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Thermodynamic and economic seasonal analysis of a transcritical CO_2 supermarket with HVAC supply through ice thermal energy storage (ITES)

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ABSTRACT

A real case of a supermarket where a CO_2 refrigerating plant also supplies heating, air conditioning and hot water is considered. Ice thermal energy storage (ITES) is used both as latent storage in summer and as sensible thermal energy storage (TES) in winter to partially cover the space cooling/heating load of the supermarket. In particular, it allows to reduce peaks in the electrical power use, when the refrigeration and HVAC systems are running at full power together with ovens and heaters for meals. A thermodynamic analysis, including a detailed theoretical model of the formation and melting of ice on the coils, is carried out to predict the behaviour of ITES during the charging and the discharging phases. A daily energy analysis for both a winter and a summer typical day, and an annual analysis are carried out for the whole system. In summer, two cases are evaluated, i.e. supplying the whole AC demand in the morning or partially covering the AC demand to reduce the design capacity of the reversible heat pump. In all cases, the use of ITES aimed at shaving electrical peaks leads to a higher electrical energy use, also on an annual basis. However, the cost analysis reveals significant benefits, including a reduction in the required capacity of the reversible heat pump, better exploitation of tariffs and the avoidance of installing an electrical transformer in a dedicated room. This results in savings €58,699 over 10 years €47,888 over 15 years, making the choice of ITES more economically advantageous within the typical lifetime of these systems.

1. Introduction

The environmental impact of refrigeration and air conditioning systems has attracted considerable attention in recent years due to the urgent need to mitigate climate change. Carbon dioxide (CO_2) as a refrigerant has emerged as a promising and environmentally friendly alternative to traditional synthetic refrigerants in various applications, particularly in the context of supermarket refrigeration. The natural abundance of CO_2, its ozone-friendly characteristics, and its low global warming potential make it a natural choice, in line with international agreements and regulations aiming to phase out synthetic fluorinated gases. Furthermore, the operational efficiency, energy savings and heat recovery potential offered by CO_2-based systems are perfectly consistent with the sustainability and cost-effectiveness goals of supermarket operators.

At higher outdoor temperatures, the refrigeration system operates in a transcritical mode, resulting in a significant reduction in energy efficiency. This necessitates the optimization of the gas cooler pressure with an increased complexity of the refrigeration cycle. To enhance the energy efficiency of the fundamental CO_2 "booster" system, several technical solutions can be employed, aiming to achieve energy performance comparable to that of traditional HFC-based systems. The most common methods are for instance the parallel compressor (Chesi et al. (2014)), the ejector (Haida et al. (2016)), and various subcooling methods such as the internal heat exchanger (Cavallini et al. (2007)) or dedicated mechanical subcooling (Dai et al. (2022); Llopis et al. (2016)) and the possibilities of overfeeding the evaporators (Minetto et al. (2014)). Numerous research papers in the literature offer comparisons among these various solutions (Gullo et al. (2016a); Sawalha S (Sawalha, 2008).Barta et al. (2021)). In spite of all the above mentioned solutions, the efficiency of a CO_2 plant remains a critical issue at warm and mild climate conditions, giving place to a yearly energy consumption which can be higher than that of a conventional and simpler HFC plant. However, this disadvantage can be strongly mitigated or even turned to advantage by performing heat recovery in favour of space heating and domestic hot water (DHW) production (Cortella et al. (2014)). The combination of the above-mentioned energy-saving solutions with heating and cooling functions in an all-in-one R744 unit (Tsimpoukis et al. (2021)) seems to be a promising technology. Nevertheless, it adds to the complexity of managing and controlling the commercial refrigeration unit to meet all required loads.

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Nomenclature

A	Area (m^2)
Cp	Specific heat (J/kg K)
e	Electricity price (€/kWh)
h	Heat transfer coefficient (W/m^2K)
k	Thermal conductivity (W/m K)
ṁ	Mass flow rate (kg/s)
M	Mass (kg)
n	Plant service lifetime (Years)
q'	Heat flow rate per unit length (W/m)
Q	Heat Flow rate (kW)
R'	Thermal resistance per unit length (W/m K)
r	Radial position (m) – Discount rate (–)
t	Temperature (°C)
U	Internal Energy (kJ)
UA	Overall conductance (W/K)
Δt	Temperature difference (K)

Acronyms

AC	Air Conditioning
AEC	Annual Energy Cost
COP	Coefficient of Performance
CC	Capital Cost
CRU	Commercia Refrigeration Unit
DHW	Domestic Hot Water
GC	Gas Cooler
GWP	Global Warming Potential
HP	Heat Pump
HPV	High Pressure Valve
HS	High Stage
HVAC	Heating, ventilation, Air Conditioning
ITES	Ice Thermal Energy Storage
MT	Medium Temperature
LS	Low Stage
LT	Low Temperature
RC	Running Cost

Greek symbols

λ	Latent heat of fusion (J/kg)
ρ	Density (kg/m^3)
τ	Time (s)
η	Efficiency (–)

Subscripts

env	environment
ev	evaporation
o	initial
r	return
ITES	refers to the Ice Thermal Energy Storage
int	internal

Thermal energy storage (TES) is therefore suggested to shift loads, in order to achieve 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. 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, around noon whereas the total electrical energy peaks earlier in the morning when ovens and heaters for meals are active. As far as air conditioning is concerned, passive (often PCM) elements can be used on the air side with the main purpose of dumping supply air temperature during on-off cycles or defrosting periods of heat pumps (Hlanze et al. (2021)). On the water side, high 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)). Additionally, an ice tank can also be used to subcool CO_2 systems as a possibility of improving efficiency (Khanloghi et al. (2023)). 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. Thus, for the typical 5K difference between return and supply temperatures from/to the AC system (i.e. supply 7 °C, return 12 °C), in absence of further heat exchanger between the storage and the AC water loop, the maximum sensible heat storage is around 20 MJ/m^3. 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); Vivian et al. (2023)) which involve a clear definition of the aim of the system and a thorough prediction of the user demand profile.

In this paper the two operating modes, wintertime and summertime, are considered. In summertime, where the air conditioning load is faced by a typical water chiller designed for 12–7 °C water temperature, the ITES is used as thermal storage taking advantage of the availability of low temperature cooling power from the commercial refrigeration unit. In wintertime, the ITES is used as storage to provide hot water (45 °C) for space heating, reclaiming heat from the commercial refrigeration unit. The purpose of these configurations resides mainly in shaving the peaks of electrical power, either by replacing the heat pump (HP) operation during some peak hours, or by operating the ITES in parallel with the HP. The system is modelled through in-house routines which are linked to a more comprehensive tool of simulation of the thermal behavior of buildings and of the commercial refrigeration plant. A theoretical model for the prediction of ice melt/growth in the ice thermal energy storage is presented.

2. Methods and case study

The case study supermarket is located near Rome, on the coast and is served by a typical CO_2 Commercial Refrigeration Unit (CRU), for the display and storage of chilled and frozen food (Cortella et al. (Cortella et al.)). There are two heat pumps operating on R-410A: one for the Domestic Hot Water DHW supply (Heat Pump 1) and one reversible for the HVAC system (Heat Pump 2) of the sales area and other warehouse, food processing, offices. The supermarket has a bakery and a deli shop, whose ovens and heaters are used to cook ready-made meals, especially early in the morning, when the air conditioning systems is also running at full power to restore indoor temperature conditions after the night pause. This results in a huge peak in electrical energy use, when fares are

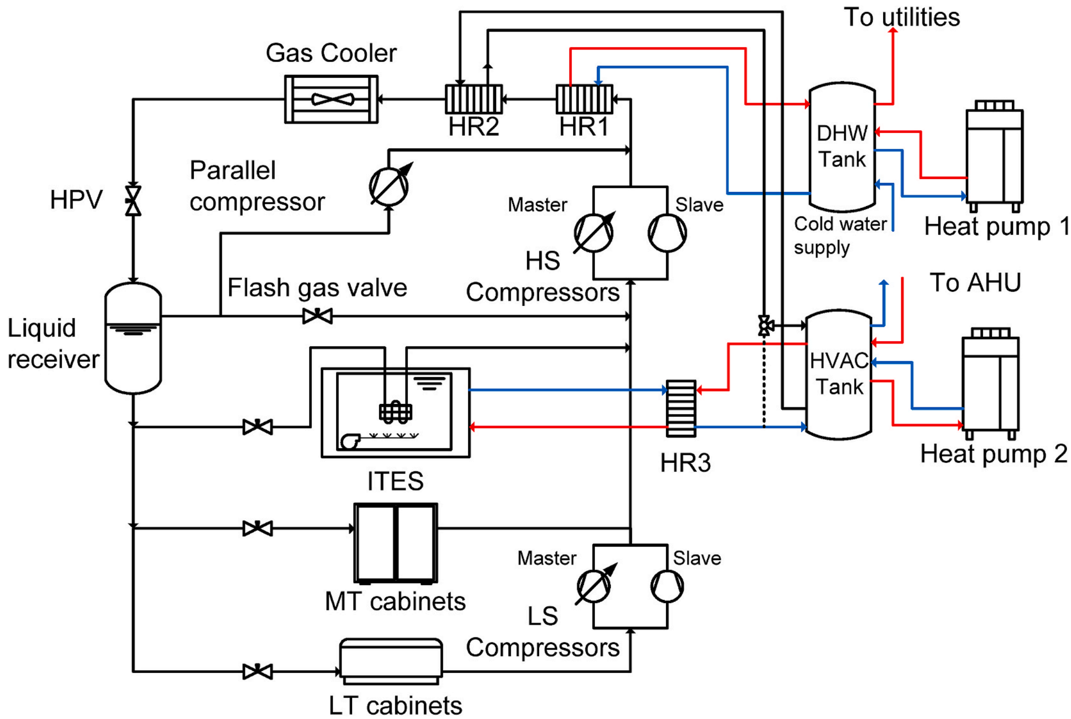

Fig. 1. Schematic drawing of the supermarket plant in summertime.

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 introduced to provide a significant energy storage in favour of air conditioning. Ice is produced by means of the CRU during night-time, when it is operating at favorable climatic conditions and at partial load. The nominal refrigerating capacity for the CRU is 22 kW at −35 °C (LT), and 118 kW at −10 °C (MT) including two 35 kW evaporators at −10 °C for the ITES.

The summertime configuration sketched in Fig. 1 is first considered. The ITES consists of a water tank with two submerged ice-making evaporators (one in the figure for ease of schematic), which have the same working pressure as the evaporators of medium temperature (MT) refrigerated cabinets.

The ITES is connected via a heat exchanger (HR_3) to a water tank (HVAC Tank) which supplies the Air Handling Units (AHU). The HVAC tank is modelled via the Trnsys Type 4, which allows a fixed number of

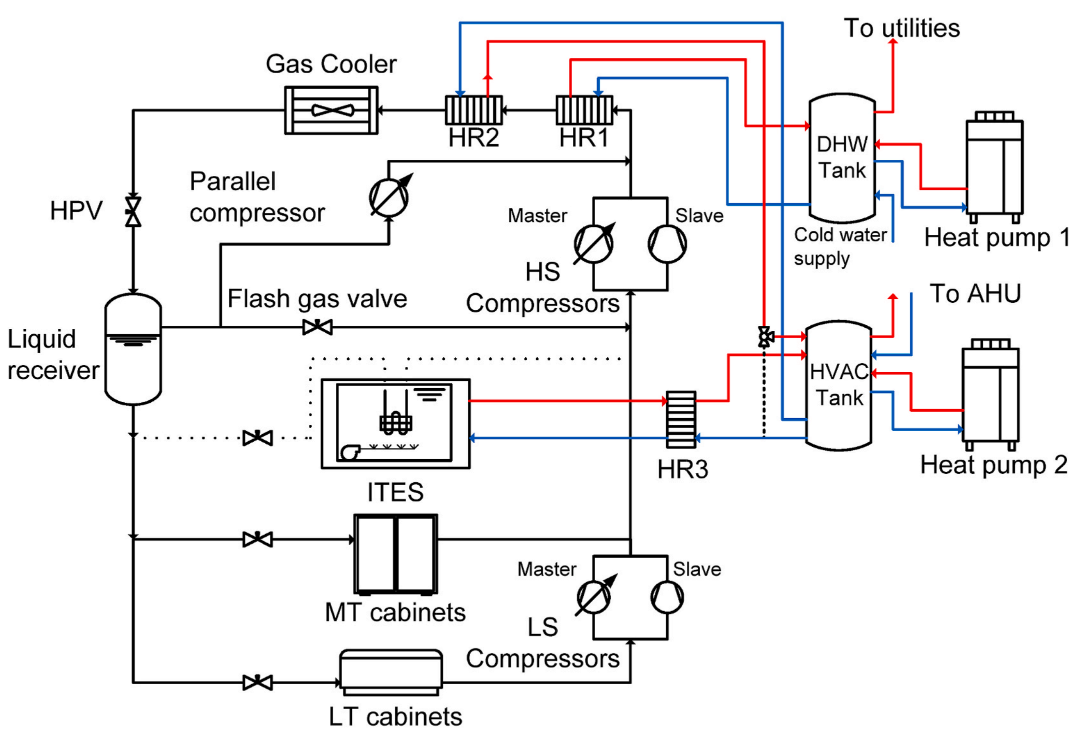

Fig. 2. Schematic drawing of the supermarket plant in wintertime.

internal nodes to take into account the stratification. The heat exchanger is needed to decouple the two mass flow rates, and allows the use of differently treated water in the two circuits. The reversible heat pump (HeatPump2), which operates as a chiller in the summer, is also connected to the same water tank, in parallel with the ITES, in order to cover the whole AC demand.

The ITES consists of a 12 m^3 water tank, thermally insulated with a 4 cm thick panel of XPS to reduce heat loss, supplied with two submerged packages of 12 coils each, with an external diameter of 21.3 mm for a total length of 720 m. They represent two separate evaporators for the CRU at −10 °C, each controlled by an electronic expansion valve. The storage is manufacturer's commercial product. About 6000 kg of ice can be formed, providing a global latent storage of 2000 MJ. An air blower supplying approximatively 100 m^3/h of air at ambient temperature distributes air at the bottom of the water tank to improve convection on the water side during ice melting for the discharge phase.

In this phase, the thermal storage is used until the water temperature raises up to 5 °C, which is the limit temperature for obtaining, at the HR_3, chilled water at 7 °C to be delivered at the Air Handling Units (AHU). The water is then supposed to return from the AHU at 12 °C, resulting in a return water temperature to the ITES of around 10 °C.

In wintertime (Fig. 2) the ITES is used as a hot water storage tank for heat recovery from the CRU, using in this case the ITES in parallel to the water tank (HVAC Tank) to allow some sensible heat storage. In this case the ITES evaporator is not used.

In this case the storage tank is heated during the night by reclaiming heat from the CRU (HR_2). In order to recover heat for both DHW and space heating, a higher enthalpy of the refrigerant is obtained at HR_1 inlet by forcing the commercial refrigeration unit to work in transcritical condition. Thus, in heat recovery mode, the pressure at the gas cooler pressure is controlled to a fixed value of 78 bar, according to in D'Agaro et al. (D'Agaro et al., 2019), which is close to the lower limit of transcritical operation.

The thermal storage tank, which is designed for summer conditions, can also be used effectively to improve the heat recovery in the winter. Thus, an analysis of possible feasibility of winter operation is also investigated with the following control rules.

- during the night, the CRU provides heat up to an ITES limit temperature of 50 °C; during its charge, water flows through HR_2 and then through the three-way valve along the dotted line as shown in Fig. 2;
- during the day, the storage provides heat for space heating until the ITES temperature is below 35 °C, or the ITES supply temperature is below the HVAC tank temperature; then the heat supplied comes straight from the CRU.

The heat pump only provides heat when the tank temperature drops below 40°, i.e. when the CRU and ITES do not meet the required heat load.

3. System modelling

3.1. Refrigeration unit and HP

To investigate the effectiveness of the ITES in terms of daily energy use or peak shifting, a comprehensive model is used. The cooling/heating load for the building is estimated hourly through the TRNSYS Type 56 dedicated to building dynamic simulation. A prediction of the annual refrigerating capacity profile with an hourly time step of display cabinets and cold rooms is then carried out as a function of the indoor climate conditions and operating conditions (defrosting, night blinds etc). The commercial refrigerating unit is modelled with in-house Types. The model can simulate refrigerating units equipped with the most widespread solutions to improve the efficiency of transcritical cycles, including subcooling via a dedicated mechanical system (Cortella et al. (2021)), parallel compression (Gullo et al. (2016b)) and also with heat recovery facilities to allow the production of domestic hot water (DHW) and space heating and cooling (D'Agaro et al. (D'Agaro et al., 2019; D'Agaro et al., 2018)) also in the view of a Demand Side Management for the electrical grid (Coccia et al. (2019)). The commercial refrigeration unit considered in this paper is a CO_2 transcritical refrigeration 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 the manufacturer data as a function of the source/supply heat exchanger temperature, and load. This model, implemented in the TRNSYS (Klein et al., 2010) simulation environment, is linked to a code in Matlab dedicated to the simulation of the ITES, as described below.

3.2. Ice thermal energy storage (ITES)

In the ITES considered, ice forms on the outside of the coils immersed in the tank (ice-on coil external melt). The charging and discharging 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. Charging phase

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

In the simulation reported in this paper, the initial water temperature is assumed to be 10 °C. The estimation of the time needed to reduce the water temperature down to the freezing point is based on a simple energy balance once the cooling capacity 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.

Ice formation over the coils is predicted through the well-known 2D 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 in series through the ice conductive resistance and the internal convection one, is:

$$q' = \Delta t_0 / \left(R'_{ice} + R'_0\right) \tag{1}$$

where Δt_0 is the temperature difference between the temperature of water at the freezing point (0 °C) and the evaporating temperature of the refrigerant, R'_{ice} is the conductive resistance of ice and R'_0 is the internal convective resistance per unit length, on the refrigerant side, assumed to be constant along 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.

The heat flow rate per unit length, neglecting the ice subcooling, provides the extraction of the latent heat of freezing at the surface $A_{ice}(t)$:

$$q' = \frac{dM_{ice}}{d\tau}\lambda_{ice} = -2\pi\rho\lambda_{ice} r\frac{dr}{d\tau} \tag{2}$$

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

$$2\pi\rho\lambda_{ice} r\frac{dr}{d\tau} = \frac{\Delta_t}{\left(\frac{\log\frac{r}{r_0}}{2\pi k} + \frac{1}{2\pi r_0 h_0}\right)} - UA_{env}\left(t_{env} - t_w\right) \tag{3}$$

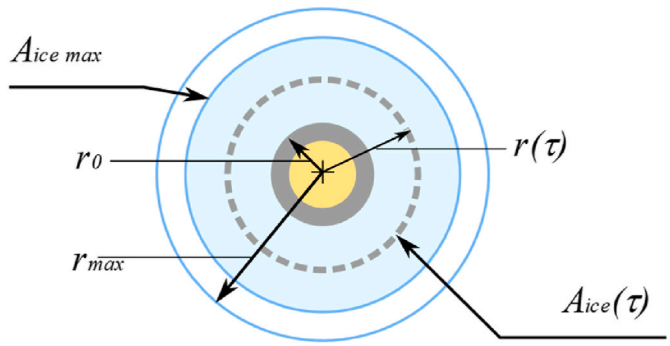

Fig. 3. Schematisation of ice formation on the coil.

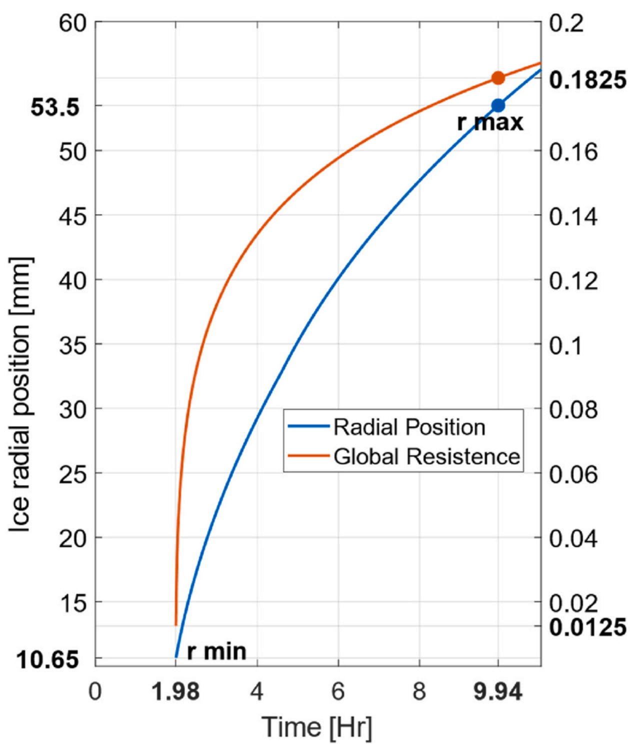

Fig. 4. Ice radius trend from the minimum (nude coil) to the maximum allowed by manufacturer and the corresponding global thermal resistance trend during the charging phase.

where r is the radial position of the growing ice and r_0 is the initial radius (corresponding to the outer radius of the cooling coil as shown in Fig. 3), t_w is the temperature of the water in the tank (assumed to be uniform) and UA_{env} is the overall thermal conductance between the ice tank and the surrounding with an U-value equal to 0.83 $Wm^{-2}K^{-1}$.

Since the radius increases with time, the thermal resistance increases with time, and consequently both the heat flow rate and 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.

A 12 m^3 ITES is considered, with water at an initial temperature of 10 °C; Figs. 4 and 5 give an example of the behaviour of the ITES during a charging phase. Fig. 4 shows the trends of the global thermal resistance and the ice front radial position over time during charging; the global resistance increases as the ice grows, since in this configuration the ice is not removed during charging. The initial radius corresponds to the nude coil radius ($r_{min} = 10.65$ mm) and the final radius is the maximum radius to allowed to prevent the ice coils from getting in touch each other ($r_{max} = 53.5$) and it is given by the manufacturer.

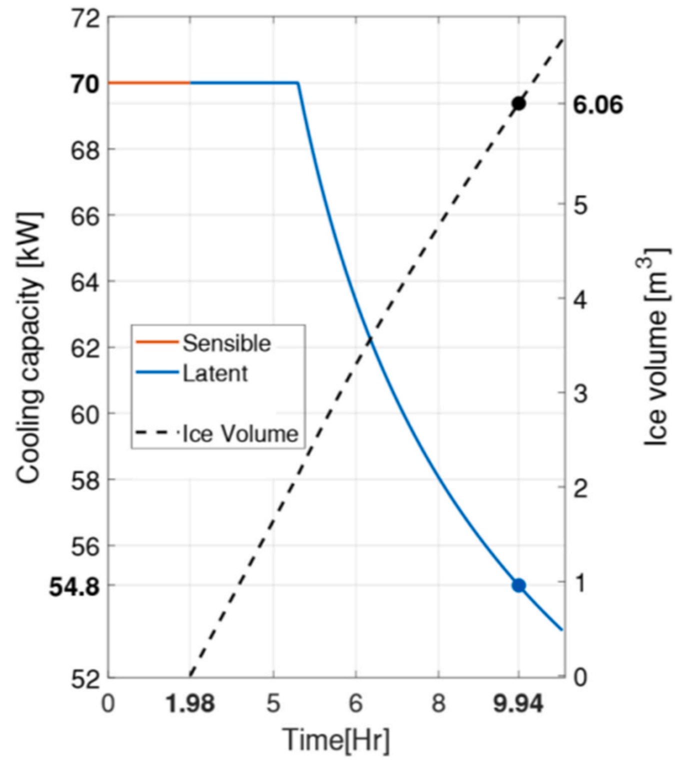

Fig. 5. Ice volume and cooling capacity trend during the charging phase.

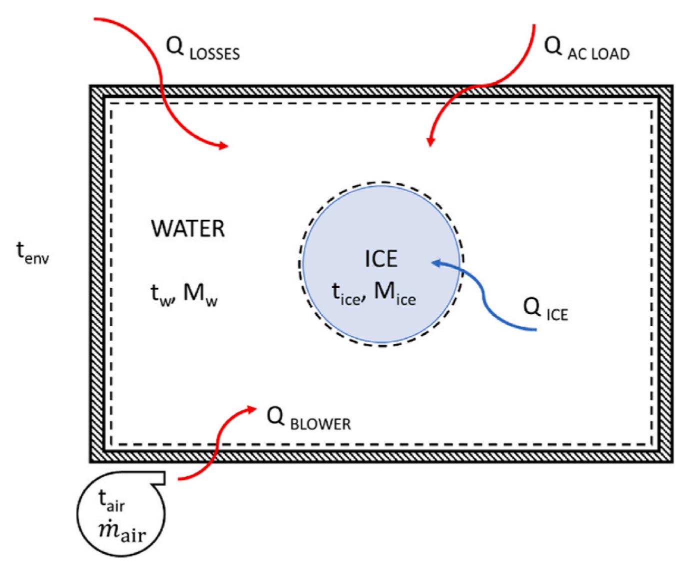

Fig. 6. Energy balance scheme during the discharging phase.

Fig. 5 shows the cooling capacity trend, divided into sensible and latent, and ice formation. At the start of the process, the cooling demand is intentionally limited to 70 kW by the size of the expansion valves of the refrigerating unit (Fig. 5), given that the coils could transfer a higher flow rate to the liquid water. Once the water is chilled to the freezing point (after 1.98 h) and ice builds up, its thermal resistance increases (Fig. 4) and when it becomes the bottleneck of the heat transfer, the cooling demand decreases below the design value. The charge must be stopped when it reaches the maximum outer radius of ice. This process takes approximately 10 h, 6.06 m^3 of ice is produced and the cooling demand at the end of the process is reduced to 54.8 kW. The total amount of energy made available in the ITES is therefore around 2100

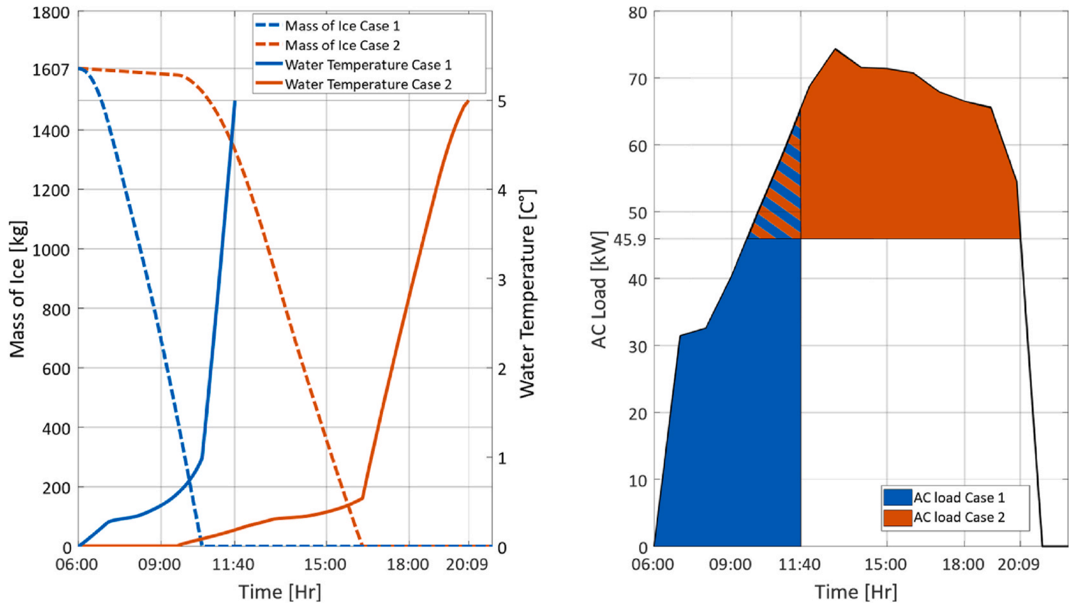

Fig. 7. Mass of ice trend and water temperature trend during discharging phase (left); AC cooling load and different control strategies (right).

MJ, considering both latent heat and the 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.

3.2.2. Discharging 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, and with the aim of increasing heat exchange at the water-ice interface, a blower is used to increase the turbulence of water in the tank during discharge. The water exiting the ITES is useful for AC purposes up to 5 °C, and this parameter determines the maximum duration of the discharging phase.

A global energy balance approach is applied to the water in the liquid phase and melting, thus the boundaries of the control volume are the water-ice interface and the tank walls. The system is sketched in Fig. 6, with the heat rates through the boundaries, that are plotted in the dashed lines The energy balance gives the following expression:

$$\frac{dU}{d\tau} = Q_{AC\ load} + Q_{losses} + Q_{ice} + Q_{blower} \tag{4}$$

where the first term on the left is the rate of change of internal energy in the water and, in order, the terms on the right correspond to: the heat flux from the heat exchanger HR_3 (i.e. the cooling demand), the heat loss to the surrounding at t_{env}, the heat exchanged by ice on water and the heat contribution due to the blowing of air at ambient temperature into the tank. The blower creates a mixing effect that allows the tank to be considered as an unstratified tank. By making explicit the terms, we obtain the following equation, similar to the equation from the model proposed by Lee and Jones (1996):

$$c_w M_w \frac{dt_w}{d\tau} = \dot{m}_w c_w (t_r - t_w) + UA_{env}(t_{env} - t_w) - h_e A_{ice}(t_w - t_{ice}) - \frac{dM_w}{d\tau} c_w (t_w - t_{ice}) + \dot{m}_{air} c_{air} (t_{air} - t_w) \tag{5}$$

where $\dot{m}_w$ and t_r are the mass flow rate of water and the return water temperature from HR_3 to the tank while t_w and M_w are the water tank temperature and the water mass respectively; regarding the heat transfer, the value of the thermal external convective coefficient h_e in the heat transfer between the water and the ice is estimated from manufacturer tests as 930 Wm^{-2}K^{-1}.It is approximatively constant despite the variation of the ice radius during melting thanks to the forced convection introduced by the blower. The ice surface (A_{ice}) is considered at t_{ice} =0°C; the fourth right 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 $\dot{m}_{air}$ is the mass flow rate. Similarly to the case of charging, the subcooling of ice is also considered negligible (London and Seban, 1943).

Additionally, the mass conservation at the ice-water interface gives:

$$\frac{dM_w}{d\tau} = -\frac{dM_{ice}}{d\tau} = \frac{q_{ice}}{\lambda_{ice}} = \frac{h_e A_{ice}(t_w - t_{ice})}{\lambda_{ice}} \tag{6}$$

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, $t_{w_{target}}$ is assumed to be at 5 °C:

$$\frac{t_{w(n+1)} - t_{w(n)}}{\Delta\tau} = -\frac{h_e\ A_{ice(n)}}{c_w\ M_{w(n)}}\left(t_{w(n)} - t_{ice}\right)\left(\frac{c_w\ \left(t_{w(n)} - t_{ice}\right)}{\lambda_{ice}} + 1\right) + \frac{m_{w(n)}\left(t_r - t_{w_{target}}\right)}{M_{w(n)}} + \frac{U_{env}A_{env}\left(t_{env} - t_{w(n)}\right)}{c_w\ M_{w(n)}} + \frac{\dot{m}_{air}c_{air}\left(t_{air} - t_{w(n)}\right)}{c_w\ M_{w(n)}} \tag{7}$$

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, for a given load profile required by AC, the model gives the trend of water tank temperature, the trend of ice quantity and thus the maximum duration of the tank.

4. Energy performance evaluation

In this chapter, an energy evaluation is conducted by considering a typical day in August and January for the summer and winter analysis respectively. Both analyses are conducted from summer/winter HVAC load profiles calculated by the dynamic simulation of the building by Type 56 in TRNSYS, taking into account the additional contribution/load of the refrigerated display cabinets (Cortella et al., 2020).

Table 1
Cooling energy values comparison in both cases.

Cooling energy [kWh/day]		
	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%

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

4.1. Option for use

In the following section, the energy and electrical power consumption performance in the summer and winter case are analysed.

4.1.1. Summer

The system shown in Fig. 1 allows various conditions of use and can supply the required AC load. The main objective in the actual plant is to shave 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 the air conditioning are switched on. Reducing the peak electric power use allows a significant reduction in the investment cost for the connection to the electrical grid. Nonetheless, there may be also other uses for the storage: 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 required capacity of the reversible heat pump and to reduce electricity power request during peak periods, thus taking advantage of the hourly tariff.

An AC cooling demand has been identified (daily summer profile in Fig. 7), and the two modes of use were analysed.

Case 1. Supply the ITES available energy in the early morning, when the supermarket's electrical demand is at its peak (blue).

Case 2. Supply the ITES available energy to cover the AC demand peak that occurs in the central hours (orange).

Through the model above described, and with the support of experimental data from the ITES manufacturer, the water temperature in the storage and the mass of ice for both cases are calculated and shown in Fig. 7. The initial amount of ice (1607 kg) is that built up in 4 h according to the charge model. In both cases, the water temperature trend (solid line) is consistent with the ice melting curves (dashed line). After an initial linear increase until low water temperature prevents effective heat transfer with ice, 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 h 41 min (until 11:40 a.m.) while in *Case 2* the ITES is used from 9:34 to 20:09 a.m. (10 h 35 min) reducing the maximum load on HP2 from 74.19 kW down to 45.9 kW, allowing a reduction in the required capacity of the reversible heat pump. The cooling capacity supplied by the ITES is always lower than the maximum value as estimated by the manufacturer for such operating conditions (163 kW at 10 °C water inlet temperature). The two colored areas represent the net available energy provided by the tank; they are slightly different because in *Case 2* the storage is used later in the day and it is subject to higher heat loss; this can also be seen in Fig. 7 where the ice trend gradually drops at the beginning of the day, when the storage is not yet used.

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

The cooling energy used in the two cases is summarised in Table 1.

Table 2
Electrical energy values and COP comparison in both cases.

Electrical Energy [kWh/day]		
	Case 1	Case 2
Heat Pump 2 alone	292.3	292.3
ITES + Heat pump 2	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 *ITES charge energy* is the energy supplied by the CRU through the evaporators during charging time, the *ITES available energy* is the energy useable for AC purposes (until 5 °C), followed by the *ITES net available energy* which is net of heat loss. An ITES efficiency is defined based on the cooling energy values as:

$$\eta_{\mathrm{ITES}} = \frac{\textit{ITES net available energy}}{\textit{ITES charge energy}} \tag{8}$$

In Table 2 are listed the electrical energy used by the HP2 (water chiller operation) to face alone the whole daily cooling demand for HVAC; the total electrical energy needed if the ITES is used; the electrical energy needed during the night by the CRU for the ITES charging process (ITES charge) and finally the extra energy to be supplied when the ITES is used, compared to an operation with the heat pump alone.

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 favorable temperature conditions during night-time, and to the efficiency of the storage process itself, estimated by Eq. (8). In fact, the ITES is charged at the evaporating temperature of −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.

4.1.2. Winter

Similarly to the summer operation mode, an analysis of a winter day is presented. It is important to emphasise that, as mentioned before, the system was designed for reducing peaks in electrical power use in summer.

The daily heat load profile for space heating (SH load), shown as black solid line in Fig. 8, has its peak in the early morning, when supermarket's electricity is also at its highest. Thus the question is whether it is profitable to use the ITES or not, since in the first case, unlike in summer, the only logic solution is to exploit the stored energy in the morning.

The comparison of daily energy fluxes and temperatures with and without ITES is shown in Fig. 8. The SH load (black solid line) remains the same in the two operating modes. In the case with ITES, the heat recovery from the CRU through HR2, shown in blue line, is used to charge the ITES exclusively during the night (from 8 p.m. to 4 am) and the temperature in the ITES (red dashed line) increases up to upper limit of 50 °C. The storage tank supplies heat (red solid line) from 6 am until 11:00 a.m., then the CRU is used directly in heat recovery mode to meet the heat demand. The two systems (CRU and ITES) do not work simultaneously: the operation control logic uses the ITES first and then the CRU. The HP goes into operation when the temperature in the HVAC tank (red dashed line) drops below 40 °C. In the case with ITES, the energy demand from the heat pump (green line) is required between 11 am and 1 pm up to 27.8 kW. In the case without the ITES, the demand

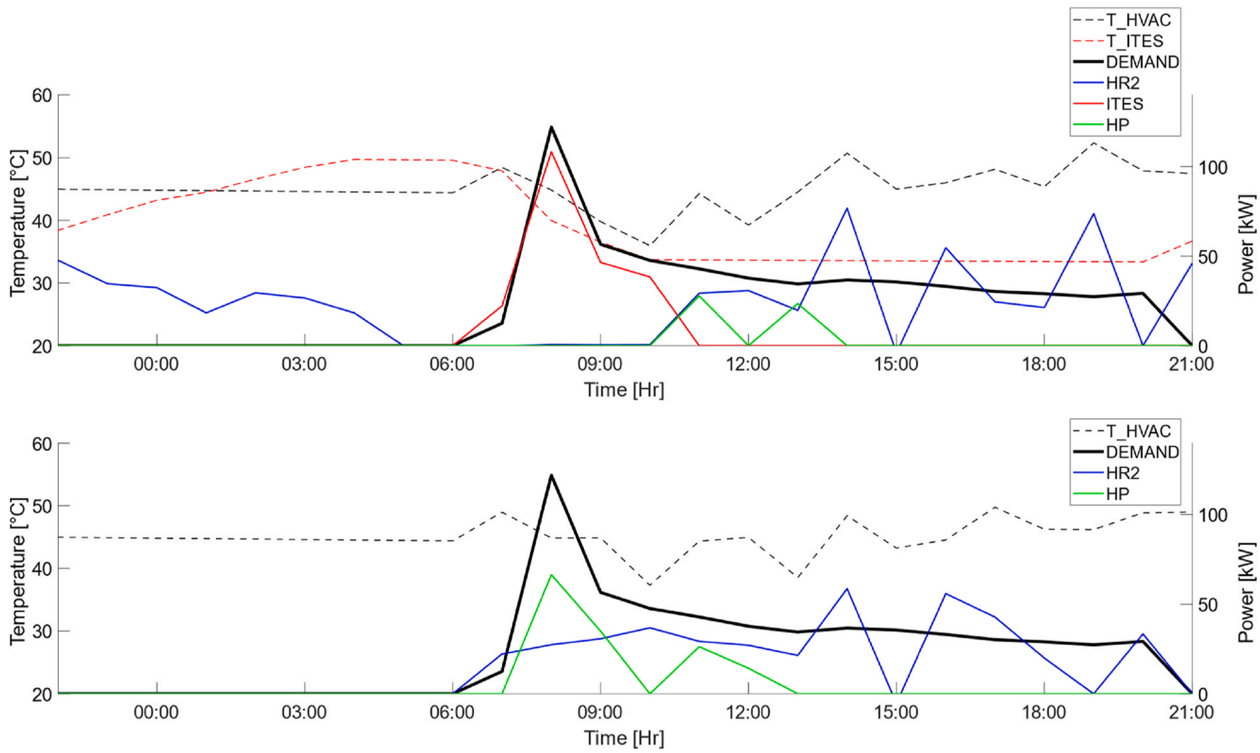

Fig. 8. Comparison of daily energy flows and water temperature trends in the winter case with storage (above) and without storage (below).

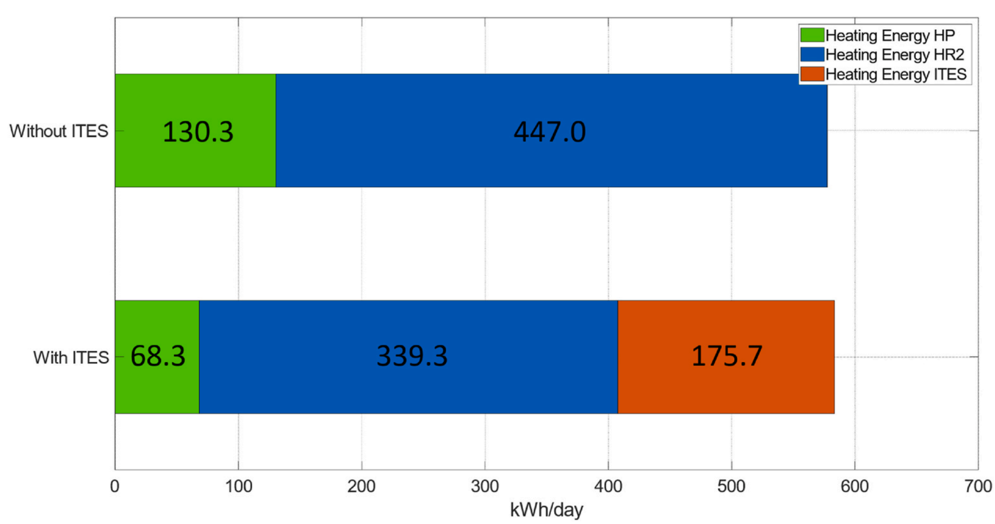

Fig. 9. Comparison between daily heating energy values in both cases.

from the heat pump is more marked and has a peak, up to 66, 5 kW, at the early morning. Similarly, to the summer case, the use of ITES allows a lower capacity of the heat pump.

Figs. 9 and 10 show a quantitative energy analysis, comparing the use of the storage in terms of electrical energy and thermal energy absorbed. The values shown are calculated from the energy balance on the HVAC tank, the two cases do not match perfectly due to the differences in heat losses in the two cases, the stratification of the HVAC tank and the fact that the storage is not at exactly the same temperature at the end of the day in the case with or without ITES. As expected, in the case without ITES, the energy use for the heat pump is higher (+91%), as is the heat supplied by the CRU (+32%). It is important to mention that ‘heating energy HR2’ only considers the heat recovered directly from the HR_2, thus excluding heat from the ITES. Approximately, in this specific supermarket, the storage tank provides up to about 30% of the heat demand.

By analysing the absorbed electrical energy, it can be seen that, similarly to the summer case, the electrical energy is higher if the ITES is used; specifically, an increase of 7 % is recorded.

In the case with ITES, electricity consumption from the heat pump is reduced by 56%, while electricity consumption from the CRU is increased by 18%. The main reason for the non-energy-efficiency, even in the winter case, is due to the fact that the CRU is forced into trans-critical operation to provide heat in winter, and the efficiencies of the commercial refrigeration unit are still lower (although less pronounced than in the summer case due to the favorable outdoor temperature) than those of the heat pump.

The real convenience may therefore once again be in the reduction of

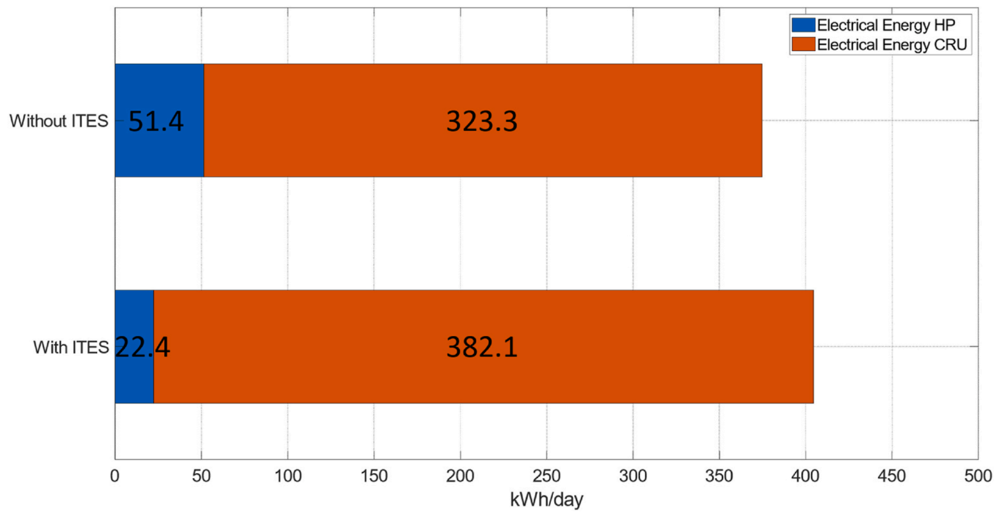

Fig. 10. Comparison between daily electrical energy values in both cases.

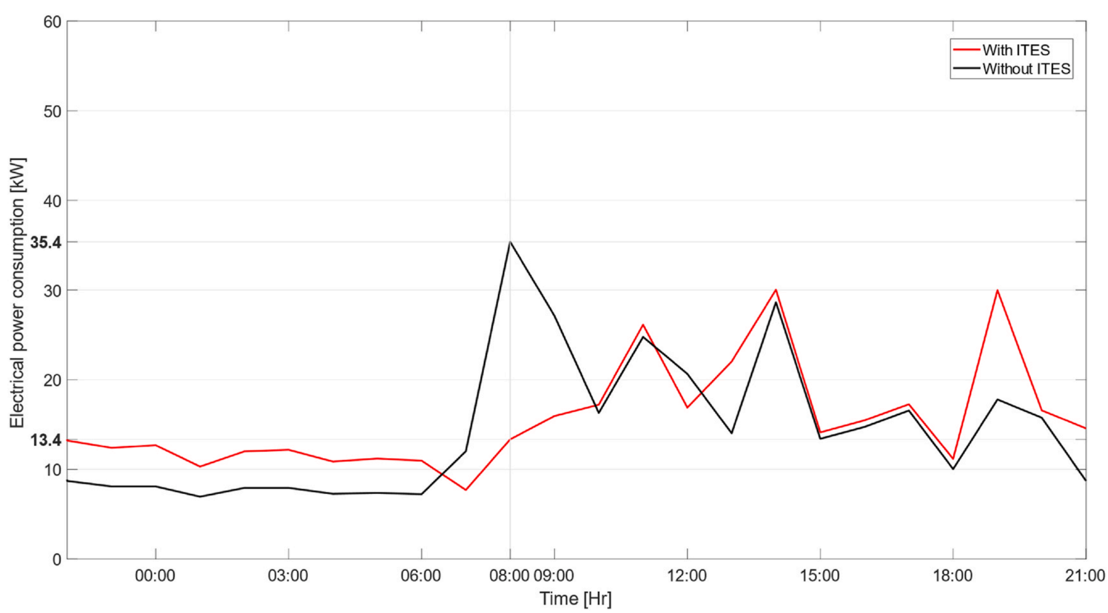

Fig. 11. Comparison of the daily trend in electrical power absorption for CRU and HP with and without ITES, winter day.

peak electricity demand in the morning. A trend of electrical power use (only for refrigeration and heating) is shown in Fig. 11, where a reduction in the morning (at 08:00 a.m.) is shown, from 35.4 to 13.4 kW.

4.2. Yearly energy evaluation

A comparison of monthly electrical energy use between ITES (right columns) and non-ITES (left columns) systems is shown in Fig. 12. As expected from the daily comparison, the solution with ITES requires more energy, with the exceptions of April, May and October, when storage is not used since the demand is not sufficient to motivate the use of ITES. The months with the highest energy requirement are the summer months when the cooling load is higher.

5. Cost analysis

Since in this specific analysis storage is used to contain costs, an economic analysis over the years of supermarket operation is conducted.

The difference between the two scenarios (with and without a storage) in terms of cost is defined as:

$$\Delta\mathrm{Cost} = (\mathrm{CC} + \mathrm{RC}) - (\mathrm{CC}_{\mathrm{ITES}} + \mathrm{RC}_{\mathrm{ITES}}) \tag{9}$$

Where CC are the capital costs and RC the running costs defined as follows:

$$\mathrm{RC} = \mathrm{AEC}\frac{(1+\mathrm{r})^{\mathrm{n}} - 1}{(1+\mathrm{r})^{\mathrm{n}}\mathrm{r}} \tag{10}$$

Where n is the lifetime of the investment in years, r is the discount rate assumed to be 6.5% (Giunta and Sawalha, 2021), and AEC are the annual energy costs calculated as:

$$\mathrm{AEC} = \sum_{h=1}^{n} E_h\, e_h \tag{11}$$

where E_h represents the hourly absorption of electricity during the year, and e_h represents the hourly cost of electricity; an Italian 3- band electricity tariff is assumed (Fig. 13). The first slot is called F1: active from Monday to Friday, from 8.00 a.m. to 7.00 p.m., excluding national

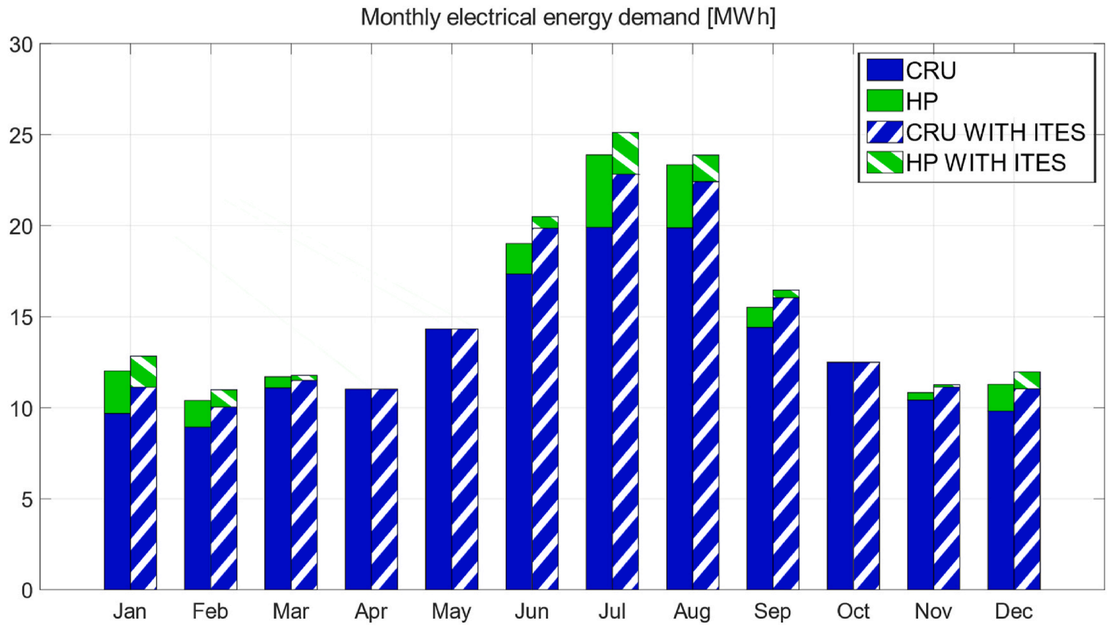

Fig. 12. Monthly electrical energy used in one year in both cases.

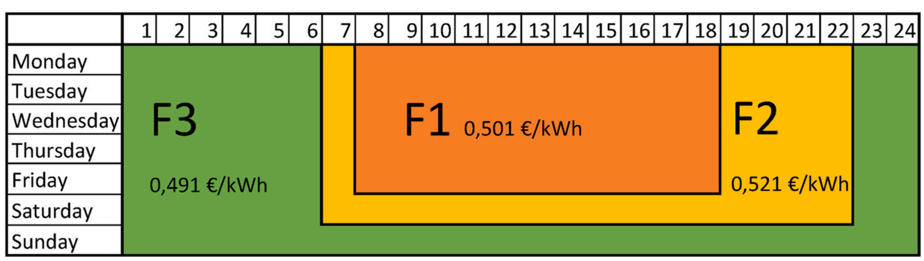

Fig. 13. Hourly tariff scheme.

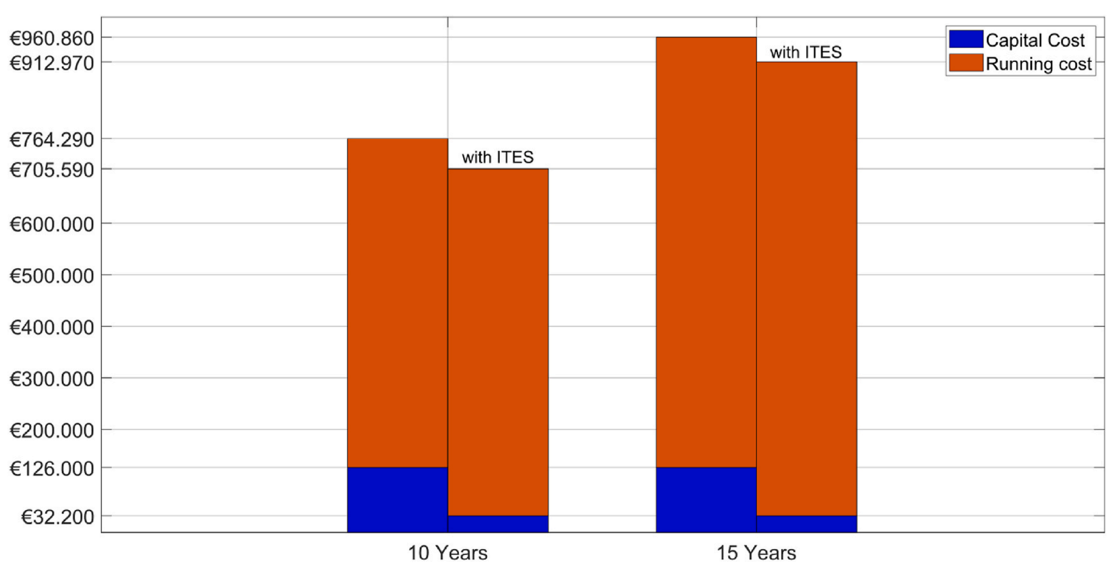

Fig. 14. Cost comparison for the lifetime of the supermarket with and without storage.

holidays. The second slot is called F2: active from Monday to Friday, from 7:00 a.m. to 8:00 a.m. and from 7:00 p.m. to 11:00 p.m., and on Saturday from 7:00 a.m. to 11:00 p.m., except on Saturdays which are national holidays. Finally, there is the third slot called F3: active from Monday to Saturday from 11:00 p.m. to 7.00 a.m. and all day on Sundays and public holidays. The prices for the three slots are 0,501 €/kWh for F1, 0,521 €/kWh for F2 and 0,491 €/kWh for F3 (Italian Regulatory Authority for Energy, 2022).

Table 3
Summary values costs for the duration of 10 and 15 year

10 Years	Without ITES		With ITES
Capital Cost			
Power supply	100.000 €	ITES storage	15.000 €
Heat Pump	26.000 €	Heat Pump	17.200 €
Running cost	638.292 €		673.393 €
TOTAL	764.292 €		705.593 €
ΔCost			−58.699 €
15 Years	Without ITES		With ITES
Capital Cost			
Power supply	100.000 €	ITES storage	15.000 €
Heat Pump	26.000 €	Heat Pump	17.200 €
Running cost	834.857 €		880.769€
TOTAL	960.857 €		912.969 €
ΔCost			−47.888 €

Regarding capital costs, in the case without storage the installation of a transformer room to supply the supermarket with medium-voltage electricity is required, due to the high absorption peak. The cost of this operation is estimated at around 100 k€, while the cost of the air-to-water heat pump for air conditioning and space heating is 26 k€ (70 kW cooling) with a total initial cost of 126 k€. With the ITES, the total cost of the tank plus the installation equals 10 k€, while the heat pump required capacity can be reduced from 70 to 50 kW reducing its cost to 17.2 k€, giving a total amount of 27.2 k€. Since the objective of this analysis is the absolute value of the cost difference, other costs are not considered in this analysis.

Given to the lower investment cost for the solution with storage, this configuration has the maximum economic advantage on the first day, and the goal is to find the time to reach the balance. This period has been found at around 30 years, longer than the usual lifetime of such refrigeration systems.

A comparison over 10 and 15 years of operation between the two solutions is therefore proposed in Fig. 14. As already mentioned, the solution with storage is the most economical over time, going from 764,29 k€ to 705.29 k€ in 10 years (8% saving) and from 960.86 k€ to 912.97 k€ in 15 years (5% saving). This analysis is done with a fixed energy price over the years, the greater the price variation the greater the potential profit due to the use of ITES could be. The values of the costs are summarised in Table 3.

6. Conclusions

A real case of a supermarket where the refrigerating unit provides also cooling and heating in favour of the building is investigated. Ice thermal energy storage (ITES) is used, to reduce electrical power peaks in the morning, when many other electrical appliances (ovens etc.) are in use. The use of heat storage in such a case shows to be detrimental for the energy efficiency, as expected. An estimation of the daily energy use is conducted for both winter and summer typical operation. in summer two cases are evaluated: Case 1, supplying the whole AC demand early in the morning, and Case 2, to assist the water chiller in covering the air conditioning peak demand, thus allowing a reduction in its design size. With Case 1 the ITES is better exploited, showing an higher efficiency (80.7%) compared to Case 2 (74.9%). However, because the storage is used when the heat pump would have had lower EER values, Case 2 shows lower electrical energy usage than Case 1. In the winter operation, an higher energy use is experienced as well. Then, an analysis on an annual basis is conducted, confirming an increase in energy need when ITES is used, for almost every month of the year. However, the cost analysis shows that the reduction in the required capacity of the reversible heat pump, a better exploitation of tariffs and the chance to avoid the installation of an electrical transformer in a dedicated room allows saving 58.699 € in 10 years or 47.888 € in 15 years, thus making the choice of ITES more profitable in the usual lifetime for these plants.

There is still room for improving the performance by acting mainly on the control of the plant, especially for the charging phase.

Furthermore, the use of ITES could allow a lower capacity of the heat pump from 66.5 to 27.8 kW of heating capacity but the summer operation forces to cover 45.9 kW of cooling capacity.

CRediT authorship contribution statement

Gabriele Toffoletti: Conceptualization, Methodology, Software, Writing – original draft. **Giovanni Cortella:** Writing – review & editing, Supervision. **Paola D'Agaro:** Methodology, Software, Writing – review & editing.

Declaration of competing interest

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

Data availability

Data will be made available on request.

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