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Demand side management through latent thermal storage in HVAC systems coupled with commercial refrigeration units

Giovanni CORTELLA*, Gabriele TOFFOLETTI, Michele LIBRALATO, Paola D'AGARO

DPIA – University of Udine, Italy.
*Corresponding author: giovanni.cortella@uniud.it

ABSTRACT

The electrical energy demand of an HVAC plant can be better managed by using latent thermal energy storage when time-of-use tariffs or peak tariffs are in force, in a view of Demand Side Management of the electrical grid. Nonetheless, air conditioning systems show a marked use of electrical power during the day, and the peak in the cooling load mostly corresponds to the lowest performance of the chiller due to outdoor conditions, thus giving rise to a marked peak in electricity use.

An HVAC plant of a supermarket is supplied with an ice thermal energy storage, with the main aim of shaving the peak in electrical power use. The latent thermal storage is charged at night-time by employing the CO_2 commercial refrigeration system of the supermarket, which is considerably part-loaded during the shop closing time. During daytime, the thermal storage can be operated in replacement of or in parallel to a reversible heat pump, operating as a water chiller for air conditioning. The same heat pump operates at wintertime for heating purposes, in parallel with heat recovery from the CO_2 commercial refrigeration plant.

A model of the whole system is presented, and possible solutions are shown for demand side management purposes.

Keywords: modelling, latent heat storage, demand side management, HVAC, commercial refrigeration

1. INTRODUCTION

The profile of electrical power use in a shopping mall is strongly uneven and subject to considerable daily fluctuations. At night-time the energy demand is very low, thanks to the reduced refrigerating capacity required by food storage equipment and to the HVAC system being idle; during the day, high refrigeration and air conditioning loads occur almost simultaneously, early in the morning or around noon.

Thermal energy storage (TES) is therefore suggested to shift loads, in order to achieve a better daily average energy efficiency, to take advantage of time-of-use tariffs, and to allow some reduction in the design capacity of the systems. Thermal energy storage for the refrigeration duty is not considered in this manuscript. As far as air conditioning is concerned, passive (often PCM) elements on the air side can be used for their simplicity and reliability, but with the main purpose of dumping supply air temperature during on-off cycles or defrosting periods of heat pumps. Higher storage capacity can be easily achieved by using water storage; in some cases, a huge water reservoir for fire prevention is available, and can be effectively used for this purpose. The authors investigated its application to subcool a refrigerating system (Polzot et al., 2015) or to act as a source for a heat pump in heating operation (Polzot et al., 2016). Instead, in the case of air conditioning the typical operating temperatures reduce the feasibility options for a TES. In fact, the usual supply water temperature for air handling units in cooling operation is around 7 °C, which is a limiting factor for the water temperature range in the heat storage, in the view of avoiding any risk of freezing. For this reason, the maximum heat storage in water is around 20 MJ/m^3, provided that no heat exchanger is inserted between the storage and the user. The volumetric storage capacity can be significantly increased by Ice Thermal Energy Storage (ITES) systems, where values around 170 MJ/m^3 can be easily reached even considering heat loss and the need to guarantee water flow within ice coils. Despite the need to perform cooling at below zero temperature, which seems ineffective when compared to the typical evaporating temperature for air conditioning purposes, such systems have been investigated from both energy and exergy points of view, and encountered some interest (Sanaye and Shirazi, 2013; Yau and Rismanchi, 2012). However, their performance and energy effectiveness are strictly correlated to their control rules, (Beghi et al., 2014; Candanedo et al., 2013) which involve a clear definition of the aim of the system and a thorough prediction of the user demand profile.

In this paper a configuration is considered where the air conditioning load is faced by a typical water chiller designed for 12 – 7 °C water temperature, but an ITES is used as thermal storage taking advantage of the availability of low temperature cooling power from the commercial refrigeration unit. The purpose of this configuration resides mainly in shaving the peaks of electrical power, either by replacing the chiller operation during some peak hours, or by operating the ITES in parallel with the chiller. The system is modelled through in-house routines which are linked to a more comprehensive tool of simulation of the thermal behaviour of buildings and of the commercial refrigeration plant.

2. THE SYSTEM

A supermarket is supplied with a typical CO_2 Commercial Refrigeration Unit (CRU), for the purpose of chilled and frozen food display and storage. A reversible heat pump (HP) operating on R-410A is used for the HVAC system of the sales area and of the back of the shop (warehouse, food processing, offices….). In the supermarket a bakery and a deli shop are available, whose ovens and heaters are operated to cook prepared meals early in the morning, when also the air conditioning systems operates at full power to restore indoor temperature conditions after the night pause. This results in a huge peak of electrical energy absorption, when fares are at the highest level.

In the search for a reduction in the peak of electrical energy use, an Ice Thermal Energy Storage (ITES) has been conceived to allow a significant energy storage in favour of air conditioning. Ice is formed by means of the CRU during night-time, when it is operating at favourable climate conditions and at partial load. The system sketched in Figure 1 is then considered. The ITES consists of a water tank with two submerged evaporators for ice formation, connected in parallel to the cabinets for chilled food.

The ITES is connected through a heat exchanger to a water tank which supplies the Air Handling Units (AHU). The heat exchanger is needed to uncouple the two mass flow rates, and allows the use of differently treated water in the two circuits. The reversible heat pump HP, in summertime operating as a chiller, is also connected to the same water tank, in parallel to the ITES, to meet the whole AC demand. Further heat exchangers are used for heat recovery from the CRU in wintertime, using in this case the ITES in parallel to the water tank to allow some sensible heat storage. However, this manuscript deals only with summertime operation when ice is stored in the ITES.

More in detail, the ITES consists of a 12 m^3 water tank, thermally insulated to reduce heat loss, supplied with two submerged packages of 12 coils each, with 21.3 mm outer diameter for an overall length of 720 m. They represent two separate evaporators for the CRU at -10 °C, each controlled by an electronic expansion valve. About 6000 kg of ice can be formed, allowing for a global latent storage of 2000 MJ. An air blower supplying around 100 m^3/h air at ambient temperature is provided, distributing air at the bottom of the water tank in order to improve convection on the water side during ice melting for the discharge phase. In this phase, heat storage is considered useful until water temperature raises up to 5 °C, so as to obtain chilled water at 7 °C at the HX, to be delivered at the Air Handling Units (AHU). Water is then supposed to return from AHU at 12 °C, thus resulting in a return water temperature at the ITES at around 10 °C.

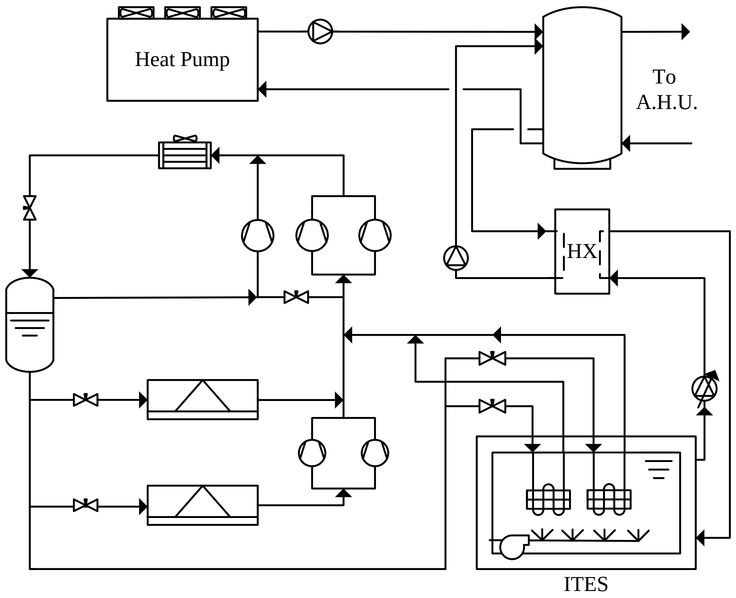

Figure 1. Sketch of the system

3. THE MODEL

3.1. Refrigeration unit and heat pump

To investigate the effectiveness of ITES in terms of daily energy use or peak shifting, a comprehensive model is used. A prediction of the annual cooling load profile with an hourly time step of display cabinets and cold rooms is performed, starting from the indoor climate conditions (which come, eventually, from another tool dedicated to building dynamic simulation). The commercial refrigerating unit is modelled in TRNSYS with in-house types for all components. The model can simulate refrigerating units equipped with the most widespread solutions to improve efficiency of transcritical cycles, among which subcooling via a dedicated mechanical system (Cortella et al. 2021), parallel compression (Gullo et al., 2016) and also with heat recovery facilities to allow domestic hot water (DHW) production and space heating and cooling (D'Agaro et al., 2018, 2019) also in the view of a Demand Side Management for the electrical grid (Coccia et al., 2019). The commercial refrigerating unit considered in this paper is a CO_2 transcritical refrigerating plant with parallel compressor, with two temperature levels, for the frozen (-35 °C) and chilled (-10 °C) food. The reversible heat pump is a commercial product whose performance is predicted from manufacturer data as a function of the source/supply heat exchanger temperature, and load. This model, implemented in the TRNSYS simulation environment, is linked to a code in Matlab devoted to the simulation of the ITES as described below.

3.2. Ice Thermal Energy Storage (ITES)

In the ITES considered, ice is formed externally of the coils submerged into the tank (ice-on coil external melt). The charge and discharge phases of the Ice Thermal Energy Storage have been simulated through heat and mass balances with a thorough heat transfer analysis, specifically adapted to the operating conditions in the water tank.

3.2.1. Charge phase

In the charge phase, water in the tank is first cooled down to 0 °C, and then ice forms externally on the cooling coils. These are fed by a mixture of liquid and vapour CO_2 in equilibrium at the evaporating temperature of -10 °C.

In the simulation reported in this paper, the initial water temperature is assumed to be 7°C. The estimation of the time needed to reduce water temperature down to the freezing point is based on a simple energy balance once the refrigerating power is known which is 70 kW in this case. In fact, measured data on the refrigerant side show that the evaporator is able to exchange the full capacity during the cooling phase. The whole load of water is considered, because at the end of the charge phase water is found experimentally to be at the freezing point in the whole volume.

Ice formation over the coils is predicted through the well-known analytical model by London and Seban (1943). Water is assumed to be at its freezing point, and the heat flow rate per unit length, flowing through the resistance composed by ice and the internal convection, in series, is:

$$q' = \Delta t_0/(R'_{ice} + R'_0) \qquad \text{Eq. (1)}$$

$\Delta t_0$is the temperature difference between the temperature of water at the freezing point (0°C) and the evaporating temperature, R'_{ice} is the conductive resistance of ice and R'_0 is the internal convective resistance per unit length, on the refrigerant side, assumed constant on the evaporator length and equal to its average value. In this model the external convective resistance (water side) is neglected because of the assumption on the initial temperature of water at the freezing point, and the conductive resistance of the evaporator tube is negligible.

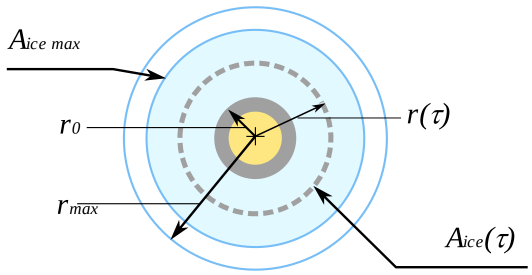

Figure 2: Schematisation of ice formation on the coil

The heat flow rate per unit length provides the extraction of the latent heat of freezing at the surface $A_{ice}(\tau)$:

$$q' = \frac{dM_{ice}}{d\tau}\lambda_{ice} = -2\pi\rho\lambda_{ice} r \frac{dr}{d\tau} \qquad \text{Eq. (2)}$$

Combining the two equations above to simplify the heat flow rate, provides the differential equation expressing radius of ice formation as a function of time:

$$2\pi\rho\lambda_{ice} r \frac{dr}{d\tau} = \frac{\Delta_t}{\left(\frac{\log\frac{r}{r_0}}{2\pi k} + \frac{1}{2\pi r_0 h_0}\right)} \qquad \text{Eq. (3)}$$

where r is the radial position of growing ice and r_0 the initial radius (corresponding to the outer radius of the cooling coil as shown in Figure 2).

Since radius increases with time, the thermal resistance increases with time, and as a result the heat flow rate as well as the cooling capacity decrease.

The analytical model is able to predict the amount of ice formed as a function of time during the charge of the ITES, and as a consequence the cooling power required.

Figure 3 gives an example of the behaviour of ITES during a charge phase, showing how the cooling capacity provided by the refrigerating unit varies during the charge time. A 12 m^3 ITES is considered, with water at 7 °C initial temperature. At the start of the process the cooling demand is limited by the refrigerating unit, since its design capacity is lower than the maximum heat flow rate which could be exchanged. Once water is chilled to the freezing point (1.39 hours) and ice builds up, its thermal resistance increases, and when it becomes the bottleneck of the heat transfer the cooling demand decreases below the design value. The charge has to be stopped when the outer radius of ice is 53.5 mm, i.e. 6.06 m^3 of ice are produced, to prevent ice coils from getting in touch each other. This process takes little more than 9 hours, and the cooling demand at the stop of the process is reduced to 54.8 kW. The total amount of energy made available in ITES is thus around 2100 MJ, considering latent heat, sensible heat up to 5 °C (maximum useful temperature for the purpose of the application) while neglecting the thermal capacity of the cooling coils. This corresponds to a volumetric storage capacity of 175 MJ/m^3.

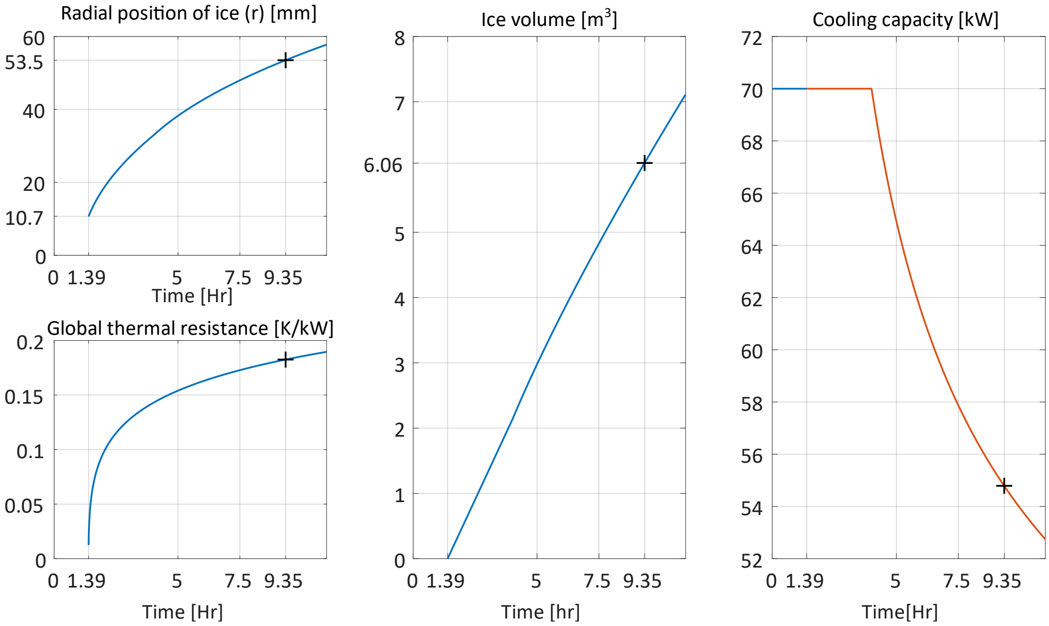

Figure 3: Ice build up and cooling demand in an example of charge phase, starting at t_w = 7°C

3.2.2. Discharge phase

For the discharge model, an analytical solution was used to predict the ice-melting performance of the ITES. The ice cylinders were melted externally by water flowing across them. Water exiting the ITES is useful for AC purposes up to 5° C, and this parameter determines the maximum duration of the discharge phase.

The model was developed from the equation proposed by Lee and Jones (1996):

$$c_w \cdot M_w \frac{dt_w}{d\tau} = \dot{m}_{\mathrm{w}} c_w (t_{\mathrm{r}} - t_{\mathrm{w}}) + U_{env} A_{\mathrm{env}} (t_{env} - t_{\mathrm{w}}) - h_e \cdot A_{ice} \cdot (t_w - t_{ice})$$

$$-\frac{dM_w}{d\tau} \cdot c_w \cdot (t_w - t_{ice}) + \dot{m}_{\mathrm{air}} c_{\mathrm{air}} (t_{\mathrm{air}} - t_{\mathrm{w}}) \qquad \text{Eq. (4)}$$

The first term at the right-hand side is the heat flux from the heat exchanger HX (i.e. the cooling demand) being $\dot{m}$ and t_r the mass flow rate of water and the return water temperature to the tank; the second term is the heat loss to the surrounding at t_{env}; the third term is the convective heat transfer between water and ice, where the value of the thermal external convective coefficient (h_e) is estimated from manufacturer tests as 930 $Wm^{-2}K^{-1}$ and the ice surface (A_{ice}) is considered at t_{ice} =0°C; the fourth term is the sensible heat for the melted water which is heated from the freezing-point $t_0 = t_{ice}$ to t_w and the last term corresponds to the heat contribution due to blowing of air into the tank, where t_{air} is the blower outlet temperature and $\dot{m}_{\mathrm{air}}$ is the mass flow rate.

Additionally, the mass conservation gives:

$$\frac{dM_w}{d\tau} = -\frac{dM_{ice}}{d\tau} = \frac{q_{ice}}{\lambda_{ice}} = \frac{h_e \cdot A_{ice} \cdot (t_w - t_{ice})}{\lambda_{ice}} \qquad \text{Eq. (5)}$$

Where λ_{ice} is the latent heat of freezing for water.

The final equation is obtained from Eq. (4) and Eq. (5) and discretized in time, where for the reasons described above, $t_{w_{\text{target}}}$ is assumed to be at 5°C:

$$\frac{t_{w(n+1)} - t_{w(n)}}{\Delta\tau} = -\frac{h_e\, A_{ice(n)}}{c_w\, M_{w(n)}}\left(t_{w(n)} - t_{ice}\right)\left(\frac{c_w\,\left(t_{w(n)} - t_{ice}\right)}{\lambda_{ice}} + 1\right) + \frac{m_{w(n)}\,\left(t_{\mathrm{r}} - t_{w_{\text{target}}}\right)}{M_{w(n)}}$$

$$+ \frac{U_{env}A_{\mathrm{env}}\left(t_{env} - t_{\mathrm{w(n)}}\right)}{c_w\, M_{w(n)}} + \frac{\dot{m}_{\mathrm{air}}\, c_{\mathrm{air}}\left(t_{air} - t_{\mathrm{w(n)}}\right)}{c_w\, M_{w(n)}} \qquad \text{Eq. (6)}$$

From the above equation, water temperature is calculated at each time step, and then the quantity of ice is determined by mass conservation.

Once the quantity of ice available is known from the cooling charge model, as well as the load profile required by AC, the model gives the maximum duration of the tank for the required load and the trend of water tank temperature and ice quantity.

4. OPTION FOR USE

The system shown in Figure 1 allows various conditions of use and can supply the AC required load. The main objective in the actual plant is shaving the peak in electrical energy use. The most critical condition in terms of electricity use for the supermarket considered in this paper is in the morning when, in addition to all the supermarket's appliances, both ovens and heaters for the deli shop and air conditioning are switched on. Reducing the maximum electric power allowed a significant reduction in investment cost for the connection to the electrical grid. Nonetheless there may be also other uses: since the reduction in the early morning peak may be necessary only some days a year, the storage could be used with other strategies. The feasibility of peak shaving is being explored, to decrease the chiller size and reduce electricity use in peak periods, thus taking advantage of hour tariff.

An AC cooling demand has been identified (daily summer profile in Figure 4), and two modes of use were analyzed:

Case 1: Supply the whole AC demand early in the morning (blue)

Case 2: Supply the AC demand undersizing the chiller (orange)

Through the model above described, and with the support of experimental data from the field, the water temperature in the storage and the mass of ice for both cases are calculated and shown in Figure 4, starting from the amount of ice built in 4 hours (1606.7 kg), previously obtained from the charge model. In both cases the water temperature trend (solid line) shows a behavior consistent with ice melting curves (dashed line). After an initial linear increase until low water temperature is the bottleneck in heat transfer, ice melting occurs and a sudden increase in water temperature coincides with the complete melting of ice. In *Case 1* the ITES is able to supply the AC load for 5 hours 41 minutes (until 11:40 AM) while in *Case 2* the ITES is used from 9:34 to 20:09 AM (10 hours 35 minutes) reducing the chiller design size down to 45.9 kW. The cooling power supplied by ITES is always lower than the maximum value as estimated by the manufacturer for such operating conditions. The two colored areas represent the net available energy provided by the tank; they are not equal since in *Case 2* the storage is used later in the day and it is subject to heat loss; this can also be seen in the ice trend where it drops slightly at the beginning of the day, when the storage is not yet used.

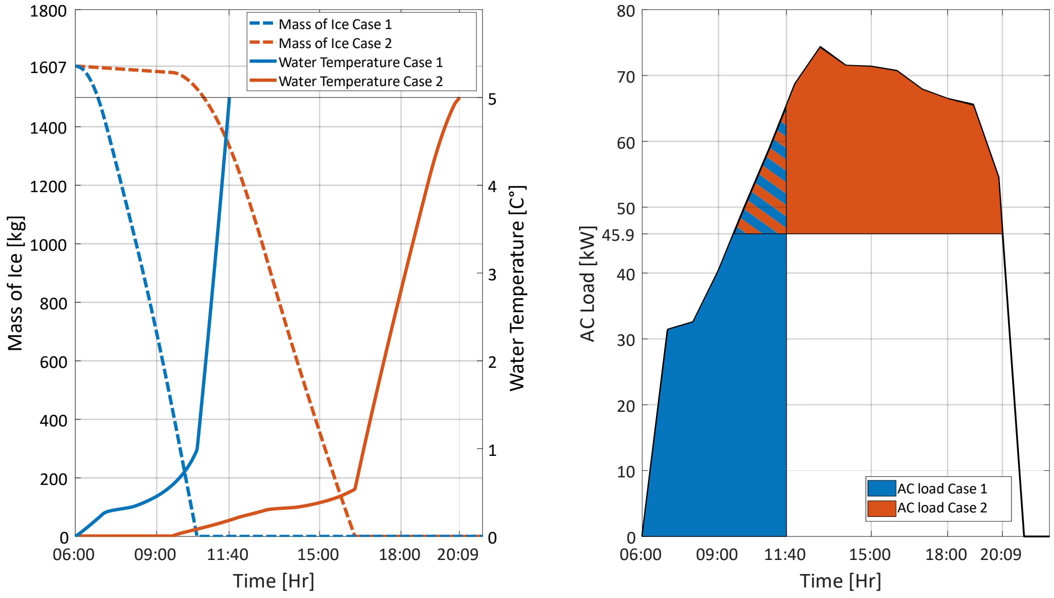

Figure 4. Different ITES control strategy

Finally, the discharge and charge models interfaced with the TRNSYS model previously mentioned for the prediction of the performance of the CRU, allow to compare the two cases in terms of cooling and electrical energy use.

The cooling energy used in the two cases is summarised in Table 1. The *ITES charge energy* is the energy supplied by the CRU during charging time, the *ITES available energy* is the energy usable for AC purposes (until 5°C), followed by the *ITES net available energy* which is net of heat loss.

Table 1 Cooling energy values comparison in both cases

Cooling energy [kWh]		
	Case 1	Case 2
ITES charge energy	280.0	280.0
ITES available energy	242.8	242.8
ITES net available energy	226.1	209.8
ITES efficiency	80.7 %	74.9%

An ITES efficiency is defined based on the cooling energy values as:

$$\eta_{\mathrm{ITES}} = \frac{ITES\ net\ available\ energy}{ITES\ charge\ energy} \qquad \text{Eq. (7)}$$

Case 2 shows a lower efficiency since the ITES is used later in the day, resulting in higher heat loss.

In Table 2 are listed the electrical energy used by the Heat Pump to face alone the whole daily HVAC cooling demand; the total electrical energy used if the ITES is used; the electrical energy needed during the night by the CRU for the ITES charging process and finally the extra energy to be supplied when the ITES is used, compared to an operation with the heat pump alone.

Table 2 Electrical energy values and COP comparison in both cases

Electrical Energy [kWh]		
	Case 1	Case 2
Heat Pump alone	292.3	292.3
ITES + Heat pump	373.8	356.9
ITES charge	145.2	145.2
Extra electrical energy	81.5	64.6
COP of CRU for ITES charge	1.93	1.93
EER of heat pump	3.55	2.60

The first and third rows have the same values since the same heat pump is used and the storage is charged at the same time (night-time) in both cases. The COP for ITES charge is calculated for the production of the "ITES charge energy" by the CRU. The EER of the heat pump is the average value of the EER of the heat pump during operation in the periods otherwise served by the ITES. The use of ITES is never energy profitable, due to the low COP of the commercial refrigeration unit even if working at favourable temperature conditions during night-time, and to the efficiency of the storage process itself, estimated by Eq. (7). In fact, the ITES is charged with the evaporating temperature at -10 °C, while the reversible heat pump is operated at an evaporating temperature of around 3 °C. Case 2 shows less electrical energy usage than Case 1 when the ITES is operated, because the storage is used when the heat pump would have had lower EER values.

5. CONCLUSIONS

Thermal energy storage has been used to overcome feasibility issues that prevented from facing high peaks in electrical power use. Modelling allows a thorough evaluation of the energy balance and a correct design of the storage. When a latent thermal storage is used, the volumetric heat storage can be significantly increased at the expense of a lower storage efficiency. This is due to the much lower evaporating temperature required to perform water freezing. To cut the gap due to extra running costs, the investment costs can be reduced. The task of charging has been assigned to a commercial refrigeration unit, already available and performing at partial load during night-time. For the same purpose, the heat storage can also be used to shave the cooling demand at the reversible heat pump, thus allowing a reduction of its size. The choice for the best sizing and operating schedule is a matter of optimization which is being investigated through this model.

ACKNOWLEDGEMENTS

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NOMENCLATURE

A_{ice}	Ice surface area (m^2)	r_0	Initial radial position (m)
$A_{ice\ max}$	Initial ice surface area (discharge) (m^2)	t_{air}	Blower outlet temperature (°C)
c_w	Specific heat of water ($J\ kg^{-1}\ K^{-1}$)	t_0	Freezing-point of water (°C)
c_{air}	Specific heat of air ($J\ kg^{-1}\ K^{-1}$)	t_{ev}	Evaporating temperature (°C)
h_0	Internal convection coefficient ($Wm^{-2}K^{-1}$)	t_{env}	Outdoor temperature (°C)
h_e	External convection coefficient ($Wm^{-2}K^{-1}$)	t_{ice}	Ice surface temperature (°C)
k	Thermal conductivity of ice ($Wm^{-1}\ K^{-1}$)	t_r	Return water temperature (°C)
$\dot{m}$	Mass flow rate of water ($kg\ s^{-1}$)	t_w	Tank water temperature (°C)

$\dot{m}_{\mathrm{air}}$	Mass flow rate of air (kg s^{-1})	$U_{env}A_{env}$	Overall conductance between ice tank and surroundings (WK^{-1})
M_{ice}	Mass of ice in the tank (kg)	Δt_0	Temperature difference (t_0-t_{ev}) (K)
M_w	Mass of water in the tank (kg)	λ_{ice}	Latent heat of fusion of ice (J kg^{-1})
q'	Heat flow rate per unit length (Wm^{-1})	ρ	Ice density (kg m^{-3})
R'	Thermal resistance per unit length (Wm^{-1} K^{-1})	τ	Time (s)
r	Radial position of growing ice surface (m)	η_{ITES}	ITES efficiency (-)

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