SIMULATION BASED PERFORMANCE ANALYSIS OF A MEDIUM TEMPERATURE SOLAR THERMAL COOLING SYSTEM FOR A HOTEL

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Abstract

A solar thermal cooling system for a hotel was designed and simulated in order to know its behaviour and to evaluate its performance and feasibility. This solar cooling system is planned to be installed during the next year in a Palma de Mallorca hotel and is going to be the first one to incorporate the 'CCstaR' collector, that consists in a solar thermal concentrator with fixed reflector and tracking absorber. Due to the typology of the hotel, with ten floors and poor isolation, solar energy can not be the main heat source; therefore a bio-mass heater was included in the design. A double effect absorption chiller was also comprised in the system to generate chilled water for the air conditioning of the hotel. The system was simulated with the Transient System Simulation program (TRNSYS). It was found that the average efficiency of the solar concentrator with respect to the beam radiation was 51.8% and that the system could provide 7.2% of the energy for the hotel space cooling if a properly sized cold storage tank and an efficient control are included. The operating temperature for the solar concentrator was around 155ºC.

1. Introduction

About 50% of the energy consumption in Europe is attributed to process heat applications and 80% of these applications need temperatures lower than 250ºC [1]. The use of mid range temperature collectors could mean a 40 to 50% saving of the primary heating and cooling energy consumption in the building sector of south of Europe and Mediterranean tourist areas [2].

Some studies demonstrate the technical feasibility of solar thermal absorption cooling, specifically those that are based on high temperature solar receivers and a double effect absorption chiller [3].

In the present study, the feasibility of a solar thermal absorption cooling system, that incorporated the technology of a solar thermal concentrator, with fixed reflector and tracking absorber, was investigated.

2. System description

The analysed system was designed to provide both the cooling and Domestic Hot Water (D.H.W.) loads of the hotel. It consists of a field of 148 m^2 of solar thermal concentrators with fixed reflector and tracking absorber, a 352 kW (steam) double effect absorption chiller, a 349 kW bio-mass boiler, a cold storage tank, a D.H.W. storage tank, several heat exchangers, circulation pumps and control valves, indicated on the process diagram of Fig. 1.

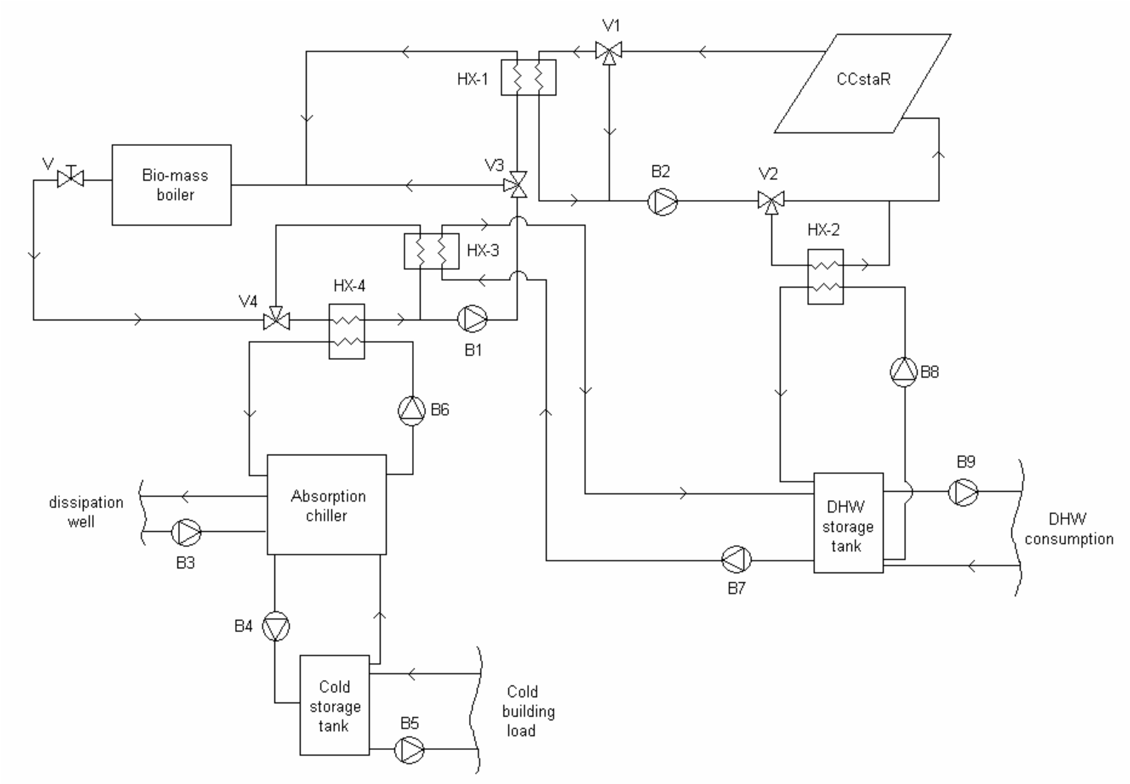

Fig. 1. Process diagram of the solar cooling and DHW system.

The solar thermal concentrator with fixed reflector and tracking absorber (‘CCstaR’ model) was developed by ‘Tecnología Solar Concentradora’ and was designed to work at mid range temperatures [4]. The optical and thermal efficiencies of the first prototype of this concentrator model were tested. Nevertheless the coefficients used in this study were based on the testing of some of the components of a new prototype that it is planed to start operation in September 2010. For the model included in this study the optical efficiency was estimated to be 0.718, the linear coefficient of thermal losses 0.123 $W/K{\cdot}m^2$ and the quadratic coefficient of thermal losses $1.034 \cdot 10^{-3}$ $W/K^2{\cdot}m^2$. The Incidence Angle Modifier (IAM) function was determined using a ‘ray tracing’ simulation. The concentrator’s surface was oriented pointing south with a slope angle of 15º.

The Double-Effect Steam-Fired Absorption Chiller LSH-S010 [5] was chosen for the system. The cooling machine was forced to operate in part load conditions. The steam flow rate and therefore the available energy input to the machine were reduced so that the required cooling load was around 79% of its nominal value. The ‘hot’ power supplied to the machine was, therefore, around 213 kW and the cooling power provided by the chilller around 277 kW. Using the absorption chiller in part load conditions has the advantages of a greater COP (around 1.32) and a longer continuous operation time. The chiller’s dissipation stream was cooled down in a dissipation well.

The (steam) bio-mass boiler that was chosen is the CS-500 model [6]. The boiler can produce both the heat flows required for cold generation and for DHW generation. It has, moreover, the solar supply. Its heat production is, therefore, quite variable depending on system conditions.

The heat rate that is assigned to DHW production (through heat exchanger HX-3) is about 60 kW when cold is simultaneously generated and 150 kW approximately when only DHW energy is required.

There is a drop in the boiler's circuit pressure after the valve V from 5 bar to 4 bar. The boiler's output temperature is 152ºC and it drops to 148ºC after the expansion. The circuit's pressure returns to 5 bar after pump B1.

In heat exchanger HX-1 part of the heat carrying medium evaporates and the rest evaporates at the biomass boiler and in HX-3the steam condensates. In HX-2 there is no phase transition. HX-4 does not exist in the real installation, but it was used in the system simulation to model the double-effect chiller's phase change. More details of the model are given in section 4.

The cold storage tank was sized with a volume of 25 m3. This size makes possible a minimum continuous operation time of the chiller of one hour (during months with lower cooling loads: April and May).

The chiller operation ranges were selected so that the lower temperature side of the cold storage tank was always kept at a temperature under 17ºC. Cold water from the tank arrives to fan-coils cooling hotel spaces. The water returns from the fan-coils to the tank at a temperature 4K higher than it entered and the flowrate of this circuit varies depending on the required cooling load.

DHW storage tank was sized with a volume of 8 m3.The output of this tank was given at a constant temperature of 45 ºC through a controlled mixing valve.

3. Operation mode and system's control

The system was controlled by means of different modules. Each module is in charge of the control (activation-deactivation) of a part of the system and can depend on other modules.

Cold generation (control3) is activated when the cold storage tank temperature exceeds a fixed value. If solar receiver's conditions enable the solar supply during cold generation (control2), there will be a fraction of solar cooling.

The boiler can produce DHW energy (through HX-3) while cold is being generated or when cold generation is not active. If DHW tank's temperature decreases under 65ºC and cold production is active, an additional amount of energy (~ 60 kW) will be assigned to the DHW tank (control4a). If, on the other hand, cold production is not active and DHW tank's temperature decreases under 55ºC, the boiler will become active (control 4b) for exclusive DHW energy production (~150 kW).

If solar receiver's conditions enable the solar supply but cold generation is not required, the solar DHW energy supply will be transferred through HX-2 (control5). Through this way of controlling the system, solar energy is used whenever it is available and adequate cold and DHW tanks' temperatures are ensured. The function of each control unit and the conditions for its activation are detailed below:

<u>control1</u>: activates circulation of the primary solar circuit: pump B2 ON (5000 kg/h).

- Collector's output temperature must be 30K higher than ambient temperature (the signal will keep ON till the temperature difference decreases under 20K).
- Total radiation on collector's surface must exceed 200 W/m^2.

<u>control2</u>: activates the solar energy supply to the boiler (through HX-1) for cold generation. Valves V1 and V3 are opened to enable the solar contribution.

- Cold generation is required (control3 ON)
- Collector's output temperature must be 15K higher than B1 pump's output temperature (the signal will keep ON till the temperature difference is less than 5K).

control3: activates cold production: bio-mass boiler, absorption chiller and pumps ON, B1 (325 kg/h), B3 (100000 kg/h), B4 (60500 kg/h), B6 (2928.5 kg/h).

- Hot side cold storage tank's temperature exceeds 19ºC (the signal will keep ON till this temperature is less than 10.5ºC)

control4a: activates DHW energy production while cold is being generated: B7 ON (3200 l/h), V4 partially open (23.5 %) towards HX-3, B1 ON (425 kg/h).

- Cold generation is required (control3 ON)
- Hot side DHW storage tank's temperature is less then 65ºC (the signal will keep ON till this temperature exceeds 70ºC).

control4b: activates exclusive DHW energy production (from boiler): bio-mass boiler ON, B7 ON (6000 kg/h), V4 open (100%) towards HX-3, B1 ON (231 kg/h).

- Cold generation is not required (control3 OFF)
- Hot side DHW storage tank's temperature is less then 55ºC (the signal will keep ON till this temperature exceeds 65ºC).

control5: activates solar contribution to DHW tank: pump B8 ON (5000 kg/h) and valve V2 open (100%) towards HX-2.

- Cold generation is not required (control3 OFF)
- Circulation of the primary solar circuit is active (control1 ON)
- B2 pump's output temperature is 12K higher than cold side DHW tank's temperature (the signal will keep ON till the temperature difference is less than 2K).

4. Model description

The software selected for modelling and simulating the system is TRNSYS [7]. Some of the elements used for the simulation of the system were standard types, but also new types were developed:

The method proposed by Yohanis et al. [8] was used to model the two-phase heat exchangers (HX-1, HX-3 and HX-4). This simplified method enables the calculation of the heat flow through a heat exchanger in which one or both heat carrying media are undergoing a phase transition. It is based on enthalpies of the heat carrying media rather than their temperatures.

Heat exchanger calculations were based on the heat transfer effectiveness 'ϵ'. The amount of heat transferred through a heat exchanger can be calculated as follows:

$$Q = \epsilon \cdot Q_{max} \qquad (1)$$

where Q_{max} is the maximum possible amount of heat transferred through the heat exchanger assuming ideal heat transfer conditions. The value of 'ϵ' for a 'well-designed' heat exchanger is close to its theoretical limit. The used model calculates three values of the heat flow through the heat exchanger (q_0, q_1, q_2) for different heat transfer considerations and the minimum of them is the maximum possible amount of heat transferred. This maximum rate of heat flow is determined from the thermodynamic properties of both heat carrying media.

Another simulation element was created to model the bio-mass boiler. This new type calculates the required power to heat the input stream to the boiler up to 152 ºC at 100% vapour fraction as a function of the input temperature, input flowrate, input vapour quality and a control signal that activates the operation of the boiler.

The (steam fired) double-effect absorption chiller was modelled using the TRNSYS Single Effect Hot Water Fired Absorption Chiller's Type 107 together with an ideal two-phase heat exchanger. The chiller's hot stream energy is given by the bio-mass boiler through the two-phase heat exchanger (HX-4, Fig. 1) to take into account the hot stream phase change (condensation) actually produced in the double-effect absorption machine.

Type 107 data file containing chiller's performance information was adapted to the LSH-S010 operation conditions. The extension of single-effect models to a double-effect absorption chiller was already suggested in other studies [9].

Standard type 71 (evacuated tube solar collector) was used for the CCstaR solar concentrator. Type 4e (stratified storage tank) was used as model for the cold and DHW storage tanks.

The hotel cooling loads were simulated with the Hourly Analysis Program (HAP) developed by Carrier. Mean meteorological data of Balearic Islands was used for the weather simulation conditions.

The hotel DHW loads were evaluated according to the criterion given at the Spanish Energy Saving Basic Document (DB HE) [10] for a 3 stars hotel.

5. Simulation results

The system was simulated for a period of 6 months: from April to September. It was found that the averaged efficiency of the solar concentrator with respect to the beam radiation was 51.8 % and that the efficiency with respect to the total radiation was 34.8 %. The system could provide 7.2 % of the cooling energy for the hotel space and 28.3 % of the required DHW energy, with the solar concentrator operating at about 155ºC.

The hot temperature supply to the absorption machine was 148ºC. The cold side of the cold storage tank was kept at a mean temperature of 11.8 ºC, with a minimum value of 6.9 ºC and a maximum of 16.9 ºC. The hot side of the DHW storage tank was kept at a mean temperature of 68.5 ºC, with a minimum value of 43.6 ºC and a maximum of 96.5 ºC. The simulation results for each month are detailed below (Table 1 and Table 2):

Table 1. Energy simulation results for each month.

	April	May	June	July	August	September
Total solar energy (GJ)	92,48	92,48	110,59	114,76	104,94	83,43
Beam solar energy (GJ)	59,29	67,67	73,35	77,39	70,86	54,21
Solar support to boiler (HX-1) (GJ)	4,42	5,30	14,99	26,83	20,39	10,12
Boiler produced energy (GJ)	96,08	94,70	267,54	396,05	379,57	258,13
Chiller consumed energy (GJ)	50,89	52,83	232,10	364,30	344,27	214,90
Chiller produced (cold) energy (GJ)	66,25	68,30	301,96	474,21	447,89	278,35

Boiler DHW produced energy (HX-3) (GJ)	49,81	47,32	50,83	59,98	56,43	53,50
Solar DHW produced energy (HX-2) (GJ)	25,91	30,41	23,15	13,43	16,36	17,38

Table 2. System performance simulation results for each month.

	April	May	June	July	August	September
Total solar efficiency	32,8%	38,6%	34,5%	35,1%	35,0%	33,0%
Beam solar efficiency	51,2%	52,8%	52,0%	52,0%	51,9%	50,7%
Fraction of solar cooling	8,7%	10,0%	6,5%	7,4%	5,9%	4,7%
Fraction of solar DHW energy	34,2%	39,1%	31,3%	18,3%	22,5%	24,5%

The graph presented below (Fig.2) shows the solar receiver and solar contribution to boiler operation temperatures. 'Tin-solar' and 'Tout-solar' refer to input and output collector's temperatures, 'Tin-HX1' and 'Tout-HX1' are the cold side (input and output) heat exchanger HX-1 temperatures, flowrate2 is the B1 pump's flowrate that goes through HX-1 and flowrate1 is the flowrate that goes directly to the boiler:

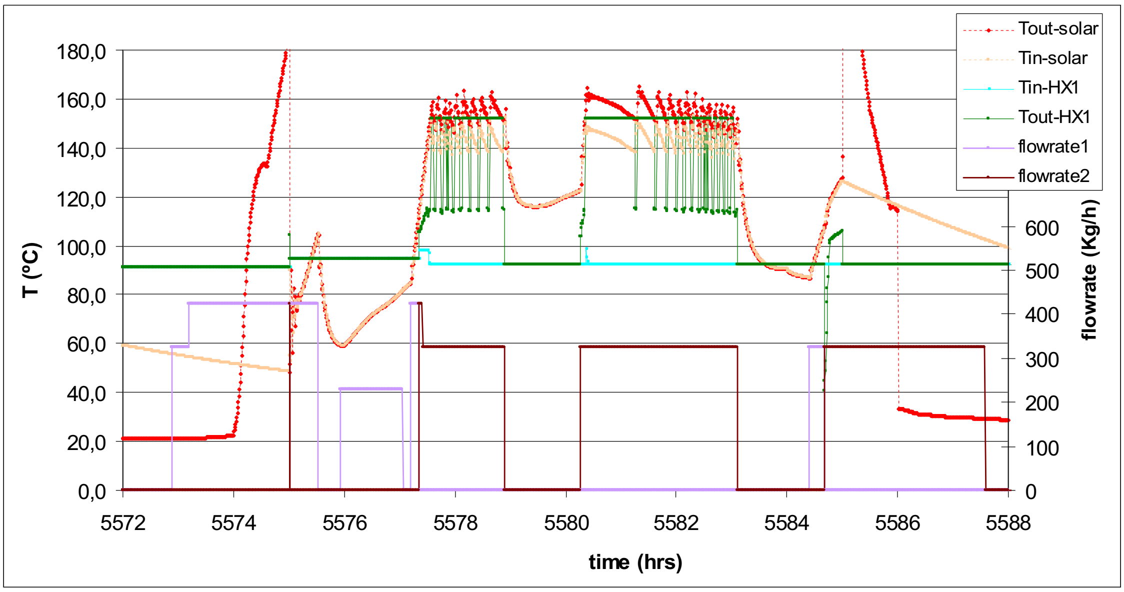

Fig.2. Simulated solar receiver and solar support operation temperatures on 20 August.

Solar concentrator operated at a temperature about 155ºC while there was a solar cooling fraction (flowrate2 not equal to zero). Heat exchanger HX-1 output temperature is higher than the input during the solar cooling contribution. Tout-HX1 temperature was kept at the saturation point (152ºC) during highest solar incident radiation (time ~ 5580.5 h), but it drop when radiation decreased. Collector's temperature oscillations are due to the fact that heat exchanger HX-1 was sized to be able to exchange the collector's maximum heat rate, but collector's received heat rate is variable along the day.

Collector's operation temperature decreased during solar DHW energy production (for example, between times 5579 and 5580 h or 5576 and 5577 h approximately).

Next graph (Fig.3) shows the different heat production situations along the day:

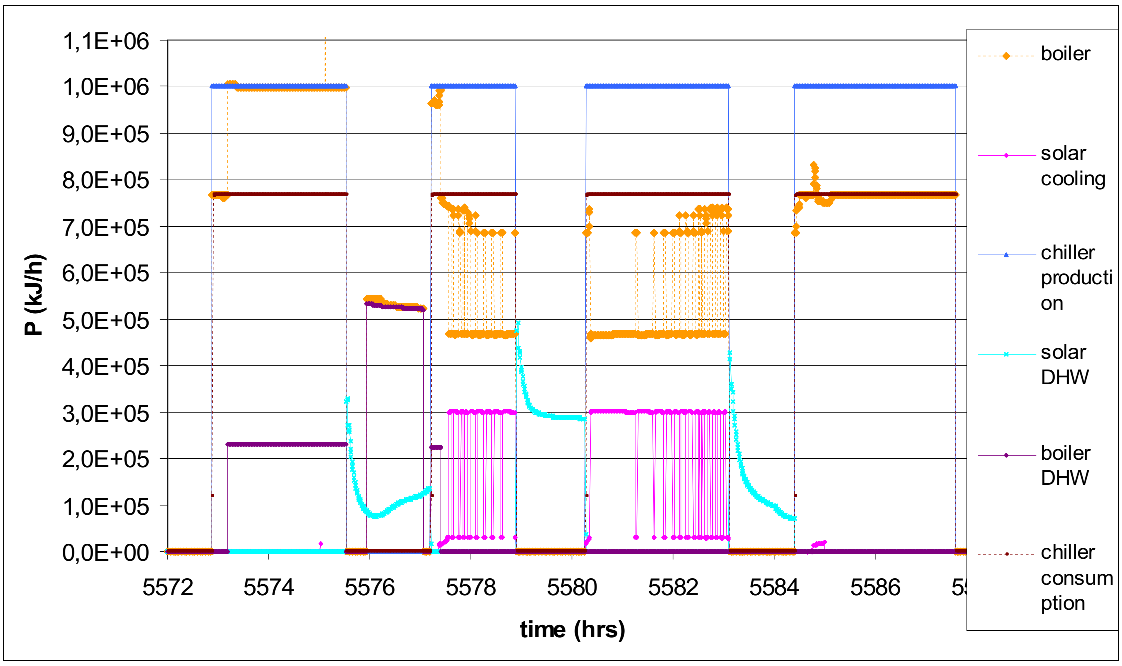

Fig. 3. Simulated system energy rates on 20 August.

Solar supplies to boiler and to DHW tank depend on time and sky conditions. The maximum reached solar power was about 83 kW.

Maximum bio-mass boiler production was 278 kW (time ~ 5574 h) when both cold and DHW energy were generated without any solar contribution. Boiler production decreased to 210 kW (time ~ 5586 h) when only cold was generated without any solar contribution and drop to a minimum value of 130 kW (time ~ 5581 h) during cold production with a maximum solar supply. The boiler generated about 150 kW during exclusive DHW boiler production (between times 5576 and 5577 h). Chiller's cold power was about 278 kW.

Even though the solar contribution to the cooling energy was 7.2%, the goal of this study was to prove the feasibility of the innovative technology of the solar thermal concentrator with fixed reflector and tracking absorber in a solar absorption thermal cooling system under limited collector area conditions.

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