EUROSUN 2010 - LATENT HEAT STORAGE INTEGRATION TO A SOLAR COOLING SYSTEM CONDENSER

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Abstract

Conventional sorption based solar cooling systems require a wet cooling tower (to dissipate the heat at condenser) which slows down their commercial development for small and mid range systems (<30 *kW*) because of a restrictive maintenance against Legionellosis risks (mainly in France and Spain). To substitute wet cooling technology by dry air cooling one generates furthers electricity consumptions (30 %) and make the cooling process less efficient during hot days with high ambient temperatures [1].

The purpose of the SOLACLIM research project is to study the addition to a drycooler (dry air cooling by fans), of a latent heat based storage tank filled with a Phase Change Material (PCM) intensified in thermal conductivity by impregnation in a Compressed Expanded Natural Graphite (CENG) matrix [2]. During the day, the heat, rejected by the sorption chiller and not dissipated by the drycooler, is absorbed by the composite which is in charging process. The sorption chiller can be still cooled down despite of high ambient temperatures. During the night, the stored heat is released taking advantage of both a cooler ambient temperature and a cheaper night-time electricity. This combination has been studied on a pre-commercial facility using an inorganic hydrated salt and first results have been exposed and discussed [3] [4].

1. Introduction

In the industrialized countries, energy consumption in commercial and residential buildings represents between 40 % and 50 % of the total primary energy consumption [5]. Heating, ventilation, and air conditioning (HVAC) represents half of the energy consumed in buildings [6]. The development of heating and solar cooling systems in buildings helps at the reduction of this consumption. The wet cooling tower, because of Legionellosis risks, unfavours the development of these systems for small power units.

In the case of the drycooler based solar cooling system (without no additional cooling component), an increase of ambient temperature from 25 to 32 °C leads to a thermal Coefficient Of Performance (COP) reduction of about 27 %. This reduction in efficiency is very critical because it occurs during high ambient temperatures when the building needs more energy to be cooled [7]. Therefore, it is necessary to study alternatives solutions to condensate properly.

The addition of a volume thermal storage to a drycooler could assist the drycooler during high ambient temperature. A tank filled by a phase change has been designed for PCM impregnated in a CENG matrix, built at Coldway Company, implemented at the condenser of the solar cooling at ProMES laboratory and monitored. The aim was to compare this system, which ran in 2009, to the system using a spray system associated to a drycooler, which ran in 2008 and presented interesting results.

2. Solar cooling facility

In July 2008, a solar cooling installation using only a drycooler (and a spray system for hot days) has been installed at ProMES laboratory in Perpignan (France). An adsorption chiller (ACS 08) from SorTech AG Company localized in Halle (Germany) produces water supply/return temperature 15/18 *°C*. This chiller has a nominal capacity of 7.5 *kW* and an actual cooling range of 5 to 10 *kW*. It is composed by two distinct tanks of silica gel working in opposition of phase to assure a quasi-uninterrupted cold production [7].

3. PCM

The paraffin RT27 from Rubitherm Company has been chosen for its commercial availability, low cost, repeatability and its easy impregnation in a CENG matrix [2].

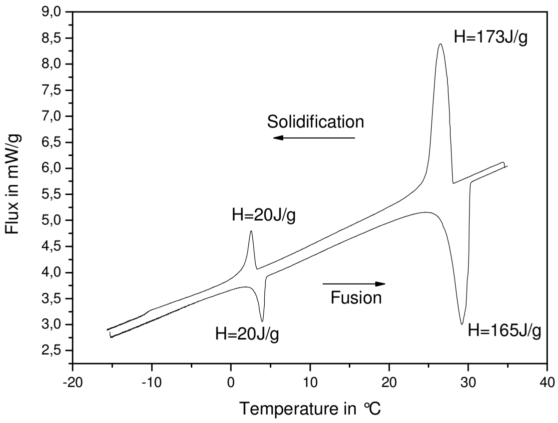


Fig. 1: RT27 thermogram

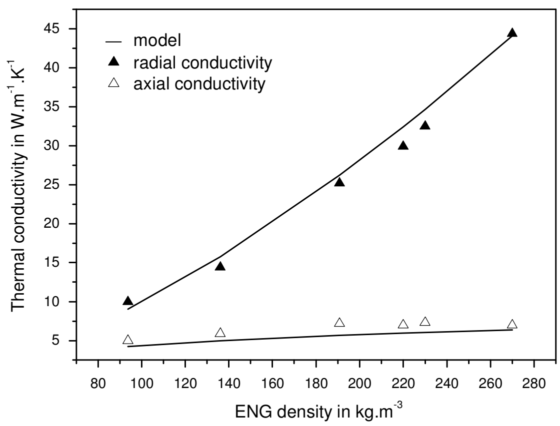


Fig. 2: CENG/RT27 thermal conductivity

As illustrated in Fig. 1, the RT27 presents sharp fusion and solidification peaks and a latent heat in the range of 170 *J/g*. The fusion temperature is 28 *°C* and the solidification is 27 *°C*, corresponding to only 1 *°C* of subcooling. Thanks to the graphite matrix and depending upon the graphite content, the thermal conductivity of the paraffin (0.24 $W.m^{-1}.K^{-1}$) can be increased significantly (Fig. 2).

4. Storage tank design

The design of the storage component was based on an optimization of the effect of the different involved thermal resistances using the Biot number. This tool has been used in order to determine the shape of the thermal heat exchanger (the available interfacial heat transfer surface area) and the quantity of ENG (leading to the available thermal conductivity of the storage composite) [7]. Two shapes of tubular heat exchanger have been considered, one for which the composite material is outside the tube (external configuration EC) and the other for which the composite material is inside the tube (internal configuration IC).

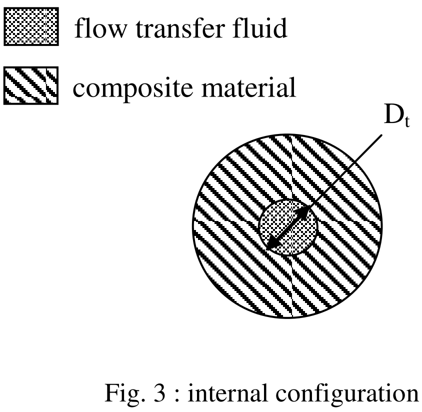


Fig. 3 : internal configuration EC

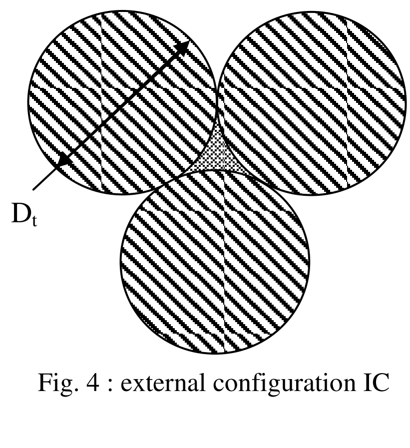


Fig. 4 : external configuration IC

In the present paper, only the IC configuration is presented, the EC configuration has been mathematically developed and already reported in a previous paper [7]. One advantage of the IC configuration is that it needs less envelopes on the Heat Transfer Fluid (HTF) side which can be contained within a conventional tank. An other major advantage is the higher heat transfer specific surface area available between the HTF and the PCM and the reduction in phase change front surface under storage or discharge while the overall heat transfer resistance increases. By symmetry, the involved thermal phenomena can be considered the same in all pseudo triangles, on which the study will be done and then extrapolated further.

The energy balance on one tube is written as:

$$\frac{\dot{m}}{N_{PT}} Cp_{tf} \frac{dT(z)}{dz} = -hP_{PT}(T(z) - T_m) \tag{1}$$

After integration between z=0 and z=L, equation (1) becomes:

$$\boxed{hP_{PT} N_{PT} = \frac{-Cp_{tf} Q_{tf} \rho_{tf}}{L} \ln\left(\frac{T_o - T_m}{T_i - T_m}\right)} \tag{2}$$

with: $P_T = \frac{1}{2} P_t$ and: $N_T \approx 1.3 N_t$

Once we define this coefficient hPN, we can calculate the number of tubes:

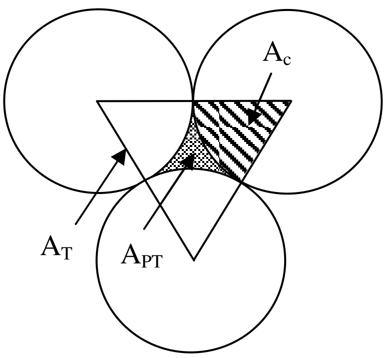
$$h = \frac{Nu \times \lambda_{tf}}{D_{HPT}} \tag{3}$$

$$\text{with: } Nu = 0.023 \mathrm{Re}^{0.8} \mathrm{Pr}^{1/3} \text{ and: } \mathrm{Pr} = \mu_{tf} \frac{Cp_{tf}}{\lambda_{tf}} \tag{4}$$

hence:

$$\boxed{N_t = \left(\frac{h \cdot P_t \cdot N_t \times D_{HPT}}{\lambda_{tf} \times \left(0.023 \times \frac{\rho_{tf} \times Q_{tf} \times 4}{\pi \times D_t \times \mu_{tf}} \right)^{0.8} \times \mathrm{Pr}^{1/3} \times P_{PT}} \right)^{\left(\frac{1}{0.2}\right)}} \tag{5}$$

The calculation of the hydraulic diameter:



$$D_{HT} = \frac{4A_{PT}}{P_{PT}} = \frac{4 \times (A_T - 3A_c)}{3\frac{2\pi r}{6}} = 0.103 Dt \tag{6}$$

Fig. 5 : Hydraulic diameter

In order to obtain a Biot number near to 1, corresponding to a same thermal resistance of conduction and of convection, CENG quantity is formulated as [8]:

$$\rho_{CENG} = 45\left(\frac{hV_c}{3N_t P_t L}\right)^{\frac{2}{3}} \tag{7}$$

The tube diameter can be chosen considering the plot (7) of needed number of tube against tube diameter for a Biot number equal to 1.

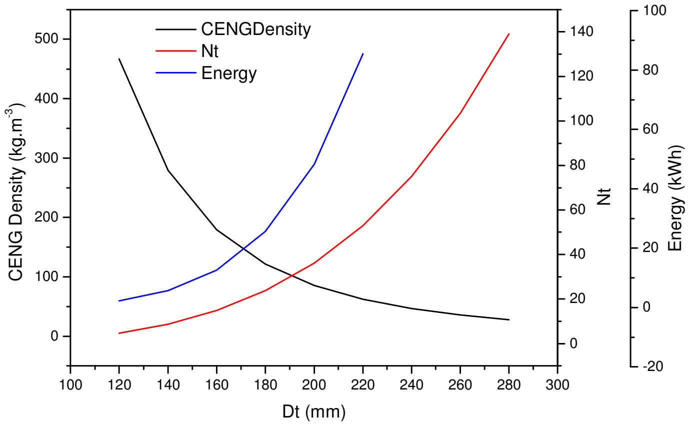


Fig. 6: Configuration 2

The available volume for the composite is directly linked to the tube diameter and the number of tubes. Therefore, to store 26 *kWh*, the design leads to a tube diameter of 0.18 *m* and a number of tubes of 24. On the storage material side, to obtain the desired effective thermal conductivity of 9.6 $W.m^{-1}.K^{-1}$ the composite has to be composed with a CENG density of 100 $kg.m^{-3}$.

For PVC tube, 0.18 m of diameter doesn’t exist. Close to this dimension, it exist tube of 0.16m or 0.2m. Diameter of 0.2 m has been chosen because it needs a lower CENG density (Fig. 6).

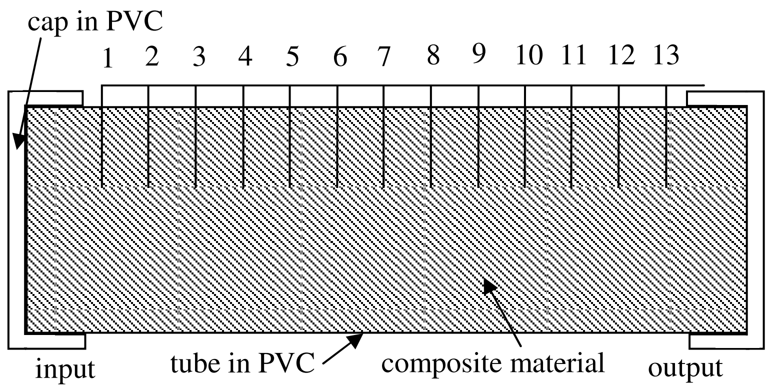
5. Implementation and results

The tubes containing the storage material were placed in a stainless steel tank. A study using C.A.D. software (Computer Aided Design) was conducted to determine the diameter of the tank and the filling up process. Indeed, the software allowed us, integrating gravity, to fix the layout of tubes. The standard diameter of the tank closest to our use is 1.219 *m*.

Fig. 7: Filling up of the storage tank

As illustrated in Fig. 7, a tank of 1.219 m allows theoretically the placement of 27 tubes. In the reality, it has been possible to place 25 tubes. The disposition of the tubes follows the C.A.D. model. The tank was placed under the drycooler and insulated against external conditions, sun and rain. The 25 PVC tubes of 1.3 *m* in length, 0.2m in diameter, have been filled with 643 *kg* of Paraffin RT27 impregnated in 98 *kg* of CENG matrix to build a storage tank which can store 26.8 *kWh* by phase change.

The storage tank has been monitored with temperature sensors and inserted in the condenser circuit after the drycooler.



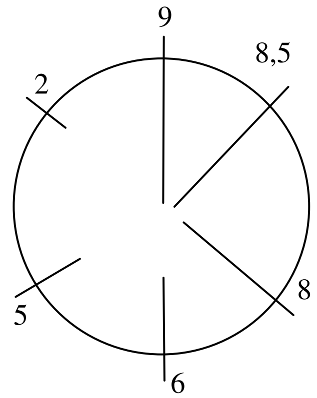


Fig. 8: axial implementation of thermocouples in axial

Fig. 9: Thermocouples in axial

Two tubes have been monitored; one with two PT1000 probes linked to the automat of the solar cooling installation for the regulation and the other one with 13 thermocouples in axial and 5 in radial positions linked to an independent acquisition system. The 13 axial thermocouples were placed at equal distances from each other and at 8.5 *cm* deep. Moreover, 5 thermocouples were placed at different depths: 2 - 5 - 6 - 8 and 9 *cm* in order to study the evolution of the melt front in time.

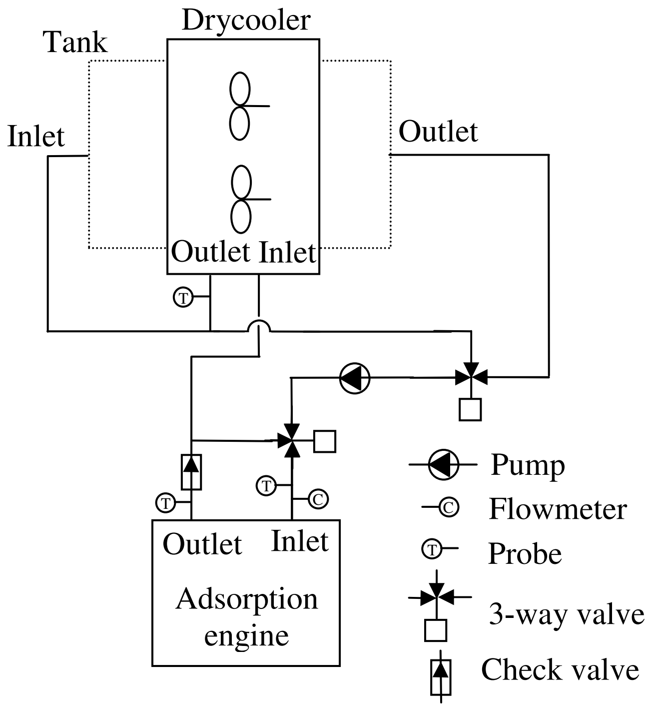


Fig. 10: Linking of the storage tank

The storage tank assists the drycooler when its efficiency is failing to sufficiently cool down the transfer fluid due to a high ambient temperature. The tank was connected to the condenser downstream of the drycooler. The flow is directed by a set of two 3-way valves to pass into the tank or not and to shunt the engine for the solidification during the night (fig. 10).

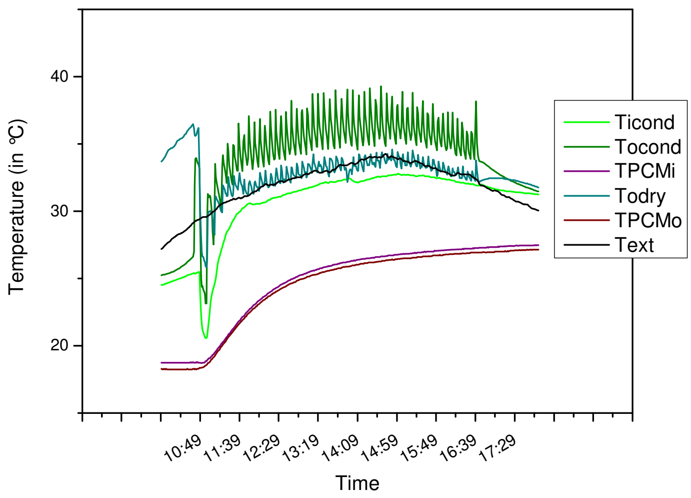


Fig. 11: Temperature at the condenser

The first results were obtained using the PCM continuously. Figure 11 presents the results of a very hot day, on the 24th September 2009, where ambient temperature reached 34 *°C*. The drycooler arrived to cool down the transfer fluid to 35 *°C*. The passage in the tank allows a smoothing of the temperature below drycooler output temperature.

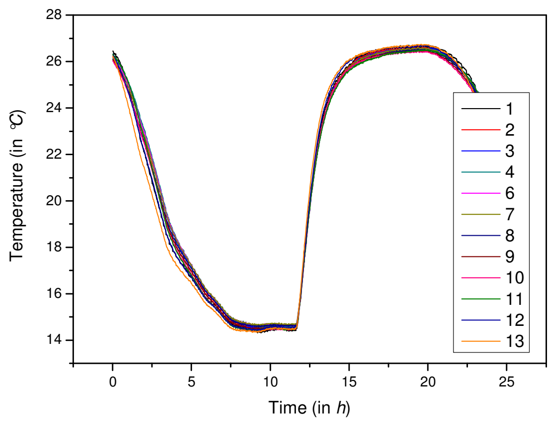


Fig. 12: Temperature in axial

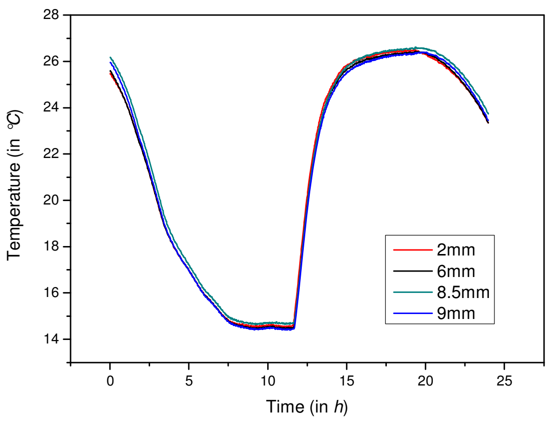


Fig. 13: Temperature in radial

According to the above illustrations Fig. 12-13 the temperatures within the composite material is very uniform both axially and radially (less than 0.5 *°C* difference) during charge and discharge steps and this even when the exchanged powers were high. This point showed that the composite material effective thermal conductivity was high enough.

The solar cooling installation system with the storage volume associated to the drycooler worked every day from the 16th September to the 16th October 2009. During this whole month, 250 *kWh* cooling were produced and 170 *kWh* have been stored by phase change. The addition has led to an average thermal COP of 0.34 comparable to the one measured in summer 2008 without any use of the spray to cool down the engine correctly. Additional test during several days without drycooler operation were made using the storage capacity and the maximum load was reached around 26 *kWh*.

The flow recorded with or without passing through the storage tank is nearly identical, therefore, the corresponding induced pressure losses are not significant.

For the days during which the temperature exceeded 30 *°C*, the electrical COP was 4.5, which is identical to the one obtained in 2008. The discharge of the storage capacity was complete.

For the days during which the temperature did not exceed 30 *°C* and when the nights were cooler (at the beginning and at the end of the season), a large thermal storage by sensible heat was possible (the tank contain 624 *kg* of water and 643 *kg* of paraffin, corresponding to 1 *kWh/°C*). For these days, an electrical COP of 5.4 has been obtained, representing an increase of 20 %.

6. Conclusion

The use of a wet cooling tower at condensing part of solar cooling systems is not applicable to small power units. As alternative approach, the combination of a drycooler and a latent heat storage tank containing an organic PCM intensified in thermal conductivity by impregnation in a CENG matrix has been theoretically and experimentally studied.

During the summer 2008, a pre-commercial adsorption solar cooling system of 7.5 *kW* in cooling power using only a drycooler including a spray system has been implemented and instrumented for the SOLACLIM project.

In parallel, a numerical tool has been develop to design properly the storage sub-system by an optimization of the geometry according to thermal exchanges, thermal resistances, thermal interfaces, effective thermal conductivity of the composite.

According to the optimization of the design, a storage tank has been built and added to the drycooler and tested from 26th September to 26th October, days during which high ambient temperatures have been observed. The obtained results showed that the chiller has worked with the same performance levels than those observe using the. The replacement of the spray system by the thermal storage component avoids the system applicability of the expensive regulation frame leading to actions against the development of legionella.

In order to study the performance of the installation during hotter days (July and August) and the stability of the PCM, the tank storage is under current operation during the whole summer 2010.

The surface area of the storage component is roughly equivalent to the size of the drycooler. In view of possible commercialization of the system, it would be accurate to bury the storage volume under the drycooler for space reasons. The tank then would be supported by the ground soil and will not need insulation and could be a rectangular shaped parallelepiped.

Acknowledgements

The authors acknowledge the Ademe agency, the French minister of the economy, industry and employment and Languedoc-Roussillon region for their financial support.

Nomenclature

$\dot{m}$	mass flow rate, $g.s^{-1}$
L	length, m
P	perimeter, m
A	area, m^2
T	temperature, K
Cp	sensible heat, J.g-1.K^{-1}
h	convective coefficient, $W.m^{-2}.K^{-1}$
N	number, dimensionless
Q	volume flow rate, $m^3.s^{-1}$
D	diameter, m
H	latent heat, $J.g^{-1}$
V	volume, m^3
Lc	characteristic length, m
U	velocity, $m.s^{-1}$
Nu	Nusselt number, dimensionless
Pr	Prandtl number, dimensionless
Bi	Biot number, dimensionless
Re	Reynolds number, dimensionless

Greek symbols

λ	thermal conductivity, $W.m^{-1}.K^{-1}$
μ	dynamic viscosity, Pa.s
ρ	density, $kg.m^{-3}$

Subscripts

m	melting
i	input
o	output
tf	transfer fluid
f	further
p	pinch
a	ambient
e	equivalent
r	recooling
c	composite
t	tube
T	Triangle
p	pseudo
cond	condenser
ext	external
dry	drycooler

References

[1] H.-M. Hellman, C. Schweigler, F. Ziegler (1998). A simple method for modelling the operating characteristics of absorption chillers. Proceedings of Seminar Eurotherm, Nancy, 6th and 7th July 1998.

[2] X. Py, R. Olives, and S. Mauran (2000). Paraffin/porous graphite-matrix composite as a high and constant power thermal storage materials. Int J Heat Mass Transf, 2727-2737.

[3] S. Hiebler, H. Mehling, M. Helm, and C. Schweigler (2009). Latent heat storage with melting temperature 29 *°C* supporting a solar heating and cooling system. Proceedings of Effstock, Stockholm, June 14-17 2009.

[4] S. Keil, S. Hiebler, H. Köbel, M. Helm, H. Mehling, H. and C. Schweigler (2007). Solar heating and cooling system with low temperature latent heat storage. Proceedings of International Congress of Refrigeration, Beijing.

[5] L. Pérez-Lombard, J. Ortiz, C. Pout (2008). A review on buildings energy consumption information. Energy and Buildings 40. 394-398.

[6] European Union Energy & Transport in Figures (2004). Part 2: Energy. European Commission, Directorate General for Energy and Transport, Brussels. 138p.

[7] B. Rodriguez, X. Py, R. Olivès, D. Mugnier (2009), Phase Change Material for optimization of solar cooling. Proceedings of Effstock, Stockholm, June 14-17 2009.

[8] R. Olives, S. Mauran (2001). A highly conductive porous medium for solid–gas reactions: effect of the dispersed phase on the thermal tortuosity. Transport in Porous Media, Volume 43, Number 2, May 2001, 377-394.