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Performance assessment of some ice TES systems

David MacPhee*, Ibrahim Dincer

Faculty of Engineering and Applied Science, University of Ontario Institute of Technology, 2000 Simcoe Street North, Oshawa, Ontario L1H 7K4, Canada

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ABSTRACT

In this paper, a performance assessment of four main types of ice storage techniques for space cooling purposes, namely ice slurry systems, ice-on-coil systems (both internal and external melt), and encapsulated ice systems is conducted. A detailed analysis, coupled with a case study based on the literature data, follows. The ice making techniques are compared on the basis of energy and exergy performance criteria including charging, discharging and storage efficiencies, which make up the ice storage and retrieval process. Losses due to heat leakage and irreversibilities from entropy generation are included. A vapor-compression refrigeration cycle with R134a as the working fluid provides the cooling load, while the analysis is performed in both a full storage and partial storage process, with comparisons between these two. In the case of full storage, the energy efficiencies associated with the charging and discharging processes are well over 98% in all cases, while the exergy efficiencies ranged from 46% to 76% for the charging cycle and 18% to 24% for the discharging cycle. For the partial storage systems, all energy and exergy efficiencies were slightly less than that for full storage, due to the increasing effect wall heat leakage has on the decreased storage volume and load. The results show that energy analyses alone do not provide much useful insight into system behavior, since the vast majority of losses in all processes are a result of entropy generation which results from system irreversibilities.

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1. Introduction

Energy storage is an extremely important part of our society. In almost every facet of science and technology, energy storage plays a significant role, whether the energy is needed in chemical, heat, mechanical, electrical or other forms. Though the motivation for the recent technological advancements in the various fields of energy storage varies, the overall impetus is the same; our energy supply – whether it comes from the earth or the sun, is never a constant. Day turns to night, winds die down, oil fields eventually run dry, and the geothermal heat from the crust of the earth, although seemingly constant, will eventually diminish. There is, then, a need to store energy, for the purpose of extracting it when it is not readily available. This is clearly evident in solar panels, which convert the sun's radiation into electricity for later use. In fact, the storage of energy thermally perhaps dates back as far as civilization itself; since the beginning of recorded history, people have been harvesting ice to keep things cool when warmer weather approaches. It is this type of thinking which has provided the desire to store many other types of energy from various sources, both for economic and ecologic purposes.

For the past few decades, the world's energy supply has not been keeping up with the increasing demand. Burgeoning countries undergoing industrial reform are consuming an increasing amount of crude oil, coal and electricity, which has been increased overall energy prices to an unprecedented level. As a result, energy conservation has been on the rise lately, and new sources to feed the human energy hunger are sought without relent. Moreover, the search for more efficient, ecologically friendly and cost effective ways to capture and store energy for later use is always a popular topic.

Thermal energy storage (TES) can be a cost effective and environmentally benign solution when dealing with these rising energy prices. There are very many ways to store thermal energy; however an extensive review is available [12] for a more complete review of the various technologies present. In short, TES can be stored in two ways: latent and/or sensible storage. Latent storage refers to the energy change in a substance as it undergoes a change in phase, say

* Corresponding author.
E-mail addresses: macphee@rohan.sdsu.edu (D. MacPhee), ibrahim.dincer@uoit.ca (I. Dincer).

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Nomenclature

A	surface area [m^2]
C	specific heat [kJ/kg °C]
E	total energy [J]
h	specific enthalpy [J/kg]
I	irreversibility [J]
L	latent heat [J/kg]
m	mass [kg]
Q	heat transfer [J]
R	thermal resistance [m^2 °C/W]
S	total entropy [kJ/K]
t	time [s]
V	volume [m^3]
W	work [J]

Greek Letters

β	coefficient of performance [–]
ρ	mass Density [kg/m^3]
ρth,max	max ice storage density [J/m^3]
Δ	"change in" [–]
Ξ	exergy [J]
η	energy efficiency [–]

Subscripts

a	first ice storage Process
b	second ice storage process
ch	charging
cond	condenser
dc	discharging
des	desired
evap	evaporator
f	final
g	glycol solution
gen	generated
i	initial
in	inlet
ice	ice property
l	leakage
out	outlet
Q	heat leaked
ref	refrigerant
req	required
room	room condition
sf	fusion
st	storage
sys	system
storage	total stored
sens	sensible
T	total
total	total amount
w	water property
1,2,3,4	refrigerant cycle stages
∞	ambient condition, dead state

from liquid to solid (as ice freezes) or from a liquid to a gas (as water boils). As the material changes state during what is called a *phase transition*, energy is released or absorbed, depending on the direction of the process. The advantage of latent storage is that the energy released/absorbed can be done so at a constant temperature, making the process easier to regulate. However, since phase transitions occur for certain substances only at a certain temperature and pressure, there is sometimes trouble in finding the right phase change material (PCM) to suit certain processes.

In contrast, sensible thermal storage is the energy stored in a change in temperature of a material. All materials have a property called a *specific heat*, which loosely means the amount of energy it takes to change the temperature of 1 kg of a substance by 1°. For example, water at 5 °C has a specific heat of 4.2 kJ/kg K. This means that it takes 4.2 kJ of energy to change the temperature of 1 kg of water by 1° at 5 °C. Thus, energy can be stored in the temperature change of any material, and since the temperature at which energy is released and stored is variable, sensible TES is applicable to almost any application. However, one of the main drawbacks of sensible TES is its large storage size. For example, in the freezing of just 1 kg of ice, about 334 kJ of energy must be released, while over 79 kg of water would be needed to store the same amount of energy at a temperature difference of 1 °C. This is why latent TES has been receiving much more attention in recent years, mainly in ice storage for use in cold TES applications.

Cold TES has become much more important in recent years, due to the increased energy demand in many parts of the world. For example, large office buildings in warmer climates can spend immense amounts of money and energy in air conditioning alone. As a result, there can be a great advantage to shift electricity usage from high demand times to lower ones, in order to save money. This is usually done by storing thermal energy from a cooling system, run with electricity at night, so that the cold thermal energy can be extracted during peak cooling periods during the day. To get a brief understanding of the electricity cost and demand relationship, in a typical August day in Ontario, Canada, night time electricity costs can be as low as 20% that of the peak demand price in the day [16]. In other, warmer climates, this disparity can be much lower, and is increasing the need for cold TES to help alleviate cooling costs.

The materials for cold TES are usually limited to water and eutectic salts. The reason for this is the incredibly low cost of water, and also because of its relatively high heat capacity, as noted earlier. Water has one of the highest latent heat of fusions and sensible heats known, and due to its extremely low cost, it is used almost exclusively in cold TES. However, since it freezes only at a specified temperature and pressure, eutectic salts are sometime used in conjunction with the water. Eutectic salts are simply a combination of inorganic salts, water and other elements, which create a mixture that will freeze at a desired temperature. Using these eutectic salts can help create materials which are ideal for each cold TES process.

The various types of cold TES can store the cold energy in either latent or sensible ways. Sensible cold TES methods are usually limited to cold water chillers. In these devices, water is chilled with a vapor-compression refrigeration cycle to cooler temperatures at night and stored for future times. Although sensible cold TES is quite simple and cost effective, the size of these devices, as noted earlier, is quite large when compared to ice storage.

The remainder of this introduction will be concerned with ice TES systems, and will include four main methods; ice slurry, ice-on-coil (both internal and external melt) and encapsulated ice TES. Each method has its own advantages, but all are similar in the sense that they can be used (depending on the operating strategy) to reduce greenhouse gases (GHG), shave peak electricity demand and reduce operating costs.

2. Operating strategies

There are three basic operating strategies when dealing with cold TES. These include full storage, load leveling and demand limiting load profiles. While these three strategies differ in the overall load profile in which they store and retrieve the cold thermal energy, they are similar in the sense that they ultimately serve to reduce costs or environmental impact.

One of the most compelling reasons to use a cold TES system is to lower operating costs. This can be achieved in using either or both of the load leveling and demand limiting strategies. The main focus here is to key on low electricity demand and cost times. Since vapor-compression cooling techniques require electricity to pump heat from a low temperature source to a warm source, electricity costs are becoming a large factor when designing cooling systems. So, both load leveling and demand limiting strategies use the chiller to produce ice during the night, and both level electricity usage, as well as limit the usage during high demand times. Since demand and cost of electricity are usually closely related, these two operating strategies often amount to the same outcome; reduced peak electricity demand as well as reduced cooling costs. In a few typical case studies provided by Dincer and Rosen [11], even the most expensive capital investments can achieve payback periods of as little as 2–7 years.

To illustrate a typical example of a load profile, one industrial case, with data taken from Carrier Corp. [4] is shown in Fig. 1. Shown in the figure is the building requirement, denoted by the "no storage" chiller load, along with the partial and full storage chiller loads. The "no storage" curve, which is typical of systems incorporating no ice storage, is at a maximum during mid-day, when electricity demand is at its highest. In contrast, the partial storage can be seen to have a heightened load during the night hours, and only slightly higher chiller consumption during peak electricity times. Full storage diverts the entire chilling load to off-peak times. These tactics serve to both lower operating costs due to reduced peak electricity consumption, as well as lower overall peak demand, by shifting loads to off-peak times. The peak demand shifting of these ice TES systems could be the most advantageous aspect, since it ultimately results in lowered GHG emissions from the burning of fossil fuels.

The reason for the reduced GHG emissions is due to the nature of supplied electricity in response to demand. In most cases, the base load is supplied by power plants working in an optimal efficiency range. These plants operate continuously, 24 h per day, since the startup costs greatly outweigh any potential benefits to slowing or stopping electricity production during low demand (night) times. However, during the day, peak load requirements must be met by other means, which include natural gas turbines and other GHG emitting sources, which may not be working at peak efficiency levels. Due to this fact, it can be summarized that shifting electricity usage from peak demand times to low demand times can allow for the base load to cover more of the total load, eliminating the need for more GHG emitting power plants covering the extra load during peak demand times.

In addition to operating strategies, there are a number of methods currently in use or in development to store ice for purposes in space cooling. These include ice slurries, ice-on-coil and encapsulated ice systems, which will be discussed shortly.

2.1. Ice slurries

In general, ice slurry refers to a mixture of ice crystals and liquid. The liquid in question is usually an antifreeze solution of water and a freezing point depressant such as ethylene glycol. Ice slurry systems come in a variety of sizes and configurations; for example, Wang and Kusumoto [31] discuss a number of ice slurry TES systems, incorporating a host of ice production methods. Kasza and Hayashi [18] present a summary of the various ice slurry storage and agglomeration techniques. However, the most widely applied ice making technology is the scraped surface process [13]. It employs a typical vapor-compression refrigeration cycle whose evaporator is located on the outside of a tube-in-tube heat exchanger. The inner tube contains the binary antifreeze solution, and the water content freezes on contact with the outer cylinder. A rotating scraper lifts off the ice, and the ice is then transported through the length of the heat exchanger with the heat transfer fluid, thereby increasing the ice content and cooling potential of the heat transfer fluid. A simplified schematic of the cross section of such a system is shown in Fig. 2.

Many experimental studies in the literature have also been conducted concerning different methods for ice production in slurries. Matsumoto et al. [25] discuss a system where an oil–water mixture is cooled while stirring, creating ice crystals with diameters of less than 3.5 mm. Performance tests were carried out to determine the effects of cooling rate and stirring wing diameter on the size and rate of ice crystal production. Yamada et al. [33] propose an oscillatory rotating cooled tube method to generate ice crystals. Briefly stated, the tube is cooled by a refrigeration cycle, and immersed in a glycol solution. Once the solution reaches its freezing temperature, the tube begins to oscillate and the angular acceleration varies in order to expel ice crystals in the "mushy"

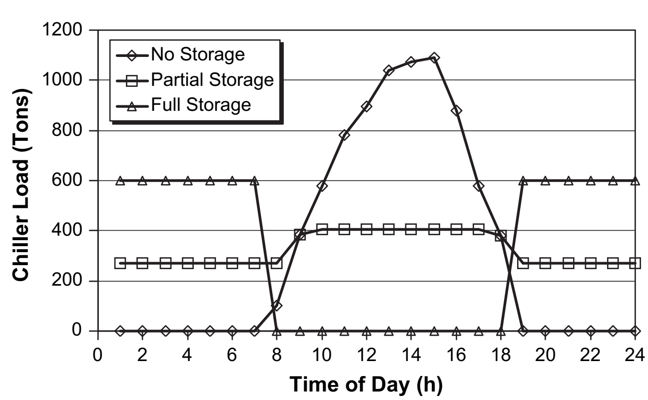

Fig. 1. Typical building load (no storage) chilling requirements, as well as partial and full storage requirements.

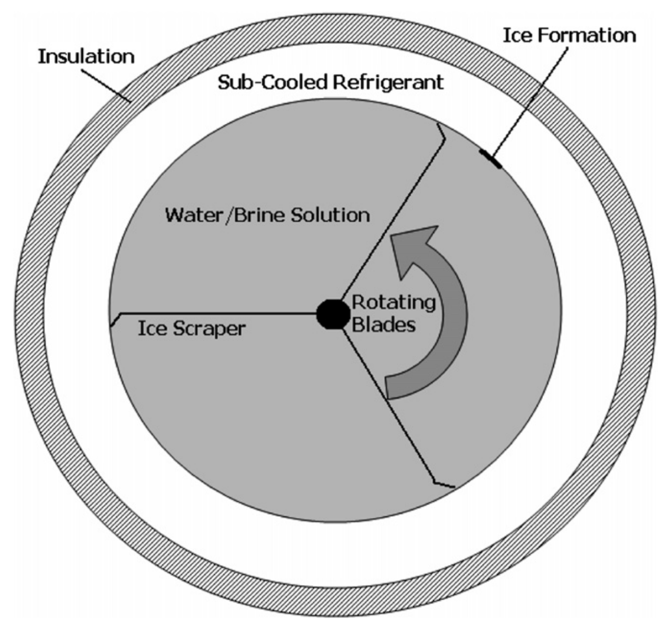

Fig. 2. Cross-sectional schematic of a scraped surface ice slurry generator.

zone; a zone where a solid matrix of crystals is mixed with interstitial liquid [1]. Tests were performed to indicate the dependence of angular acceleration, rotation angle of oscillation and initial composition of the solution, and compared to present models, with varying success. Koszawa et al. [21] propose a numerical method to predict the effects of ice content and mass flow rate on the storing characteristics of the dynamic-type system. Their model agrees well with experimental values, and is considered to be an acceptable design tool when conceptualizing systems such as these.

Other works in the literature are concerned with analyzing the effect of the ice carrying solution. For example, Guilpart et al. [15] propose an analytical method to determine the effect of the secondary refrigerant on ice slurry performance. The influence of operating temperature, volume enthalpy drop and relative viscosity on system performance is analyzed. Their results show that inorganic mixtures provide the best combination of the above three variables in order to achieve better system performance.

The advantage of an ice slurry based cool TES system is the fact that the ice storage density can be relatively large when compared to other systems. In addition to this, the ice slurry can be applied directly to any cooling load, since additional heat transfer fluids are not needed as in other ice cooling systems. However, one major drawback of the ice slurry technique is the costly manner in which the ice is produced. A high amount of energy is required to drive the ice scrapers in the scraped ice technique, but new ice making procedures currently being studied may hopefully reduce this drawback.

2.2. Ice-on-coil

Ice-on-coil systems can effectively solve some of the cost and energy density problems associated with ice storage. In fact, systems like these require very little maintenance and can be successfully operated for years. Chang and Nixon [5] describe such a system; one which has been operating in an army camp in Arizona for over 12 years. These easy to operate, yet low-cost systems typically come in two types; internal and external melt. Internal melt systems use a sub-cooled brine solution, most likely a refrigerant running in a vapor-compression refrigeration cycle, which runs through coils immersed in a tub of water. The coolant effectively freezes the water during charging times and during discharge periods, the ice extracts heat from the brine solution, cooling it for use in air conditioning applications. Kiatreungwattana and Krarti [20] discuss such a system. External melt systems employ the same procedure for freezing the water, but during discharge periods the ice is melted from outside the coils as in the system shown in Fig. 3.

Though simple in concept, the attempts to predict behavior of such systems through numerical and analytical modeling can prove difficult. Erek and Ezan [14] undergo a numerical and experimental study of the charging process in an external melt TES system. The numerical procedure was made much simpler by considering a small section of the tank and by considering a few symmetry assumptions. The control volume approach used in this study provided good results into system dynamics, and could accurately predict the effects of HTF flow rate and inlet temperature on the cool storage characteristics of the tank. Such characteristics included heat transfer rate, total stored energy and energy efficiency. Fig. 3 shows a typical arrangement of an ice-on-coil storage system.

Various other models, which attempt to predict system behavior and operating modes, are also in the literature. A few analytical models were developed by Lee and Jones [23] which includes a myriad of simplifying assumptions, while Zhu and Zhang [35] model the melting and solidification processes, accounting for density differences in ice/water mixtures to correct heat transfer rate approximations. Ihm et al. [17] have used the EnergyPlus software to simulate various systems integrated with cooling applications including ice-on-coil (both internal and external melt) as well as an ice-harvesting system. Control of these systems is addressed, as is the expected input and output predictions.

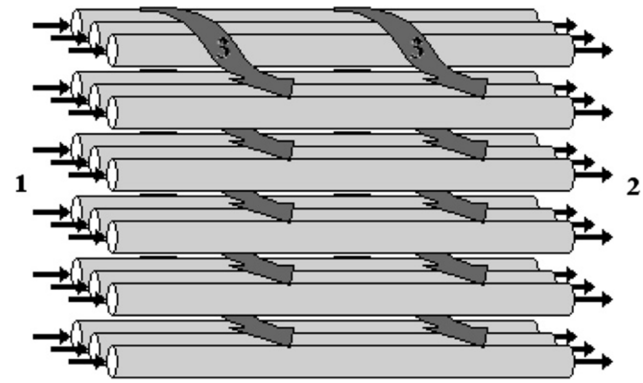

Fig. 3. Typical arrangement of an ice-on-coil storage system. In an internal melt system, refrigerant enters the coils (1–2) and cools the surrounding water (3), and the cold energy can be extracted via the same coils (1–2). During external melt, the ice must be harvested by flowing water or coolant (3).

Though these systems are quite easy to operate and have low maintenance costs, the installation costs can still be quite high, and have limited large-scale applications, due to the immense network of coils which must be constructed for the storage tank. This has led ways to other means of storing the PCMs, where the heat can be stored and retrieved with relative ease and at low manufacturing and operating costs.

2.3. Encapsulated ice

In this method of ice storage, the water is packed into capsules, which in turn are packed into a storage tank. A heat transfer fluid can then be run through the storage tank when heat extraction or input is desired. The simplicity in design in this case occurs where the capsules (usually spherical, but can be of any geometry) are mass-produced, and used to fill any sized storage tank to meet any cooling load requirements. Typically, the storage tank will be of a cylindrical shape, for the following reasons: the cylinder is a relatively low-cost shape to produce which can withstand high pressures, and the surface area-to-volume ratio is lower than most other geometries, allowing for less heat penetration or leakage from the system.

There have been a considerable amount of experimental works in the literature which analyze packed bed encapsulated TES (for example Ref. [6]) and most of them consider the storage tank as a whole. In other words, inputs and outputs to the system (flow rate, inlet HTF temperature, void fraction, etc.) are monitored to discover their effects on system efficiency, as well as charging and discharging characteristics. The wealth of information regarding packed bed investigations is immense, however most are concerned with high velocity gas flowing through a bed with particles of small diameter; for example Refs. [32,9]. These studies, although helpful, usually have applications in reactor beds [34] or adsorption beds. Due to the small particle diameters, they cannot correctly analyze the important phenomena which are inherent to encapsulated packed bed TES. For these larger capsules, a number of studies have been undertaken in order to analyze thermal dispersion [27] and viscous dissipation [24] [Lee and Kamiuto, 2002]. A critical review of the thermal dispersion in various packed beds was done by Delgado [10].

Numerous experimental investigations regarding packed bed flows have been conducted over the years (for example Refs. [30,2,26,7]) to determine the effects of flow rate, void fraction and capsule geometry on the overall performance of packed bed latent TES, but it is usually much easier and cost effective to consider a numerical investigation to simulate flow and heat transfer phenomena. This is because the flow or temperature fields do not need to be solved a priori; the temperature and flow fields are simplified into a finite number of nodes or volumes and relations between them are approximated.

Some of the works in the literature are concerned with modeling existing systems, such as the one presented by Kerslake and Ibrahim [19]. Here, a two-dimensional axisymmetric model is used to discuss the role of free convection on the heat transfer performance of the TES storage tank.

Of the many numerical procedures available in the current literature concerning packed bed, encapsulated TES, most are concerned with warm TES in paraffin waxes. Zukowski [36] analyses the heat transfer characteristics in a ventilation duct filled with encapsulated paraffin wax in rectangular configurations. They consider a three-dimensional transient model, which is then used to predict the effect of capsule geometry and configuration on heat storage. It was also found that introducing parallel connectors downstream from the inlet could greatly assist in making the heat storage or retrieval more uniform. Benmansour et al. [3] provide a two-dimensional transient analysis of a cylindrical storage tank filled with uniformly sized spherical capsules. The paraffin wax in the randomly packed capsules exchanges heat with air, acting as a heat transfer fluid, and the resulting model is found to agree favorably with expected results.

Kousksou et al. [22] propose a two-dimensional approach to solve for the temperature field in a cylindrical container containing spherical capsules used for ice storage. The porous medium model was used, and along with Churchill [8], who proposed the average Nusselt number for such flows, the entire charging and discharging processes could be evaluated. Density variations within the HTF were considered, and the system was run in both the vertical and horizontal positions. It was determined that the optimal case occurred with the tank in the vertical position, when the natural convective currents coincide with forced convection currents.

Although the above is a shortened list of the immense amount of work in the area of ice TES, it will suffice as a foundation as the performance criteria for the various methods will be introduced in the next section.

3. Analysis

The following includes both energetic and exergetic analyses of the charging, storage and discharging processes of ice storage systems. Since the ultimate goal is to identify the energy and exergy efficiencies associated with each process, there must be a number of assumptions which will be addressed in order to properly compare the processes. A few briefcase studies will follow the analysis in order to achieve this.

In the process of charging the ice storage, a vapor-compression refrigeration cycle will be used, since the majority of all refrigeration and ice making techniques make use of this type of system [28]. If pipe losses and expansion valve heat losses are assumed negligible, the four cycle process is as displayed in Table 1.

To more easily view the performance characteristics inherent in each system, the discharge process will involve the cooling of an antifreeze solution. The analysis will also incorporate a storage tank of varying size in order to more realistically approximate real-world scenarios. However, the analysis can be greatly simplified if the chiller and storage tank are lumped together as a single process, so that efficiency calculations can be presented with more ease. More specifically, the process assumptions incorporated are as follows:

- During charging the only input to the system will be the compressor work. Other interactions with the ambient atmosphere include heat leakage into the tank, condenser heat transfer to the ambient and energy lost due to inefficiencies in the refrigeration cycle.
- During storage, there is no input to the system; the only interaction with the ambient is heat leakage into the ice storage tank.
- During discharge, the ice storage is used to cool an ethylene glycol solution. Therefore, the flow difference between inlet and outlet states must be considered, as will be the heat leakage from the ambient atmosphere.
- In terms of specific analysis, the following assumptions are made:
  - All kinetic and potential effects are neglected.
  - All piping losses and viscous dissipation losses assumed negligible.
  - The storage tank is cylindrical, with diameter equal to height.
  - The thermal energy stored in the HTF is negligible – all thermal energy is stored in the water/ice medium.
  - Tank is considered to be of constant temperature, changing according to the specified system process.
  - All thermophysical properties are assumed constant at their prescribed values.

Now that the basic assumptions for the investigation have been addressed, it is possible to address the energy and exergy analyses which provide the basis for the performance parameters.

3.1. Energy analysis

An energy balance on the entire process results in the following:

$$\Delta E_{sys} = E_f - E_i = E_{in} - E_{out} \tag{1}$$

The energy efficiencies associated with the charging, discharging and storing cycles are defined as the ratio of the desired energy to the required energy contents.

$$\eta = \frac{E_{des}}{E_{req}} \tag{2}$$

However, since the charging, storage and discharging processes are quite different; the analyses will be handled separately.

3.1.1. Charging

For the charging process, the interactions crossing system boundaries include the work done on the compressor of the refrigeration cycle, heat leakage from the ambient as well as heat given to the ambient from the condenser of the refrigeration cycle. So, the energy balancing equation in (1) can be reduced to:

$$Q_{evap} = Q_{cond} + Q_{l,ch} - W_{in} \tag{3}$$

In other words, the change in energy of the storage system is exactly the energy absorbed in the evaporator of the refrigeration cycle, while the right hand side of the above equation represents the one input and two outputs across system boundaries. However, since the total amount of stored energy is known, it can be assumed that

$$Q_{evap} = E_{total} \tag{4}$$

Table 1
Cycle description for an ideal vapor-compression refrigeration cycle.

Component	Process	Inlet	Outlet	Notes
Compressor (1–2)	Compression into superheated vapor region	Saturated vapor	Superheated vapor	Adiabatic
Condenser (2–3)	Condensation of refrigerant across vapor region	Superheated vapor	Saturated liquid	Isobaric
Expansion valve (3–4)	Expansion of refrigerant to a two-phase liquid–vapor mixture	Saturated liquid	Two-phase liquid–vapor mixture	Isenthalpic
Evaporator (4–1)	Evaporation of refrigerant	Two-phase liquid–vapor mixture	Saturated vapor	Isobaric

where E_{total} is the known total amount of cold energy stored in the charging process.

The vapor-compression refrigeration cycle used in this ice storage process must be pumping heat from a cold reservoir T_{ch} to the ambient hot reservoir T_∞, with the specific cycle descriptions shown in Table 1. So, the desired energy content in the charging case is simply the amount of heat energy pumped from the cold to the warm reservoir.

$$E_{des,ch} = Q_{cond} - Q_{evap} \tag{5}$$

And, in the same fashion, the work input to the system will be simply the work input required to drive the compressor.

$$E_{req,ch} = W_{in} \tag{6}$$

However, there are still three unknowns in the above equations; namely the condenser and wall heat terms, as well as the compressor work input. Since the coefficient of performance for the refrigeration cycle will be known, the work input can be solved by the definition of the coefficient of performance as follows:

$$\beta = \frac{Q_{evap}}{W_{in}} \tag{7}$$

The condenser heat output can now be solved since the work input is known. The work input depends on the enthalpy change over the compressor, so the total mass of refrigerant used over the charging process is:

$$m_{ref} = \frac{W_{in}}{(h_2 - h_1)} \tag{8}$$

However, there are two unknowns in Eq. (8); the total mass of refrigerant m_{ref} and the enthalpy value at stage 2 of the refrigeration cycle (all thermophysical data at points 1, 3 and 4 are known by the assumptions in Table 1). So, to solve for the mass of refrigerant used, the evaporator heat input is taken:

$$Q_{evap} = m_{ref}(h_1 - h_4) \tag{9}$$

And, applying Eq. (4) with (9), Eq. (8) can be solved for the unknown enthalpy value, h_2. Due to this, the condenser heat input can now be found and is as follows:

$$Q_{cond} = m_{ref}(h_2 - h_3) \tag{10}$$

All terms required for a complete energy analysis have now been addressed, except for one last term; the wall heat leakage Q_l.

Since the temperature distribution is assumed constant within the storage tank, this heat leakage will depend on the tank inner temperature, ambient temperature, wall area and thermal resistance. For the charging process, this average storage temperature is assumed to be T_{st}, so that the wall heat leakage is:

$$Q_l = A\frac{T_\infty - T_{st}}{R_T}\Delta t_{ch} \tag{11}$$

Here, A is the surface area of the tank where heat leakage from the ambient is occurring. The thermal resistance, R_T, of the tank, as well as the charging time Δt will be known. So, it follows that in order to correctly evaluate the heat leakage equation, the surface area A of the storage tank should be evaluated. However, another readily obtainable characteristic about many ice storage techniques is the maximum ice storage density or thermal density, $\rho_{th,max}$, which can be used to find the total volume, and hence, the surface area of the storage tank.

$$V = \frac{E_{storage}}{\rho_{th,max}} \tag{12}$$

Since the storage tank will be of the assumed cylindrical shape, due to the minimized surface area due to heat loss in a given volume, the surface area of the storage vessel is:

$$A = 6\pi\left(\frac{V}{2\pi}\right)^{2/3} \tag{13}$$

So, finally, the heat leakage terms are known, which suffices to obtain all relevant data for the efficiency equation (2) for the charging process.

3.1.2. Storage

There are sometimes two storage processes for ice storage during a daily cycle. Though very short in duration and usually highly efficient, they nonetheless play a role in the overall efficiency. The first storage period occurs when switching from cold storage to cold discharge, while the second usually occurs when switching back from cold discharge to storage (see Fig. 1). For each case, the desired cold energy is the cold energy stored below the ambient temperature at the start of the storage process, minus the wall heat leakage term Q_l, while the required energy is this same stored energy. In other words, if the two storage periods are labeled "a" and "b", the desired and required energy contents are as shown below.

$$E_{des,a} = E_{total} + E_{sens} - Q_{l,a} \tag{14}$$

$$E_{req,a} = E_{total} + E_{sens} \tag{15}$$

$$E_{req,b} = E_{sens} - Q_{l,b} \tag{16}$$

$$E_{req,b} = E_{sens} \tag{17}$$

It should be noted that the sensible energy contents, E_{sens}, exist to liken the thermal energy storage to the ambient temperature, and represent the amount of useful thermal energy the cold storage has when the ice storage has been discharged.

$$E_{sens} = m_w C_w(T_\infty - T_{dc}) \tag{18}$$

In addition to this, the wall heat terms still must be evaluated. However, since the geometry of the storage tank has already been determined, the average temperature in the storage tank is all that is needed.

$$Q_{l,a} = A\frac{T_\infty - T_{st}}{R_T}\Delta t_a \tag{19}$$

$$Q_{l,b} = A\frac{T_\infty - T_{dc}}{R_T}\Delta t_b \tag{20}$$

Now, enough information is known that Eqs. (14)–(17) can be used to obtain the storage efficiencies when inserted into Eq. (2).

3.1.3. Discharging

Since there are no heat or pipe friction losses during discharging, except for the wall heat leakage, the desired and required energy contents for the discharging process are:

$$E_{des,dc} = E_{total} - Q_{l,ch} - Q_{l,a} - Q_{l,dc} \tag{21}$$

$$E_{req,dc} = E_{total} - Q_{l,ch} - Q_{l,a} \tag{22}$$

And, once again, the heat leakage term is as follows:

$$Q_{l,dc} = A\frac{T_\infty - T_{dc}}{R_T}\Delta t_{dc} \tag{23}$$

3.2. Exergy analysis

The exergy balance for any of the charging, discharging and storage processes is as follows:

$$\Delta\Xi_{sys} = \Xi_f - \Xi_i = \Xi_{in} - \Xi_{out} - \Xi_{Q,ch} - I \tag{24}$$

So, the change in exergy, Ξ, of the system from the initial to the final state is the difference between inlet and outlet exergy flows, minus the exergy leaked from heat penetration, Ξ_Q, minus the irreversibilities I which occurs during all processes due to entropy generation. Similar to the energy analysis, the exergy efficiencies associated with the charging, discharging and storing cycles are defined as the ratio of the desired exergy output to the required exergy input.

$$\psi = \frac{\Xi_{des}}{\Xi_{req}} \tag{25}$$

Throughout the remainder exergy analysis, it will be quite apparent that the calculations are very simple, since exergy views “cold” as a useful, and hence, positive, commodity. In contrast, the energy analysis was carefully constructed to ensure all terms remained positive, and that a proper efficiency was obtained in lieu of the coefficient of performance. The exergy analysis allows for a much more direct approach to the system efficiency.

3.2.1. Charging

In charging, the exergy balance equation in Eq. (24) can be written as:

$$\Delta\Xi_{sys,ch} = W_{in} - \Xi_{cond} - \Xi_Q - I \tag{26}$$

The actual energy attained in the storing process, in contrast to the energy value, which was defined as the actual heat *pumped* from the cold to the hot source, can now be described simply as the stored exergy in the tank in the form of ice. In other words, it can be directly related to the total cold storage as follows:

$$\Xi_{des,ch} = \Delta\Xi_{sys,ch} = \Delta E_{sys,ch} - T_\infty \Delta S_{sys,ch} \tag{27}$$

So, all else that is needed is to determine the rightmost term; the entropy change during charging from T_{dc} to T_{st}. Since all system energy is assumed to be contained within the water/ice portion of the tank, this entropy change will consist of both sensible (both liquid and solid) entropy change, as well as the entropy of fusion as the water solidifies. If the energy storage term is kept negative, then the entropy change is as follows:

$$\Delta S_{sys,ch} = m_w\left[C_w \ln\left(\frac{T_{sf}}{T_{dc}}\right) - \frac{L}{T_{sf}} - C_{ice}\ln\left(\frac{T_{st}}{T_{sf}}\right)\right] \tag{28}$$

The middle term in the above equation arises from the entropy of fusion rule, which states that the change in entropy during fusion is the change in entropy divided by the temperature of fusion. The required exergy input for the charging process is even simpler; it is the work added to the system in order to produce the desired exergy storage.

$$\Xi_{req,ch} = W_{in} \tag{29}$$

The exergetic efficiency for the charging process can now be evaluated. However, the exergy balance equation in (26) cannot be solved in full until two more terms are known. These are the exergy accompanying heat transfer in the condenser and the storage tank itself.

$$\Xi_{cond} = Q_{cond}\left(1 - \frac{T_\infty}{T_\infty}\right) = 0 \tag{30}$$

$$\Xi_{Q,ch} = Q_{l,ch}\left(1 - \frac{T_\infty}{T_{st}}\right) \tag{31}$$

So, it is apparent that the condenser temperature is ideal, since no exergy is lost by the condenser heat output. While this is not an exact assumption, there is no data to determine without bias the condenser temperature in each case, so the inclusion of this ideality should not affect the comparisons between the case studies that are to follow.

3.2.2. Storage

For the storing process, the exergy analysis is once again much simpler than its energy counterpart. For example, for any storage process, the exergy balance equation is:

$$\Delta\Xi_{sys,st} = -\Xi_{Q,st} - I \tag{32}$$

For example, if there are two storage cycles that are split up into “a” and “b” as done earlier, each of the desired and required exergy contents are as follows:

$$\Xi_{des,a} = \Xi_{des,ch} - \Xi_{Q,a} - I_a \tag{33}$$

$$\Xi_{req,a} = \Xi_{des,ch} \tag{34}$$

$$\Xi_{des,b} = \Xi_{sens} - \Xi_{Q,b} - I_b \tag{35}$$

$$\Xi_{req,b} = \Xi_{sens} \tag{36}$$

Here, $\Xi_{Q,a}$ and $\Xi_{Q,b}$ represent the exergy loss from heat transfer to the tank, I_a and I_b denote the irreversibilities present in the system due to entropy generation, while Ξ_{sens} denotes the sensible exergy in the system after discharging. In order to solve the above equations, the following are needed:

$$\Xi_{sens} = m_w C_w\left(T_\infty - T_{dc} - T_\infty \ln\left(\frac{T_\infty}{T_{dc}}\right)\right) \tag{37}$$

$$\Xi_{Q,a} = Q_{l,a}\left(1 - \left(\frac{T_\infty}{T_{st}}\right)\right) \quad (38)$$

$$\Xi_{Q,b} = Q_{l,b}\left(1 - \left(\frac{T_\infty}{T_{dc}}\right)\right) \quad (39)$$

And, to evaluate the irreversibilities which arise from entropy generation, the temperature change as a result of heat addition will be calculated as:

$$\Delta T = \frac{Q_w}{m_w C} \quad (40)$$

So, for each of the storage processes, the sensible temperature change may be evaluated once the proper specific heat is inserted into (40); C_{ice} and C_w for storage processes a and b, respectively. This finally allows for the calculation of the irreversibilities present in the system in each process:

$$I_a = m_w C_{ice} T_\infty \ln\left(\frac{T_{st} + \Delta T_a}{T_{st}}\right) \quad (41)$$

$$I_b = m_w C_w T_\infty \ln\left(\frac{T_{dc} + \Delta T_b}{T_{dc}}\right) \quad (42)$$

3.2.3. Discharging

For the discharging process, there will be a glycol solution entering the storage vessel at room temperature and cooled by the ice storage. The glycol solution is assumed to leave at a specified discharge temperature. For the discharge process, the exergy balance equation is as follows:

$$\Delta\Xi_{sys,dc} = \Xi_{in} - \Xi_{out} - \Xi_{Q,dc} - I \quad (43)$$

The desired exergy output for the discharge process is the difference in flow exergy in the glycol solution as it gives heat to the storage tank.

$$\Xi_{des,dc} = \Xi_{in} - \Xi_{out} = m_g C_g\left(T_{in} - T_{out} - T_\infty \ln\left(\frac{T_{in}}{T_{out}}\right)\right) \quad (44)$$

However, in order to evaluate the above equation, the total mass of glycol used will depend on the total energy transferred to the glycol, as well as the temperature change:

$$m_g = \frac{E_{req,dc} - E_{Q,dc}}{C_g(T_{room} - T_{dc})} \quad (45)$$

Now, the exergy transferred from the system from heat leakage will be addressed in similar fashion as was done earlier:

$$\Xi_{Q,dc} = Q_{w,dc}\left(1 - \frac{T_\infty}{T_{dc}}\right) \quad (46)$$

The final step in solving the exergy balance equation lies with solving the irreversibility term I which results from entropy generation.

$$I_{dc} = T_\infty S_{gen,dc}$$

If the total mass of glycol used is taken as a closed system with the storage tank, then:

$$S_{gen,dc} = \Delta S_{sys,dc} + \Delta S_{g,dc}$$

So, the generated entropy is simply the difference between the entropy increase in the storage tank (system, for less confusing notation) as it loses its cold storage, plus the entropy decrease realized by the glycol solution. The two are calculated as follows:

$$\Delta S_{sys,dc} = m_w\left[C_{ice}\ln\left(\frac{T_{sf}}{T_{ch} + \Delta T_a}\right) + \frac{L}{T_{sf}} + C_w \ln\left(\frac{T_{dc}}{T_{sf}}\right)\right] \quad (47)$$

$$\Delta S_{g,dc} = m_g C_g \ln\left(\frac{T_{dc}}{T_{room}}\right) \quad (48)$$

So, finally, all terms and efficiency equations can be calculated for the discharging case.

4. Case studies

The variables used in the case studies here are obtained from Dorgan and Ellison [13], where the operation principles of many ice TES systems are discussed in detail. Many of the systems used in these case studies are also found in Wang and Kusumoto [31].

There will be four case studies for investigation as follows:

(I) Ice slurry storage
(II) Ice-on-coil storage (internal melt)
(III) Ice-on-coil storage (external melt)
(IV) Encapsulated ice storage

To compare the above ice TES methods, a number of universal criteria must be outlined. To begin with, a typical building daily air conditioning load will be considered, with data taken from Carrier Corp. [4]. There will be two separate storage methods – full and partial storages – and the resulting performance criteria will be investigated. Both ice storage and retrieval data can be found in Fig. 1, and are shown quantitatively in Table 2.

The data in Table 2 will give the approximate cold storage times in each process, as well as the thermal energy stored during each process. The refrigerant used in this process will be R134a, a commonly used material for such purposes. The refrigeration cycle will operate between two thermal reservoirs, T_∞ and T_{ch}. The ambient temperature will be set at $T_\infty = 25\,°C$, while T_{ch} will depend on the process in question and outlined in Table 3. The refrigerant properties can be obtained via thermodynamic tables once any two of the temperature, pressure, enthalpy or liquid fraction are known. The coefficient of performance β must also be known, and can also be found in Table 3 for each case study. The total thermal resistance of the storage module is $R_T = 1.98\ m^2\,K/W$, a value deemed reasonable in Rosen et al. [29]. The latent fusion of water (L) is 334 kJ/kg, while the thermophysical properties of liquid water and ice are taken at 5 °C and −5 °C, respectively. Therefore the density of ice will be $\rho_{ice} = 917.4\ kg/m^3$, solidifying at $T_{sf} = 0\,°C$, while the specific heat of both ice and water will be $C_{ice} = 2106$ J/kg °C and $C_w = 4200$ J/kg °C. For the storage process, the storage temperature will be T_{st}, while the maximum ice density or thermal energy density of the module will be given as $\rho_{th,max}$; both values are once again dependent on the case study in question and will be outlined in Table 3. For the discharge process, the glycol solution (30% by mass) has a specific heat of $C_g = 3574$ J/kg °C, and will enter the storage module at the room temperature, set at $T_{room} = 20\,°C$. The discharge temperature is once again dependent on the case, and is shown in Table 3 with the other values for each case.

In Table 3, all data for the ice slurry system were taken from Wang and Kusumoto [31] while the remainder is found in Dorgan and Ellison [13], but displayed in Wang and Kusumoto [31]. The coefficients of performance and maximum storage densities are taken as the average of the range reported, while the evaporator temperature is equal to the maximum charging temperature

Table 2
Cooling loads for both the partial and full storage strategies in the present case study.

Time of day (h)	Partial storage				Full storage			
	Process	Storage (tons)	Building (tons)	Chiller (tons)	Process	Storage (tons)	Building (tons)	Chiller (tons)
1	Charging	270	0	270	Charging	598.85	0	598.85
2	Charging	270	0	270	Charging	598.85	0	598.85
3	Charging	270	0	270	Charging	598.85	0	598.85
4	Charging	270	0	270	Charging	598.85	0	598.85
5	Charging	270	0	270	Charging	598.85	0	598.85
6	Charging	270	0	270	Charging	598.85	0	598.85
7	Charging	270	0	270	Charging	598.85	0	598.85
8	Charging	170	100	270	Discharging	0	100	0
9	Storing	0	385	385	Discharging	0	385	0
10	Discharging	175	580	405	Discharging	0	580	0
11	Discharging	375	780	405	Discharging	0	780	0
12	Discharging	490	895	405	Discharging	0	895	0
13	Discharging	635	1040	405	Discharging	0	1040	0
14	Discharging	670	1075	405	Discharging	0	1075	0
15	Discharging	685	1090	405	Discharging	0	1090	0
16	Discharging	475	880	405	Discharging	0	880	0
17	Discharging	175	580	405	Discharging	0	580	0
18	Storing	0	380	380	Discharging	0	380	0
19	Charging	270	0	270	Charging	598.85	0	598.85
20	Charging	270	0	270	Charging	598.85	0	598.85
21	Charging	270	0	270	Charging	598.85	0	598.85
22	Charging	270	0	270	Charging	598.85	0	598.85
23	Charging	270	0	270	Charging	598.85	0	598.85
24	Charging	270	0	270	Charging	598.85	0	598.85

Note: 1 ton of refrigeration = 3.517 kW.

reported. The storage temperature will be chosen as the average charging temperature reported, while the discharging temperature is the averaged maximum discharging temperature.

5. Results and discussion

For the above prescribed cases, the results will be compared by storage strategy, and data will be presented from both partial and full storage cases. Table 4 lists the efficiency values, along with percentage losses due to heat leakage and irreversibilities from entropy generation. To get a better understanding of overall system performance, Table 5 lists the *overall efficiency* corresponding to each case, and is a result of multiplication of the charging and discharging efficiencies.

The most apparent aspect of the data in Tables 4 and 5 is that all energy efficiencies are over 98%, meaning that energetically, all processes are extremely efficient. This is due to the fact that the only losses in the energy sense are the heat leakage into the storage module. This loss is displayed in Table 4 as a percentage of total energy recovered as Q_w, and are usually well below 1%.

However, in exergetic analysis, the efficiencies are vastly lower; ranging from 69% to over 76% for the charging cycles, and from 18% to over 24% in the discharging cycles. The reason for this much lower efficiency is due to the inclusion of the irreversibility term I in the exergy balance equation, and results from entropy generation due to heat transfer between the storage module and its surroundings. In inspection of Table 4, it is quite apparent that the vast majority of losses in all cases are due to this irreversibility, while exergy loss due to heat leakage is quite low comparatively; in all cases it is less than 1%. This demonstrates the differences between energy and exergy analyses, since energy calculations alone do not allow for a proper measurement of the quality of the ice storage. For comparison purposes, the differences between total energy and exergy efficiencies for the full storage processes are shown below in Fig. 4.

This extra loss term, I is, as stated previously, a result of entropy generation. Entropy generation is a representation of irreversibilities present in a system. For example, a cube of ice sitting in a glass of water will obviously melt if left alone. This symbolizes an irreversible process, in the sense that the volume of water representing the cube will not cool and re-solidify unless work is done on that system to drive heat out of it (for example, in a refrigerator). In the same sense, in the discharging process, the thermal exergy contained in the ice is transferred to the flowing glycol solution, and exergy is forever lost due to the irreversibilities present in this exergy transfer. As a result, the exergetic efficiency in the discharging process is quite low, much lower than in the charging process.

From Table 4, the charging exergy efficiencies are still much lower than the energy efficiencies, and once again this is due to the entropy generation accompanying irreversibilities. The refrigeration cycle is assumed ideal, and as a result the losses experienced

Table 3
Temperatures, coefficients of performance and maximum storage densities used in the case study.

Method of ice storage	Coefficient of performance, β	Storage temperature, T_{st} [°C]	Evaporator temperature, T_{ch} [°C]	Storage density, $\rho_{th,max}$ [MJ/m^3]	Discharge temperature, T_{dc} [°C]
(I) Ice slurry	2.4	−11	−12	167.4	2
(II) Ice-on-coil (internal melt)	3.3	−4.5	−6	172.98	6.5
(III) Ice-on-coil (external melt)	3.5	−6.5	−7	156.6	5.5
(IV) Encapsulated ice	3.5	−4.5	−6	172.98	7

Table 4
Efficiency and losses data for the full storage cycle.

Case	Energy efficiency [%]	Q_w [%]	Exergy efficiency [%]	Ξ_Q [%]	I [%]
Charging					
I	99.2	0.8	46.93	0.23	52.83
II	99.06	0.94	73.65	0.14	26.21
III	98.99	1.01	69.61	0.17	30.21
IV	99.06	0.94	75.98	0.14	23.88
Case	Energy efficiency, η	Q_w [%]	Exergy efficiency, ψ	Ξ_Q [%]	I [%]
Discharging					
I	99.82	0.18	24.51	0.30	75.18
II	99.86	0.14	19.09	0.23	80.68
III	99.84	0.16	19.82	0.26	79.92
IV	99.86	0.14	18.10	0.22	81.68

are lower than in the discharging process, resulting in much higher exergy efficiencies; however still very much lower than the energetic charging efficiencies. The variance in this case is due in large part to the variance in COP used for this study. For example, the case studies which incorporate the lowest charging COP, the exergetic efficiency is much lower (as in the case of the ice slurry). The reason for this lower COP in the case of the ice slurry is most likely due to the extra work input needed to drive rotating scrapers, as well as a less efficient evaporator tube than can be achieved with the other ice making procedures. Another interesting aspect of Table 4 is the percentage of exergy lost due to heat leakage. This value is well below 1% in all cases, suggesting that it is far more important to address the method of storage and retrieval rather than insulating the module itself.

During discharging, all exergy efficiencies were much lower, with the highest efficiency being around 25%. The reason for this much lower efficiency is due once again to the entropy generation during ice melting. The exergy value of the ice storage is much higher when in a colder, solid form, and after melting it is a lesser quality fluid. This loss of quality is reflected by the irreversibility I, and is a result of varying temperature of the exit fluid from the ethylene glycol recovering the cold storage. The systems which have a colder exit temperature, which results in a more quality cold recovering, have higher efficiencies.

So, while the energy efficiencies would lead one to believe that the process is almost ideal, it is in fact far from it; total exergy efficiencies are less than 15% in all cases. This is due to a number of factors, including poor COP's, but a large portion of irreversibility comes from the discharging process, where the discharge temperature plays a large role. Exergetically, it is more desirable to have both a storage temperature and the evaporator temperature closer to the solidification temperature, a high thermal density, and a lower discharge temperature. This latter point may seem contradictory, since higher discharge temperatures mean more energy difference in the flow. However, if the discharge temperature is decreased closer to the solidification temperature, the process becomes much more ideal, which is shown by the discharging exergy efficiency of the ice slurry system.

Table 5
Total energy and exergy efficiencies for the four ice storage systems (full storage).

Case	(I) Ice slurry	(II) Internal melt	(III) External melt	(IV) Encapsulated ice
Total energy efficiency, η_{tot} [%]	99.02	98.92	98.83	98.92
Total exergy efficiency, ψ_{tot} [%]	11.50	14.05	13.79	13.75

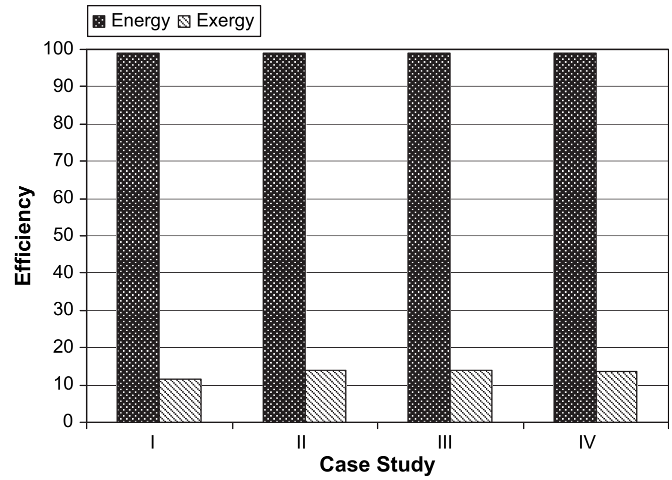

Fig. 4. Comparison between total energy and exergy efficiencies for the full storage case.

For the partial storage systems, the same arguments can be made. For example, all efficiencies (both energetic and exergetic) are very similar to the energy case, despite the lessened storage load and the inclusion of the storage cycle. This is exactly as expected, since the only effective difference between the two in terms of performance is the heat loss equation – which accounts for very little of the overall energy stored. However, since the partial load systems need less ice, and therefore smaller storage tanks, they will lose less heat to the surroundings – a seemingly desirable trait. But, when looking at the total stored energy/exergy decrease in the efficiency equations, this actually amounts to a decrease in efficiency in both the energy and exergy sense. As a result, it is always more desirable, from an energy or exergy standpoint, to store as much of the building load as possible to maximize performance. All efficiencies associated with the charging, storage and discharging processes in the partial storage scenario are shown in Table 6, and can be compared to the full load shifting scenario in Table 4. The total efficiencies are also displayed in Table 7.

Once again, in the exergy sense, the external melt system is the most efficient, followed by both the internal melt system and the encapsulated ice system, while the ice slurry is the least efficient. Energetically, all systems operate at over 98% efficiency, a value that could be deemed almost ideal if not for exergy analyses.

Table 6
Efficiency and losses data for the partial storage cycle.

Case	Energy efficiency [%]	Q_w [%]	Exergy efficiency [%]	Ξ_Q [%]	I [%]
Charging					
I	98.86	1.14	46.87	0.34	52.79
II	98.64	1.36	73.56	0.22	26.23
III	98.49	1.51	69.51	0.28	30.21
IV	98.64	1.36	75.89	0.21	23.90
Storage					
I	99.95	0.05	99.99	<0.01	<0.01
II	99.95	0.05	99.98	0.01	<0.01
III	99.95	0.05	99.98	0.01	<0.01
IV	99.95	0.05	99.98	0.01	<0.01
Discharging					
I	99.83	0.17	24.23	0.28	75.49
II	99.87	0.13	18.91	0.22	80.88
III	99.85	0.15	19.60	0.24	80.16
IV	99.87	0.13	17.94	0.21	81.86

Table 7
Total energy and exergy efficiencies for the four ice storage systems (partial storage).

Case	(I) Ice slurry	(II) Internal melt	(III) External melt	(IV) Encapsulated ice
Total energy efficiency, η_{tot} [%]	98.64	98.46	98.29	98.46
Total exergy efficiency, ψ_{tot} [%]	11.36	13.90	13.62	13.61

The effect of reference environment is another interesting aspect of this study. Though the reference temperature is assumed to be the same temperature at which the condenser is working, it is nonetheless important to address the impact of varying this parameter. To determine the effect of the reference environment temperature, T_∞, on system performance, this parameter was varied from 15 °C to 50 °C in increments of 5 °C, and the resulting changes in efficiency were monitored. The energy efficiency dependence on this parameter can be seen graphically in Fig. 5, while the exergy efficiency response due to changing atmospheric temperature is shown in Fig. 6. Note that due to similar profiles, only the case of partial storage will be shown to avoid duplication.

The most important aspect of Figs. 5 and 6 is the rate at which the efficiency changes when dead state temperature is varied. In Fig. 5, the energy efficiency changes only by a few percent over the course of the temperature range. This is due to the fact that the only effect the dead state temperature has in terms of energy is from the heat leakage equation. In Eq. (23), if the dead state temperature is larger, there will be more of a gradient between the storage tank and its surroundings, and thus there will be more heat leakage, leading to lower efficiencies.

However, in the exergy analysis, the opposite is true; efficiency actually increases as dead state temperature increases. While the amount of exergy destroyed due to heat leakage will actually increase with the amount of heat leakage – see Eq. (31) – the reason for the large exergetic efficiency increase along with increased temperature is due to the increased storage exergy. When the dead state temperature is heightened, the ice storage is of a much higher quality – and so is the recovering glycol solution. Though the recovered energy remains the same, in an exergy sense the quality of the recovered fluid is much closer to the storage quality, which makes it more efficient exergetically. The reader should also note that the profiles in Cases II and IV are strikingly similar in Figs. 5 and 6, which is due to the similar assumptions in Table 3 used in this analysis. Although these two cases exhibit similar performance

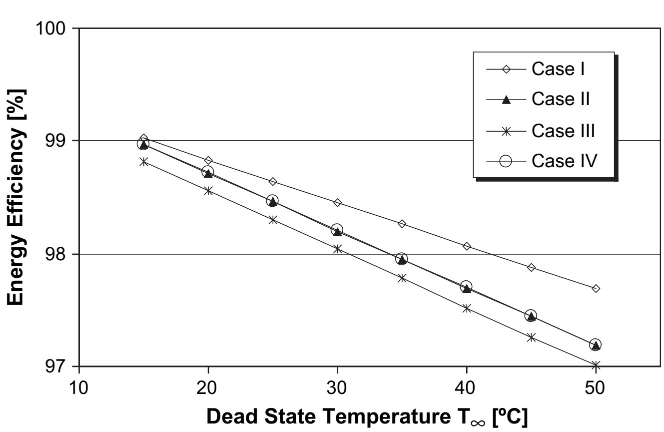

Fig. 5. Energy efficiency for the various cases according to dead state temperature [partial storage].

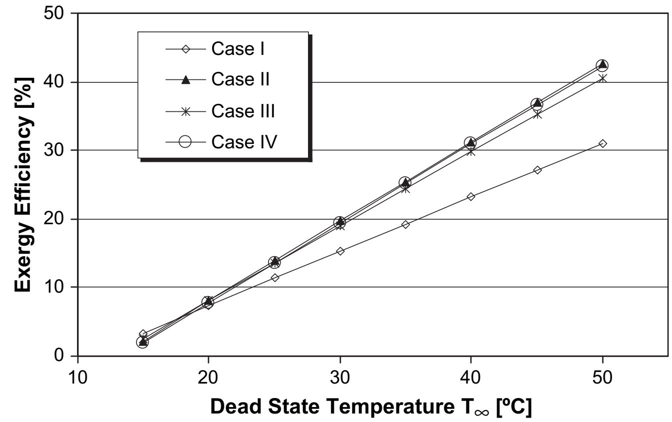

Fig. 6. Exergy efficiency for the various cases according to dead state temperature [partial storage].

trends, they vary considerably in cost, energy requirements, ease of maintenance and installation, and are used in different capacities depending on the load, space and cost constraints.

This is why exergy plays an important role in cold storage; because it treats cold as a useful commodity, and because the locations and magnitudes of losses in the system. For example, if an energetic analysis alone is undertaken in order to determine the feasibility of any of the above systems, the ice slurry system would be chosen as the most efficient. However, exergetically, it was found that the most efficient system in both the partial and full load systems was the ice-on-coil systems (internal melt), followed closely by the encapsulated ice systems, the external melt system and lastly, the ice slurry storage system.

Though the internal melt ice-on-coil system has shown to be the most efficient, the storage tank and chiller system required would be far more complicated than that for encapsulated ice storage, and for larger applications it is less economically viable. For this reason, ice-on-coil systems would be better suited for smaller applications; with the internal melt being more expensive, while the external melt system would be less efficient, but also less costly due to the fact that the cold storage can be directly extracted from the coils without need for a more complicated refrigerant/ice retrieval system. The encapsulated ice system is most likely the best suited for larger applications, since it achieves relatively high efficiency values, and is also relatively inexpensive when compared to the other methods, especially when the capsules are spherical and can be packed randomly into an unstructured storage module. The ice slurry system is most likely the least efficient for this type of application, and this would be reflected in overall operation and energy costs. For this reason, both energy and exergy analyses should be performed to get a more clear understanding of the overall performance of a cold storage system.

6. Conclusions

In this study, a comparison of four main types of ice storage techniques for space cooling purposes is conducted. The systems studied include ice slurry systems, ice-on-coil systems (both internal and external melt), and encapsulated ice systems.

A detailed analysis, coupled with a case study based on values found in the literature follows this review. The four ice storage and retrieval techniques are compared on the basis of energy and exergy efficiencies according to the charging, storage and discharging process. A vapor-compression refrigeration cycle with R134a as the working fluid provides the cooling load, according to a prescribed coefficient of performance. In addition to this, the storage and discharging temperatures, ice storage density, as well as the ambient temperature are specified in order to complete the analysis. The storage module itself is assumed to be cylindrical, and its height equal to its diameter. The analysis is performed for both a full storage and a partial storage load cycle, with all data taken from works in the literature.

For the case of full load shifting to off-peak times, all energy efficiencies were over 99% for both the charging and discharging cycles, while the most energy efficient scenario was realized with the ice slurry method of storage, with an energy efficiency of over 99%. However, this is an unrealistically high value, which can be more apparent when viewing the exergetic efficiencies. The exergy efficiencies for the charging process were found to vary between 46% and 76%, with the most efficient scenario being the encapsulated ice system. For discharging, the efficiencies were much lower, resulting from entropy generation due to heat transfer and phase change, and varied from 18% to 24%, with the ice slurry technique having the highest discharge exergy efficiency. However, in terms of total exergy efficiency, the most desirable scenario was the ice-on-coil storage system, utilizing an internal melt storage tank, with an overall efficiency 14.05%. In the partial load shifting scenario, the performance values obtained were extremely close to that found for the full load cycles. However, it was found that the partial load efficiencies were slightly lower than that of the full load efficiencies, which was attributed to a greater effect of heat leakage in the partial load scenario.

In summary, the results indicate that the energy efficiencies are misleadingly high, and only when the exergy analysis is undertaken is a more realistic view of the system performance achievable.

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