ANALYSIS OF SOME CONFIGURATIONS FOR RESIDENTIAL SOLAR COOLING SYSTEMS

Natividad Molero Villar, José M. Cejudo López, Fernando Domínguez Muñoz, Antonio Carrillo Andrés

Energy Research Group, University of Málaga

Calle Arquitecto Francisco Peñalosa, 29071 Málaga (SPAIN)

1. Introduction

The intensive use of conventional electrically driven chillers in the residential sector has been one of the main factors behind the continuous increase in the electricity consumption and peak electricity demand. Solar-driven cooling systems are a promising alternative to conventional systems. The most common technology for residential applications is the solar-driven single effect absorption chiller.

There are many possible configurations for this kind of system. For instance, the thermal storage could be in the 'hot' side of the absorption machine (between the solar field and the generator), and/or in the 'cold' side of the machine (between the evaporator and the load). Most of the published works on these systems use only hot water storage (Florides et al., 2002; Atmaca et al., 2003; Joudi et all, 2003; Assilzadeh et al., 2005; Zambrano et al., 2008; Eicker et al., 2009), whereas a few studies consider both cold and hot water storages (Li et al., 2001; Marc et al., 2010).

As any other solar system, a conventional backup is required for cases of insufficient solar contribution. There is no consensus in the literature about which is the best type of backup system. Some authors like Calise et al., 2010 argue that the configuration with compression chiller has better energetic performance than the configuration with boiler. Henning (2004) also mentions the advisability of using a cold backup instead of a hot backup.

Many ways exist for coupling the backup system with the solar system. In case of using a hot backup (i.e., a boiler), a parallel connexion is preferred in order to avoid higher temperatures of the water returning to the solar collectors, which would lower the collector efficiency. In case of using a cold backup, it could be installed in the room to be serviced, or coupled with the solar system. In the first case, two different terminal units are required. In the second case, when the auxiliary system is integrated as part of the solar system, the chiller can be installed in parallel or in series connexion. The series connexion is only advisable when the auxiliary system is able to modulate the cooling capacity. When cool water storage is present, the electrical chiller can be installed in series, parallel or mixed connexion.

The purpose of this paper is to investigate if there are significant differences in the thermal behaviour of three different configurations of solar-driven absorption systems. All these configurations include 'hot' and 'cool' water storages, and use a reversible compression chiller as backup system. In configuration 1, the backup chiller is independent of the solar system. In configuration 2, the backup chiller is connected in parallel with the cool water storage. In configuration 3, the backup chiller is connected to the cold storage.

The configurations will be simulated with TRNSYS 16. Some issues are:

a) Two locations in Spain are considered, Málaga (south cost) and Madrid (centre)

b) A single effect absorption chiller is employed (nominal cooling capacity 10 kW), with characteristics well suited for residential applications

c) Solar flat-plate collectors are used

d) Terminal units are fan-coils

e) A standard house is defined and simulated in order to determine typical heating (space + DHW) and cooling demands.

The main figures of merit to compare configurations will be the solar fraction and the auxiliary energy demand, although other operational parameters were also considered. In particular, the electric consumption of each configuration was estimated from the characteristics of the equipment and the number of working hours. However, the results of the simulations showed that the electric consumption of the three analyzed configurations was very similar and, consequently, this factor was not used for comparison.

2. System description

The base case has 30 m^2 of flat plate solar collectors tilted 30º, 300 litre hot water storage and 900 litre cool water storage. The system provides space heating, space cooling and DHW to a 128 m^2 floor area standard house. Figure 1 shows simplified schematics of the systems in summer mode. The common part of the scheme and the variations between configurations are shown. The differences between configurations are found, as explained before, in the arrangement of the backup system. Configuration 1 has an independent chiller that requires its own terminal unit, different from the one used by the solar system. Configurations 2 and 3 integrate the chiller into the solar system. In configuration 2, the chiller directly services the load when required. In configuration 3, the chiller is connected to the cold storage tank.

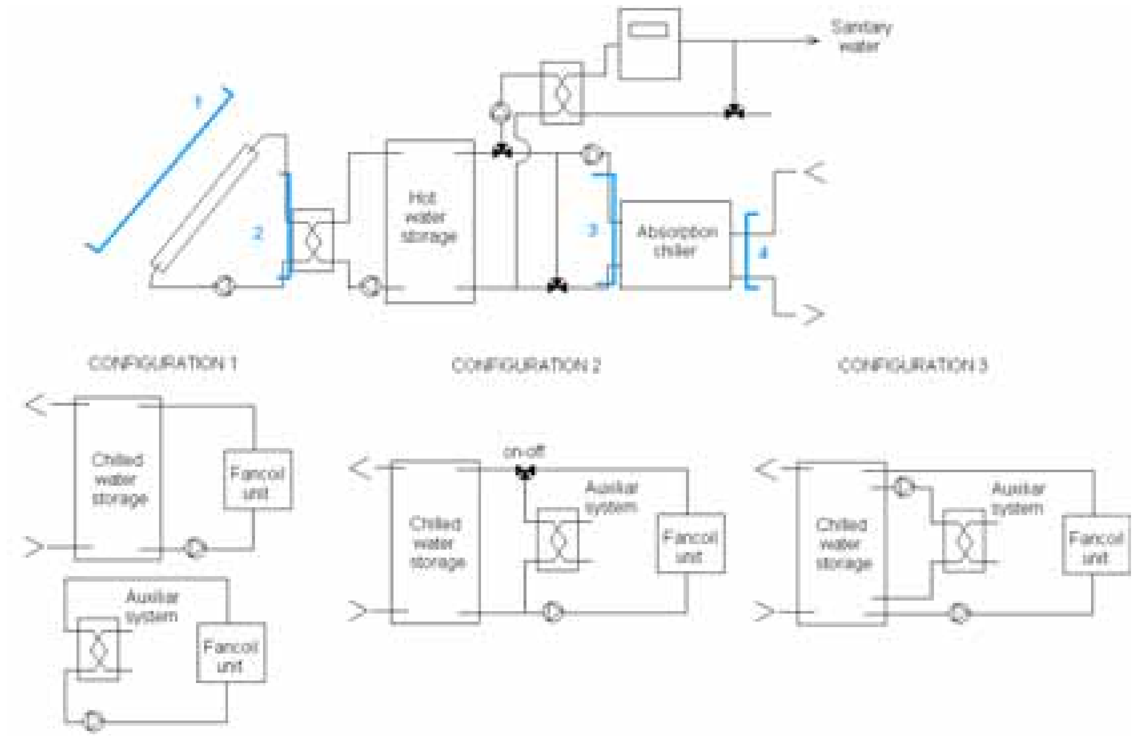

Fig. 1: Schematic of solar cooling configurations (summer season)

In winter mode, the auxiliary system works as a heat pump, the absorption machine is bypassed, and the cold storage is used as extended hot water storage. All over the year, domestic hot water is preheated in a heat exchanger in series with the hot water tank, and finally heated (if necessary) by an independent modulating boiler. This boiler is exclusive of the DHW circuit, that is, independent of the space heating function.

2.1 Component models

The flat plate solar collectors have been modelled with the TRNSYS standard component Type1b. The efficiency curve of the collector is shown in equation 1.

$$\eta = 0.75 - 3\frac{T_m - T_a}{G} - 0.015\frac{(T_m - T_a)^2}{G} \qquad \text{(eq. 1)}$$

T_m and T_a represent the average temperature of the fluid and the ambient temperature (ºC), respectively. G is the total radiation on the collector surface (W m^{-2}).

All the storage tanks are a height of 2.5 m and have been simulated using the standard TRNSYS component TYPE 60. This component models a water-filled sensible energy storage tank, subject to thermal stratification. In the simulations, five fully mixed nodes have been used to account for vertical stratification effects. The thermal loss coefficient of the tanks is 0.7 W m^{-2} K^{-1}, which lies between 0.5 and 0.9 W m^{-2} K^{-1} as recommended by Syed (2005). The pipes of the collector field have been modelled with the standard TRNSYS component TYPE 31, which considers both the thermal losses from the pipes to the ambient and the thermal inertia of the fluid in the pipes.

The absorption chiller has been modelled using the non-standard TRNSYS component TYPE 680 (TESS, 2007). This steady-state model simply interpolates the characteristic curves of the chiller in order to determine the available capacity and the COP at the current operating conditions. The COP and the cooling capacity of the chiller are defined as tabulated functions of several temperatures: generator water inlet, chilled water outlet and cool water inlet to the absorber and condenser. The functional form of these curves was derived from manufacturer data for the commercial chiller Yazaki SC30. The chiller used in the simulations has a nominal capacity of 10 kW, and a nominal coefficient of performance (COP) of 0.695. The cooling tower is simulated assuming that the tower outlet water temperature is 5 ºC above the ambient wet bulb temperature.

The fan-coil unit and the back-up chiller have been modelled using manufacturer curves from CIAT Spain. The chiller model corrects the nominal energy efficiency ratio (EER) and the nominal capacity as functions of the part load ratio and the operational temperatures (chilled water temperature, dry and wet air temperature of the zone, ambient temperature). The fan-coil model considers the effect of the operational temperatures (chilled water temperature, zone conditions) on the sensible heat ratio. Other (only sensible) heat exchangers are modelled assuming a constant efficiency of 0.8

2.2 Energy demands

A 128 m^2 floor area, two storeys high, standard house was simulated with EnergyPlus (2011) in order to determine the space heating and cooling demands. The ratio of glazed area to total façade area is 23.4%. The orientation of the main façade is south, and all walls are exterior. The maximum number of occupants is 5 people, the lighting load is 20 W m^{-2}, and the power installed on electric equipment is 1250 W. Appropriate schedules have been applied to all internal gains. The ventilation level is fixed at 3 litre s^{-1} per person, and the assumed infiltration rate is 1 air changes per hour. The total annual demands (including DHW heating) are summarized in figure 2 for Málaga and Madrid.

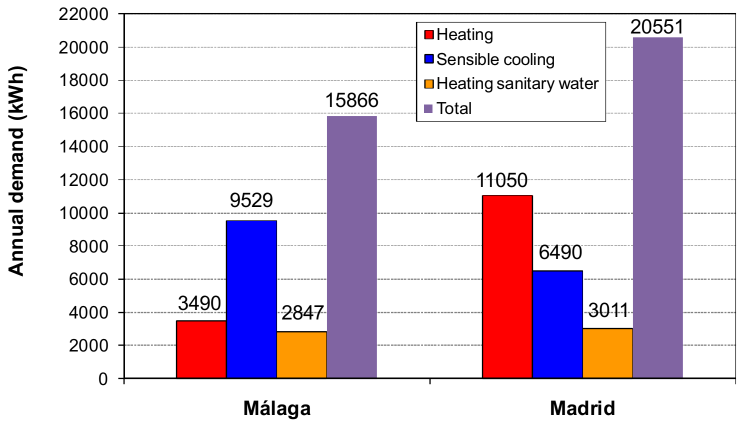

Fig. 2: Total annual demand

2.3 Control strategies

The primary and secondary pumps are activated when the outlet temperature of the collectors is 7 ºC above the temperature at the bottom of the hot storage. A hysteresis window of 5 ºC is applied. The primary loop mass flow rate is set at 50 kg h m^{-2}. To activate the absorption chiller, the temperature at the top of the hot storage tank must be 7 ºC above the minimum recommended temperature for the water entering to the generator (75 ºC). A hysteresis window of 5 ºC is also applied to limit the number of on/off cycles. A modulating three ways valve is used to prevent temperatures higher than 95 ºC from entering the absorption chiller. The temperature of the water leaving the cooling tower must be below 32 ºC when the absorption chiller works.

The set point temperature of the absorption chiller is 7 ºC, but if the machine does not have enough capacity, the outlet temperature of the chiller increases. It depends on the available hot water temperature, and the ambient conditions.

In configuration 1 and 2, the set point temperature of the backup system is 13 ºC, slightly higher than the set point of the absorption chiller. In these two configurations, the system can regulate capacity to provide the energy required. However, in configuration 3, when water temperature at the bottom of the cool storage is higher than 13 ºC, the backup system is activated. Therefore, the outlet chilled temperature is variable, and it depends on the entering water temperature (top of the cool storage) and ambient conditions. With these restrictions, it is possible to cover the peak demand at any time in any of the three configurations, because terminal units are sized to cover peak cooling demand with water entering at 13 ºC.

The signal to activate the auxiliary system depends on the configuration. In configuration 1, the activation happens when solar system can not cover the total cooling demand. In configuration 2, it happens when cool water storage is not able to provide chilled water at the desired temperature, less than 13 ºC. In configuration 3, the activation happens whenever the temperature at the bottom of the cold storage tank is higher than 13 ºC and it does not stop until this temperature is 2 ºC below.

In winter season, the cold storage tank is used as an extended hot tank, connected in series with the buffer solar tank. In that season, electrical-driven reversible heat pump, the backup system, provides hot water to a set point of 40 ºC in configuration 1 and 2. In configuration 3, the backup is activated when temperature at the top of the second hot storage is below 40 ºC.

3. Simulation results

The annual results for the base cases in Málaga and Madrid are shown in figure 3 for cooling and figure 4 for heating and DHW. The “thermal auxiliary energy” is the energy supplied by the backup system.

The total solar fraction in configuration 1, 2 and 3 (total means space heating + space cooling + DHW) is 71.1%, 63.6% and 62.5% respectively. But although configurations 2 and 3 present similar results for solar fraction, they present different results for the auxiliary thermal energy that backup system has to provide.

In Málaga the auxiliary thermal energy in cooling mode in configuration 2 and 3 is 16.4% (602 kW.h) and 30.5% (1118 kW.h) higher than in configuration 1 respectively, and the auxiliary thermal energy in heating mode is 89.2% (603 kW.h) and 123.1% (832 kW.h) higher too.

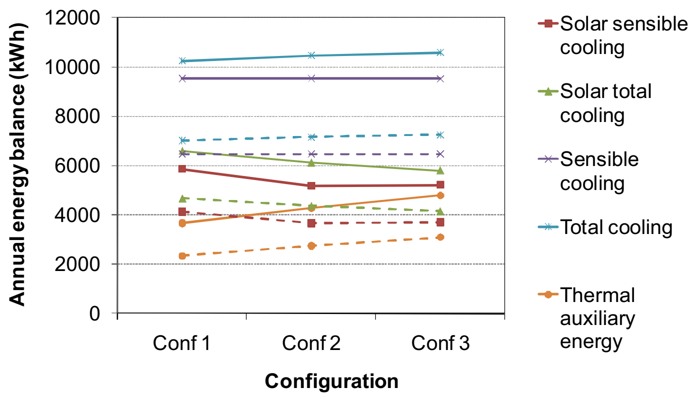

Fig. 3: Annual cooling energy balance. Thermal energy. — Málaga, - - - Madrid

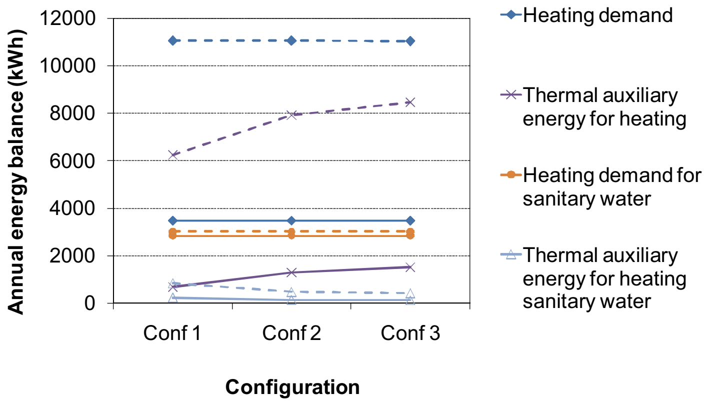

Fig. 4: Annual heating energy balance. Thermal energy. — Málaga, - - - Madrid

From a thermal point of view, configuration 1 performs the best in both locations, followed by configurations 2 and 3 in this order. In configuration 1, the auxiliary system is fully independent of solar system, and consequently it has its own fan-coil unit. This allows using chilled water produced by solar energy even if its temperature level is not low enough for satisfying the total cooling demand. In addition, configuration 1 involves a higher energy extraction rate from hot water storage to the absorption chiller during the early and last hours of the day. The average temperature of the solar field is thus reduced, improving efficiency.

When the auxiliary system is integrated with the solar installation, configuration 2 is better than configuration 3. Two factors explain this result. First, the storage tank in configuration 3 experiences higher thermal gains because its temperature is lower. It happens because the backup system ensures that temperature at the bottom of the storage does not exceed 13ºC. Therefore, part of the conventional energy used to cold down the water in this storage is lost. Besides, the lower temperature influences in the behaviour of the absorption chiller. Secondly, the effect of the latent to sensible heat ratio of the terminal unit is important (see figure 5). In configurations 1 and 2, the set point temperature for the backup system is 13 ^{o}C, but in configuration 3 it is not fixed, and it is usually smaller than 13 ºC. This means that in configuration 3 the fraction of conventional energy used to lower the zone humidity is higher than in the other two configurations.

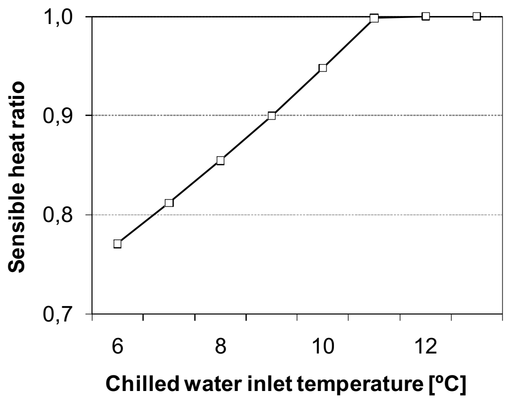

Fig. 5: Sensible heat ratio of the fan coil unit as a function of the chilled water temperature

The comparison in heating mode is similar. Configuration 1 is better because it allows using solar energy even if the temperature of the solar hot storage is not large enough to meet the total heating demand. Configuration 2 is better than configuration 3 because part of the energy produced by the auxiliary system, which is stored in the extended hot tank, is lost to the environment.

Just to have an order of magnitude for comparison in terms of primary energy consumption and cost, we will assume a boiler efficiency of 80%, a COP of 3 for auxiliary heat pump, and a conversion value of 0.33

between primary energy and electrical. The primary energy required for each configuration is represented in figure 6. The annual operating cost for each system, assuming a cost of 0.11 c€ kW^{-1} h^{-1} for electricity and 0.74 c€ kg^{-1} (13.8 kW h kg^{-1}) for gas, is shown in figure 7. The total savings are presented in table 1.

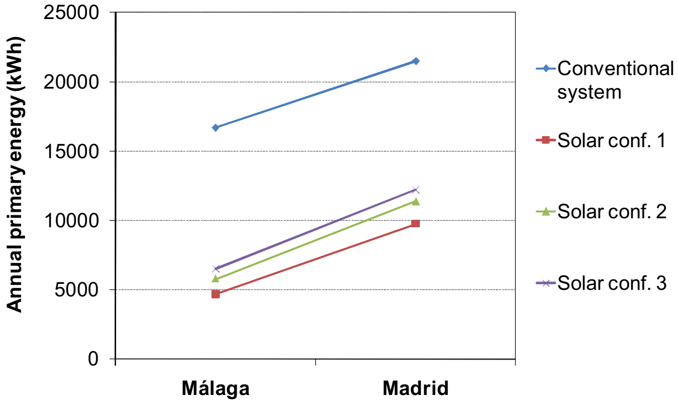

Fig. 6: Annual energy balance. Primary energy

Tab. 1: Annual operating saving cost (€)

	Solar conf. 1	Solar conf. 2	Solar conf. 3
Málaga	499.7	461.7	435.0
Madrid	480.1	427.6	398.1

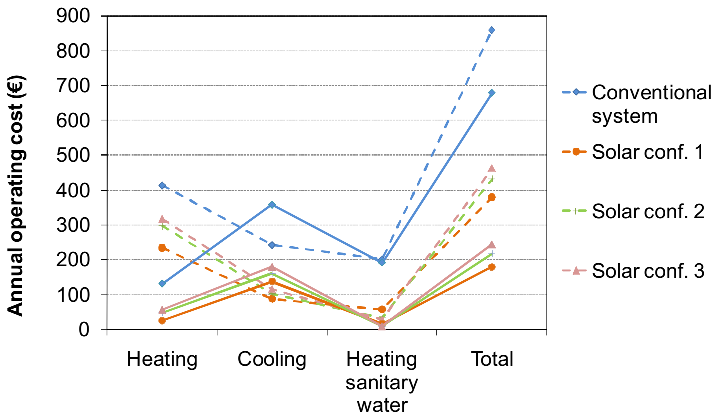

Fig. 7: Annual operating cost for the different systems in both locations

In the following subsections, the influence of different design parameters is analysed: collector area, collector type and hot and cool water storage volumes.

3.1 Collector area

Solar fraction is very dependent on the collector area. An increase from 20m^2 to 40m^2 in Malaga implies an increase of the total solar fraction (total means space heating + space cooling + DHW) from 57.2% to 81.0% for configuration 1, from 52.3% to 72.0% for configuration 2, and from 50.5% to 70.4% for configuration 3; see figure 8.

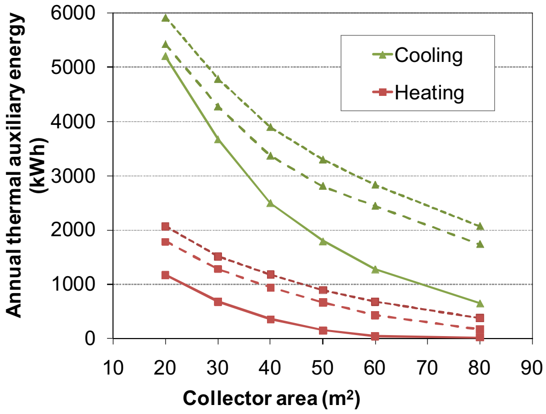

Fig. 8: Influence of the solar collector area in auxiliary thermal energy required. Málaga . (——) Conf 1, (— —) Conf 2, (- - - -) Conf 3

The number of on/off cycles of the absorption chiller increases when the solar collector area decreases. This behaviour can be observed in figure 9, which represents the hourly values of different energy flows during a typical summer day (10th July) for configuration 1 for three different collector areas. The operation of the absorption chiller becomes more continuous as the collector area increases.

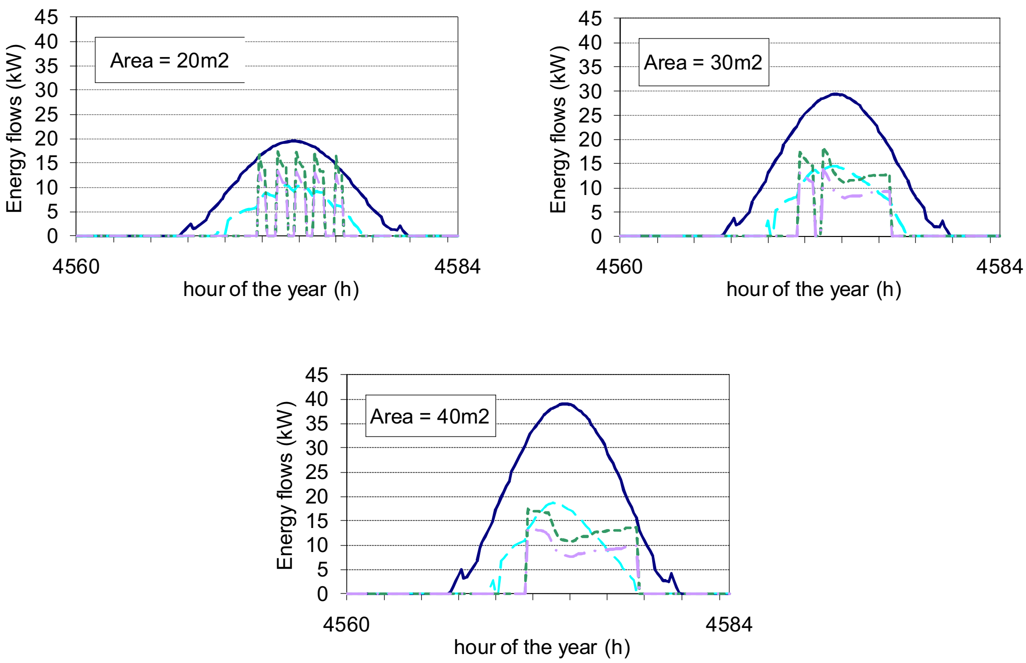

Fig. 9: Representative energy flows for several collector area. Málaga. Energy flows . (—) Incident radiation, (— —) power from solar collectors, (· · ·) inlet power to the generator of the absorption chiller, (— · —) power from the evaporator of the absorption chiller

3.2 Collector efficiency

Three different collector models are considered; table 2 shows the coefficients of the performance curves of each collector. The base case is type 1. Figure 10 shows how the solar fraction changes with the collector type. The main influence is observed in the solar cooling fraction. During the heating season the three alternative collectors perform the same effect.

Besides, an increase of on/off cycles in absorption machine occurs if the efficiency curve of the collector is worse. It is the same effect observed when the solar collector area is reduced (figure 9).

Tab. 2: Efficiency curves for collectors considered

	Type0	Type1	Type2
a0 (-)	0.82	0.75	0.75
a1 (W m^{-2} K^{-1})	2	3	4
a2 (W m^{-2} K^{-2})	0.015	0.015	0.015

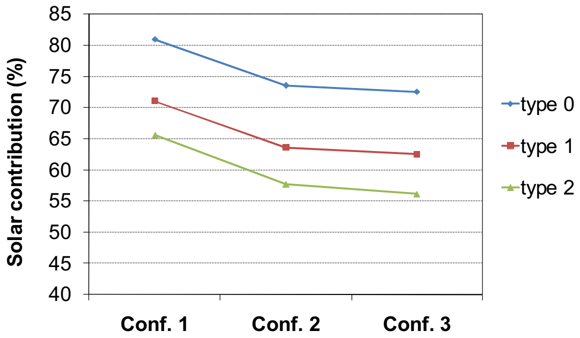

Fig. 10: Influence of collector efficiency curve in total solar contribution (%). Málaga

3.3 Storage size

The effect of the storage size is analysed by changing the volumes of the hot storage from 10 litre m^{-2} to 30 litre m^{-2}, and the total volume from 20 litre m^{-2} to 100 litre m^{-2}. Figure 11 plots the auxiliary annual energy required to meet the cooling demand. For all configurations, the higher is the total volume, the higher is solar fraction. In addition, the solar fraction increases (or auxiliary energy decreases) as the hot water storage volume decreases for the same total volume.

Configuration 1 is more sensitive to the cold storage size. When the cold storage volume increases, the temperature of the water entering the fan-coil unit also increases, thus increasing the fan-coil sensible heat ratio. Consequently, part of the energy that was spent in drying the zone air is now dedicated to reduce the air temperature. This effect is not observed in the other two configurations because the temperature entering the fan-coil unit is almost constant. Trends remain the same for Madrid. When the ratio of hot storage volume to total storage volume increases (less cold storage volume), the advantages of using configurations with cold storage vanish.

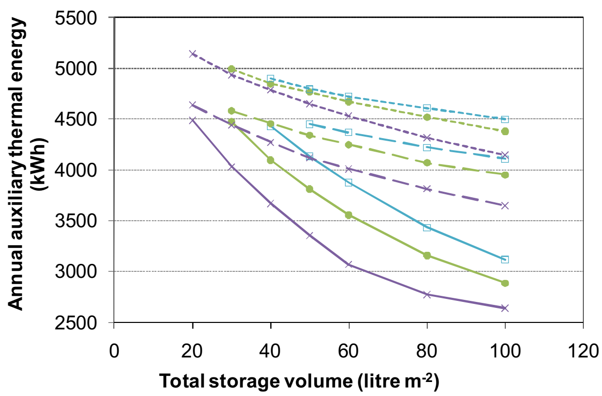

Fig. 11: Influence of the solar total volume in auxiliary thermal energy required for cooling. Málaga . (——) Conf 1, (— —) Conf 2, (- - - -) Conf 3. Hot water size: (x) 10 litre m^{-2}, (●) 20 litre m^{-2}, (□) 30 litre m^{-2}

In relation with solar heating contribution, the important parameter is the total volume of storage, no matter where it is located in the hot or cold side of the system. Influence in both locations is shown in figure 12.

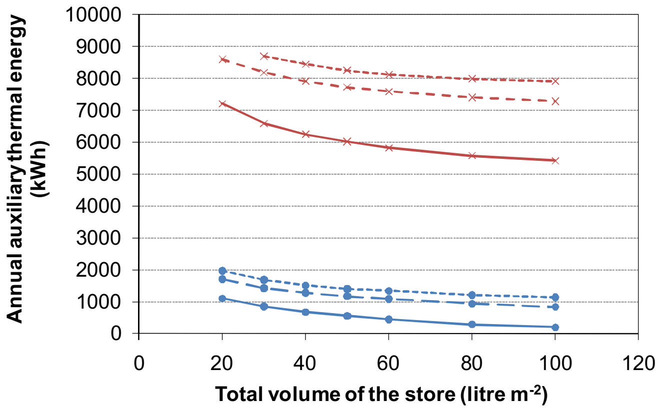

Fig. 12: Influence of the solar total volume in auxiliary thermal energy required for heating. Málaga

4. Conclusions

Simulations with TRNSYS have been performed to compare three configurations of a solar system that provides cooling, heating and domestic hot water to a standard house. The house, with 128 m^2 of useful floor area, is located in Málaga or Madrid, both in Spain. The main parameters of the base-case system are 30 m^2 of flat plate solar collectors tilted 30°, 10 litre m^{-2} of hot water storage and 30 litre m^{-2} of cold water storage. The terminal units in the zone are fan-coils. Configurations differ in the arrangement of the auxiliary system. It is independent of the solar system in configuration 1. In configurations 2 and 3, the auxiliary system is integrated into the solar system, in parallel with the cold water storage in configuration 2, and connected to the cold storage in configuration 3. A sensitivity analysis of the main design parameters has also been performed.

From an economic point of view, the main drawback of configuration 1 is that the investment cost is higher due to the need to use two terminal units, one for the solar system and another for the backup chiller. Concerning configuration 3, the installed capacity of the auxiliary chiller can be reduced compared to configurations 1 and 2, which results in a lower investment cost.

From a thermal point of view, configuration 1 performs the best because there are no interferences between the solar and the conventional system. The solar system works even in the case of insufficient power to meet the full cooling demand. When the auxiliary system is integrated with the solar installation, configuration 2 is better than configuration 3. Two factors explain this result. First, the cool storage tank in configuration 3 experiences higher thermal gains because its temperature is lower. Part of the conventional energy used to cool down the water in this storage is lost. Secondly, the effect of the latent to sensible heat ratio of the terminal unit is important too. The leaving water temperature to fan-coil is the lowest of the three configurations. Therefore, the fraction of conventional energy used to lower the zone humidity is higher than in the other two configurations. The comparison in heating mode is similar. Configuration 1 is the best, following by configuration 2 and 3 respectively.

The sensitivity of the results (solar fraction and consumed auxiliary energy) with the solar collector surface, the collector type and the storage size is analysed. The size of the collector field is the dominant parameter. A secondary effect of the collector area is to increase or decrease the number of on/off cycles of the absorption chiller. If the collector surface is reduced, the number of cycles increases. The same effect is observed when the quality the collector (efficiency) is worse. Regarding the storage sizes, a general rule can be stated: the higher is the total volume, the higher is the solar contribution. Besides, the relation between hot storage volume and cool storage volume is an important factor in cooling mode. As this ratio decreases, the performance of the system increases (higher solar fractions and lower auxiliary energy consumption).

5. Acknowledges

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