



## **Monitoring and simulation of an existing solar powered absorption cooling system in Zaragoza (Spain)**

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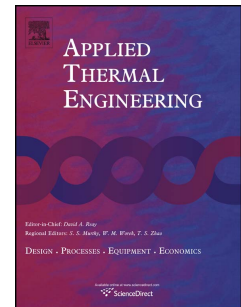


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## TITLE

Monitoring and simulation of an existing solar powered absorption cooling system in Zaragoza (Spain).

**Abbreviated title:** ‘Solar cooling simulation in Zaragoza’

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## ABSTRACT

In 2007 and 2008 the performance of a solar powered absorption cooling installation was analyzed. The solar cooling system consists of 37.5 m<sup>2</sup> of flat plate collectors, a 4.5 kW, single effect, LiBr-H<sub>2</sub>O rotary absorption chiller and a dry cooler tower. The performance analysis of the solar driven chiller shows average values of COP close to 0.6 in 2007 and between 0.46 and 0.56 in 2008. Concerning to the average cooling power, the chiller reaches values between 4.0 and 5.6 kW in 2007 and between 3.6 and 5.3 kW in 2008.

During the analysis phase, a detailed model of the solar cooling system was developed using the simulation environment TRNSYS. The results of the simulation were validated with the experimental ones, and presented in this paper. The measured data as well as the simulation results of the installation show the strong influence of the cooling water temperature and the generator driving temperature on the COP. For this reason an alternative heat rejection sink was design. Among the alternatives, a geothermal sink was chosen since there is a water well located in the surroundings of the solar cooling installation. The first results of the studies carried out show an improvement on the COP up to 42%.

**KEYWORDS:** Solar energy, Absorption cooling, Solar cooling, Energy saving.

## MAIN TEXT

### 1. Introduction

Nowadays, the majority of buildings have been designed with heating installations but they do not have refrigeration equipments. Refrigeration needs are covered by air conditioning machines usually fed with electricity, and these equipments are becoming very common in houses. Due to the operation of these air conditioning machines, in summer, electricity consumption reaches high values, overcoming the electricity generation limits, causing technical problems and increasing electricity prices [1].

Therefore, alternative refrigeration equipments driven by renewable or residual energies have been developed. One of these alternatives is the solar absorption refrigeration system, which uses solar thermal energy integrated with an absorption cycle to refrigerate. This arrangement is especially interesting because the hours in which more solar radiation is available, are the hours with more refrigeration needs [2].

Comparing absorption refrigeration systems with mechanical compression ones, the first are not competitive regarding to economy. Furthermore, low power absorption

machines are not widely commercialized, they are very expensive, and most of them need refrigeration towers to operate [3]. For that reason, research and development activities are necessary in order to reduce the cost of using solar assisted air conditioning in buildings [4-5]. There are a lot of researches referred to theoretical studies [6-9], but the information of experimental studies in this kind of installations is scarce [10]. Away other reasons about this, is the difficulty for designing solar cooling systems. There are not many tools available for accurate dimensioning cooling systems and/or to evaluating the solar thermal contribution to the total energy requirements [11]. One of these available tools is the dynamic simulation used in this work. This paper describes the model and shows the results and the validation with the experimental data of the model of a solar absorption cooling installation based on the performance of an absorption chiller. The experimental installation is located in Zaragoza (Spain). The installation supplies the cooling demand of a gymnasium placed in the University of Zaragoza. To carry out the installation analysis, the system was consequently monitored.

The installation was monitored and analyzed in 2007 and 2008. During this period of time, a model of the installation was carried out with TRNSYS. Monitored climatic data of the installation location was used to develop the model. Furthermore, a detailed model of the rotary absorption chiller was implemented to be included in the whole installation model. The dynamic simulation model was validated and compared with the measured data. The simulation results show a good agreement with the measurements. Thus the model will be used for the evaluation of further investigations and improvements in the installation. The simulation results also show the improvement

capacity of the installation performance with other control strategies. After the experimental and simulation evidence of the influence of the cooling water temperature, a geothermal cooling system has been studied and installed. The installation with the new modification started to work in 2009. With the geothermal cooling system, it is expected that the COP of the chiller will be improved up to 42% more than the current one.

The aim of this paper is to evaluate the potential of the developed dynamic simulation model to design and analyze solar absorption cooling installations.

## **2. Scope of the work**

In the year 2006, a solar thermal installation went into operation to supply the domestic hot water to a building placed in the University of Zaragoza. The building contains a sports pavilion, several offices, and a gymnasium. During the first year of operation, the solar field suffered overheating problems (during the summer months) due to its oversized design. The solution was to use this surplus of thermal power for driving a new refrigeration system. The problem was solved with the integration of an absorption chiller. The chiller transforms the energy from the oversized solar field to supply the cooling demand of the gymnasium. An additional reason for the absorption chiller integration was the possibility to study and analyze the performance of this kind of chillers. The analysis results of the first summer showed the very strong influence of the cooling tower temperature, the chilled water and the generator driving temperatures levels on the COP, (common characteristic of absorption cooling chillers [12-15]).

Based on these results, a model of the solar cooling installation was developed in order to evaluate several possibilities to improve the performance of the solar cooling system. Finally, a new heat rejection system was proposed to improve the installation.

### 3. Description of the installation

The components and the absorption cooling machine used are described below. (Figure 1)

A commercial Rotartica 4.5 kW, air-cooled (Rotartica 045 [16]), single effect, LiBr-H<sub>2</sub>O absorption chiller has been used (3, Fig. 1). It has a rotary drum in which the single effect absorption cycle is carried out with the drum rotating at 400 rpm. The rotation favours mass and heat transfers. Inside the drum are placed the evaporator and the condenser and instead of a traditional compressor fed with electricity, there are a chemical absorber and a generator which reduce the electricity consumption.

The solar collector field (1, Fig.1) has 37.5 m<sup>2</sup> of useful area. Solar radiation is absorbed and transformed in thermal energy to feed the absorption machine. The solar collectors and the heat exchanger (7, Fig.1) form the primary circuit. The installation contains a hot water tank (2, Fig. 1) and auxiliary boiler (6, Fig. 1) but both are not used. So that, the absorption chiller will only work when the solar field can provide it enough energy.



A dry cooling tower (4, Fig. 1) evacuates the residual heat from the condenser and the absorber to the outdoor air, and there is no need of a wet refrigeration tower. Although the COP would be bigger with a wet cooling tower, a dry cooling tower is selected in order to avoid the usual problem of the legionella of the wet cooling towers. Two fan coils (5, Fig. 1) with 6.21 kW of chilling power transfer the chilling power from the evaporator of the absorption machine to the gymnasium air.

#### **4. Simulation model and validation**

In order to evaluate possible changes in the system and its improvement potential, a model of the installation was developed. The TRNSYS software tool [17] was used as a basis for assessment. Besides defining and using TRNSYS existing system component types, experimental data taken from the real installation have been used.

For the gymnasium simulation, a TRNSYS type 56 component was used in the model. To develop the required information for the type 56, the TRNSYS application, TRNBUILD was used. The 215 m<sup>2</sup> gymnasium model considers overall the internal gains, since it is located in the underground of the building. To model the performance of the two fancoils placed in the gymnasium, a simplified cooling coil was used (type 32). The nominal air flow of each fancoil is 1185 m<sup>3</sup>/h and the nominal water flow is 1.29 m<sup>3</sup>/h. Other parameters of the fancoils were obtained from the technical information of the manufacturer [18].

For the simulation of a flat-plate collector, TRNSYS Type1b was used. This type allows the simulation of a flat-plate collector. The characteristic parameters of the collector, referring to its absorber area, are shown in the Table 1.

Although in the normal operation the solar storage does not operate, the 700 litre stratified tank with uniform heat losses of the system was modelled with the help of the Type4c. In this way, new strategies in the storage operation could be evaluated in order to increase the performance of the installation. The same kind of type was used to model the geothermal water tank which is supplied by the water well. This second type 4c, was linked to the type 501, which models the ground temperature. The ground conductivity according to the experimental data is  $1.1 \text{ W/m}\cdot\text{K}$ . The water well temperature shows a constant value of approximately  $17^\circ\text{C}$  in summer time.

To simulate the performance of the rotary absorption chiller, type 107 was used. Since this type uses a catalogue data lookup approach to predict the output of the absorption chiller, and the default data file of the type differs from real operation of the installation, several data files of absorption chillers were created, according to the manufacture technical data and the monitored values of the experimental installation, monitored during the operation chiller in the summer of the year 2008. To model the chiller, the fraction of full load capacity and the fraction of design energy input for various values of fraction of design load, chilled water set point, entering cooling water and the entering hot water temperature have to be defined [17]. To calculate these parameters a

multi-regression fit was carried out from the experimental steady state values of the chiller [19]. The numerical fit of these values results in the following equations:

$$W_g = 0.2508 \cdot T_{g,i} - 0.4654 \cdot T_{c,i} + 0.3836 \cdot T_{ev,o} - 3.5420$$

$$W_{ch} = 0,1452 \cdot T_{g,i} - 0,3846 \cdot T_{c,i} + 0,2735 \cdot T_{ev,o} - 1,0231$$

The multiple correlation coefficient of this regression algorithm was 0.91, showing a good fit between the experimental data and the method results.

Applying this equation and the nominal values of the chilling capacity and the COP from the manufacture's catalogue, the data file of the absorption chiller type was created.

Since the model has to show the real behaviour of the installation, and the performance of the solar cooling chiller depends strongly from the climatic conditions, the monitored meteorological data were implemented in the model of the installation using the type 62, in order to able to utilize them in the simulation.

An auxiliary cooling device, type 92, was chosen to model the performance of the dry cooling tower. The most important parameter of this type is the overall loss coefficient between the cooling device and its surroundings during the operation. The value of this parameter was defined according to the experimental data of temperatures of the water and air side (log mean temperature difference) and of the cooling capacity.

Due to the auxiliary system, two gas boiler, have not supplied any thermal energy to the generator of the absorption chiller, they have not been modelled for the simulation.

Type 5, a steady-state heat exchanger model, was used to model the 34 kW solar flat plate heat exchanger and the 20 kW geothermal flat plate heat exchanger. The overall UA values of both heat exchangers according to the technical information of the manufacturer are  $5037.5 \text{ W/m}^2\cdot\text{K}$  for the solar heat exchanger and  $5506.4 \text{ W/m}^2\cdot\text{K}$  for the geothermal one). [20].

Others TRNSYS components were used: type 3b for simulations of pumps, type 31 for the different kind of pipes and type 2 for the control of the installation (Figure 2).

Once the whole model was finished, it was simulated in TRNSYS with the following parameters according to the operation of the installation and its monitored system:

- Simulation start step: 1<sup>st</sup> of June
- Simulation end step: 15<sup>th</sup> of September
- Simulation step time: 5 min
- Tolerance integration: 0.01
- Tolerance convergence: 0.01

## 5. Methodology

The installation is completely monitored. There is a PLC unit and a web controller which form the controlling and recording values system. As well as this, there are temperature, humidity, and radiation sensors, flow meters and energy meters. In this way, the outdoor and indoor conditions of the installation are well defined.

The monitoring system consists mainly of temperatures probes and flow meters located in the three water flows that enter the absorption machine. Two temperature probes (near to the absorber and the condenser of the absorption machine) measure the inlet ( $T_{c,i}$ ) and outlet ( $T_{c,o}$ ) temperature of the flow between the absorption chiller and the dry cooling tower (finned tube heat exchanger). Two temperature probes measure the inlet ( $T_{ev,o}$ ) and outlet ( $T_{ev,o}$ ) temperature of the flow between the absorption chiller and the fan coils (the measure is taken near the evaporator in the absorption machine). Two temperature probes measure the inlet ( $T_{g,i}$ ) and outlet ( $T_{g,o}$ ) temperature of the flow between the absorption chiller and the heat exchanger (the measure is taken near to the generator in the absorption machine). Furthermore, there is a flow meter in each one of these loops; the water flow that goes to the generator ( $m_g$ ), the water flow that goes to the fan coils ( $m_{ch}$ ) and the water flow that goes to the finned tube heat exchanger ( $m_c$ ).

The methodology to analyze the results of the installation model is the same as the employed in the experimental analysis [21-23]. Four different types of graphics have been done in order to compare and relate variables: 'Rank Graphics', 'Phase Graphics', 'Stationary Graphics' and 'Trend Graphics'.

## 6. Results

### a. Data analysis

The installation was analyzed in 2007 and in 2008, thus the principal experimental values of the installation were taken. The Table 2 shows the mean values of several variables of the installation measured in these years: the outdoor temperature ( $T_{dbo}$ ), the inlet temperature at the generator ( $T_{g,i}$ ), the outlet temperature at the evaporator ( $T_{ev,o}$ ),

the chilling power ( $W_{ch}$ ) and the coefficient of performance of the absorption machine (COP). Both COP and  $W_{ch}$  are calculated from other measured variables.

Some days with different climatic conditions have been simulated and compared with the measured data. In Figure 3 the measured performance of the solar cooling system is shown together with the simulation results for one day in July 2008. A comparison of the measured outlet temperatures of the generator, condenser and evaporator of the rotary absorption chiller clearly show that the performance of the installed system is very well described by the developed simulation model.

The mean values of three representative days of the chiller performance that have been compared are presented in the Table 3. The day 11/07/2008 represents a very hot day (outdoor temperature between 28°C and 40°C), meanwhile the 29/07/2008 and 08/08/2008 represent warm (outdoor temperature below 30°C) and hot (outdoor temperature between 24°C and 35°C) days respectively. The measured and simulated data show a good agreement with an acceptable deviation [24].

Several analyses have been carried out from the point of view of the steady-state of the performance of the absorption chiller. The Figure 4 represents an example of the definition of the steady situation interval. To analyze this steady performance of the chiller, only the values of the diverse variables located inside the interval have been considered.

In the Figure 5 is represented the influence of the ambient temperature in the COP of the chiller. The trend of the simulated results follows the trends of the measured data of both monitored years (2007 and 2008). The three trend lines show a decrease in the COP when the ambient temperature increases. In this configuration of the installation, the outdoor ambient is the sink of the cycle where the surplus heat is rejected. As the outdoor temperature rises, it is more difficult to reject this heat to the ambient. As a consequence, the thermal energy that doesn't go out of the chiller overheats the flows inside the machine. Finally, this situation leads to a worse performance of the machine that is shown on its COP value.

The Figure 6 indicates the evolution of the chiller power concerning to the ambient temperature. The simulation results show almost a perfect agreement with the experimental results of the year 2008. The trends of the three performances present a decrease in the delivered power of the chiller in the way the ambient temperature is increasing. This effect has a direct relation with the explained above. As it has been explained, if the outdoor temperature increases, the absorption chiller can not reject the heat correctly. As a consequence, the water leaving the condenser is hotter than it should be. When this water enters into the evaporator, its capacity to absorb heat is lower. As a result, the chilling effect decreases when the water flow in the evaporator increases its temperature.

Finally, the results of the mean values of the total cooling period in 2007, 2008 and the mean values of the results of the installation model also show a good agreement with an acceptable deviation. These mean values are shown in the Table 4.

### **b. New heat rejection system**

During this first two years, the installation worked with the initial dry cooler tower. The studies have shown the great influence of the temperature of the heat rejection sink on the machine performance. Therefore, the substitution of the initial heat rejection system was proposed in order to improve the performance of the absorption chiller. Two options were proposed, a wet cooling tower and a geothermal system. With both of them, the performance of the installation would improve but with the first alternative, the replacement of the dry cooler with a wet cooling tower was ruled out because its high requirements of maintenance. Besides that, due to there is a water reservoir close to the sport centre building, finally, it was decided to use a geothermal system using this water well, because the chiller will operate with a constant temperature in the heat rejection sink. This temperature, according to others water wells placed close was around 17 °C. This water well supply water to a 25 m<sup>3</sup> water tank, in which the heat produced in the absorption cycle, will be rejected. The tank is used to irrigate the sport grounds placed in the surroundings of the solar cooling installation in summer. This use means the contained water of the tank is renewed every day, so, daily, the water temperature will be constant, resolving the possible problems of thermal saturation in the tank.

In order to estimate the improvement of the performance of the chiller with the new heat rejection conditions, the TRNSYS model presented previously was used. In Figure 7 it can see the evolution of COP and chilling power in function of the cooling power rejected operating with the dry cooling tower. As it can be seen, with the new



operational conditions, the COP would achieve a value around 0.72 and the chilling capacity would be 7.6 kW. That represents a theoretical improve of a 42% in the COP respect the mean value in 2008.

The scheme of the geothermal installation can be seen in Figure 8. It is connected to the cooling loop of the absorption chiller by means of a 20 kW plate heat exchanger. There is a by-pass that leads the water to the plate heat exchanger avoiding the dry cooling tower.

Another feature of the geothermal system is the following. The overall length of the new circuit is 190.5 m, of which 90.5 m have been divided into three pipes with a diameter smaller than the rest of the circuit. This has been done in order to increase the heat exchange surface between the pipes and the ground. Hence the rejection of the heat generated by the absorption machine will take place in two places sinks: the water tank and the geothermal horizontal exchanger.

## **7. Conclusions**

A dynamic simulation model for a solar powered absorption cooling system was developed and validated by measured data. The simulation results show a good agreement with the measurements. Thus the model will be used for the evaluation of further investigations and improvements in the installation. In other terms, this work

confirms that the dynamic simulation provides very useful help to design and develop these systems.

After the experimental and simulation evidences of the influence of the cooling water temperature in the performance and in the COP of the absorption chiller, several options have been analyzed in order to improve the efficiency of the installation. Finally a geothermal cooling system has been studied and installed in the initial solar cooling system explained in this paper. The TRNSYS model of the installation, described previously was used to estimate the new performance of the chiller working with the water tank as a heat rejection sink.

The installation with the new variation starts to work the present year 2009. Preliminary studies have shown that with the geothermal cooling system, the COP of the chiller can be improved up to 42% over than the current one [25, 26].

## ACKNOWLEDGMENTS

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## APPENDIX

### Nomenclature

$A$	solar collector area ( $\text{m}^2$ )
$COP$	coefficient of performance (-)
$C$	specific heat capacity of Tyfocor HTL ( $3.65 \text{ kJ kg}^{-1} \text{ K}^{-1}$ )
$c_p$	specific heat capacity of water ( $4.18 \text{ kJ kg}^{-1} \text{ K}^{-1}$ )
$I$	irradiation on the collector surface ( $\text{W m}^{-2}$ )
$N_c$	collectors number (-)
$m$	flow rate ( $\text{l min}^{-1}$ )
$SCOP$	Solar COP (-)
$T$	temperature ( $^{\circ}\text{C}$ )
$UA$	overall loss coefficient ( $\text{W K}^{-1}$ )
$W$	power (kW)
$\eta$	efficiency (-)

### Subscripts

$c$	cooling
$ch$	chilling
$dbo$	dry bulb outdoor
$e$	electricity
$ev$	evaporator
$g$	generator
$he$	finned tube heat exchanger

i	inlet
o	outlet
s	solar
sc	solar collector

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## FIGURE CAPTIONS

Figure 1. Scheme of the solar absorption cooling installation.

Figure 2. Trnsys model of the installation.

Figure 3. Measured and simulated absorption chiller temperatures.

Figure 4. Temporary values vs stationary values. Stationary Graphics.

Figure 5. COP vs  $T_{db0}$ .

Figure 6.  $W_{ch}$  vs  $T_{db0}$ .

Figure 7. Estimation of the COP and chilling power in the new dissipation system.

Figure 8. Scheme of the new dissipation system.

## TABLES

Parameter	Value
$\eta_0$	0.850
$a_1(\frac{W}{m^2 \cdot K})$	4.07
$a_2(\frac{W}{m^2 \cdot K})$	0.0070

Table 1. Experimental mean values of the installation in the years 2007 and 2008.

Year	$T_{\text{dbo}}$ (°C)	$T_{\text{gi}}$ (°C)	$T_{\text{ev,o}}$ (°C)	$W_{\text{ch}}$ (kW)	$COP$
2007	27.7	91.0	11.5	5.7	0.57
2008	31.2	92.5	11.5	4.4	0.51

Table 2. Experimental mean values of the installation in the years 2007 and 2008.

		$T_{\text{dbo}}$ (°C)	$W_{\text{ch}}$ (kW)	$W_{\text{c}}$ (kW)	$W_{\text{g}}$ (kW)	COP
<b>11/7/2008</b>	Measured Data	35	4.02	12	7.9	0.48
	Simulated Data	35	4.5	13.5	8.7	0.52
	Dev.(%)	-	10.7%	11.1%	9.2%	7.7%
<b>29/7/2008</b>	Measured Data	30.4	4.5	12.8	8.2	0.53
	Simulated Data	30.4	5.1	13.8	8.5	0.58
	Dev.(%)	-	11.8%	7.2%	3.5%	8.6%
<b>8/8/2008</b>	Measured Data	29.4	4.8	13.3	8.7	0.53
	Simulated Data	29.4	5.1	13.8	8.5	0.58
	Dev.(%)	-	5.9%	3.6%	2.4%	8.6%

Table 3. Summary of the measured and simulated results for several days.

Year	$T_{\text{dbo}} (^{\circ}\text{C})$	$T_{\text{g,i}} (^{\circ}\text{C})$	$T_{\text{ev,o}} (^{\circ}\text{C})$	$W_{\text{ch}} (\text{kW})$	$\text{COP}$
2007	27.7	91.0	11.5	5.7	0.57
2008	31.2	92.5	11.5	4.4	0.51
Model	31.2	91.3	11.3	5.5	0.60

Table 4. Comparison of the measured and simulated results.

1. Solar collector field.
2. Hot water tank.
3. Absorption chiller.
4. Finned tube heat exchanger and fan.
5. Fancoils.
6. Auxiliary boiler.
7. Exchanger.

