OPTIMIZED DESIGN OF HOT WATER STORAGE IN SOLAR THERMAL COOLING FACILITIES

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Abstract

Solar thermal installations can provide a significant contribution to the energy needs of cooling demand of single family buildings. Unfortunately oversizing facility elements is not uncommon. Most of the design flaws concern collector field or auxiliary elements, such as backup boilers or electric resistances inside storage devices. This leads to lower than expected facility *COP* and *SCOP*, but also higher cost. Customer dissatisfaction is the result.

This paper presents a numerical model of the multiple purpose solar thermal facility installed at *Universidad Carlos III de Madrid* (UC3M) using the TRNSYS® tool. The solar cooling facility (http://termica.uc3m.es/solar.htm) is completely monitored for its performance characterization in the production of AC, DHW and heating. Operational data for various summer seasons have been recorded, simultaneously with 7 meteorological variables. The experimental facility includes a single effect BrLi absorption chiller working at part load under summer season.

TRNSYS is a completely feasible platform for simulating solar facilities and is commonly used by researchers and planners, for its simplicity and ease manipulation. This simulation tool contains general solar cooling elements found in most experimental facilities and has been kept as simple as possible. The model developed aims at analyzing facility elements in order to resize collector field and storage volumes. Furthermore it allows studying different configurations of the facility and the control schemes. These configurations include different hot water storage capacities within the facility allowing comparing with the facility without any kind of storage excepting its own thermal inertia. The simulation has been validated with instantaneous and seasonal experimental data for different summer seasons including 2003, 2004 and 2005.

Simulation results show that there is a hot storage tank capacity that optimizes the facility in terms of *COP*, *SCOP* and total cold produced. Even with no storage at all, the facility still improves its behavior from current operating conditions. Simulation and experimental results are compared and an optimum configuration of the facility is proposed.

1 Introduction

Solar cooling with absorption technology offers a less polluting and less expensive alternative for cooling compared to vapor-compression chillers under some conditions. Those are driven by electric energy, thus making that choice polluting and also with higher cost with nowadays fossil fuels prices, (Rodriguez et al. 2007). Nevertheless the initial investment of solar cooling is higher. Its availability in continental warm climate zones makes solar cooling even more interesting because it offers an opportunity to utilize the excess heat produced in solar thermal facilities (e.g. domestic hot water (DHW) and space heating) during summer season, when cooling demand and radiation reach their highest values.

Optimizing the design of solar thermal facilities for acclimatizing plays an important role in reducing elements sizes and in increasing its benefits (reduction of CO_2, ozone layer depletion and energy and cost savings). One of the elements that seem to need more optimization in this kind of facilities and, sometimes not given the necessary attention, is the thermal storage tank volume. Recent works by Zambrano et al. (2007), Asdrubali et al. (2006), Salgado et al. (2006), Sumathy et al. (2002) and Izquierdo et al. (2005), among others, presented experimental results of working facilities using hot storage tank working either in well mixed regime (Rodriguez et al. 2005, Venegas et al. 2005) and in stratified regimes (Syed et al. 2003). In real applications its design is frequently based on limited empiric criteria. Because of the low implementation of solar cooling facilities, those criteria are less than accurate and unspecific.

In the present work an experimental validation of a TRNSYS simulation is accomplished for the *Universidad Carlos III de Madrid* (UC3M) facility under cooling operating mode for summer season 2005. It consists on a multipurpose solar thermal facility completely monitored for its performances characterization in the production of AC, DHW and space heating. Operational data for various summer seasons has been simultaneously recorded along with 7 meteorological variables. TRNSYS is a convenient platform for simulating solar facilities and is widespread, because of its simplicity and ease of use. The aim of the simulation is to optimize the UC3M current facility. Two different schemes of the solar cooling plant are proposed in its simplest way, remembering that this work is focused on domestic cooling. These working schemes are no storage and hot storage tank (HST).

Once the model has been validated with experimental results from summer season 2005, the different configurations of thermal storage tank have been evaluated for different collector surfaces and storage volumes. To complement this, the simulation has been benchmarked with past measuring campaigns on the facility for past summer seasons, including 2003 and 2004. Performance curves under current operating conditions have been traced jointly with simulation results, leading to optimum hot storage volumes for a single family house acclimatizing application in Madrid. The results are discusses in section 4.

2 Experimental set-up

The experimental solar facility exposes $50m^2$ of flat-plate collectors to solar radiation. The field is connected to a HST of 2,000 liters capacity by means of a plate heat exchanger. This way the heat produced at the collector field is stored and made available to the hot water driven absorption chiller generator when its temperature is adequate. The product *UA* for the heat exchanger has been determined using experimental data gathered from the facility. This value has been calculated making an energy balance at the hot side of the heat exchanger. The average value for the season 2005 corresponds to 2400 W/K.

$$\dot{m}_1 C_{p,1}(T_{C,o} - T_{C,i}) = (UA\Delta T_{lm})_{HX} \quad (1)$$

The absorption chiller installed in the facility is a BrLi-H_2O Yazaki WFC10 with a nominal cooling capacity of 35 kW. At the time of the experimental facility construction, this machine was the lowest capacity model available in the market. A wet cooling tower connects to the absorption chiller to dissipate heat from the absorber and condenser.

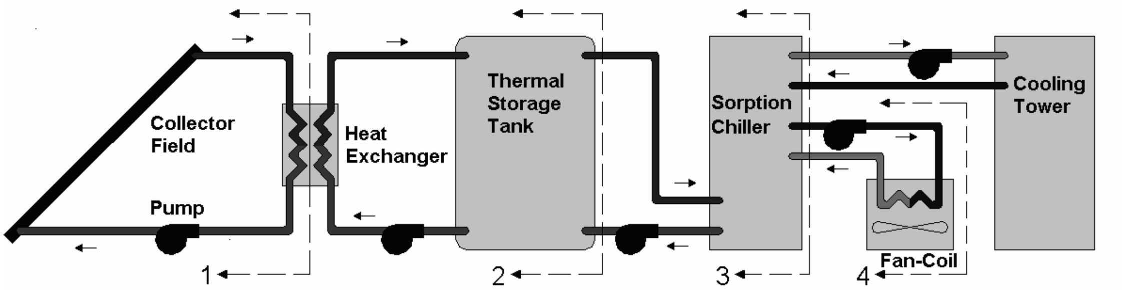

Figure 1: Current configuration of the solar facility. Numbers refers to main circuits of the facility.

The cold water produced at the chiller evaporator is sent to a fan-coil, producing the cooling load effect. A more detailed description of the facility configuration can be found in Rodriguez et al. (2005) and Salgado et al. (2006). Figure 1 presents the current configuration of the solar cooling facility.

3 TRNSYS simulation

No simulation model of the components used to describe the facility has being modified from its original definition; only the absorption chiller model needed a user defined data file to reproduce its particular behavior. To fulfill the data file some characteristic working temperatures and part load behavior information of the absorption chiller operating at current conditions are required. They have been constructed calculating averaged values from experimental data of 2005 summer season. The data gathered for 2005 correspond to the months of July and August. As expected from experimental measurements, some few days are missing because of diverse incidents during operation. These days were not taken into account. Table 1 resumes the current operating conditions of the facility elements. These values were used as inputs for the model. Meteonorm® weather file for Madrid – Barajas was used. To validate the current simulation a representative working day has been selected. This day correspond to July 12, 2005. Figures 2 through 4 depicts instantaneous working temperatures, instantaneous power provided to the generator, instantaneous cooling power and daily *COP* values for both experimental and simulated results for the representative day.

Figure 2 shows curves for collector inlet and outlet temperatures, generator inlet and outlet temperatures, ambient temperature, cooling stream temperature and chilled water outlet temperature. Slight discrepancies between simulated and experimental values can be appreciated. This can be explained in terms of differences between real and averaged weather conditions, as the weather generator uses averaged values from past decades in its calculating procedure.

Table 1: Current operating values for UC3M solar facility.

Primary circuit (collectors) fluid	33% propylene glycol mixture
Fluid of the rest of the facility	water
Primary circuit (collectors) mass flow rate	0.54 kg/s
Secondary circuit (storage) mass flow rate	0.45 kg/s
Tertiary circuit (chiller) mass flow rate	0.45 kg/s
Quaternary circuit (load) mass flow rate	0.30 kg/s
Cooling tower mass flow rate	1.10 kg/s
Heat exchanger UA	2400 W/K
Hot storage tank capacity	2000 liters
Absorption chiller capacity	35 kW
Evaporator set temperature	5 ºC
Fan-coil air flow rate	650 kg/h
Collector area	50 m^2
Collector slope	40º
Normalization curve parameters	0.85; 4.07 W/m^2K; 0.007 W/m^2K^2

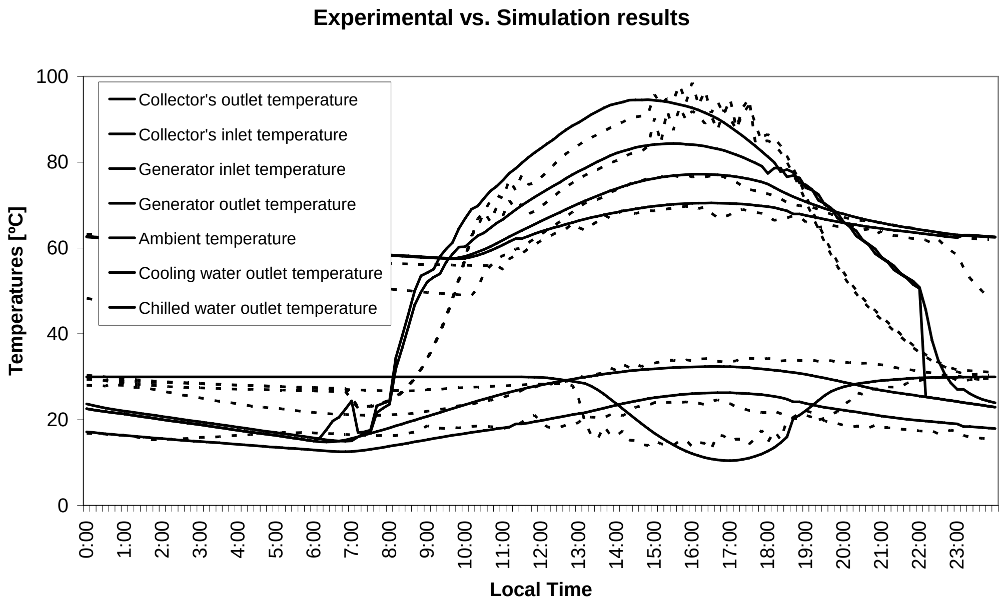

Figure 2: Characteristic temperatures of the solar cooling facility corresponding to July 12, 2005. Continuous lines correspond to simulated results and dashed lines correspond to experimental values. The figure legend is defined from top to bottom of the graph, at 16:40.

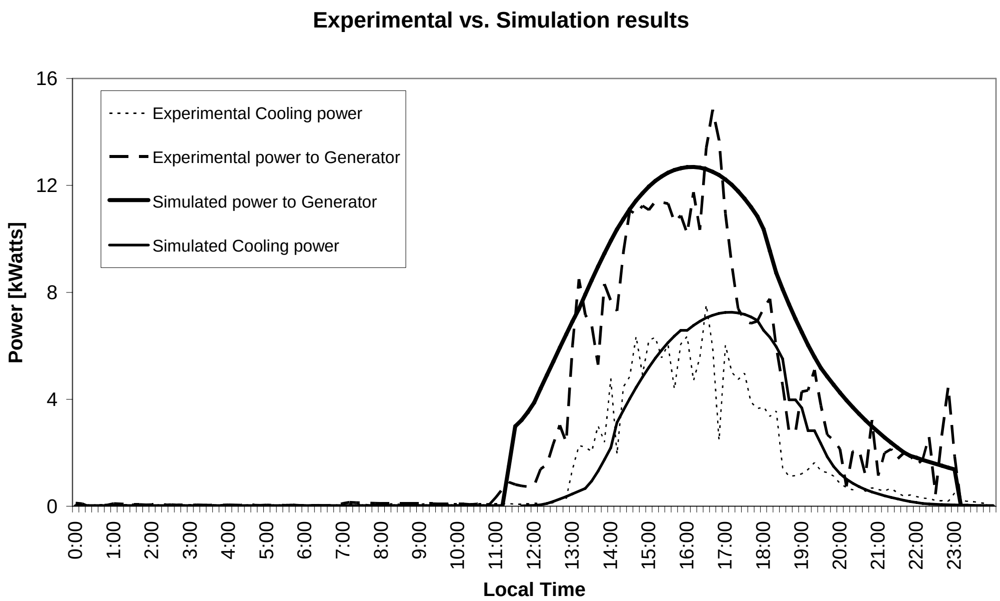

Figure 3: Instantaneous power delivered to the generator and instantaneous cooling power produced by the facility corresponding to July 12, 2005.

The instantaneous power delivered to the generator $\dot{Q}_g$ and the instantaneous cooling power $\dot{Q}_e$ produced by the facility are shown in Figure 3. These curves have been constructed defining:

$$\dot{Q}_g = \dot{m}_3 \, c_{p,3} (T_{G,i} - T_{G,o}) \tag{2}$$

$$\dot{Q}_e = \dot{m}_4 \, c_{p,4} (T_{F,o} - T_{F,i}) \tag{3}$$

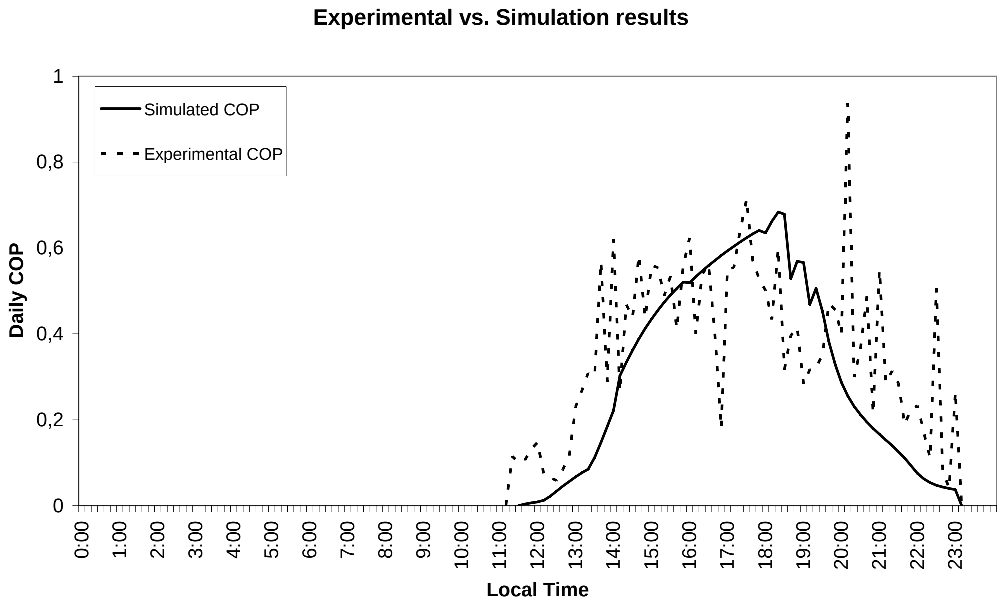

Figure 4: Daily simulated and experimental COP corresponding to July 12, 2005.

Integrating these curves the total amount of energy delivered and produced over de cooling period is obtained. The daily *COP* has been calculated as the ratio of the total cooling energy produced and the total energy delivered to the absorption chiller for the day.

$$COP = \frac{Q_e}{Q_g} \quad (4)$$

It is important to point out that this *COP* value does not coincide with the daily averaged *COP*. The difference comes from de fact that days with lower *COP* have the same weight than with a higher one. The parameter that informs about the performance of the whole facility for cooling applications is the Solar *COP*. The *SCOP* has been calculated as the ratio of the total energy produced to the total incident radiation energy Q_r.

$$SCOP = \frac{Q_e}{Q_r} \quad (5)$$

Table 2 resumes the results of the simulated against experimental facility. It shows that for this representative day the simulation predicts reasonably well the yield of the facility.

The next step is to simulate the whole summer season behavior of the facility for the current operating configuration. Additionally, different collector field surfaces have been simulated for different HST volumes. The HST has two possible extreme working modes, well mixed and stratified. Because of the mass flow rates, working temperatures and the tank aspect ratio *D/L*, the stratification in the thermal storage tank is almost completely lost. Fortunately, well mixed configuration of the HST has proven to be a better solution compared to the stratification regime for cold production purposes, as established by Salgado et al. (2006) for current operating conditions at the UC3M solar plant. Furthermore Li and Sumathy (2002) reported that for creating the stratification phenomena a value below 0.014 kg/s per square meter of collector area is necessary,

Table 2: Validation summary for July 12, 2005.

	Experimental	**Simulated**
Total energy provided to the generator	66.78 kWh	84.41 kWh
Total cooling energy produced by the facility	29.04 kWh	34.03 kWh
Total incident radiation energy	386.76 kWh	369.77 kWh
Daily COP	0.43	0.40
Daily SCOP	0.08	0.09

working with a similar storage volume. Nevertheless, a stratified storage tank simulation for 50m^2 of collector surface has been conducted for information purposes.

HST simulation results versus no storage simulation results are presented in Figure 5, as a function of collector surface. In order to make a correct interpretation of these results, it is necessary to detail how the hot tank is used in the UC3M solar plant: when the HST reaches 60 ºC the tertiary loop pump begins feeding the absorption chiller. This is accomplished by a control action, triggered by the average temperature in the tank. The figure shows that the *COP* value tends to increase for lower values of thermal storage tank volumes and for smaller collector surfaces, less than about 50 m^2. An interesting behavior is that volume values between 100 to 500 liters give better results in comparison to the facility without storage, for collector surfaces less than 65 m^2. In a facility working without thermal storage, high temperatures are reached and delivered earlier to the generator than in a facility with thermal storage, making the cooling process begin earlier. But late in the afternoon, where the cooling demand remains significant, there is no sufficient energy to satisfy the thermal load. Normally, absorption chillers have a set temperature of 5 ºC to prevent chilled water freezing and no matter how much energy is supplied to the generator, the cooling energy remains constant, as a consequence making the facility *COP* to descend during the time interval where the set temperature is reached. Even thermal storage as small as 100 liters helps to reduce these time intervals where set temperature is reached, acting as a temperature buffer, implying an increase in the *COP* value. On the same figure experimental values for past seasons are depicted for benchmarking purposes. Dashed line refers to the simulation with stratified storage tank.

4 Analysis of results

The no storage configuration (NSC) offers the opportunity to begin earlier the cold production process and thus attending the cooling demand earlier. Another positive behavior is that the absorption chiller arrives at its working generator temperature earlier and in addition to that higher cooling power peaks are achieved. A drawback of this configuration is that late in the afternoon,

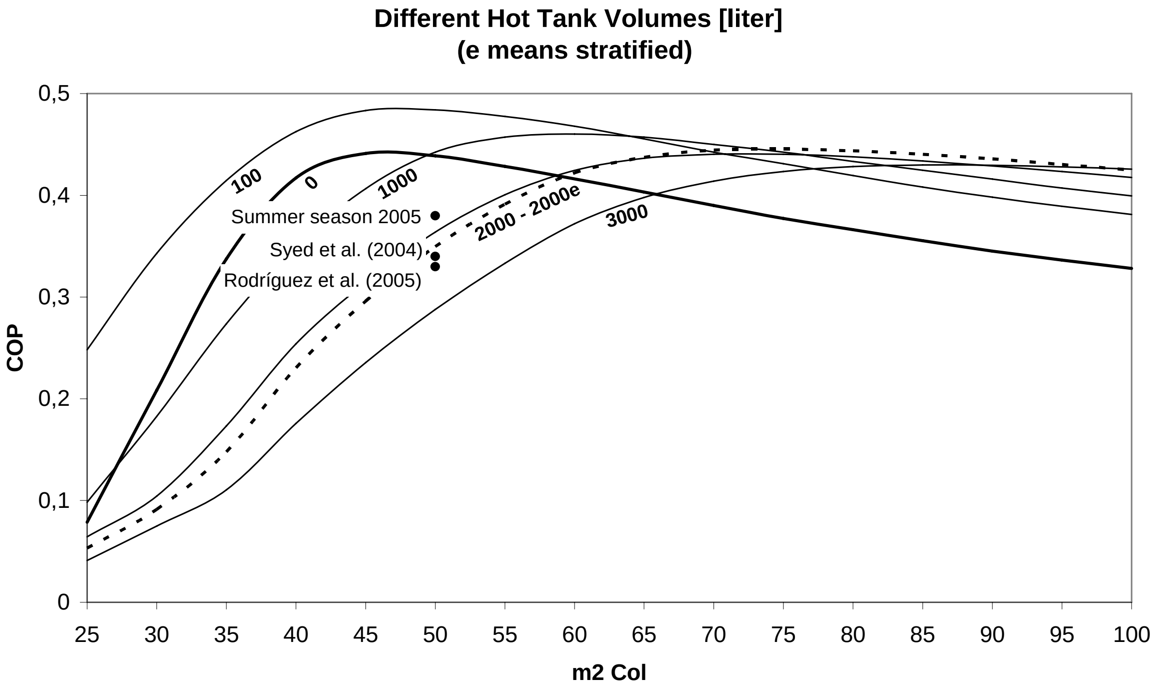

Figure 5: COP values for different collector surfaces and hot storage tank volumes. Results of TRNSYS simulation, compared with several experimental seasonally averaged results.

when there is a significant value of the instantaneous thermal load, the absorption chiller can not attend it because of lower generator temperatures. Nevertheless, in the overall behavior, this configuration gives satisfactory results from the point of view of *COP*, *SCOP* and total cooling energy produced. Besides that it is less costly, simpler and smaller.

The hot storage tank configuration (HSTC) presents an improvement compared to the NSC because it acts as a temperature buffer and additionally, energy is being stored to its later use for cold production. Acting as a buffer, the cooling process begins later than the NSC, because the generator temperature increases but at a slower rate. Late in the afternoon, the facility can still produce a considerable amount of cooling.

A problem appears when selecting the adequate thermal storage capacity. Most of the times this selection is based on scarce empirical data, not being possible to accurate determine the capacity if detailed weather data of the facility location is not available. Surprisingly from simulation results it has been appreciated that a 100 liters thermal storage capacity gives best results in the overall behavior of the facility, compared to the NSC. This storage capacity has been selected as the optimum capacity working under HSTC. An improvement is achieved in the *COP* value of facilities with less than 65 m^2, while the *SCOP* and total energy produced reflects an increase up to collector surfaces of 45 m^2. Table 3 resumes the simulation results for the optimum HSTC and for the NSC. From the point of view of *COP*, *SCOP* and total energy produced, the 100 liters HSTC has been selected as the best configuration with a hot storage tank. Nevertheless, working at current conditions, 50 m^2 and a hot storage tank of 2,000 liters, it is a better choice to work under NSC. For comparison purposes, Figure 6 presents the total cooling power produced with different hot storage tank capacities for July 12, 2005.

Table 3: Seasonal simulation and experimental results for the different configurations with a collector surface of 50 m^2.

Configuration	Q_e (kWh)	Q_g (kWh)	*COP*	*SCOP*
NSC	2696	6143	0.44	0.13
HSTC (100 liters)	2710	5598	0.48	0.13
Current operating conditions	1322	3448	0.38	0.06

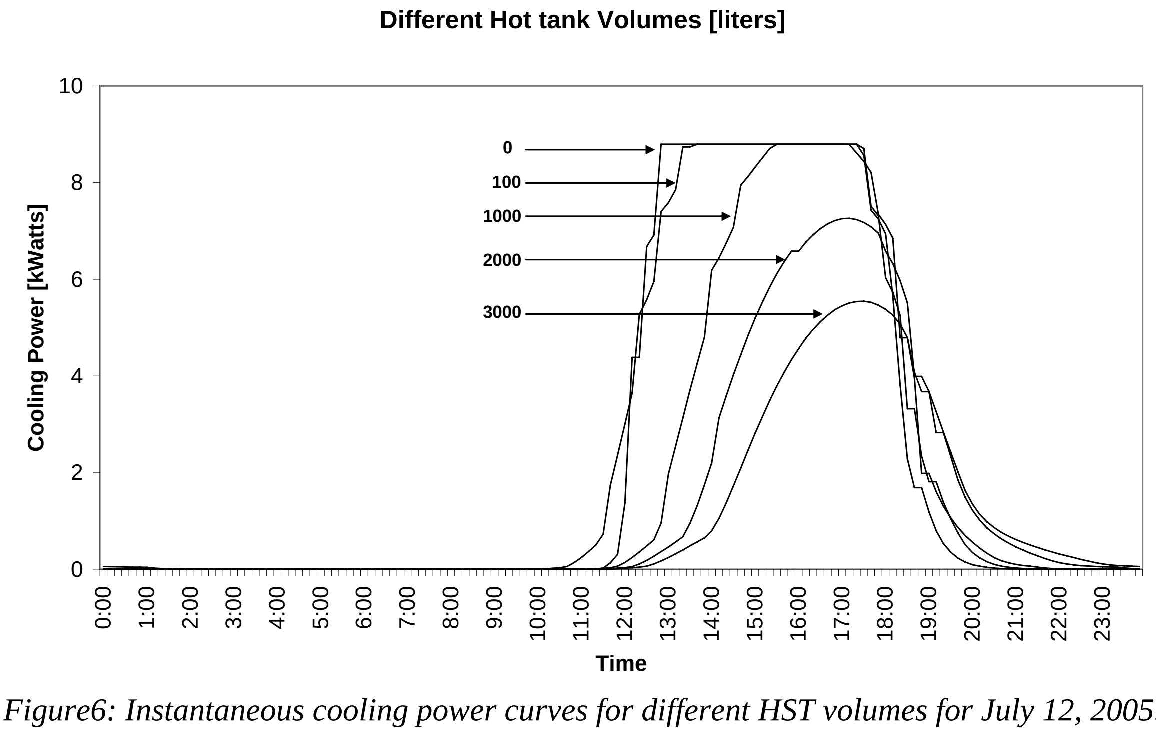

Figure6: Instantaneous cooling power curves for different HST volumes for July 12, 2005.

5 Conclusions

- A simulation of the experimental solar cooling facility installed at Universidad Carlos III de Madrid using the TRNSYS computer code has been conducted, obtaining a good prediction for the instantaneous working variables and for seasonal performance parameters; *COP*, *SCOP* and total energy produced.
- The simulation has been oriented towards optimizing the current experimental facility in its application to domestic cooling. An optimum capacity value for hot storage tank has been found to be 100 liters.
- Nevertheless, when comparing with current conditions working, the facility improves its behavior without any kind of storage.

6 Acknowledgements

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