Experimental investigation of the temperatures and performance of a commercial ice-storage tank

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Abstract

This paper presents the results of an experimental installation with an internal melt-ice-on-coil tank which has a total capacity of 172 kWh. The aim of this work is to analyse the freezing process in a tank with counter-current spiral-shaped coils immersed in around 1855l water. An experimental campaign has been performed with different inlet temperatures and mass flow rates of the heat transfer fluid. This study analyses (i) the chiller performance, (ii) the ice-formation process and (iii) the energy consumption of the installation. Supply temperatures between -2.5ºC and -5.2ºC have been sufficient to charge the tank without using any nucleating agents. The lowest energy consumption has been achieved for the fastest charging tests.

Keywords: Thermal storage, experimental installation, ice formation, energy consumption

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NOMENCLATURE

Re	Reynolds number	D	Diameter of the coils (cm)
$\dot{m}$	Mass flow rate, kg h^{-1}	R	Thermal resistance, K W^{-1}
$\overline{COP}$	Mean coefficient of performance	$\dot{W}_{elec}$	Electric consumption of the chiller and primary circuit pump, kW
ΔT'	Temperature difference brine/wall as in Fig. 4, K	ΔT	Temperature difference brine/PCM as indicated in Fig. 4, K
$\dot{Q}_{LHTS}$	Thermal power as calculated from the supply and return temperatures, kW		

Subscripts

LHTS	Latent Heat Thermal Storage	SHS	Sensible Heat Storage
PCM	Phase-Change Material	HTF	Heat Transfer Fluid
HEX	Heat exchanger	fc	Forced convection
evap	Evaporator	cond	Condenser
in	Inlet	out	Outlet

supply	Supply temperature of the ice-storage tank	return	Return temperature of the ice-storage tank
n	Nucleation	t	Tube
c	Cooling	h	Heating

1. INTRODUCTION

In recent years, Thermal Energy Storage (TES) has become a key issue to widen the integration of renewable energies and compensate the mismatch between the energy generation and supply. In comparison with sensible heat storage (SHS), latent heat thermal storage (LHTS) is attracting an increasing interest among other reasons because of the high volumetric capacity and low heat transfer losses (Dincer and Rosen, 2011; Ercan Ataer, 2006; Mehling and Cabeza, 2008; Paksoy, 2007).

In the past 20 years, a significant effort has been applied in the research of LHTS systems. Today, more than 150 Phase Change Materials (PCMs) have already been studied (Zalba et al., 2003).

Nevertheless, the most commonly used PCM remains ice/water (ASHRAE, 2007). Despite the disadvantage of the low melting temperature, ice/water is the most extended PCM due to the combination of reliability, stability, low cost, high latent heat capacity of 335 kJ/kg, high specific heat, high density, and safety (ASHRAE, 2007).

Ice/water systems have been analysed in different configurations such as with encapsulation (Chen and Lee, 2002; Eames and Adref, 2002; Tan et al., 2009), ice slurries (Egolf and Kauffeld, 2005; Guilpart et al., 2005; Kauffeld et al., 2010) and internal/external melt-ice-on-coil systems (Badr, 2007; Ezan et al., 2011; Fertelli et al., 2009; Fertelli, 2011; Wang et al., 2003).

In external melt-ice-on-coils (Shi et al., 2005; Wang et al., 2003) the discharging is achieved by circulating warm water directly through the ice layer, whereas in internal melt-ice-on-coil systems, the brine always flows through the coils during the charging and discharging of the tank. It has been demonstrated that natural convection can be a key heat transfer mechanism (Ezan et al., 2011) although research on internal melt-ice-on-coils has been generally carried out on systems with one single coil (Badr, 2007; Ezan et al., 2011; Kayansayan and Ali Acar, 2006; Zhu et al., 2001). Nevertheless, many commercialized ice-storage tanks are internal melt systems (CALMAC, 2011; BALTIMORE, 2011) with a significant PCM volume and more complex coil geometries. The consequent effect on natural convection or nucleation is hard to predict and deserves a more specific attention, which is the aim of this work.

In practical terms, two key points in ice-storage installations are the energy efficiency and the electricity rates (Andrepont, 2011). In this work, the energy consumption of the installation has been measured and analysed. As in other studies (Chen and Lee, 1998; Fertelli, 2011; Wang et al., 2003) the temperatures in different points of the PCM (water) have been measured. A new aspect which has been introduced is the measurement of the heat transfer fluid temperature as it flows through the coils of the tank. These measurements have helped to better understand the effects of natural convection in the PCM and the consequent vertical stratification within the storage tank.

2. EXPERIMENTAL SET-UP

This paper presents the experimental results of an ice-storage installation (López-Navarro et al., 2011) from the Polytechnic University of Valencia (Spain). The installation (shown in Fig.1) has been designed to characterize LHTS systems within a temperature range between -10ºC and 100ºC. In order to carry out the charging and discharging tests, an external heat sink and source is required. The heat transfer fluid (HTF) in the primary circuit is a 32% mass water-glycol mixture which flows through the ice-storage tank. Mass flow rates between 1000 kg/h and 4000 kg/h of the HTF are achieved by means of the variable-speed pump 1. The latter is a CRE 5-5 pump from GRUNDFOS, with a nominal power consumption of 750W.

The LHTS system is a CALMAC ICEBANK tank model 1098C with a nominal capacity of 350 kWh. In this study, the tank has been tested at a partial load of 172 kWh. Fig. 2 shows a detailed scheme of the ice-storage tank, which consists of 34 counter current spiral-shaped tubes immersed in the phase change material (tap water). During the charging process, the ice builds concentrically around the coils. In order to test the tank at a partial load, the top 16 coils have been blocked using plumbing accessories for polyethylene pipes and thus, the HTF only circulates through the bottom 18 coils, which are the only ones which are surrounded by water.

The charging of the ice-storage tank requires inlet temperatures of the HTF below 0ºC. The necessary cold is produced by means of an 8 kW vapour-compression chiller working with R22. The 3-way valve V_{c1} can regulate the mass flow rate of the cold water/glycol mixture which flows through the plate heat exchanger HEX_c. As the capacity of the chiller is small, in the set of tests which are discussed in this paper, the entire flow rate was sent directly to the heat exchanger in order to fasten the charging process. The compressor of the chiller switches off once the temperature in the outlet of the circulation pump 2 reaches the desired set-point temperature. A 200 l storage tank has been added to avoid having a frequent on/off operation of the compressor.

The chiller is a laboratory prototype which has a semi-hermetic BITZER compressor. The superheat is controlled by means of an electronic expansion valve with proportional modulation. In the tests, the total superheat has been fixed to 7ºC.

In order to discharge the tank and melt the ice, a heat source is required. This is achieved using the water from the building condensation ring (at around 22ºC) which heats the primary fluid by means of the plate heat exchanger HEX_h.

As may be inferred from Fig. 1, several thermocouples (T_{PCM1}-T_{PCM6}) have been placed to measure the PCM temperature in different vertical and radial positions.

Additionally and as shown in Fig. 3, four thermocouples have been stuck to the external surface of the coils and they have been isolated in order to measure the HTF temperature. The measurement principle is illustrated in Fig. 4.

A theoretical calculation of the thermal resistances shows that the insulation resistance is much higher than the rest of thermal resistances. Thus, the thermocouples which are attached to the tube wall practically measure the temperature of the HTF inside the coils, as can be inferred from Eq. (1):

$$\frac{\Delta T\prime}{\Delta T} = \frac{R_{fc}+R_t}{\sum_i R_i} \approx 0.032 \qquad (1)$$

Where ΔT' is the temperature difference between the brine and the external wall of the coil and ΔT is the temperature difference between the brine and the PCM, as shown in Fig. 4.

All of the thermocouples inside the tank (T-type) have been first calibrated in an ice/water mixture with an accuracy of ±0.2 K. The inlet and outlet temperatures of the HTF have been measured with RTD thermal resistances PT100 1/10 DINB. The mass flow rate in the primary circuit is measured with a SIEMENS CORIOLIS flow meter. The accuracy of the complete measurement equipment is summarized in Table 1. The installation is fully monitored by means of a datalogger Agilent 34970A and a three 22 channels multiplexer HP 34901A with a recording interval of one minute. The instrumentation of the tank is located in one of the multiplexers, whereas the other 2 multiplexers are used to monitor temperature measurements in the heating and cooling loops.

3. RESULTS AND DISCUSSION

An experimental campaign has been carried out to analyse the ice formation process. In total, 9 tests have been performed under different operating conditions (Table 2) which are basically the mass flow rate and supply temperatures of the HTF. The supply temperatures vary between -2.5ºC and -5.2ºC whereas the tested mass flow rates range between 1574kg/h and 3856kg/h. As may be inferred from the low Reynolds numbers, all of the tests have been carried out with a laminar flow of the HTF. The mass flow rates are mean measurements during the latent heat transfer region. During the initial part of the tests, as the supply temperature cannot be held constant, the viscosity of the HTF changes and consequently there is a small variation of the mass flow rate for a same frequency of the circulation pump. The maximum variation of the mass flow rate during a test is 12.6% and takes place in the initial part of the test until the supply temperature is stabilized.

The initial temperature of the thermocouple T_{PCM5} is also given in Table 2. At the beginning of each test there is inevitably a vertical thermal stratification within the PCM so the reference temperature has been chosen as T_{PCM5} with a target value of 12ºC.

In the next paragraphs, the results are explained in detail for one typical charge test (C7). The temperature measurements in the cooling loop and in the ice-storage tank are discussed in paragraphs 3.1 and 3.2 respectively. Finally, in section 3.3, the energy consumption of the installation is analysed.

3.1 Analysis of the cold generation

Fig. 5 shows the temperature profiles which have been obtained in the cooling loop. The cold fluid heats up as it circulates through the heat exchanger HEX_c and then returns to the evaporator. In test C7, the chiller set-point is -11ºC. As inferred from Fig. 5, this temperature is only achieved at the end of the test, once the latent heat transfer is finished. The chiller has a dead-band between -9ºC and -11ºC.

In Fig. 6, the operating pressures of the chiller have been plotted on the left y-axis. The chiller consumption is given on the right y-axis. Test C7 has been carried out with a chiller pressure ratio of 7 (condensation pressure at 14 bar, evaporation temperature at 2 bar). The chiller power consumption is constant and equal to 3.13 kW as for the rest of the charging tests. At the end of the charging test (1450 min to 1600 min) the chiller is operating very close to the set-point. The PID controller which controls the expansion valve induces the pressure oscillations which are observed in Fig. 6.

3.2 Analysis of the ice formation process

Figs. 7 to 9 help to analyse the ice formation process which takes place inside the LHTS system. In Fig. 7, the temperatures of the 6 thermocouples T_{PCM1}-T_{PCM6} and of the two RTD thermal resistances have been plotted. 3 regions are clearly observed, a central region corresponding to the latent heat process, and two sensible heat transfer regions. Since the beginning of the charging process, the temperature of the water decreases progressively. The density inversion of water at 4ºC (Melinder, 2010) is often observed in the tests in agreement with published literature on ice-storage systems with a significant capacity (Fertelli, 2011).

When the PCM thermocouples reach the phase change temperature (0ºC), a meta-stable state starts where the water remains in liquid state until the nucleation and crystallisation of ice occurs. Nucleation is heterogeneous as it initially begins on the surface of the coldest coils. This helps to achieve nucleation with a smaller subcooling (or metastability degree) than with homogenous nucleation. The thermocouples T_{PCM1}-T_{PCM6} present different nucleation temperatures, for instance, in test C7 nucleation starts within a temperature range between -0.6ºC and -3ºC.

Due to the release of the latent heat, right after the nucleation the thermocouples T_{PCM1}-T_{PCM6} almost instantaneously reach the phase-change temperature (0ºC) which corresponds to the thermal equilibrium. PCM 3, which is the lowest thermocouple in the central region, does not nucleate during the entire test. The temperature reduces very slowly in comparison with the other thermocouples because it is located very far from any coil.

Fig. 8 shows a comparison between three tests which have been carried out with different supply temperatures. All three tests start with a PCM temperature of 12ºC, and the lowest inlet temperatures are achieved for test C9. In the latter test, the latent heat transfer region finishes quicker than in the other tests. From the entire series of tests C1 to C9, it can be clearly observed that the higher the heat transfer fluid flow rate and the lower the supply temperature, the faster the charging process.

Fig. 9 provides relevant results on the temperature evolution of the HTF as it circulates through the coils of the tank. The even rows of the tank (Fig. 3) have a centrifugal flow, whereas the uneven rows have a centripetal flow. With the measurement set-up shown in Fig. 4, the thermocouples in centripetal direction (rows 3 and 17) measure the temperature of the HTF just before the heat transfer with the central volume and the circulation through the common collectors.

For water temperatures above 4ºC (for instance after 50 min), the warm water has a lower density and is hereby collected in the top of the tank. As may be inferred from Fig. 9, the heat transfer fluid becomes warmer as it circulates through the upper coils (row 17).

After around 215 minutes, the centripetal return temperatures in rows 3 and 17 cross, and the heat transfer fluid becomes warmer as it flows through the bottom coils (row 3). This is due to the density properties of water (Melinder, 2010), as the warmest water is denser and is thus collected in the bottom of the tank.

Table 3 shows the nucleation temperatures which have been measured by the thermocouples T_{PCM1}-T_{PCM6}. Nucleation takes place within a very short period of time (around 5 minutes) in all of the thermocouples. In the entire series of tests nucleation generally takes place between -1ºC and -3.1ºC. For low subcooling degrees of less than 1K as for thermocouple T_{PCM6}, the nucleation rate is sensibly slower and is probably due to a larger induction time (Qin et al., 2003).

Several aspects help reduce the subcooling degree (Bedecarrats, 1993) in this tank; the significant volume of water (around 1855l) and the tap water impurities (e.g. lime, nitrates) which are rather high in Valencia and may enhance the nucleation.

Nucleation is a very random process, for instance it has been demonstrated that water inside spherical capsules does not always nucleate, even for same coolant temperatures (Chen et al., 1998). Thus, it is interesting to compare charge tests which have been carried out under similar operating conditions, which is the case of tests C1/C4 and C3/C7. The mean nucleation temperature in those tests is similar, but the results clearly show that nucleation is rather random in the different measurement points. This aspect would require some more specific tests as done in literature on encapsulated systems demonstrating that the lower the coolant temperature the greater the nucleation probability (Chen et al., 1998; Chen and Lee, 1998). A more detailed experimental campaign will be carried out in future work to correlate the nucleation temperatures with the mass flow rate and supply temperature of the HTF.

3.3 Power consumption of the components

During the ice formation process, the main power consumption derives from the compressor of the chiller and the circulation pump of the HTF. The specific power consumption of the system has been calculated as in Eq. (2).

$$\frac{1}{\overline{COP}} = \frac{\int_{t0}^{t\infty} \dot{W}_{elec}}{\int_{t0}^{t\infty} \dot{Q}_{LHTS}} \quad (2)$$

Fig. 10 shows the specific power consumption in the latent heat storage region. In practical installations there is only one hydraulic loop and hence, the energy consumption which has been measured is the consumption of the chiller and of the circulation pump 1 in the primary circuit.

The compressor consumes from 87.5% to 96.7% of the total power consumption depending on the test. Low mass flow rates help to decrease the inlet temperature in ice-storage tank, but reduce the forced convection inside the coils. The highest specific energy consumption (tests C8 & C9) yields a mean COP of around 2.1 and has been achieved with a relatively low mass flow rate in the primary circuit and a low set-point temperature of the chiller. For low mass flow rates, the temperature difference between the inlet and outlet of the heat exchanger HEX_c is higher. Thus, given the power limitations of the chiller, low mass flow rates help to achieve lower temperatures and consequently fasten the charge process. As the main consumption of the installation derives from the compressor, the faster the charge tests, the lower the overall energy consumption. These results are specific to this installation as it has not been possible to vary independently the mass flow rate and supply temperature.

4. CONCLUSIONS

In this work, the charging of an ice-storage tank has been analysed. Different tests have been performed to investigate the effect of the mass flow rate and supply temperature of the HTF. The following conclusions have been obtained:

- During the charging tests, the HTF has to be supplied at temperatures below the nucleation temperature. A maximum subcooling degree of 3K has been observed. In future work, a more specific experimental campaign will be performed to analyse the effect of the operating conditions on the crystallization rate and nucleation temperatures.
- The lowest energy consumption has been reached for low mass flow rates and cold supply temperatures. The chiller consumption is much higher than the circulation pump. Thus, fast charging is desirable as it decreases the total energy consumption. Low mass flow rates help decrease the supply temperature, but below a certain value (in this case 1915 kg/h) the low forced convection of the HTF can lead to longer charging tests.
- Natural convection is the predominant heat transfer mechanism inside the tank. The effect of the density inversion at 4ºC can be observed not only in the temperatures of the water, but also on the HTF. Before the nucleation, the warmest and denser water is collected in the bottom of the tank. Thus, the HTF becomes warmer as it flows through the spiral-shaped coils in the bottom.

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