) analyzed the effect of applying ultrasonic (US) waves to a sample of pure, non-stirred, supercooled water via the walls of the water containing vessel. The experiments were conducted with US at a frequency of 45 kHz. The US device was switched on permanently during each experiment. Depending on the experimental design and materials in contact with the water the effectiveness of initializing freezing was varying. Best results were obtained with a metal stick immersed in water and intensities of the US of 0.13 and 0.28 W/cm. The probability of freezing at a certain degree of supercooling was higher for the high intensity of 0.28 W/cm. Sometimes supercooling rates of 2 - 4 K could be ended. However, as the process is stochastic, rates up to 20 K could occur in pure water during treatment of the water with the release methods. The effects of higher US intensities were not analyzed. The authors mention that the probability of initiating freezing at low US intensities is higher if air bubbles are present at the surfaces between water and materials in contact with the water. A reason for this seems to be that the air bubbles are easing cavitation which in turn can initiate nucleation.

In Hozumi et al. (2003), the authors investigate the effect of applying an electric charge at 50 Volt DC for 30 seconds to pure, non-stirred water. The resulting current was in a range of 2 to 5 μA. Thus, only very low electric powers were used. Different materials were used for the electrodes immersed in water: aluminium (Al), copper (Cu), silver, gold, platinum and carbon. The authors showed that freezing starts several seconds after applying the electric charge and the probability of freezing corresponds to the tendency of the electrodes to ionize. Therefore, Al and Cu show the highest probability for initiating freezing. With the conditions in the experiments supercooling rates of 2 K and 4 K could be ended with probabilities of 0.3 and 0.9 respectively during the experiments. The authors discuss as mechanism of initiation of freezing that hydrates of the oxidized electrode material are formed at the anode. If Al is used as material for the electrodes poly-nuclear complexes of hydrates of $[Al(H_2O)_6]^{3+}$ an OH^- are formed that contain oxygen groups (O-O) with distances of 2.8 Å. This distance is close to the distance of 2.76 Å that O-O groups have within ice. The hypothesis is that due to the similarity of the distances the poly-nuclear complexes serve as excellent nuclei for freezing.

The literature review on methods used to initiate cristalization of supercooled water shows that both inducing cavitation in water with ultrasonic waves and applying electric current to immersed aluminium electrodes provide a high probability of starting freezing. However, the experiments presented by the different authors were all done with purified water and at high supercooling degrees. It remains unclear to which extent the results can be transferred to the envisaged use of tap water for ice slurry applications. Furthermore, it remains unclear to which extent the release of ice can be started with the envisaged low supercooling degree of approximately 2 K.

No publications could be found on Peltier elements as release device. Peltier elements generate temperature differences between two metallic surfaces which are part of the element. Commercial available peltier elements can generate temperature differences up to approximately 70 K with an electric power consumption below 100 W. If such a device is immersed into supercooled water in a way the cold side of the Peltier element is not heated too much by the water it might be possible to generate sufficient low temperatures on the cold surface of the Peltier element to force the water to freeze.

3.3. Market available ice slurry supercooling systems for cooling applications

Only few cold thermal energy storage systems using supercooling ice-slurry technology are installed worldwide (Kauffeld et al., 2005) and all of them are used for cooling applications. None of these systems have so far exploited the potential of supercooled water in heat pumping applications. Moreover, all installed systems are designed for medium and large scale applications with custom made refrigeration cycles developed by Japanese companies. By 2015, Mycom Mayekawa[1] has installed 46 supercooling ice-slurry systems with refrigeration capacities in the range of 50 kW to 2 MW. Shinryo Corporation[2] has installed 11 refrigeration systems using a dynamic ice storage system, called *The Jiyu Sekkei*, with a supercooler heat exchanger with capacities up to 1.8 MW. Tagasako Thermal Engineering has installed another 29 systems with a total refrigeration capacity of 35 MW. Other manufacturers are for example Taikisha[3] and Mitsubishi Heavy Industries (Kawada et al., 1998).

In Europe, the potential of the supercooled water method has not yet been exploited in commercial ice-slurry applications because so far energy efficiency plays a minor role in the current European ice slurry market. Instead, the operators of such ice-slurry systems focus rather on efficient distribution of cooling or direct contact cooling for food industry. In contrast, in heat pump applications energy efficiency plays a major role.

3.4. Supercooling heat exchanger design

Ideally it would be desirable to use standard heat exchangers as supercoolers. For example, brazed flat plate heat exchangers, as the one shown in Fig. 3.2 are very common as evaporators and condensers of heat pump cycles. However, the corrugated design may negatively effect stable operation of supercooling.

From the literature it seems that typically used supercooling heat exchangers are either coaxial, although shell and tube should work too. Examples of those heat exchangers are shown in Fig. 3.3.

A Summary of heat exchangers used and operating conditions applied for supercooling experiments are presented in Table 3.1. From those results, it seems possible to target a supercooling degree between 2-2.5 K. Other supercooling experiments than those shown in Table 3.1 have been reported. However, these are not listed because they were carried out without circulating water. On those experiments, water or an aqueous solution is inside a bath were it is subcooled and therefore the interesting conditions for the supercooling heat exchanger are not met. With these supercooling degrees around 2.5-3.1% of the mass will be crystallized.

[1] `http://www.mayekawa.com/products/cooling_systems`
[2] `https://www.shinryo.com/en/tech/thejiyusekkei.html`
[3] `http://www.taikisha-group.com/service/strathermi.html`

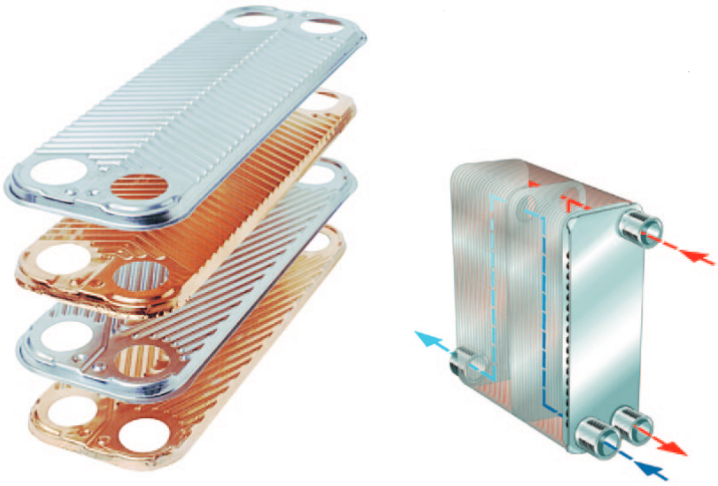
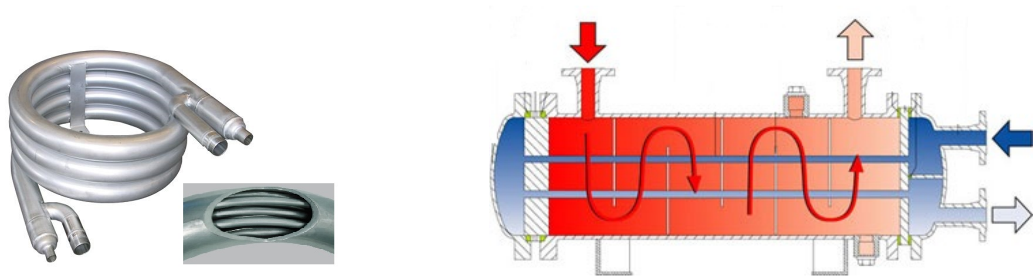


Figure 3.2: Example of a brazed flat plate heat exchanger with a multipass design (`http://www.alfalaval.com`).



(a) (b)

Figure 3.3: Schematic of different heat exchangers that can be used as supercooling heat exchangers (a) coaxial (`http://www.anthermo.de`) and (b) shell and tube heat exchangers (`http://www.funke.de`)

Table 3.1: Heat exchangers and operation conditions of supercooling studies for circulating fluids. d is the diameter and the subscripts w and Ref stand for water and refrigerant, L_{hx} lenght of the tubes, A_{hx} the heat exchanger area, r_a tube roughness, $\dot{m}$ mass flow rate, ΔT_s degree of supercooling, U heat transfer coefficient, Re Reynolds number and t_s time with stable supercooling.

Ref		Ernst and Kaufeld (2016)	Bédécarrats et al. (2010) [a]	Wang et al. (2012)	
Type		coaxial-copper	coaxial-copper	coaxial	coaxial
Fluid		tap water	tap water	tap/pure water	tap/pure water
d_w	[mm]	8	17.4	12	12
d_{Ref}	[mm]	-	11 ($\delta = 0.7$mm)	22	22
L_{hx}	[m]	9.6	5	1.2	1.2
A_{hx}	[m^2]	0.24	0.17	0.05	0.05
Coating		None	None	None	fluorocarbon
r_a	[μm]	0.4-2.1	-	0.75	0.27
$\dot{m}_{Ref}$	[kg/h]	-	-	-	-
$\dot{m}_w$	[kg/h]	265	432-648	90	90
ΔT_s	[K]	3-2.3	1.3-1.9	0.9-1.2	1.4-1.6
U	[kW/K]	3.8-2.9	4-5.4	2-2.7	3.2-3.6
Re	[-]	6500	2900-4500		
t_s	[min]	stable for hours	30	6	9

[a] Same heat exchanger as Castaing-Lasvignottes et al. (2006)

3.5. Condenser subcooler in the heat pump

Water pumped from the ice slurry storage to the supercooler must be free of any ice particles to avoid freezing of water inside the supercooler. Therefore, it is recommended to heat the water before entering the supercooling heat exchanger to ensure no ice particles are present in the water. Assuming preheating from 0 °C to 0.5 °C, and supercooling of 2 K, 20% of the thermal power of the evaporator is used for the preheating. It is hence of key importance that preheating is provided for free or at very low energetic cost.

There are different ways to preheat water at low energetic cost. In state-of-the-art ice slurry cooling systems this is achieved by extracting heat from the ventilated air of the building which is therefore a cooling gain for the system. However, in the present case a heating demand is targeted and it is assumed that no cooling needs will be present.

One option is the use of solar collectors for preheating the water. This could be achieved by a small storage of few liters charged by solar collectors to 10 °C. The problem of this approach is that solar energy is not always available and adding an extra storage makes the control more complex. Another option is the use of the lower part of a combi-storage which is only loaded by solar collectors and not by the heat pump. This concept needs a hydraulic loop from the combi-store to the heat pump and it could be that the bottom part of the combi-store is cooled too much. For cold and cloudy winters, preheating can not be guaranteed by solar energy. Therefore, a third option, using the refrigerant circuit of the heat pump has been considered. The latter option is analized hereafter.

In refrigeration, it is common to subcool the condensed refrigerant, which increases the refrigerant effect and the COP of the heat pump. For heating applications the refrigerant in the condenser is subcooled only in some large heat pumps in the order of hundred kW of heating power. For small and medium scale heat pumps, the efficiency increase by subcooling the refrigerant usually cannot be compensated by the extra cost of either an enlarged condenser or an extra subcooler which can be a further plate heat exchanger implemented into the refrigerant cycle. An important factor is that usually there is no extra external heat sink for subcooling the condenser below the temperature of the water entering the condenser, and therefore the subcooling effect is somehow limited. A method to solve this is the use the refrigerant at the outlet of the evaporator in the so-called liquid suction heat exchangers. However, this method increases the compressor work. Depending on the ration between the increase of condenser heat and compressor power, the COP can increase or decrease.

Pottker and Hrnjak (2015) analyzed the increase of COP as a function of the refrigerant and of the degree of subcooling by means of simulations. It was found that as subcooling increases, the COP reaches a maximum as a result of the ratio between the increase of the evaporation heat and the compressor power. The improvements of COP were a function of the refrigerant, specifically to the ratio of specific heat capacity of the liquid refrigerant and the heat of vaporization. The normalized COP reached a maximum at around 9 K of condenser subcooling for all refrigerants. Maximum COP increase for optimized condenser subcooling was in the order of 6-8% when appropriate refrigerants were used, i.e. with low heat of vaporization and not too low specific heat capacity such as R404A and R1234yf. Under these conditions, the increase of compressor power was in the order of 2%. A theoretical T-h diagram of cycles with and without condenser subcooling is shown in Fig. 3.4. If the power of the condenser is kept constant, the condenser's saturation temperature needs to rise lowering the heat of vaporization to compensate the subcooling heat included. Ideally, the subcooling should be provided avoiding that the condensing temperature of the refrigerant increases.

From the diagram one can see that as the degree of subcooling $\Delta T_{c,sub}$ increases, the condenser and compressor power increases, and therefore an optimum trade-off between the increase of these three values will reach the highest COP. The definition of the COP of the heat pump for heating applications is:

$$COP = \frac{q_c}{w_{comp}} \tag{4}$$

where q_c is the condenser heat and w_{comp} is the compressor work. Usually, the heat from subcooling the refrigerant in the condenser Δq_c is used as useful heat for the system and therefore:

$$COP_{sub} = \frac{q_c + \Delta q_c}{w_{comp} + \Delta w_{comp}} \tag{5}$$

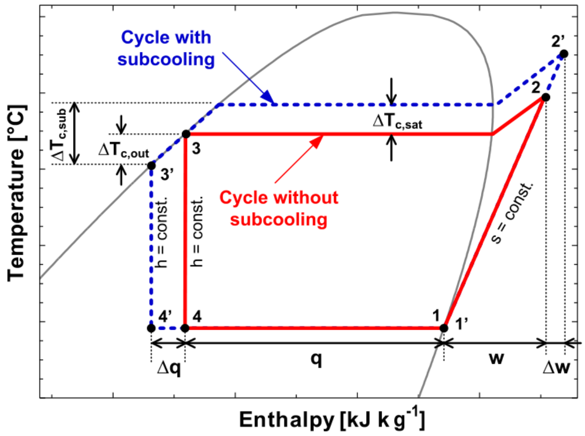


Figure 3.4: T-h diagram for cycles with and without considering condenser subcooling from Pottker and Hrnjak (2015).

where Δw_{comp} is the extra compressor work due to subcooling. From Eq. 4 and 5 one can derive:

$$COP_{sub} = COP\left(\frac{1 + \Delta q_c/q_c}{1 + \Delta w_{comp}/w_{comp}}\right) \tag{6}$$

In the present case Δq_c is to be used to heat the entering fluid at the evaporator and, therefore, it is not longer a gain into the system. Therefore, assuming that $\Delta q_c = 0$, one gets:

$$COP_{slurry} = COP\left(\frac{1}{1 + \Delta w_{comp}/w_{comp}}\right) \tag{7}$$

Therefore, the COP_{slurry} will depend on the increase of work of the compressor.

Results obtained in Pottker and Hrnjak (2015) were for an air to water chiller used for air conditioning. Therefore, it is of interest to do a similar analyses for a water to water heat pump under the relevant conditions of interest for the provision of space heating demand. As discussed before, 20 % of the heat absorbed in the evaporator needs to be provided by the subcooling heat exchanger. High subcooling degrees could be achieved without compromising the COP of the heat pump as long as the condenser saturation temperature does not increase. In the slurry system proposed, a low heat sink of energy is available at 0 °C which would allow to achieve the degree of subcooling needed without increasing significantly the compressor work. In order to support this statement stedy state calculations using a semi-physical heat pump model have been carried out with a $T_{evap,in} = 0.5$ °C. The heat pump model used does not have implemented the possibility of the subcooler heat exchanger and therefore to estimate the effect of it, two fluid inlet condenser temperatures have been used $T_{cond,in} = 35$ °C and $T_{cond,in} = 25$ °C. The outlet of the condenser has been fixed to $T_{cond,out} = 40$ °C by adapting the condenser mass flow rate. Results from Fig. 3.5 show that a lower temperature source in the condenser water allows for subcooling the refrigerant efficiently. A degree of subcooling of 14 K provided the maximum COP with very low compressor power increase and a share between subcooling and evaporator power of around 17%, which is close to what would be needed to heat the slurry water to 0.5 °C assuming a supercooling degree of 2 K. In the slurry system, the sink will be at 0 °C and to achieve a higher supercooling degree would be possible. The only practical limitation would be that some vapor exist at the entrance of the evaporator.

As a summary, it seems feasible to use condenser subcooling as heat source for preheating cold water coming from the ice storage before it is entering the evaporator, with a very low decrease in the overall heat pump COP.

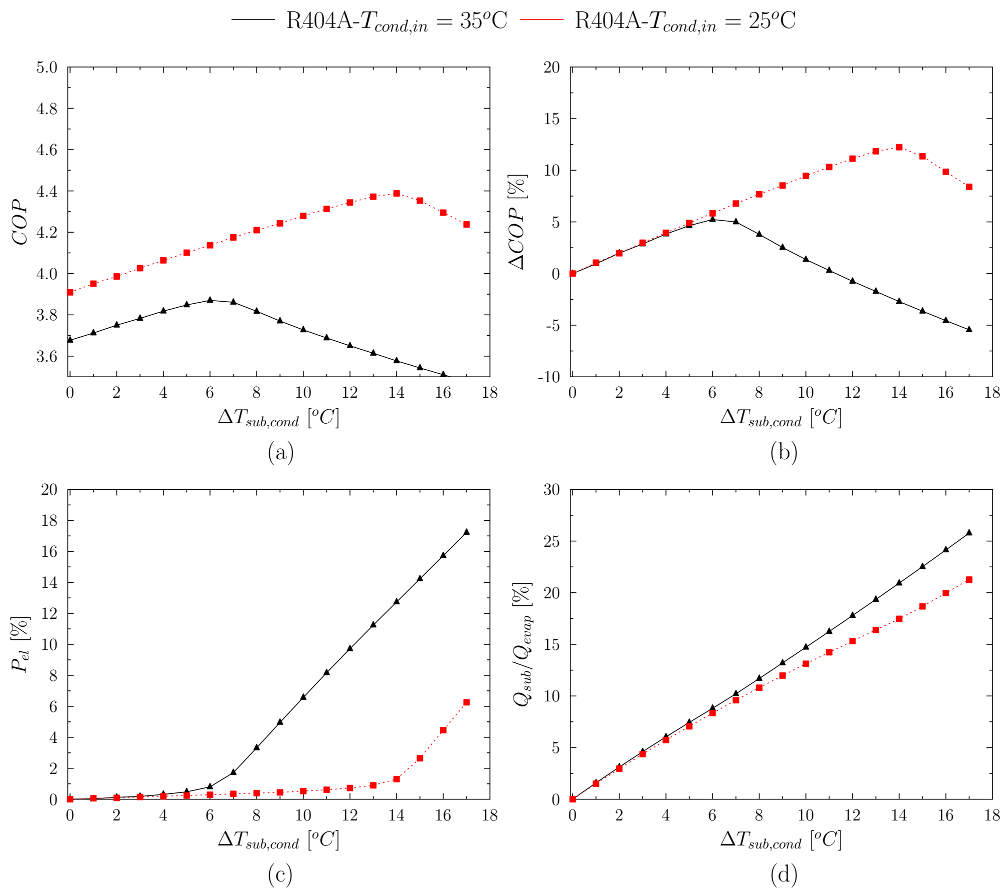


Figure 3.5: Simulation results with varying refrigerant subcooling in the condenser by using two different condenser water inlet temperatures of 25 °C and 35 °C.

4. Concepts for solar ice-slurry systems with supercooling

In the following three concepts for solar-ice slurry systems based on the supercooling principle are proposed. It was aimed to find simple and robust system designs that can result in low investment costs and easy installation and maintenance.

An important difference between the concepts is the system integration of the supercooler. Two ways are suggested for the integration:

- **Indirect supercooling**: The supercooler is an extra heat exchanger (not part of the heat pump unit). Water from the ice storage is supercooled in the primary side of the supercooler. Brine is used as a heat transfer fluid in the secondary side where it is pumped from the supercooler to the evaporator. A standard brine to water heat pump can be used.
- **Direct supercooling**: The water from the ice storage is supercooled directly in the evaporator of the heat pump. Thus, only water is used as a heat transfer fluid between the water from the ice storage and the evaporator of the heat pump. A heat pump with a special evaporator-supercooler has to be designed.

The two different types of supercooling result in different working temperatures on the evaporator side. For indirect supercooling a second heat source can be integrated into the brine cycle via a switching valve. This allows for using heat from the solar collectors on the evaporator also at temperatures considerably below -2 °C. While being unfavourable for the COP, this strategy helps reducing component sizes and cost of the system and can be even advantageous for the system seasonal performance factor as the use of the back-up system can be reduced or even avoided when no sufficient solar irradiation is available as heat source for running the heat pump (Carbonell et al., 2017). For direct supercooling, where the evaporator is used as supercooler, the temperature of the water entering the evaporator has to be kept

≥ 0 °C. This greatly reduces the solar gains in the critical periods but increases the performance of the heat pump and obviates the need for one additional heat exchanger.

For all three concepts the solar collectors are the only heat source of the system. In two of the concepts, solar heat can be used directly on the evaporator without an extra heat exchanger in the hydraulics. This ensures a high efficiency of the solar collectors when delivering heat to the evaporator. In all concepts the solar heat can be used also directly for heating the combi storage. Otherwise, the system performance would always be below the heat pump performance.

Preheating of water before it is entering the supercooler is done in all concepts with a subcooler that is integrated into the heat pump (see section 3.5). The fact that solar energy is not always available, hinder the possible use of it for this purpose. Also the use of the lowest (solar) part of a combi-storage for preheating the water is not considered further because during cold weather periods with low solar irradiation also the stored solar heat in the combi-storage will not be sufficient for the preheating.

Table 4.1 is listing the main specifications and differences of the three system concepts that are presented in the following chapters.

Table 4.1: Main differences of the proposed designs for solar ice-slurry systems.

Type of system	Type of supercooling	Fluids	T_{min} (HP inlet)	Solar as evap. source
Slurry-indirect	Indirect	water/brine/refrigerant	$\geq$ -8 °C [a]	direct
Slurry-direct	Direct	water/refrigerant	≥ 0 °C	via HX
Slurry-direct-water	Direct	water/refrigerant	≥ 0 °C	direct

[a] depending on the heat pump specifications

4.1. Concept with indirect supercooling

In the slurry-indirect system, the heat pump can be used until the minimum source temperature according to the design of the specific heat pump is reached. This is possible thanks to an external supercooler that transfers the thermal energy of the water to an intermediate brine cycle. The hydraulic scheme is presented in Fig. 4.1.

Standard brine to water heat pumps that are available on the market usually accept source temperatures down to -5 °C and can often work also with lower temperatures of approx. -10 °C. Specially designed brine to water heat pumps for heating applications can use heat sources of approx. -15 °C.

Due to the indirect supercooling, it is possible to bypass the supercooler and use heat from the (unglazed) collectors on very low temperature levels in the evaporator.

All further components needed such as supercooler, valve and pump (see box below the heat pump in Fig. 4.1) can be integrated in a pre-fabricated casing that can be installed next to the heat pump. The ice releaser should be close to the ice storage.

The ice storage is loaded by the solar collectors via heat exchanger and brine is used as heat transfer fluid in the collectors. The heat exchanger can be an immersed tube installed in the ice storage or an external plate heat exchanger.

4.2. Concept with direct supercooling and solar brine cycle

In the Slurry-direct concept, the evaporator of the heat pump is used as supercooler (Fig. 4.2). This simplifies the design of the system and reduces the number of components. However, because water is used as heat transfer fluid, the minimum temperature in the circuit is determined by the degree of supercooling that can be achieved with the supercooler (approx. 2 K).

A further consequence of the direct-evaporator concept is that the (unglazed) collectors cannot be used directly as heat source for the evaporator since brine would be mixed with water. Because collectors are connected only to the ice storage and combi-store, the working temperature in the collector loop will always be ≥ 0 °C. This will lead to lower solar gains for the direct compared to the indirect concept. Depending on the sizes of collector field and ice storage for a specific heat demand, the approach of direct

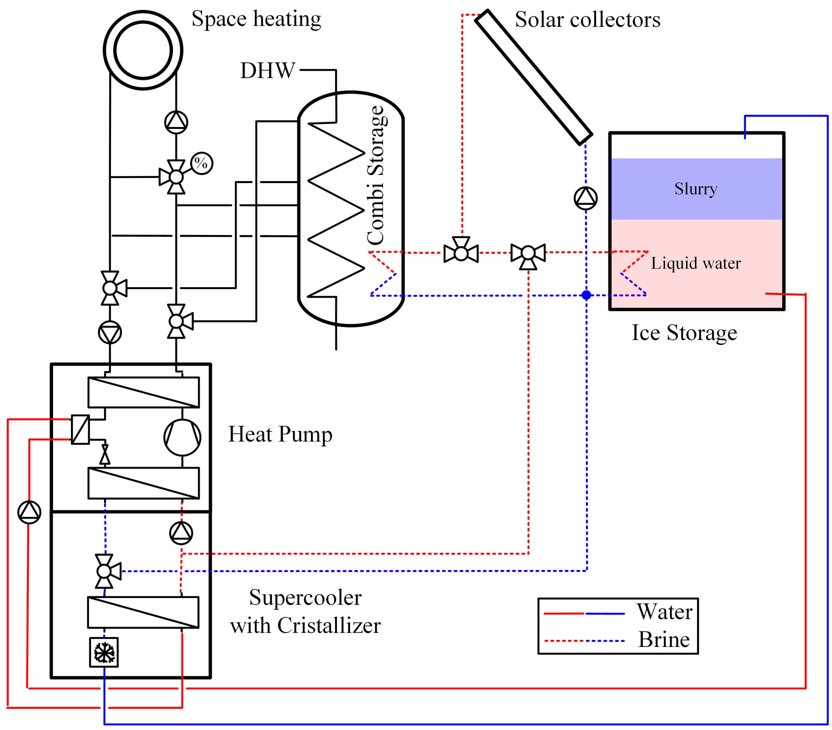


Figure 4.1: Slurry-indirect - Hydraulic scheme.

supercooling can strongly reduce the availability of heat for the evaporator in winter time which will, in turn, increase the electric back-up.

Next to the "standard" heat pump that is extended with the subcooler, as further component only the ice releaser (crystallizer) has to be installed between heat pump and ice storage. Like in the indirect concept the ice storage is loaded by solar heat via a heat exchanger.

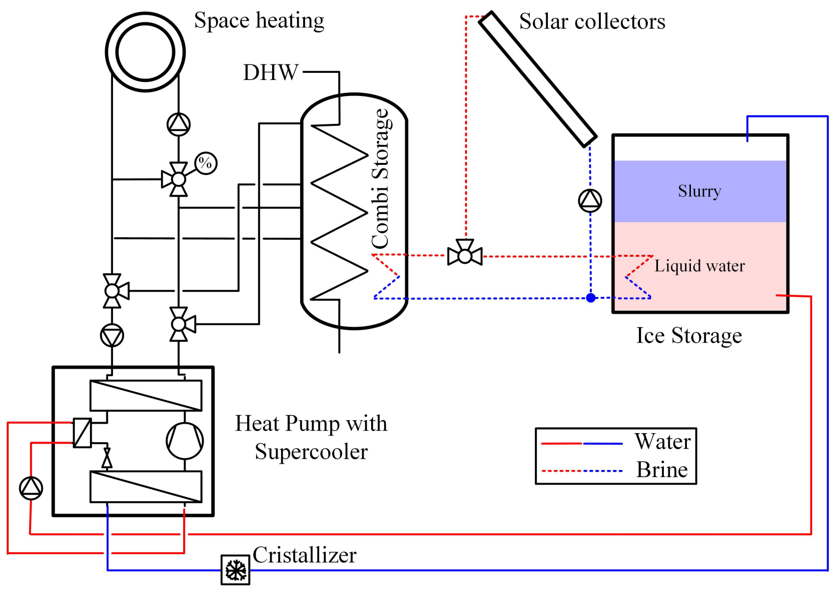


Figure 4.2: Slurry direct - Hydraulic scheme.

4.3. Concept with direct supercooling and solar water drain back

In the slurry-direct with water drain back system concept a further reduction of system components is achieved by omitting the solar heat exchanger used for loading the ice storage. As a consequence water is used as heat transfer fluid in the collectors and a drain back function has to be implemented. During times with no solar irradiation and ambient temperatures near or below 0 °C, the water in the collector cycle is drained back into the ice storage. The drain back avoids freezing of the water in the collector cycle. As no heat exchangers are implemented in the hydraulics between collectors and ice storage, high solar gains can be expected when loading the ice storage. The direct concept with drain back function has the same limitation as the concept without drain back regarding the working temperature level of the heat pump, i.e. it needs to be above 0 °C.

When starting the filling process of the drained collector cycle in the morning, the temperature of the (empty) collector pipes outside the building might be below 0 °C in winter time. For pipe lengths that are common for normal collector fields, it is expected that the latent and sensible heat capacity of the water coming from the ice storage is sufficiently high to avoid that the return pipe to the collector field is blocked with ice. However, this will only be possible if the water is above 0 °C which will not be true in winter. Drain back systems are not very common and a further technological challenge will be faced here, i.e. the combination with an ice slurry system might cause critical situations due to icing in the collector loop.

If the slurry is not stirred inside the ice storage it will accumulate an agglomerate in the upper part of the storage. If the outlet pipe in direction of the collector field is placed at the bottom of the ice storage, it can be expected that only water and no ice will be pumped to the collectors. However, pumping slurry into the collectors can be favorable as the working temperature of the collectors is reduced. The effect on the efficiency of the collectors when pumping different concentrations of slurry through the collectors is analyzed below.

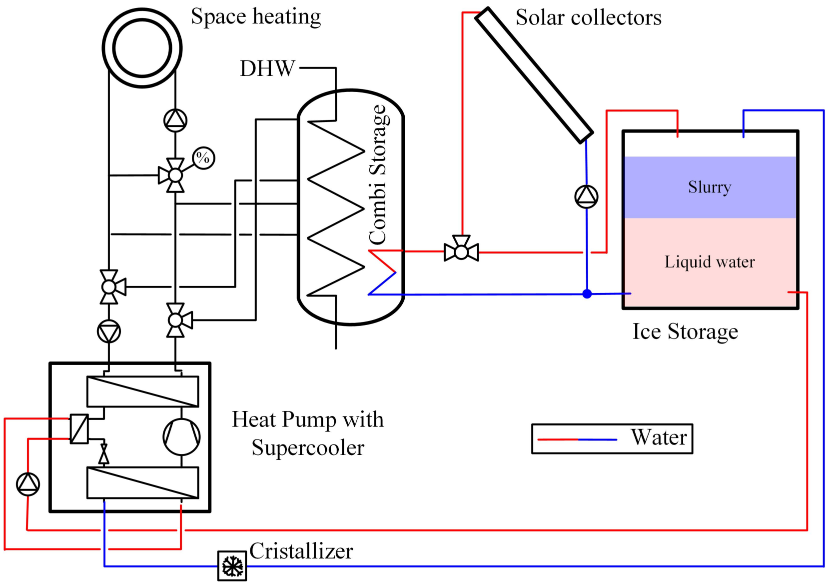


Figure 4.3: Slurry direct with water drain back – Hydraulic scheme.

Pumping slurries through the collector field

One possibility to increase the solar collector yield in winter and thus the system efficiency is to pump the ice slurries stored in the ice storage through the collectors. The heat transfer fluid including solid ice particles in this case benefits of the latent heat of fusion of water. The melting of the ice can be used to enhance the heat transfer between the fluid and the collector panel and to reduce the working temperature and thus the heat losses of the collectors.

Flat plate collectors with two phase flows have been analysed in the past. For example, collectors under boiling conditions have been studied in Abramzon et al. (1983) and El-Assy and Clark (1989).

Some applications of collectors with boiling conditions are two phase thermosyphon systems and direct expansion heat pumps, where the refrigerant is being evaporated directly in the collectors. No research has been conducted for ice slurries in solar collectors but there is one ongoing research activity using other PCM slurries to increase the solar collector efficiency. Serale et al. (2016) analyzed theoretically the improvements of the solar collectors by using different concentrations of the PCM. The carrier fluid was a mixture of water with a concentration of 40% of glycol. The PCM slurry used was n-eicosane with a melting temperature of 37 °C and a heat of fusion of 247 kJ/kg [4]. This work showed improvements of the instantaneous collector efficiency in the order of 5% for a 30% concentration of PCM. The performance of the collectors is increasing with the PCM concentration and is more prominent for low solar irradiation conditions. In order to evaluate the benefits a function as shown in Fig. 4.4 has been obtained based on results from Serale et al. (2016). The solar ice slurry based system is likely to be used in cold climates

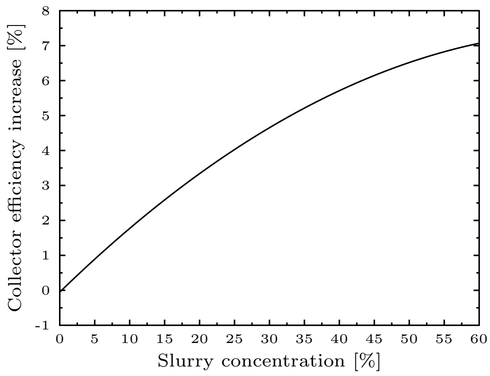


Figure 4.4: Increase of collector performance as a function of PCM slurry concentration from Serale et al. (2016).

and an increase in the range of 5% would be a benefit for the system if the increase of cost is minimal or zero. In principle, common pumps for solar loops can be used for pumping slurries with a concentration below 40%. The cited theoretical work neglected the increase of viscosity and pumping power needed by the use of the PCM, which would reduce the expected improvements. However, in an ice slurry system higher gains compared to the n-eicosane could be expected due to the 35% higher latent heat of fusion of water.

Theoretically, it should be possible to pump ice slurries though the absorber pipes of collectors which have normally inner diameters of 8 mm. Old slurry particles normally reach maximum sizes of around 1.5 mm. The pumpability of slurries depend on the size and structure of the particles. Pure water shows a dentritic structure and therefore pumping pure water is not easy. The size and structure of the ice particles are influenced by the type and concentration of additives. Therefore, an approach to pump slurries could be to add sugar in the mixture, which smooths the ice particle shape, reducing the viscosity of the resulting slurry, but with a slight decrease of the freezing point. In order to pump ice slurries the concentration of the ice particles needs to be kept below 40%. Therefore, another technological difficulty will arise when the ice storage is at 60% ice fraction, which is the maximum ice slurry concentration that can be achieved. Moreover, for this specific application, it might be necessary to add a stirring device to avoid that slurry particles agglomerate.

4.4. System extensions with melting function

One of the challenges of the supercooling technology is to keep the supercooler free from ice. The heat pump parameters and the conditions in the supercooler have to be kept within a range where a stable operation of the supercooler can be assured. If operating conditions are not within this range, a crystallization of the supercooled water within the supercooler will occur and ice will attach to the heat exchanger surface. As soon as some ice particles are formed they will serve as seeding for the supercooled water, the ice layers will grow and the supercooler will be blocked. Hence, if the process of supercooling cannot be controlled in a appropriate way, the supercooler has to be heated up periodically to melt the

[4]Heat of fusion of water: 333 kJ/kg

ice inside. Consequently, a second supercooler may be needed, which would allow to run the heat pump even if one of the two supercoolers needs to be regenerated after being blocked.

Within this feasibility study it cannot be answered if this system extension with a second supercooler will be necessary to guaranty a proper system operation. Hence, two scenarios are analysed in this study where i) the second supercooler is not needed (optimistic scenario) and ii) this extension is necessary (pessimistic scenario). The two scenarios will help assessing the potential of the technology regarding possible effort and costs with more confidence. Results for the scenarios are discussed in section 5.2.2.

With the concept of indirect slurry production a melting of blocked supercoolers with solar heat (from the lowest part of the combi storage or from the collectors directly) can be done. A possible hydraulic integration of two supercoolers with the melting function is shown in Fig 4.5. In order to operate the two supercoolers independently, a considerable number of extra components like a pump and five valves are needed.

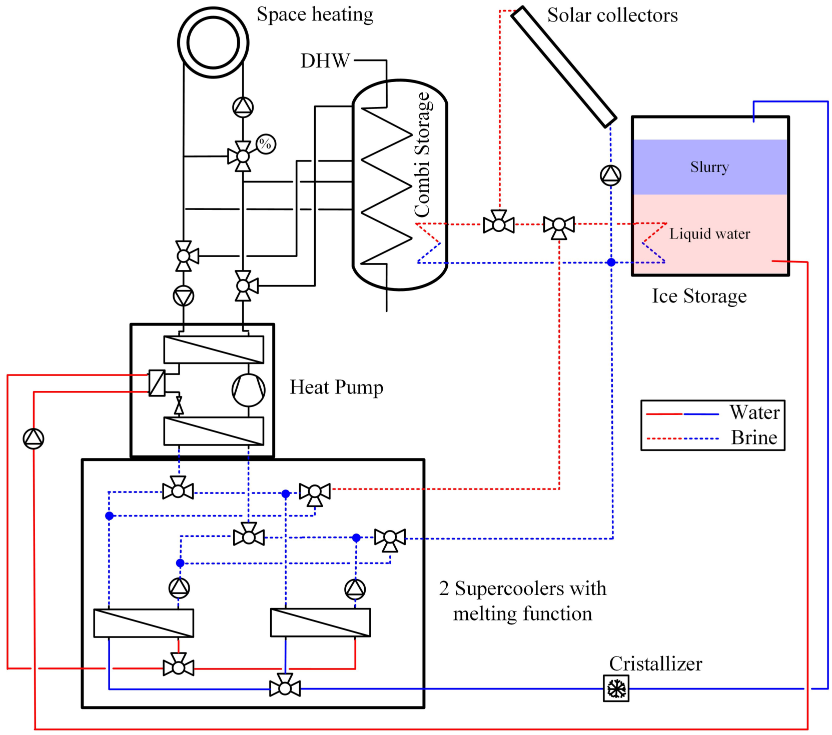


Figure 4.5: Slurry-indirect - System extension with a second supercooler and a function for melting ice in blocked supercoolers with solar heat.

With direct slurry production it is assumed that the blocked supercoolers have to be heated with electrical heating rods that are coiled around the supercoolers. A melting with solar heat is not possible with the existent hydraulics of this concept. If one of the supercooler is blocked, the refrigerant and water is directed to the other supercooler, and the blocked one is heated up by an electric heating rod. As valves are integrated into the refrigerant cycle, the heat pump has to be built accordingly (no standard heat pump). On the other hand, no further parts are needed outside the heat pump casing.

4.5. Conclusions on system concepts

In the sections above three systems have been presented. Only two of them will be further analysed. The slurry-direct system with water drain back seems to face too many technological difficulties that will hinder all expected improvements in terms of simplicity of system hydraulics and high energetic efficiency. Further, it does not seem feasible to pump ice slurries through the collectors and this option is also disregarded for the present application.

The two remaining system concepts will be analysed by means of simulations in the next chapter, one indirect with brine as heat transfer fluid in the collectors and heat pump evaporator and the direct system

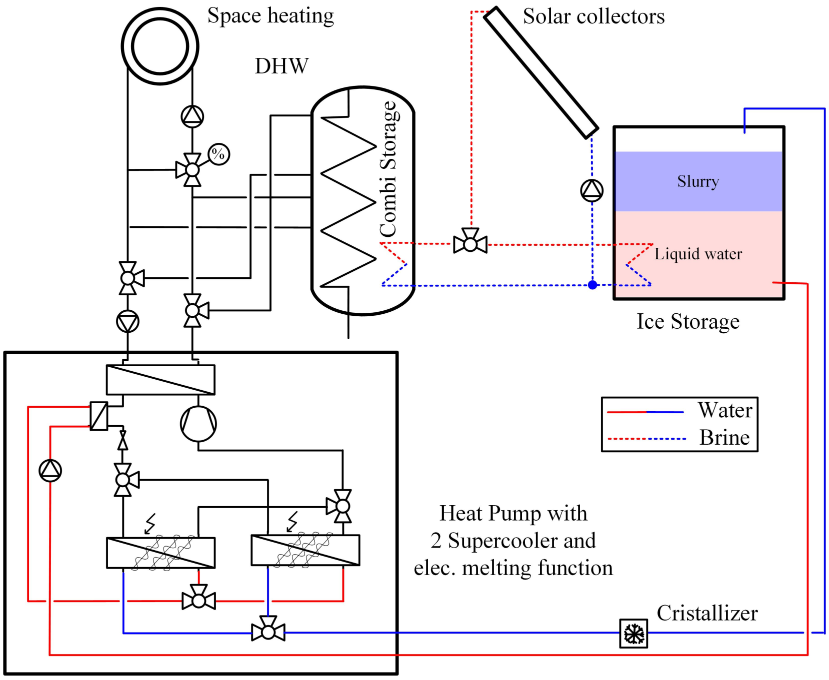


Figure 4.6: Slurry-direct - System extension with a second supercooler and a function for melting blocked supercoolers with electric rods.

with pure water in the evaporator but a brine solution in the collector loop. Both systems will use a condenser subcooler as preheating technology for the inlet water of the evaporator in winter times (icing periods). The use of heat from the collectors or from the lower (solar) part of the combi-store for this purpose seems to be less feasible due to the availablility of these heat sources. However, further studies would be necessary to confirm this assumption.

The need of a second supercooler or the frequency of the melting function described in section 4.4 can not be answered in this report. However, personal communications with experts on ice slurry allow to assume that a supercooling rate of 2 Kelvin can be stable and due to this a second supercooler might not be needed and the melting function would be used seldomly.

5. Energetic and economic system analyses

5.1. Methodology

The system simulation is based on the TRNSYS template developed within the SFOE funded project Ice-Ex (Carbonell et al., 2017). The simulated building is a single family house located in Zurich with a heat demand of 55 kWh/(m^2/a) for space heating and 15 kWh/(m^2/a) for domestic hot water. Weather data from meteorological station of Zurich Fluntern for the years 2000 and 2001 is used. Year 2000 is used to initialise the ice storage for the second year and results for the year 2001 are reported. In previous works (Carbonell et al., 2016a) it was found that results from simulations using the weather data of year 2001 were closest to the average over twelve years.

In order to consider the ice-slurry concept several simplifications have been used, as no modelling work could be done within this pre-study :

- The need of preheating the water before entering the evaporator has been neglected
- The ice storage model used has no direct ports, instead the heat exchangers implemented in the ice storage model had to be used and the ice grows on their surfaces. The conductivity of ice is set to a very large value to model the free-of-ice heat exchanger surface. Thus, in the model ice is built on the heat exchanger but the resistance of ice is negligible.

- The higher melting rates of the slurry method are considered by setting a very high heat transfer coefficient of the heat exchangers under these conditions.
- The maximum mass of ice is limited to 60% which is the limit which can be achieved with an ice slurry system.
- It is assumed that water can be supercooled without any blockage. Therefore the energy to melt a blocked supercooler has been neglected.

The main indicator to assess the energetic performance is the system performance factor calculated as described in Malenkovic et al. (2012):

$$SPF_{SHP+} = \frac{Q_{DHW} + Q_{SH}}{P_{el,T}} = \frac{Q_D}{P_{el,T}} \tag{8}$$

Q is the yearly heat load energy and $P_{el,T}$ the total yearly electric energy consumption. The subscripts SHP, DHW, SH and D stand for solar and heat pump, domestic hot water, space heating, and total demand, respectively.

The total electricity consumption is calculated as:

$$P_{el,T} = P_{el,pu} + P_{el,hp} + P_{el,cu} + P_{el,aux} + P_{el,pen} \tag{9}$$

where the subscripts *pu*, *hp*, *cu*, *aux* and *pen* refer to circulation pumps, heat pump, control unit, auxiliary and penalties, respectively. The symbol "+" in the $SHP+$ from Eq. 8 refers to the consideration of the heat distribution circulating pump in the electricity consumption.

Two reference systems are used for a comparison of the ice slurry systems. The reference systems are a typical ground source heat pump system (GSHP) and a solar-ice system where flat plate heat exchangers are used in the ice storage that are not de-iced. The heat exchanger area for the reference system is based on results from Carbonell et al. (2017), where optimum areas were found between 10-14 m^2 of heat exchanger area per m^3 of storage volume.

5.2. Economic system analyses

The cost of the systems are evaluated with the heat generation cost using the method of VDI (2012) and Bangerter (1985) with some simplifications. The net present value is used to calculate the heat generation cost. To calculate the heat generation costs the investment for the system are needed. Details about the cost and assumptions used for the specific components for the ice slurry system are provided below. Details of the calculation method and their assumptions for solar-ice systems are described in Philippen et al. (2015).

5.2.1 Investment costs for supercooling components

In the following, the cost per component needed for the ice slurry approach are listed. The values are estimations that were determined either by using real offers, by discussing with manufacturers or using data found in the literature.

Supercooler

For the supercooler two types of heat exchangers are used in the following as variants. The heat exchangers considered are i) a standard plate heat exchanger commonly used as evaporator in heat pumps, and ii) a shell in tube heat exchanger (details see Chapter 3.4). Within this feasibility study it cannot be answered which one is more applicable for the use as supercooler. A coaxial heat exchanger is also a possibility, however, reliable cost were not found.

Based on the information of two companies, the cost of the plate heat exchanger is estimated as 70 Fr./kW. The price of a shell and tube heat exchanger is significantly higher and is in the range of 300 Fr./kW.

According to a heat pump manufacturer the labour cost for mounting an evaporator in a heat pump of approx. 10 kW in the factory will be approx. 475 Fr. (at end consumer level).

Coating

Costs were estimated for a highly hydrophobic coating based on materials that contain halogens. According to companies that offer coating, PTFE (Polytetrafluoroethylene) or FEP (fluorinated ethylene propylene) could be applied. Both materials can be poured liquidly into the part of the heat exchanger that has to be coated. After dripping off, a thin layer remains that needs to be sintered (baked) at approx. 260 °C (for PTFE) or 340 °C (FEP). Hence, for brazed plate heat exchangers an appropriate plumb with melting temperature above the sintering temperature has to be used.

Depending on the roughness of the surface that is coated and the mechanical stress the fluid motion puts onto the coating, the surface of the heat exchanger has to be treated by sandblasting. Through this treatment the mechanical adhesion of the coating onto the heat exchanger surface after sintering is enhanced. Most likely, sandblasting is only possible if the plate heat exchanger is bolted and thus, the plates would need to be disassembled for pre-treatment with sandblasting. Sandblasting of the plates of the heat exchanger is labour-intensive and thus costly.

In the following it is assumed that no pre-treatment with sandblasting is needed for applying a coating on the supercooler surface. Cost offers from manufacturers for coating showed quite a high scattering, and an average cost for coating of 300 Fr. per supercooler is assumed in the cost calculations.

Crystallizer / Releaser

It is assumed that a Peltier element can be used to initiate the crystallisation of supercooled water when needed (triggered only several times per year). Cost for Peltier elements are in the range of 50 Fr. It is assumed that its hydraulic integration into the pipe between supercooler and ice storage or into the ice storage itself costs 250 Fr. for material and work. Hence, the total cost of the ice release sums up to 300 Fr.

Subcooler

According to heat pump manufacturers, inside the casing of a standard heat pump enough space is available to mount a further heat exchanger that is used as subcooler. It is assumed that adding a subcooler in heat pumps with sizes appropriate for single family houses costs 250 Fr. for labour and 42 Fr./kW for the material.

Further parts

Building up the hydraulics of the concept with indirect supercooling needs further components (see differences between Fig. 4.1 and Fig. 4.2). The components and their estimated costs are a valve (230 Fr.), a brine circulation pump (500 Fr.), and some short pipes (150 Fr.). The cost for a casing is neglected as the components could be installed next to the heat pump in the cellar like a common hydraulic installation. Two working hours are estimated for mounting (160 Fr.). Thus, the total extra costs for the indirect supercooler sum up to 1040 Fr.

If the solar loop is operated with brine (and not water), a heat exchanger has to be mounted to be able to load the ice storage with solar heat. As heat exchanger a corrugated stainless pipe can be used and the total costs for installing are estimated to be 50 Fr./m^2 of collector area.

5.2.2 Costs scenarios for the solar ice-slurry systems

The ice slurry supercooling technology faces several challenges that need to be addressed. There are several ways to overcome the challenges that may lead to different solutions, namely the kind of heat exchanger applicable as supercooler, the possible need for coating of the supercooler surface to avoid ice attachment, and a possible need for melting ice when it blocks the supercooler. All these possibilities affect the investment cost and therefore a significant uncertainty exist. For this reason two scenarios are defined with different investment cost for the solar ice slurry system:

Optimistic cost scenario:

- A plate heat exchanger can be used as supercooler.
- No coating of the supercooler surface is necessary.
- Ice blocking of the supercooler will occur seldomly and using a single supercooler is sufficient.

Pessimistic cost scenario:

- A shell and tube heat exchanger is used as supercooler.
- A coating is necessary to ensure that the supercooling state is preserved inside the supercooler.
- Ice blocking will occur frequently and thus two supercoolers (i.e. shell and tube hx) with a melting function have to be used.

Different extra investment costs are assumed for the pessimistic cost scenario, depending on the system concept (see hydraulic schemes of Fig. 4.5 and Fig. 4.6). For the pessimistic scenario of the indirect supercooling, further costs for the hydraulic connections (1050 Fr.), four additional valves (930 Fr.), extra piping (450 Fr.), and 1 day of labour cost (640 Fr.) are assumed. In total, 3060 Fr. extra costs are taken into account.

In the pessimistic scenario of the direct supercooling, extra costs for two valves in the water cycle and two valves in the refrigerant cycle are needed (in total 670 Fr.). Moreover, two electric rods for melting water in blocked supercoolers (200 Fr.), and labor costs (300 Fr.) are needed. Thus, for direct cooling total extra costs of 1170 Fr. are assumed.

The investment costs of the GSHP system are assumed as 54530 Fr. with an SPF_{SHP+} of 4.0. The investment costs and the SPF_{SHP+} of the solar-ice system are varying with respect to the system size (see Fig. 5.2).

5.3. System simulation results and cost analyses

Results for dynamic yearly simulations are shown in Fig. 5.1 for three ice storage volumes of 3 m^3, 4 m^3 and 5 m^3 and three collector areas of 15 m^2, 20 m^2 and 25 m^2. These ice storages have sizes that could fit in the cellar of a single family house. Larger ice storages of 10 m^3 are usually buried in the ground if they are used in single family houses. A relevant indicator of the size of the storage is given by the ration between the ice storage volume and the total heating demand V_{ice}/Q_D. The total yearly heat demand of the simulated building considering space heating and domestic hot water is around 9.5 MWh. Therefore, for the storages of 3 m^3, 4 m^3 and 5 m^3, the ratio V_{ice}/Q_D is 0.3, 0.4 and 0.5 m^3/MWh. The typical ice storage volume of 10 m^3 offered by Isocal for a single family house has a ratio of 1.1 m^3/MWh for the climate of Zurich and the building under consideration. Large ice storage volumes in the range of 1 m^3/MWh are typically installed in multi-family buildings since it is usually possible to find space available in the cellars, parking, etc. Then, for multi-family or tertiary buildings, the ice storage is not buried in the ground but usually built inside the building using concrete as material for the storage walls.

The yearly performance of the heat pump (SPF_{HP}) is shown in the upper graphs of Fig. 5.1. The SPF_{HP} increases linearly with the collector area. The highest SPF_{HP} are obtained with the slurry-direct system due the higher temperature source provided by the ice slurry (0 °C). The heat pump performance for the slurry systems is independent from the ice fraction of the storage. The dependency of the SPF_{HP} with storage volume, is linked with the temperature level provided by the collectors, i.e. the larger the ice storage, the lower the need of the collectors at very low temperature. The heat exchanger area in an ice-on-hx case scales with the storage volume, otherwise part of the volume would not be used for icing. Therefore, for a given mass fraction of ice, the larger the ice storage volume, the higher the heat exchanger area and the lower the ice thickness, which results in higher temperatures in the evaporator and better SPF_{HP}.

In slurry systems, the outlet temperature from the ice storage is always at 0 °C and thus independent of the actual ice fraction of the storage. The indirect slurry system has an SPF_{HP} lower than the direct system because there is a heat exchanger (supercooler) between the evaporator and the ice storage, which

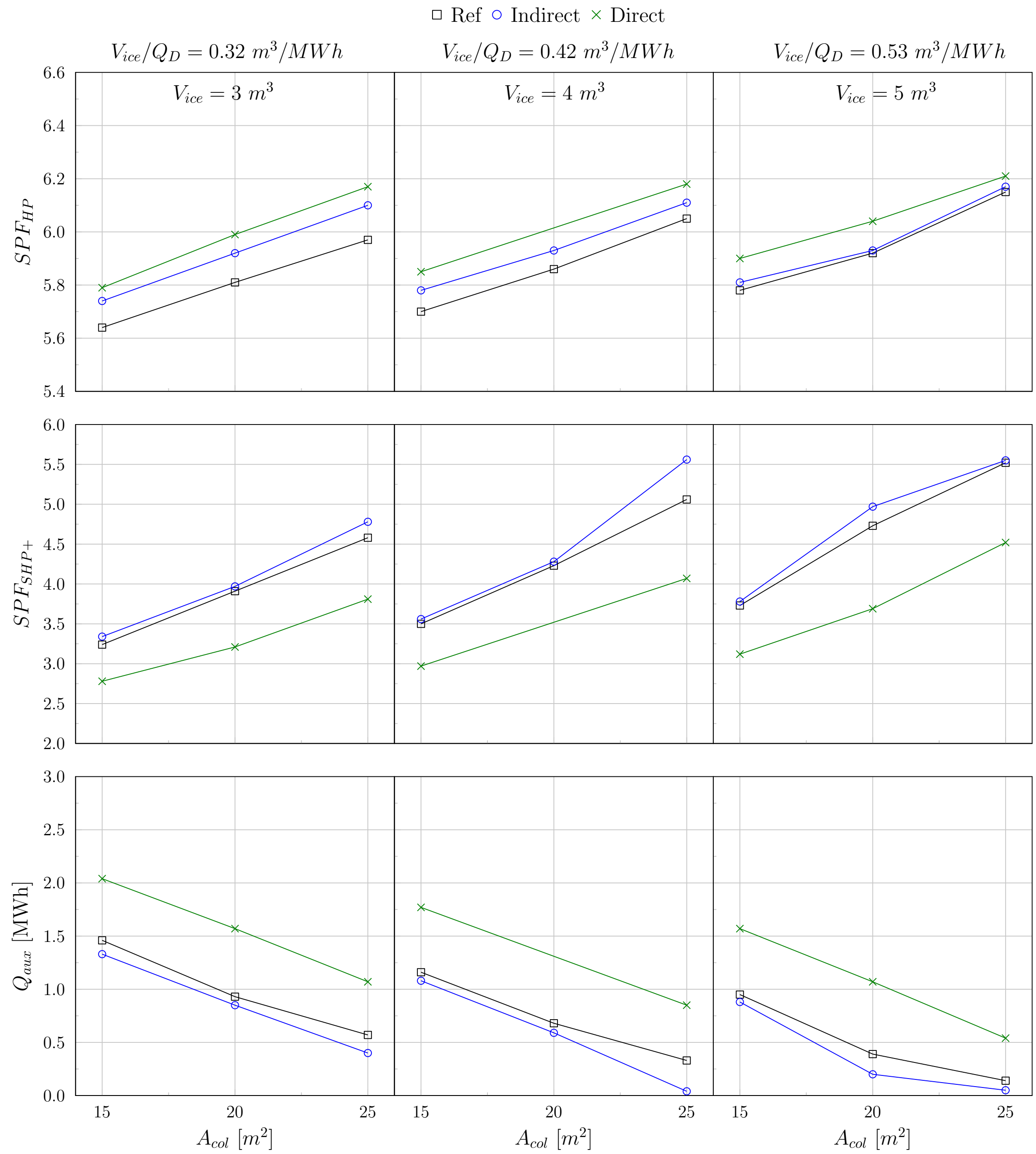


Figure 5.1: Comparison of (up) yearly heat pump performance, (mid) yearly system performance and (down) electric back up for Reference (ice-on-hx), Slurry direct and indirect systems for (left) V_{ice}=3 m^3, (mid) V_{ice}=4 m^3 and (right) V_{ice}=5 m^3.

is not needed in the direct system where the evaporator is the supercooler. Nevertheless, the $\mathrm{SPF_{HP}}$ of the indirect-slurry system is higher than that of the ice-on-hx solution. Besides the ice storage, the collectors are used as source for the heat pump. Thereby, the higher the collector area the higher the temperature in the evaporator and the higher the $\mathrm{SPF_{HP}}$. This applies for both, reference and slurry systems.

The yearly performance of the system $\mathrm{SPF_{SHP+}}$ is shown in the middle graphs of Fig. 5.1. Though the slurry-direct system has the higher heat pump performance $\mathrm{SPF_{HP}}$, the system performance $\mathrm{SPF_{SHP+}}$ is the lowest for the component sizes shown in the graph. The main reason is that this system uses more electric back-up due to the lower solar gains achieved by the limitation of having the whole circuit with temperatures ≥ 0 °C (a more detailed explanation is given in the next paragraph). The best results are obtained with the indirect system with increases of $\mathrm{SPF_{SHP+}}$ just slightly above the ice-on-hx reference system. Therefore, it seems that the increase of performance at the heat pump level achieved from the indirect system compared to the ice-on-hx case does not bring a significant improvement at system level for the component sizes analysed.

The direct electric back-up is shown in the bottom part of Fig. 5.1. The lines for electric back up look the same as the $\mathrm{SPF_{SHP+}}$ but with opposite slope. This basically means that the most important factor for the system performance is the reduction of the electric back-up. This is the main reason why the better $\mathrm{SPF_{HP}}$ observed in the direct system does not lead to a better $\mathrm{SPF_{SHP+}}$ of the system. The direct system needs more back-up because all solar and ambient energy that could be used at temperature levels below 0 °C are lost, while in the indirect system low temperatures ($\leq$ -2 °C) from the collectors can be used as low-grade source for the heat pump evaporator. This reduction of the back-up is the most important factor when designing a solar-ice system with relatively small component sizes. For large component sizes, where the direct electric back-up is not used, there are other factors that have a high influence on the performance of the system, such as $\mathrm{SPF_{HP}}$, use of direct solar heat in winter periods, etc.

Ice slurry systems will benefit more at increased volume ratios $\mathrm{V_{ice}/Q_D}$ because the heat exchanger size is not related to the storage volume[5].

The share of the investment cost for a solar-ice system is shown in Fig. 5.2 for the reference (ice-on-hx) and the indirect system for the three ice storage volumes and a collector area of 20 m^2. System performances $\mathrm{SPF_{SHP+}}$ are provided above each column of Fig. 5.2. For the calculation of the investment

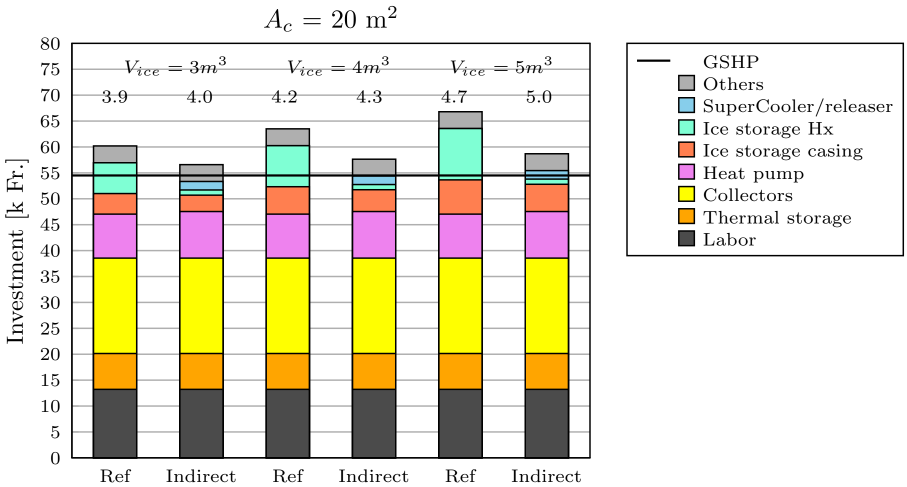


Figure 5.2: Comparison of investment cost between a reference case (ice-on-hx) and a slurry-direct solutions for cases with $A_c = 20$ m^2. Values above each column are the $\mathrm{SPF_{SHP+}}$. GSHP investment cost are shown as a solid horizontal line.

cost of the indirect system, the optimistic approach is used (see chapter 5.2.2). The comparison between optimistic and pessimistic calculations are provided below. The ice-on-hx system is assumed to use a stainless steel cylinder as storage casing while the slurry-indirect concept is using a polypropylene tank used for oil storage. Notice that ice-on-hx solutions need a robust storage able to support a structure that holds the ice below the water surface. This is not necessary in the slurry storage. The investment cost of the indirect-slurry system is always below the ice-on-hx system and the improvement of cost increases by increasing the ice storage volume. For all these cases, besides having a lower investment cost, the indirect-slurry system has a slightly higher $\mathrm{SPF_{SHP+}}$ than the reference ice-on-hx solution. However, as discussed before, the increase of $\mathrm{SPF_{SHP+}}$ is usually not very significant. An important conclusion from these results is that higher volumes can be used in the slurry system and thus the $\mathrm{SPF_{SHP+}}$ can be improved, without increasing significantly the system cost. For example, from Fig. 5.2, it can be seen that the investment cost of the indirect-slurry system with $\mathrm{V_{ice}} = 5$ m^3 is lower than that of the ice-on-hx solution with $\mathrm{V_{ice}}$ of 3 m^3 while the $\mathrm{SPF_{SHP+}}$ reaches 5 instead of 3.9. Therefore, for ice slurry systems it is advisable to increase the ratio of $\mathrm{V_{ice}/Q_D}$.

[5]Some area is still needed to bring the solar heat into the storage, but this area is related to the collector field and not to storage volume.

The investment cost of a GSHP system are shown as a solid horizontal black line in Fig. 5.2. Even though the investment cost for the slurry systems are always above the GSHP system, this can be compensated by increasing the V_{ice}/Q_D and thus the SPF_{SHP+}. As it will be shown afterwards, increasing the storage volume has the potential to reduce the heat generation cost during the life time of the system with respect to the GSHP solution by reducing the electricity demand.

Simulations of heat generation costs and SPF_{SHP+} using higher storage volumes in the order of 8 m^3, 10 m^3 and 12 m^3 are shown in Fig. 5.3 as a function of the collector area, along with the corresponding reference cost of the GSHP system. For these storage sizes, the cost of the ice-on-hx systems are usually above the plotted range and thus not shown. For this cost comparison, sizes used in commercially available state-of-the-art solar-ice systems are used. The ice storage volumes correspond to ratios of V_{ice}/Q_D between 0.8-1.3 m^3/MWh. As references, ratios of V_{ice}/Q_D of 2 m^3/MWh were installed in the pilot plants for a kindergarten (Philippen et al., 2014) and for a multi-family building recently set into operation. Both pilot plants are located in Rapperswil-Jona and are using a thermal de-icing concept (Philippen et al., 2012).

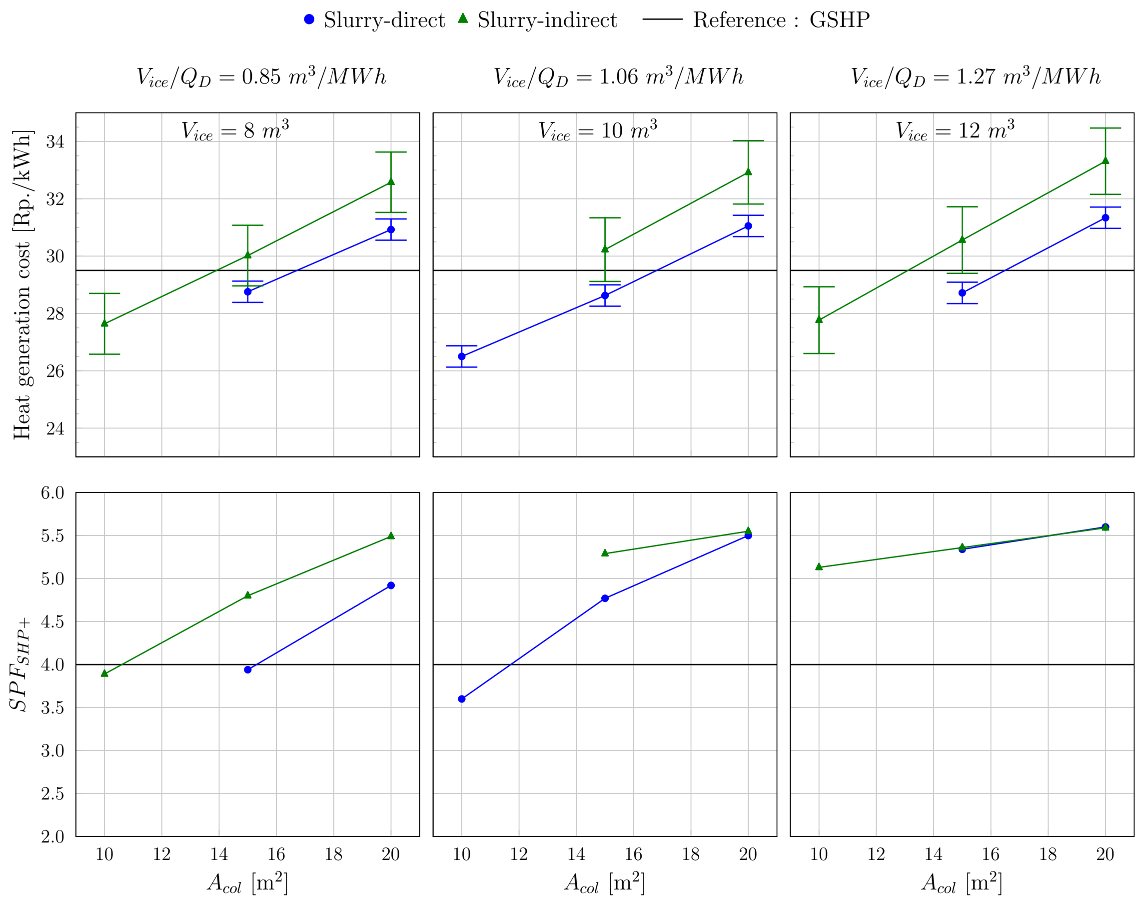


Figure 5.3: Comparison of the heat generation cost between a reference (GSHP), slurry-direct and slurry-indirect system.

As explained in section 5.2.2, in order to calculate the cost of the slurry systems, two scenarios are considered, an optimistic and a pessimistic one. These scenarios are considered as an uncertainty in the cost calculations and shown as error bars in the upper graphs of Fig. 5.3. The pessimistic scenario is shown with the upper end of the error bar with higher costs.

In the bottom graphs the performance of the SPF_{SHP+} is shown. The main interest relies on systems with an $SPF_{SHP} \geq 4$. Simulations with 10 m^3 of storage volume and 15 m^2 of collector area show an SPF_{SHP+} of 4.7 and 5.3 for the direct and indirect system, while heat generation costs are in the order of a GSHP system. For the direct system, both the optimistic and pessimistic scenario show a heat generation cost below the GSHP reference cost. For the indirect system, the optimistic is below and the pessimistic above the costs of the GSHP system, but in the same order of magnitude. Increasing the ice storage volume to 12 m^3 and reducing the collector field to 10 m^2 allows to achieve values of SPF_{SHP+} in the order of 5.2 for the indirect slurry systems with costs that are 2 Rp./kWh below the GSHP. As said before, increasing the storage volume in the ice slurry concepts proposed here, is of great benefit from an economic and energetic point of view.

For the simulated cases of Fig. 5.3, the storage casing is assumed to be built-on-place inside the building, which can not be realised in a single family house but it is common in multi-family buildings, where sizes of V_{ice}/Q_D around 1 m^3/MWh or higher can be used. It is clear that for multi-family buildings the reference cost of the GSHP will be lower, but also the cost of the slurry-system will be lower, since here, all cost are assumed for a single family size and, therefore, the economy of scale will apply to both systems.

It is also worth to discuss results for solar-ice systems obtained in the SFOE project Ice-Ex (Carbonell et al., 2017). In this project different heat exchanger types in ice storages were experimentally analysed. The most promising heat exchangers, capillary mats (CM) and flat plates of stainless steel (FP-SS) were simulated at system level. The system performance SPF_{SHP+} and the heat generation cost obtained in the Ice-Ex project are shown in Fig. 5.4. Results from Ice-Ex are slightly better than those obtained in this report due to one main reason, i.e. the maximum ice fraction was estimated to be 80 %. This assumption was based on the fact that experimental results showed that ice fractions up to 90 % were possible in the laboratory without damaging the ice storage casing. However, the state-of-the-art systems do not usually ice as much as 80 %, with the exception of one system of one manufacturer with a very small ice storages of 300 l. In any case, it is worth to compare a possible *future* ice-on-hx storage able to ice 80 % with the slurry systems able to ice only to 60 %. As can be observed in Fig. 5.4, the heat generation cost of ice-on-hx solutions can be below the GSHP solution. There are two combinations using capillary mats and 15 m^2 of collectors with storage volumes of 3 m^3 and 4 m^3 able to have a lower heat generation cost than the GSHP with $SPF_{SHP+} \geq 4$. None of the solutions using flat plate heat exchangers made of stainless steel were able to reduce the cost with respect to the GSHP solution, although they were not too far from the reference, i.e. about 2 Rp./kWh higher than the GSHP for V_{ice} = 4 m^3 and A_c=20 m^2. The lowest heat generation cost was obtained with a capillary mat with V_{ice} = 4 m^3 and A_c=15 m^2 with 29 Rp./kWh and an SPF_{SHP+} of 4.1. However, the slurry system shown in Fig. 5.3, achieved a heat generation cost of around 28 Rp./kWh with an SPF_{SHP+} in the range of 5.2. Therefore, even comparing with the best simulated solar-ice system using capillary mats (not realized yet in practice), the slurry system would be quite advantageous if a larger ice storage can be installed.

6. Conclusions and discussion

A feasibility study analysing the potential of supercooling ice slurry systems for solar heating applications has been carried out. A literature review of several ice slurry system concepts was used that confirmed that the supercooling method is, amongst the slurry technologies, the most appropriate method for the provision of space heating and domestic hot water using solar energy as a main source.

Due to its high efficiency and simplicity, the supercooling method is a very promising technology. Three system concepts using the supercooling method have been discussed. Among them, the direct system with water as heat transfer fluid and with a drain back function was estimated to be difficult to realise. The two other systems using a glycol solution in the collector loop were analysed from an energetic and economic point of view. Dynamic yearly simulations using TRNSYS were used to analyse the energetic performance. The installation cost of each system was estimated and the heat generation cost was evaluated for all simulated cases. As references, a solar-ice system using ice-on-plate heat exchangers and a ground source heat pump (GSHP) system were used. The proposed ice slurry systems showed to be advantageous with respect to both reference cases when sized appropriately.

The first system simulated, called slurry-direct, uses water as heat transfer fluid between the ice storage and the heat pump, but a glycol mixture between the solar collectors and the ice storage. Therefore, the ice storage has a coil heat exchanger to transfer heat from the solar collectors to the storage water. For the slurry-direct concept, the evaporator of the heat pump is the supercooler, i.e. only one heat exchanger exist (the evaporator of the heat pump) between the refrigerant and the water of the ice storage. A second system, called slurry-indirect has also been analysed. In the indirect system, a further heat exchanger is used as supercooler between the ice storage and the heat pump evaporator. A brine loop is used between

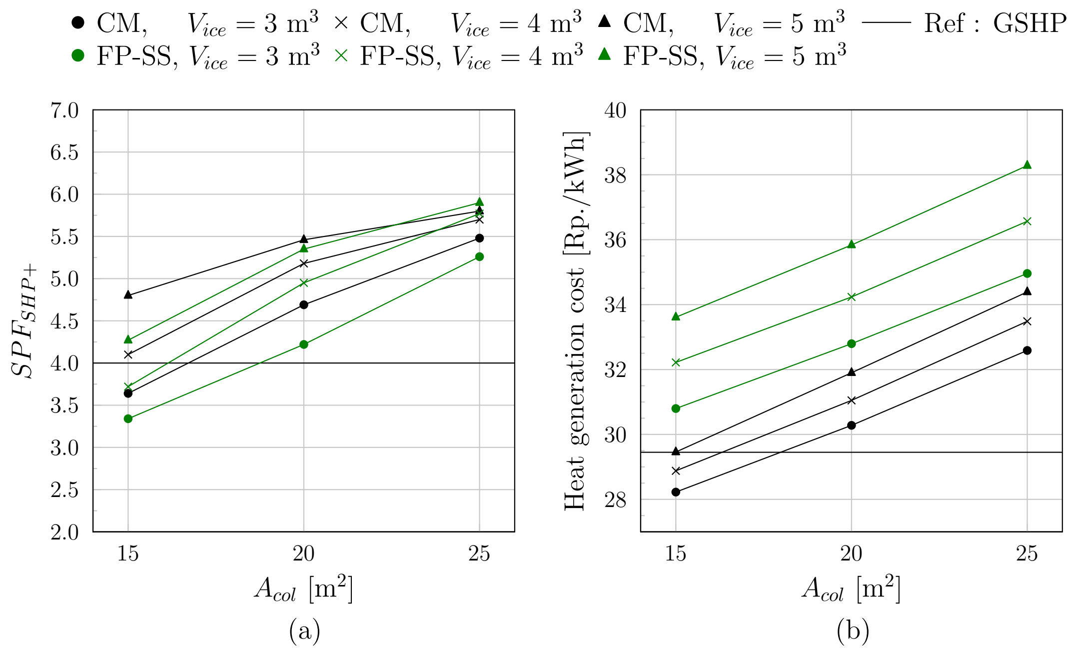


Figure 5.4: Comparison of cost heat generation cost between a Reference (GSHP), and two ice-on-hx storages, i.e capillary mats (CM) and flat plates of stainless steel (FP-SS).

the supercooler and the heat pump with a three way valve which connects the solar circuit. Water is used as a heat transfer fluid between the supercooler and the ice storage. The main difference is that the direct system needs working temperatures ≥ 0 °C[6] because water is used. In the indirect system it is possible to use solar heat at temperatures well below 0 °C. The use of very low-temperature solar and ambient heat allows the indirect slurry system to perform better than the direct system for relatively small sizes of ice storage and collector field. However, for ice storages that are large enough, the direct slurry system, which is simpler in terms of hydraulics and components, is more cost economic.

Simulations and cost analyses of the supercooling method were carried out and compared to ice-on-hx and GSHP solutions. Higher efficiencies of the slurry-indirect systems compared to ice-on-hx systems were obtained using low ice storage volumes in the order of 0.3-0.5 m^3/MWh, i.e. 2 - 12 m^3. However, differences in $\mathrm{SPF_{SHP+}}$ alone are probably not large enough to justify the risks associated to the supercooling method. The slurry supercooling method has its main potential in the reduction of the cost. The cost savings are related to the simple storage casing and the very low heat exchanger area needed in the ice storage, since only a coil to transfer the solar heat is necessary. This potential for cost reduction is largest for ratios of $\mathrm{V_{ice}/Q_D}$ in the order of 1 m^3/MWh or higher, where results of the ice slurry systems were found to be most satisfactory. Heat generation cost around 28 Rp./kWh, which is 1.5 Rp./kWh below the reference cost of GSHP system, were obtained for system configurations able to achieve an $\mathrm{SPF_{SHP+}}$ of 5.2 with an ice storage volume of 1.2 m^3/MWh and a collector area of 1 /MWh. Therefore, higher system performances can be achieved at competitive cost if the cellar allows for a storage in the range of 1 m^3/MWh or higher, which is possible on multi-family or tertiary buildings, but may be difficult in single family homes. Therefore, for multi-family or tertiary buildings, the ice slurry solution using the supercooling method seems to be a promising technology.

For single family houses where ice storages in the size of 3-4 m^3 can be placed in the cellar, an ice-on-hx solution using a very cheap heat exchanger type such as capillary mats could be the most economical solution while achieving a good system performance. However, this is true if high ice fractions in the order of 80% or higher, can be achieved in the ice storage[7]. Furthermore, achieving high ice fractions within ice-on-hx solutions, although experimentally demonstrated (Carbonell et al., 2017), face some practical/engineering challenges:

[6] Right after the supercooler the temperature will decrease to -2 or -3 °C

[7] The ice slurry solution has a maximum ice fraction of approx. 60 %

- Find a prefabricated casing that fits the specific heat exchanger design (coil, capillary mat, flat plate) is often not easy and most probably the geometry of the casing leads to regions in the storage where latent heat can not be extracted. If a casing is designed for a specific heat exchanger, higher cost can be expected.
- Another aspect is that storages have discrete sizes. Given a specific manufacturer, it is possible to find a storage casing with, e.g. 2 m^3 or 4 m^3, but not necessarily 3 m^3. Because all the volume needs to be filled with heat exchangers, the cost may increase by using a storages that are larger than the optimal size.
- For an efficient use of the space of the cellar, a rectangular cross section with a height slightly below the height of the cellar is often the best solution for an ice storage. However, the rectangular casing may not fit well the heat exchanger design, e.g. coils. Therefore, there is a solution of compromise between the efficient use of the cellar volume and the ratio of storage volume used for icing.

All these practical problems discussed above are elegantly solved by the supercooling slurry solution, since ice generation and storing are decoupled in two different devices, supercooler-ice releaser and ice storage respectively. The ice storage can be adapted to the available space in the building and it is not restricted by any heat exchanger size or design. Therefore, the 60 % ice fraction can be achieved independently of the storage shape. For this reason, even for single family houses, the ice slurry super-cooling method can be an attractive solution.

The potential benefits of the slurry supercooling method for solar-ice applications will only be possible if the technological barriers are solved. The main technological challenges for the supercooling method consist in achieving a stable and sufficiently high supercooling degree without cristallisation of the ice particles in the supercooler or before entering the ice releaser. In order to avoid spontaneous nucleation of ice particles in the supercooler or to reduce its frequency, several aspects need to be considered:

- Design a robust supercooling heat exchanger with suitable surface properties (smooth surfaces and/or suitable coatings).
- Controlling the minimum wall temperature of the supercooler in a narrow range (e.g. 0.5 K).
- Keep the water cycle clean in order to avoid fouling or scaling on the supercooler surface.
- Ensuring a robust method to preheat the water before entering the evaporator.

Besides ensuring a stable degree of supercooling, other challenges of the supercooling method are:

- Ensuring a large heat transfer area while being compact in order to provide the heat demands of the evaporator.
- Find an optimum ratio between the degree of supercooling and the mass flow rate of the supercooled water in order to reach high energy efficiency.
- Design a defrosting mechanism to solve the blockage of the circuit when spontaneous nucleation occurs.

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