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Experimental analysis of a solar absorption system with interior energy storage

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Abstract
This study examines experimentally the cooling application of a solar absorption system with interior energy storage that uses two different auxiliary systems. The experiments were performed at Uludag University, Bursa, Turkey on the 3rd and 4th of August 2010 that had the approximately same average outdoor temperature, 31°C. A solar hot water was delivered via a 40 m2 array of flat plate solar collectors that drove a lithium chloride (LiCl) absorption heat pump with a cooling power peak of 20 kW. A solar-powered air conditioning system was designed for heating and cooling in a test room that had a total floor space of 30 m2. Chilled water produced in the evaporator was supplied to the fan coil units, and the heat of condensation and absorption was rejected by means of a wet cooling tower. An electric heater and an air source heat pump were used as auxiliary systems for the absorption cooling application for two different cases when the solar energy was insufficient. Temperature variations were recorded for the absorption machine components, the test room, and the outdoors. The cooling energy, thermal energy, and daily average coefficient of performance (COP) of the absorption system were calculated for two days. Solar absorption cooling was considered for two different auxiliary systems and is presented in this manuscript. The results showed that the daily average COP of the absorption system was 0.283 for Case 1 and 0.282 for Case 2. For both cases, the interior energy storage of the absorption system enabled it to satisfy the cooling demand during the night while solar energy was not available.

Keywords: solar energy, absorption system, heat storage, solar cooling

Introduction

Many problems have arisen from the increased usage and demand of energy; therefore, the use of clean and renewable energy sources is becoming more important. Because the energy requirements for heating and cooling applications account for a large share of global energy consumption, renewable energy sources appear to be an appropriate solution for addressing the existing problems. Currently, several solar-driven systems exist for heating and cooling buildings. The operating costs of these systems are low, and they are driven by clean and renewable energy sources. Energy consumption and carbon emissions during the summer season rise due to the cooling demand, and this causes many technical and environmental problems. Thus, solar cooling applications have become more attractive. Hidalgo et al., (2008) developed an experimental facility with a 50 m^2 flat plate solar thermal collector and fully monitored for housing an air conditioning application using Li-Br/H2O absorption technology. The experimental results indicated that, satisfying 100% of the housing air conditioning demand (56% by solar energy and 44% with a natural gas boiler), the energy cost would be 62% and the CO2- emission savings would be 36% respectively. The International Energy Agency Task 25 "Solar Assisted Air Conditioning of Buildings" (IAE, 1999), and Task 38, "Solar Air Conditioning and Refrigeration" (IAE, 2006), published several reports regarding solar assisted cooling and heating applications. Models provided by simulation tools (Mateus and Oliveira, 2009; Casals, 2006) and design guidelines (SEAL, 2005a, b; Henning, 2007) help to design more reliable systems. However, solar energy is affected by weather conditions and seasonal changes and can be harnessed only during clear sky hours. Therefore, to utilize solar-assisted heating and cooling applications, energy must be stored, or auxiliary systems must be employed. Because the first cost of the system components is high and energy must be stored, the total cost of the system is higher than that of alternative approaches (Li and Sumathy, 2000) and return on investment is prolonged. As a consequence, solar-assisted systems are not competitive with conventional heating and cooling systems.

Solar absorption heat pumps have high system efficiencies, low operating costs, and can be used for both heating and cooling; therefore, they are more promising than other solar applications. Sanjuan *et al.,* (2010) reported that generally, the coefficient of performance (COP) is in the range of 0.6-0.7 for single-effect absorption machines and in the range of 1.2-1.5 for double-effect absorption machines. They also reported that absorption cooling systems require thermal energy supply, which means a solar collector field if looking for renewable energy systems that can also provide daily hot water and heating energy. The efficiency of solar absorption cooling systems depends not only on the COP of the chillers, but also on the efficiency of the solar collector field, the losses at the distribution systems and the profiles of the cooling loads (Sanjuan *et al.,* 2010).

Many authors have worked on absorption systems and the system performance. Karamangil *et al.* (2010) reviewed the studies related to absorption refrigeration systems using various refrigerant-absorbent pairs, which focused on cycle designs and an appropriate working fluid selection. Syed *et al.* (2005) worked on a solar-driven single-effect LiBr/H_2O absorption system of 35 kW nominal cooling capacity. They reported that the maximum instantaneous, daily average and period average COP were 0.60 (at maximum capacity), 0.42, and 0.34, respectively. Izquerdo *et al.,* (2008) carried out an experimental study to determine the performance of a commercial (Rotartica 045v) 4.5 kW air-cooled, single-effect LiBr/H_2O absorption chiller for residential use in Madrid. Three typical August days with different outdoor temperatures were chosen for the study. They reported that the average COP for the period, when auxiliary equipment was included into the calculations, was 0.37. The solar absorption system requires energy storage. Löf and Tybout (1974) reported that the optimum storage volume is about 50 kg/m^2 of collector area. Kreider and Kreith (1981) suggested that the nominal storage amounts for cooling purposes range from 80 kg/m^2 of collector area to 200 kg/m^2. The main problem resulting from hot water storage is the heat loss that occurs during the process. The temperature difference between the surrounding area and the hot water storage tank is high, which increases the heat loss. However, external hot water storage systems increase the first cost of the solar absorption systems and require large amounts of space. At this point, new generation solar absorption heat pump systems with interior heat storage address the problems related to energy storage. Since absorption systems with interior heat storage have become commercially available (ClimateWell, 2011), significant improvements have been made in solar cooling and heating applications for buildings.

Soutullo *et al.* (2010) presented a comparative study of the performance of absorption cooling systems with internal storage and with external storage. A full dynamic simulation model including the solar collector field, the absorption heat pump system and the building loads was performed. They reported that the comparison between both absorption chillers indicates that, in order to reach similar values of storage energy, a conventional system has a greater room requirement than four units with internal storage working in parallel, requiring an external water tank of at least 15 m^3. Sanjuan *et al.* (2010) focused on the optimization of the performance of a solar absorption cooling system composed of four units with interior energy storage. They developed a full dynamic simulation model that includes the solar collector field, the absorption heat pump system, and the building load calculation. Their study indicated how strong the influence of the control strategies in the overall performance is, and the importance of using hourly simulations models when looking for highly efficient buildings.

This paper presents the experimental results of the implementation of a cooling application with a solar absorption system with interior energy storage that uses two different auxiliary systems. The aim of the study is to show the effect of the parameters on the system performance and to compare the two different auxiliary systems that were driven by electric power.

Methods

The experimental facility is set up to investigate solar-powered heating and cooling of a test room

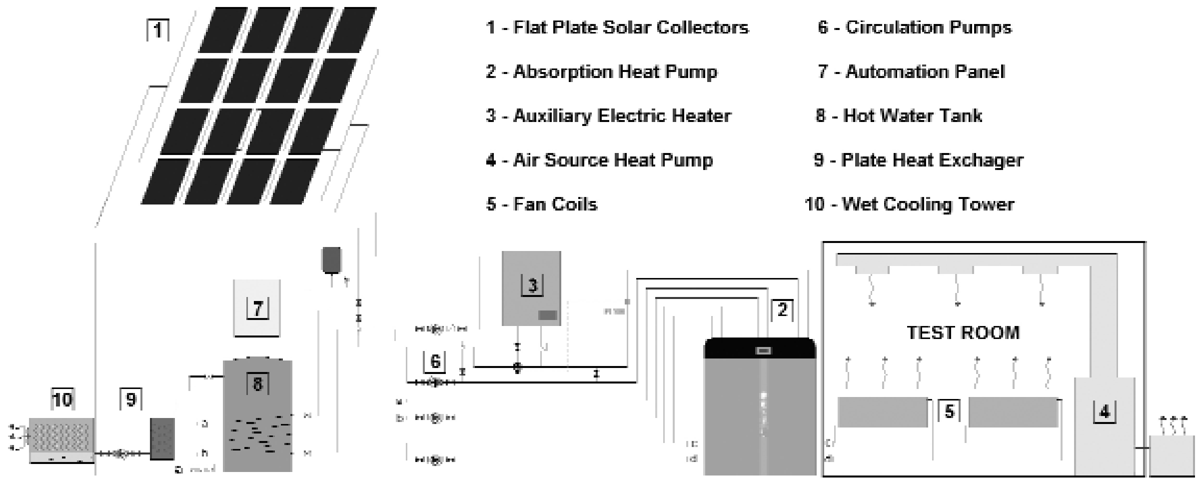

Figure 1: Schematic diagram of the experimental facility

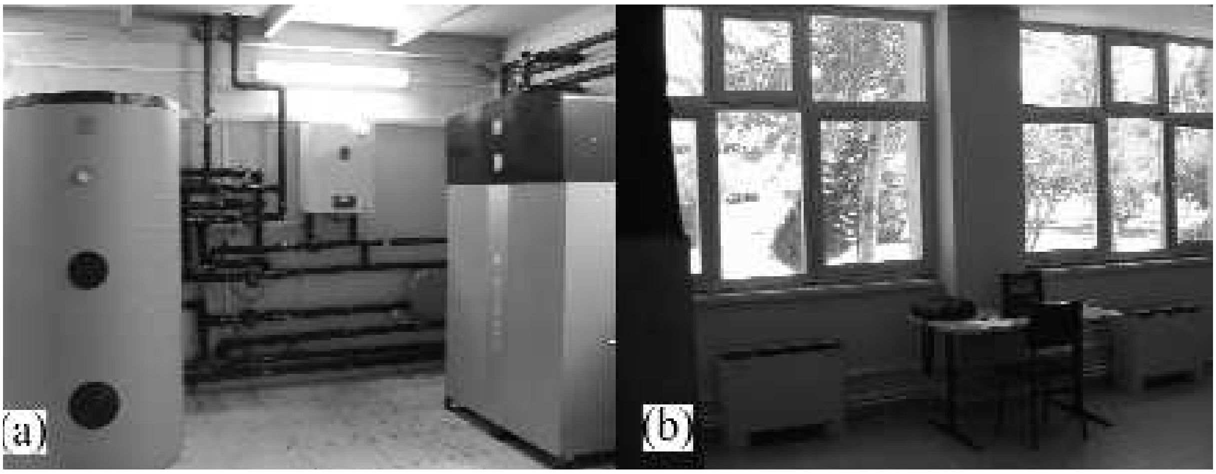

Figure 2: Absorption system (a) and test room (b)

with two different auxiliary systems at Uludag University in Bursa, Turkey, using commercially available components. Figure 1 shows the schematic diagram of the experimental facility. It has been designed to operate in a flexible way so that various experiments can be reproduced. The test room has a net floor space of 30 m^2 and a volume of 84 m^3. The room has a 9m^2 double-glazed window that faces south-east. The internal faces of the test room walls are thermally insulated with polystyrene foam. The cooling load of the room has been calculated as 4.2 kW, using a conventional methodology and local data (Yamankaradeniz *et. al.,* 2008). The distribution system consists of two fan coil units, each of which has a cooling capacity of a 2.2 kW. The test room has an internal auxiliary system that consists of an air source heat pump (ASHP) that has a 7 kW nominal cooling capacity and an air duct distribution. Figure 2 shows a picture of the test room and absorption system.

The solar absorption system comprised the commercially available ClimateWell SolarChiller (CW 20 model) with a cooling power peak of 20 kW (ClimateWell, 2009). A small-capacity absorption system with interior energy storage is preferred for taking advantage of a smaller installation area and for reducing the first cost of the system. This system is different from conventional absorption heat pumps in some important aspects. The machine consists of three parts: two twin barrels and one plumbing unit connecting both of the barrels to external circuits. The machine is connected to three external circuits: the thermal energy supply, the heat sink, and the cooling/heating distribution. The barrels (Barrel A and Barrel B) work independently, whether they are charging energy into the salt functioning only as a storage tank, or discharging the energy stored in the salt form of cooling energy. Swapping the barrels between charge and discharge can be controlled manually or automatically by the various modes of the machine (ClimateWell, 2009).

Figure 3 shows the working principle of the absorption system. Each barrel has two separate bowls: one that is filled with the salt (reactor-absorber) and another that is filled with water (evaporator-condenser). The salt is lithium chloride (LiCl), which never changes its position during the

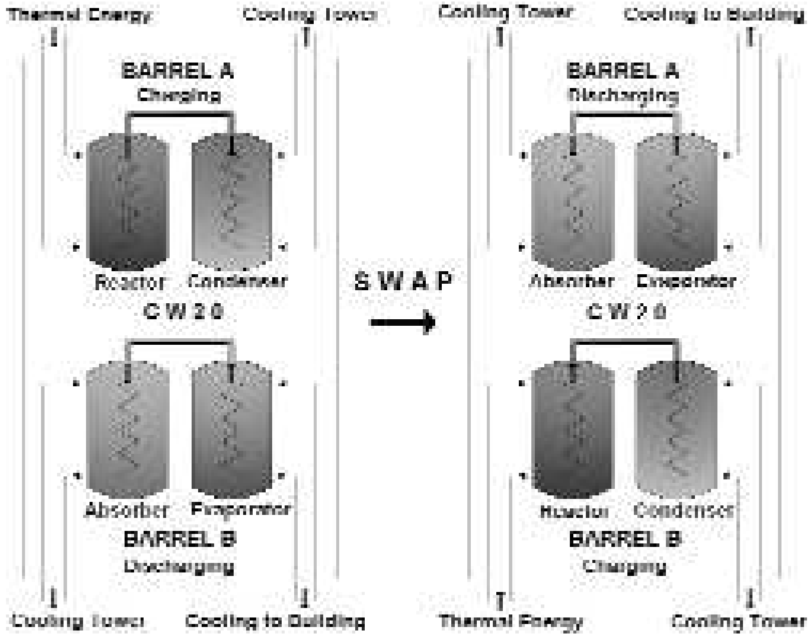

Figure 3: Working principle of the absorption system

process. During the charging process, the LiCl salt in the reactor is dried by the thermal energy, and the water returns to the condenser in Barrel A. The heat of condensation in the condenser is rejected by the heat rejection system. During the discharge process, the salt in the absorber absorbs the water vapour from the evaporator, which provides cooling at the necessary locations by the distribution system. The heat rejection system enables the absorber to reject the heat of absorption. When the barrels are swapped, Barrel A switches from charge mode to discharge mode and provides cooling energy, while Barrel B switches from discharge mode to charge mode and stores energy.

The solar collector field, which is the main heat source of the absorption system, consists of 16 flat plate solar collectors with titanium-coated selective absorber areas of 2.5 m^2. The solar collector system is arranged in four rows that are connected in parallel. Each row consists of four solar collectors that are connected in series (Figure 4). The solar collector system is mounted on the roof of the test room and faces southeast at a 30° tilt angle. The heat of condensation and absorption in the absorption system are transferred to a wet cooling tower of nominal capacity 35 kW via a plate heat exchanger. The operating fluid in the heat source, the distribution, and the heat rejection circuit are a mixture of 15 % propylene glycol and water and are circulated by three-stage centrifugal pumps. Within the wet cooling tower circuit, the operating fluid is water, and it is circulated by a single-stage centrifugal pump. All of the pumps in the system are controlled by an automation panel.

An electric heater with 9.6 kW (Figure 1, Component No. 3) is used as the auxiliary heat source when the solar energy is insufficient for the absorption system. A PT 100 sensor is positioned at the absorption machine inlet, and it measures the temperature in order to control the electric heater. The operating temperature of the electric heater can be determined and controlled from the automation panel.

The solar collector, fan-coil, test room and outdoor temperatures were measured and recorded by thermometers with the data logger. Absorption system temperatures (reactor, absorber, condenser and evaporator temperatures) and flow rates (water, solution) were measured and recorded by the internal data logger devices of the absorption machine. Measurement devices are given Table 1. In order to determine the value of the experimental error, an uncertainty analysis is carried out using equations proposed by Moffat (1998). Maximum uncertainties in experimental results were found to be with in ±1%.

Figure 4: Solar collector system

Table 1: Measurement devices

Measured quantity	*Measurement device*	*Measurement range*	*Accuracy*
Solar collector, test room, fan coil temperatures	Testo177, Thermocouple (K type)	(-200) – (+1000) °C	±0.5%
Outdoor temperature	Testo 454/350 with 0635 1047 Probe	(-20) – (+70) °C	±0.5%
Absorption system temperatures (reactor, absorber, condenser, evaporator)	CW 20 Internal Thermometer NTC Sensor	(0) – (+120) °C	±0.5%
Absorption system flow rates (water, solution)	CW 20 Internal Flowmeter Grundfos VFS	2– 40 l/s	±1.5%

Experimental results and discussion

This study was performed to investigate a solar absorption cooling application that utilizes two different auxiliary systems when solar power is insufficient. Experiments were performed on two different August days (3rd August 2010 for Case 1 and 4th August 2010 for Case 2), with approximately the same average outdoor temperature, 31°C. The temperature of the test room ranged from 25 to 27°C. The cooling application began when the indoor temperature exceeded 27°C and ended when the outdoor temperature was equal to that of the test room.

The experiment consisted of two cases: Case 1 and Case 2. For Case 1, the absorption machine was fired with an auxiliary electric heater between 07:30 and 09:00, which was when cooling was required, but also when the solar power was insufficient for driving the absorption system. The entire cooling demand of the test room was satisfied by the absorption system. For Case 2, under the same indoor conditions, an ASHP was used as an auxiliary system in order to support the cooling demand of the test room between 09:05 and 11:05.

The following results have been recorded for the two cases.

Case 1

For Case 1, the absorption system achieved 11.7 kWh of cooling energy during the day. The fan coil systems were controlled by the room thermostat in order to distribute cooling and stabilize the room temperature between 25 and 27°C. The barrels of the absorption machine were swapped three times at 09:05, 11:45, and 15:00. Swaps were performed manually once the discharging Barrel was incapable of cooling and thus could not satisfy the existing cooling demand. Figure 5 shows the cooling energy for the test room within a daily period. As a result of the interior energy storage, the absorption system satisfied the cooling demand during the night while solar energy was not available. Figure 6 shows the thermal energy that was inputted into the reactor of the absorption machine.

The auxiliary electric heater achieved 8.105 kWh of thermal energy from 07:30 to 09:00. This amount of energy was equal to 20.718% of the total thermal energy input of 39.120 kWh. Temperature of the hot water supplied by the solar collectors was appropriate for charging the reactor at 09:25. Solar power was used for driving the absorption system until 16:15 and supplied 31.020 kWh of thermal energy. The results showed that during the midday, the peak values of the thermal energy resulted from energy delivered from the solar system, and the peak values of the cooling energy were also reached. After 16:15, the temperature of the hot water from the solar collectors decreased, and the solar power was insufficient to drive the absorption system. The solar power was used for a domestic hot water system that consisted of a 1000-L hot water vessel with a spiral tube heat exchanger (Figure 1, Component 8.).

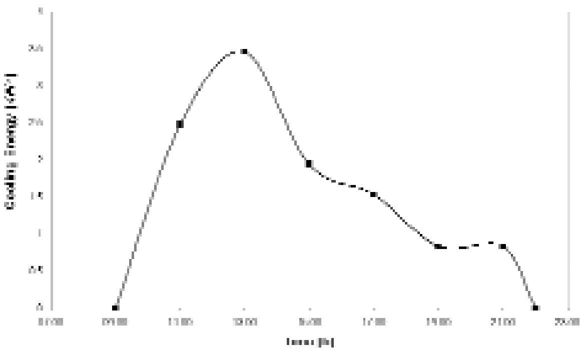

Figure 5: Case 1 – Cooling energy

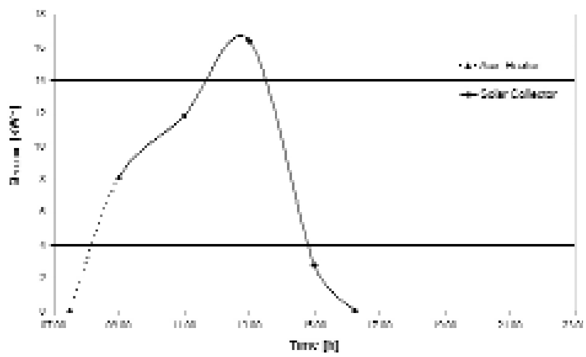

Figure 6: Case 1 – Thermal energy input into the reactor

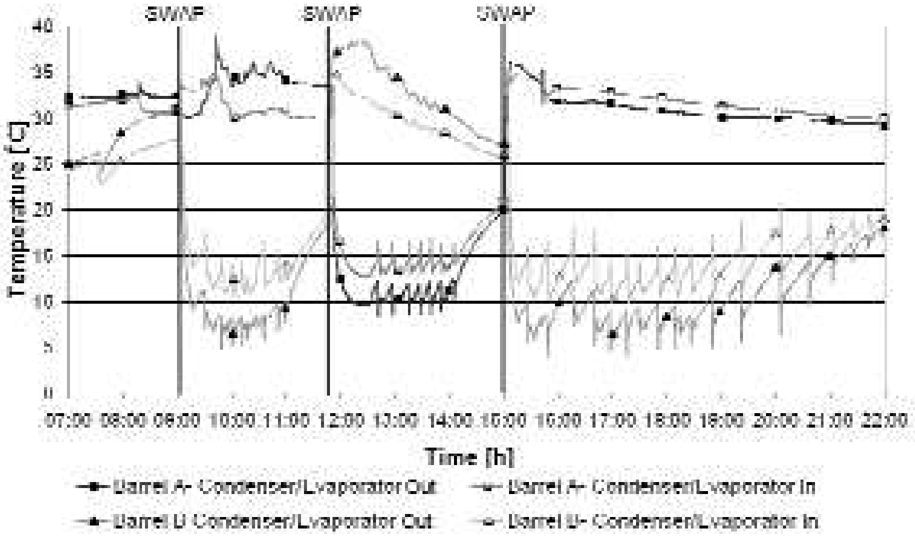

Figure 7: Case 1 – Condenser-evaporator temperatures

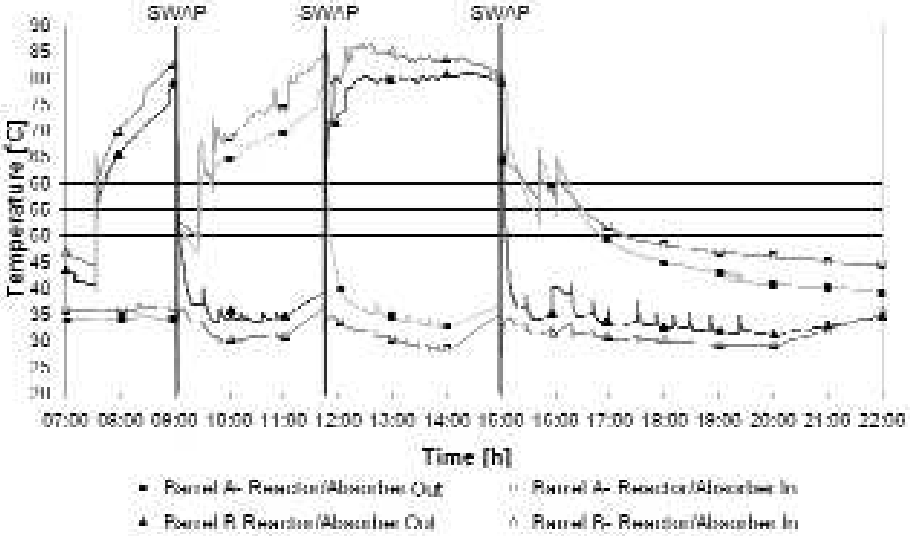

Figure 8: Case 1 – Reactor-absorber temperatures

Figures 7 and 8 show condenser-evaporator and reactor-absorber temperatures for Case 1, respectively.

The condenser temperature varied between 30 and 35°C during the cooling period; the absorber temperature varied within the same range. The evaporator temperature was determined as 7 - 12°C. The reactor inlet temperature was rapidly increased from 07:30 to 09:00 by an electric heater and exceeded 80°C. Barrel B was charged via auxiliary energy when the solar power was insufficient for charging. At 09:05, there was a cooling demand, and Barrel B began to discharge once the barrels were swapped. The absorption system was turned on for cooling, and the auxiliary heater was turned off. The temperature of the hot water from the solar collectors was appropriate at 09:25, and the absorption system was fired by the solar energy. The reactor inlet temperature exceeded 85°C during the midday. Figures 9 and 10 show solar collector and fan-coil temperatures for Case 1, respectively. Figure 11 shows the test room and outdoor temperatures during the cooling period. This figure indicates that the cooling demand of the test room was satisfied by the absorption system and that the test room temperature was maintained at desired values during the night, as well.

The COP for the absorption system is the ratio of the cooling energy ($Q_{evaporator}$) provided to the thermal energy ($Q_{reactor}$) supplied and can be calculated according to the following equation:

$$COP = \frac{Q_{evaporator}}{Q_{reactor}} \tag{1}$$

Hence, the COP of the absorption system for Case 1 is:

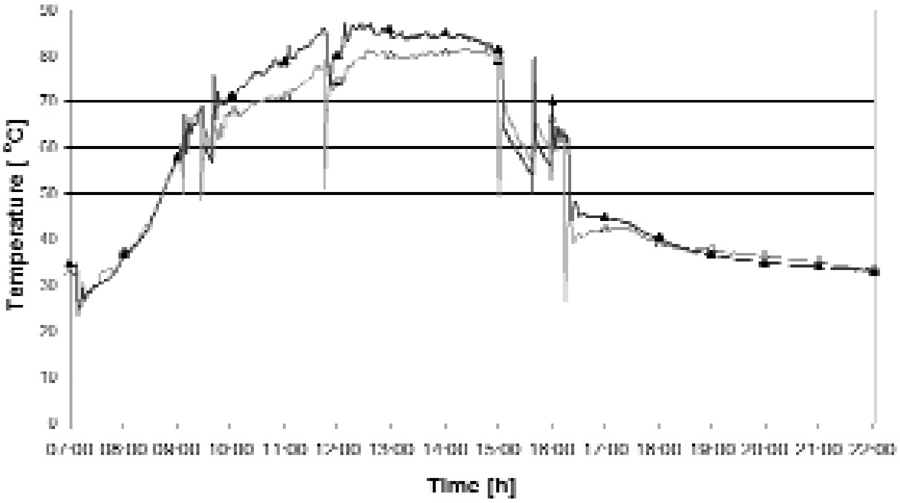

Figure 9: Case 1 – Solar collector temperatures

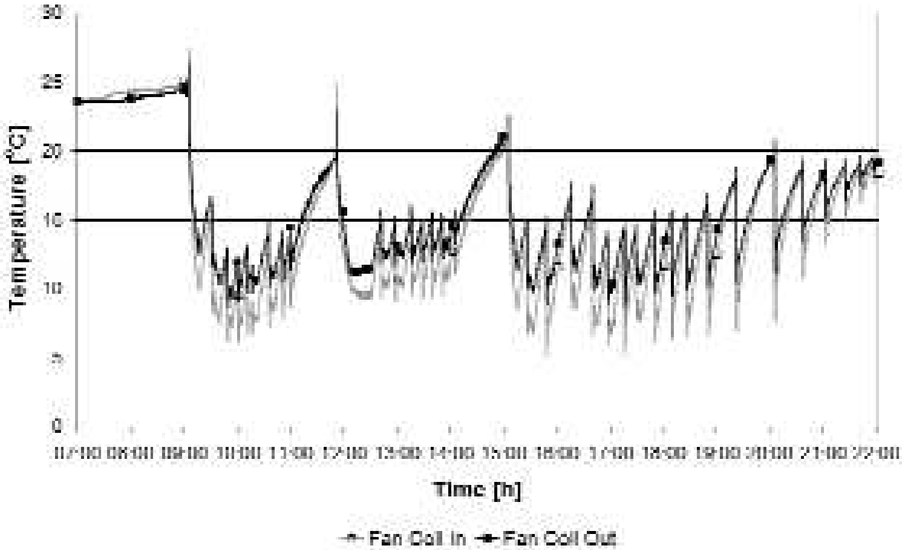

Figure 10: Case 1 – Fan coil temperatures

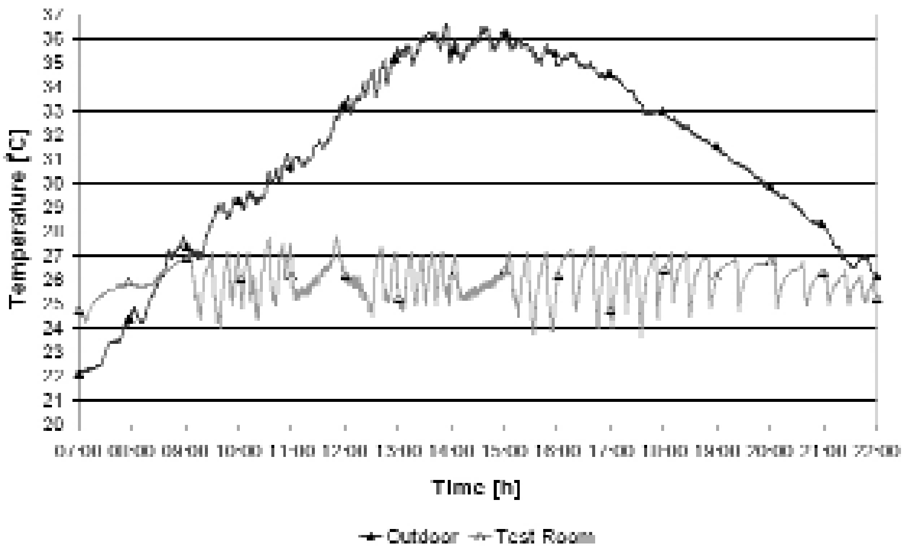

Figure 11: Case 1 – Test room and outdoor temperatures

$$\text{COP} = \frac{11.079}{39.130} = 0.283$$

Case 2

For Case 2, the ASHP with an air duct distribution system was used as an auxiliary system to satisfy the cooling demand between 09:05 and 11:05. The ASHP was controlled by the thermostat, and the electric power consumption for the ASHP system was recorded as 0.116 kWh. Because the solar power was sufficient, the absorption system was turned on between 11:05 and 22:00, and satisfied the cooling demand by using solar power. The absorption system achieved 10.190 kWh of cooling energy within the test room during the day (Figure 12). The absorption machine was manually swapped three times at 11:05, 13:15, and 16:30.

Figure 13 shows the thermal energy that was inputted into the reactor, which was all obtained by solar power. Solar power was used for driving the absorption system between 09:30 and 16:30. After 16:30, it was used for the domestic hot water system, just as in Case 1.

Figures 14 and 15 show the condenser-evaporator and reactor-absorber temperatures for Case 2, respectively.

Figures 16 and 17 show solar collector and fan-coil temperatures for Case 2, respectively. The test room and outdoor temperatures for Case 2 are shown in Figure 18. This figure shows that the cooling demand of the test room was satisfied by the ASHP during the day, while the solar power was insufficient to drive the absorption system. During the night, the absorption system satisfied the cooling demand by using the stored energy within its barrels.

For Case 2, 36.120 kWh of thermal energy was supplied completely by solar energy and inputted into the rector; the absorption system achieved 10.190 kWh of cooling energy for Case 2. The COP of the absorption system for Case 2 is:

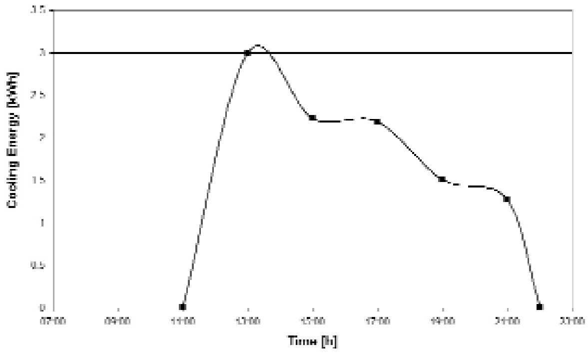

Figure 12: Case 2 – Cooling energy

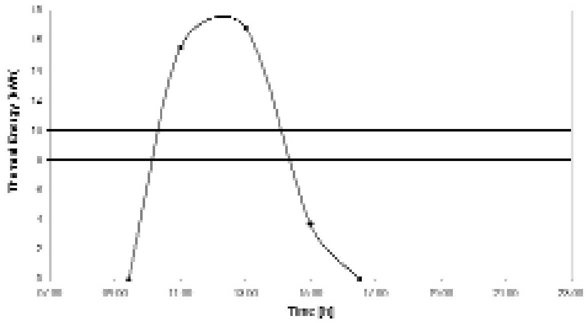

Figure 13: Case 2 – Thermal energy inputted into the reactor

$$\text{COP} = \frac{10.190}{36.120} = 0.282$$

Experimental studies were performed on two different August days with approximately the same

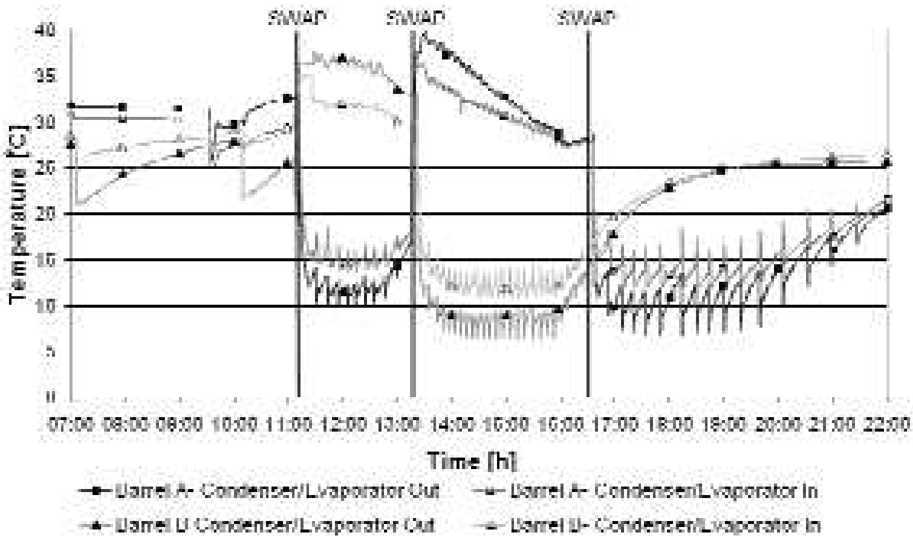

Figure 14: Case 2 – Condenser-evaporator temperatures

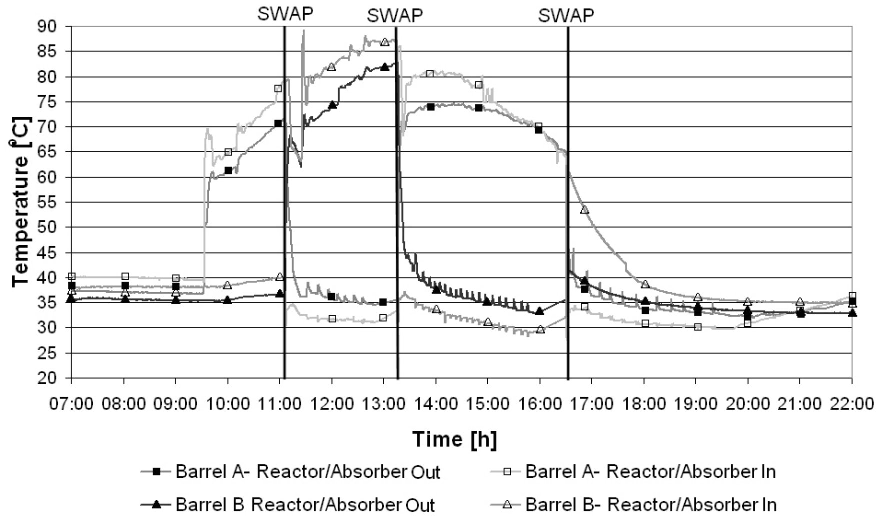

Figure 15: Case 2 – Reactor-absorber temperatures

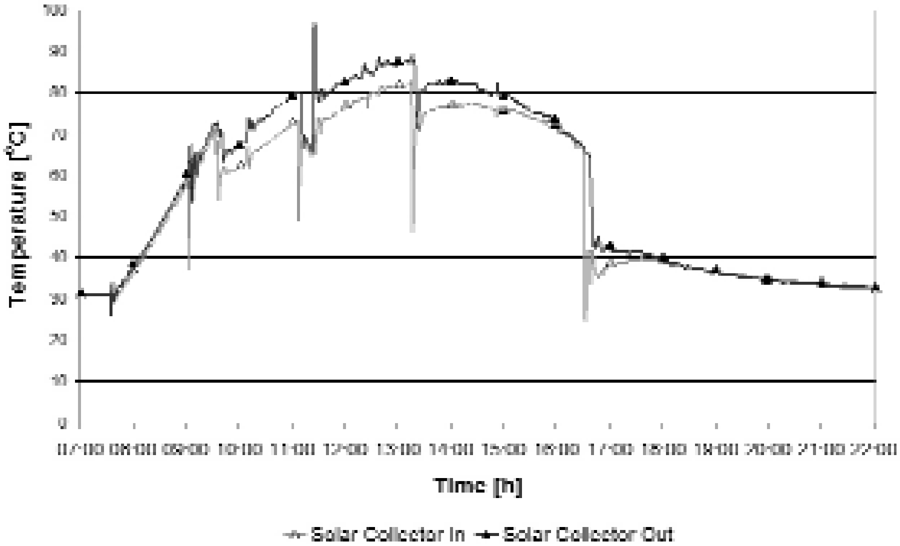

Figure 16: Case 2 – Solar collector temperatures

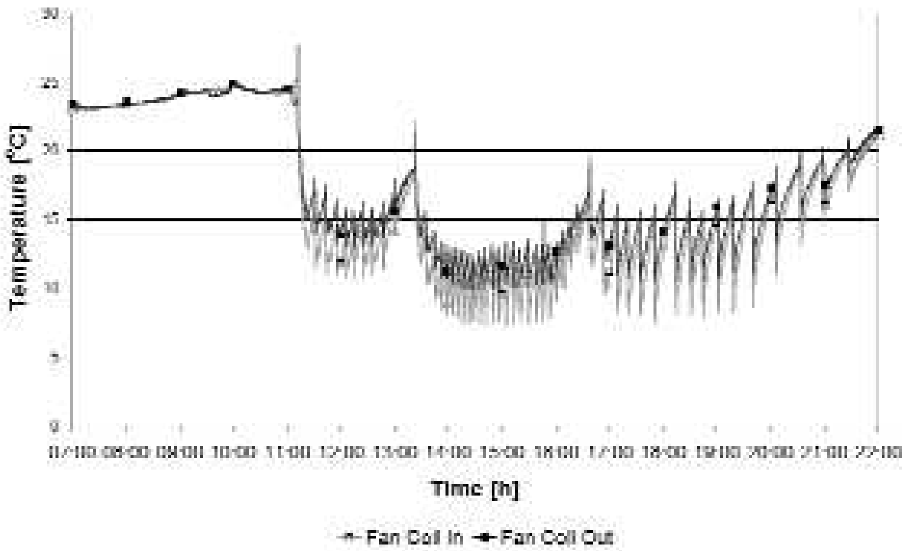

Figure 17: Case 2 – Fan coil temperatures

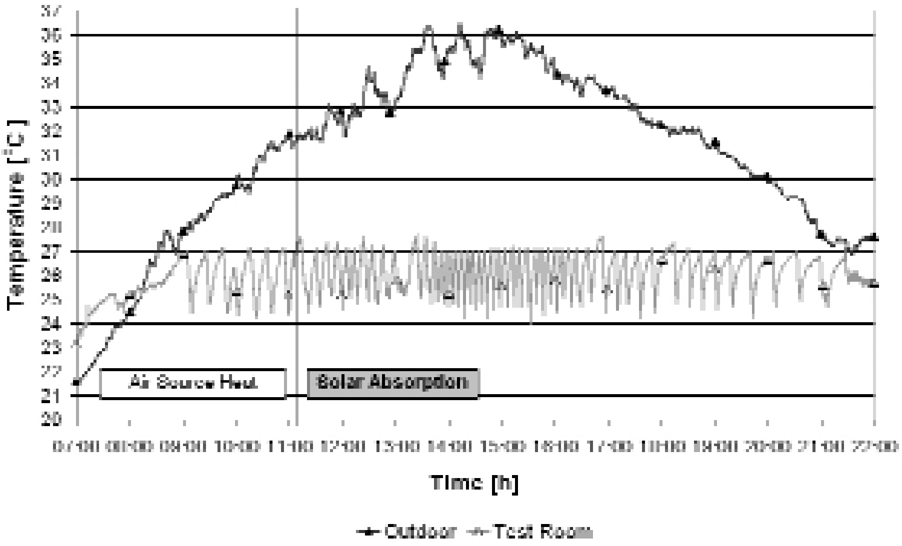

Figure 18: Case 2 – Test room and outdoor temperatures

outdoor temperatures, with two different systems driven by electric power used as auxiliaries for the absorption system. The COP values for the absorption system were calculated and the results were almost same for both cases. Because the walls of the test room were thermally insulated along their interior faces and solar heat gain is high, fan coils were operated intermittently and very frequently (Figures 10 and 17). Therefore, the efficiency of the distribution system was affected negatively, and the COP of the absorption system decreased.

For water-cooled air conditioning systems, Li and Sumathy (2000) reported that from the point of view of improving the COP, it is better to use cooling water of low temperature (below 25°C) when the normal cooling water temperature is about 25 - 32°C. For both cases, the absorber and condenser temperatures varied within the range of 30 - 35°C (Figures 7, 8, 12 and 13). The COP of the chiller decreases with decreasing evaporation temperature. Therefore, some have suggested that the chilled water temperature should be maintained above 5 - 7 °C (Li and Sumathy, 2000). The manufacturer of the absorption machine, ClimateWell A.B (ClimateWell, 2008) has also reported that the absorption unit is designed to work with high temperature systems ranging from 10 - 16°C at the outlet of the absorption unit and that performance rapidly decreases below this range (ClimateWell, 2009). Because the distribution system of the experimental facility consisted of fan-coils, the evaporator temperature was determined as 7 - 12°C, which resulted in lower COP values during the day. This result indicates that the determination of the components that work with the absorption system affects the system performance.

For Case 1, the total cooling demand was satisfied by the absorption system. An electric heater was used as an auxiliary system during the day while the solar energy was insufficient. Using an electric heater as an auxiliary system is more cost effective than using an ASHP. Because the absorption system can be used as a heat pump, it can be used during the heating season along with an auxiliary system. The electric heater can be used as a boiler for heating and supplying domestic hot water, and it does not require an extra distribution system. Hence, choosing the electric heater as an auxiliary system decreases the payback time of the solar absorption system.

For Case 2, an ASHP was used as an auxiliary system during the period in which the absorption system was incapable of cooling. An ASHP was used directly to satisfy the cooling demand, and it was operated intermittently via the thermostat control. Because the ASHP has high COP values in the range of 3-3,5 (Sanjuan *et al.,* 2010) and can be used for both heating and cooling the energy consumption of an ASHP is less than that of an electric heater when an ASHP is used as an auxiliary system for an absorption system. This means that the ASHP system has a lower operating cost than that of an electric heater; therefore, ASHP systems are more attractive than their alternatives. However, despite this cost advantage, the first cost of the ASHP is higher than that of the electric heater. An ASHP also requires an extra distribution system and cannot be used for supplying domestic hot water. Because it operates intermittently, its payback time as an auxiliary system is prolonged. However, the payback time of the solar absorption system is prolonged as well, because the absorption system does not work while the ASHP is being used.

Conclusion

Widespread use of solar absorption systems for cooling and heating applications is important to addressing global environmental and economic problems. These systems require auxiliary systems when solar energy is insufficient. This study presents the comparative results of an experimental study that examined a solar absorption cooling application with two different auxiliary systems that were driven by electric power. The advantages and disadvantages of the two systems were investigated. Consequently, the choice of the proper auxiliary system and the proper components for the existing conditions will play an essential role in the design of more efficient solar absorption systems.

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