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Thermal Analysis of an Absorption and Adsorption Cooling Chillers Using a Modulating Tempering Valve

Jesús Cerezo Román, Rosenberg Javier Romero Domínguez, Antonio Rodríguez Martínez and Pedro Soto Parra

Abstract

The energy consumption for space cooling is growing faster than for any other end use in buildings, more than tripling between 1990 and 2016. The efficient use of energy is important to reduce the consumption of electricity of conventional air conditioning. This chapter presents a thermal analysis of absorption and adsorption chillers for conditioning the airspace in a building, controlling the hot maximum temperature at the generator input with a modulating tempering valve (MTV) programmed in TRNSYS and Excel software. The energy performance of the system was maximized based on the tilt of the solar collector, storage tank specific volume, and input generator temperature. The results showed that 35 and 27 l/m² of specific volume is a good choice for absorption and adsorption chiller without MTV, and 23 and 22 l/m² were selected absorption and adsorption chillers using the MTV at a fixed tilt angle of 7° of the solar collector and selecting a minimum temperature at the generator input of 111 and 109°C for absorption chiller without and with MTV, respectively, and 75°C for adsorption chiller without and with MTV. The use of MTV represented a significant reduction of the heater energy for both chillers, mainly for absorption chiller.

Keywords: adsorption chiller, absorption chiller, modulating tempering valve, TRNSYS, air conditioner

1. Introduction

The use of air conditioner and electric fans in buildings around the world is nearly 20% of the total electricity. This consumption has a trend to grow up due to the demographic growth, and more people will naturally want to live in thermal comfort [1]. Solar cooling is a promising and clean alternative equipment which has the advantage for cooling demand in buildings [2]; however, the heat and mass transfer process between the refrigerant and sorption solution and the number of heat exchangers of this kind of equipment (condenser, evaporator, generator, and sorption) reduce the performance of the thermodynamic cycle. The use of new configurations to improve the efficiency of energy has been an important topic of study to reduce the consumption of fossil fuels [1].

The absorption and adsorption cooling system technologies are heat-activated based on the liquid and solid sorption process, respectively. The absorption is based on the absorption and desorption of a working fluid named refrigerant in an absorbent. The absorption cooling system (ACS) consists of four main components: generator, condenser, evaporator, and absorber components as shown in **Figure 1**. The thermodynamic cycle is described as follows: Heat energy is added to the generator to vaporize the refrigerant from the strong solution (high absorbent concentration). The vaporized refrigerant goes to the condenser where it is condensed delivering an amount of heat (Q_{COOLING}). The refrigerant leaving the condenser flows through an expansion valve to reduce the pressure and goes to the evaporator; the refrigerant absorbs the heat of the room ($Q_{\text{AIR CONDITIONING}}$) vaporizes producing the cooling effect. Then, the generated vapor goes to the absorber where it is absorbed by the poor solution of the absorbent coming from the generator, delivering heat (Q_{COOLING}), which is dissipated to the ambient to keep the absorption process at a desirable temperature. Finally, the mixture refrigerant-absorbent is pumped to the generator to restart the cycle.

The adsorption system has similar components of the absorption system. However, the cooling effect can be carried out in separate two basic phases (**Figure 2**). In the first phase when the heat is removed ($Q_{\text{AIR CONDITIONING}}$) from the chilled water, the refrigerant vapor leaves the evaporator and enters the adsorber, where the refrigerant vapor is adsorbed in the adsorbent bed. The second phase is the desorption process as the result of heating up (Q_{HEATING}) the adsorbent bed to release adsorbate (refrigerant) from it. The vapor moves to the condenser, and after throttling liquid refrigerant

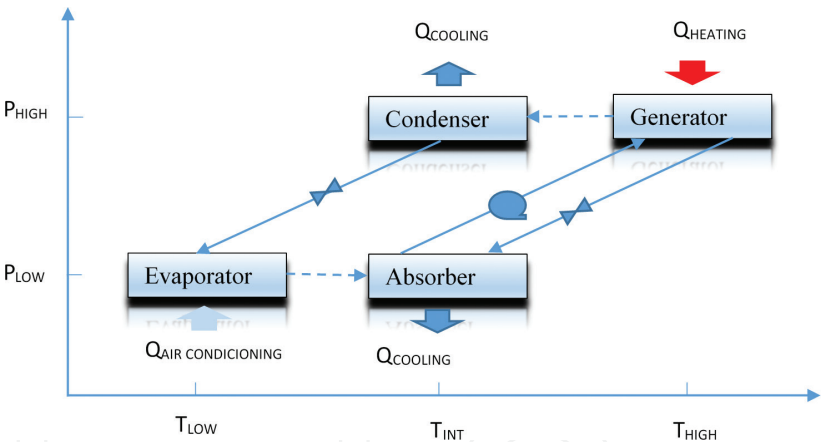


Figure 1.
Schematic diagram of a single-stage absorption cooling system.

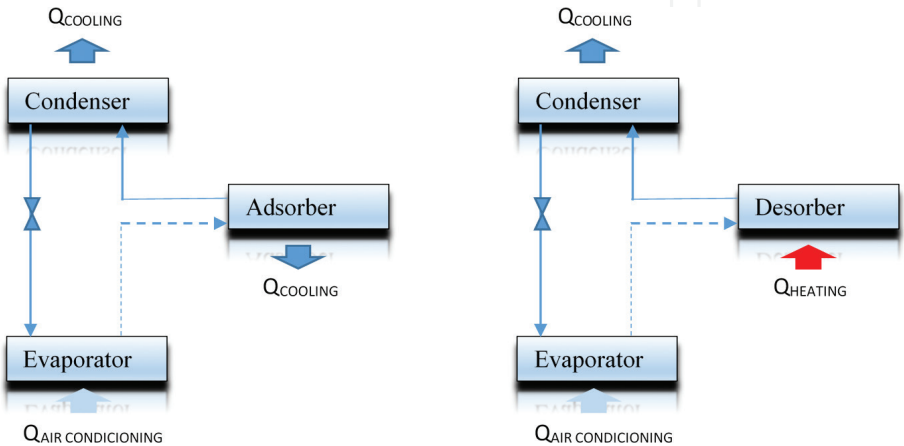


Figure 2.
Adsorption and desorption phases of an adsorption cycle.

	Absorption system	Adsorption system
Generator temperature, °C	85–180	50–100
Coefficient of performance (COP)	0.6–1.1	0.4–0.6
Working fluid	H ₂ O–LiBr or NH ₃ –H ₂ O	H ₂ O silica gel

Table 1.
Typical operation conditions for the sorption systems.

flows to the evaporator. The adsorption system has two beds to ensure continuous operation which adsorption and desorption processes occur in the same phase as the switching of the chambers [3].

The advantages of the adsorption on absorption chiller are the relatively low temperature to separate the refrigerant in the generator as shown in **Table 1**; however, the coefficient of operation is lower than absorption system [3].

2. Description of the system

The TRNSYS [4] software was used to simulate the solar chillers (SC) for cooling a building located in Temixco City, Mexico. **Figure 3** shows the schematic diagram of the process. There are three main circuits: the solar collector (red line), chilled water (blue line), and cooling (green line). In the solar collector circuit, a heating fluid is pumped (Type 3d-3) from a storage tank (Type 4a) into the evacuated solar collector (Type 71) and returned to the tank. This pump is controlled by Type 2b, and it is turned on when the output is higher than the input temperature of the solar collector or when the temperature of the tank does not reach the maximum temperature of operation in the generator of the SC; otherwise, it is turned off. On the other hand, the heating fluid is pumped (Type 3d-4) to the CC (Type 107-2) to supply energy to the generator. When the temperature is lower than the minimum temperature of the working operation of the SC, a heater (Type 6) is turned on.

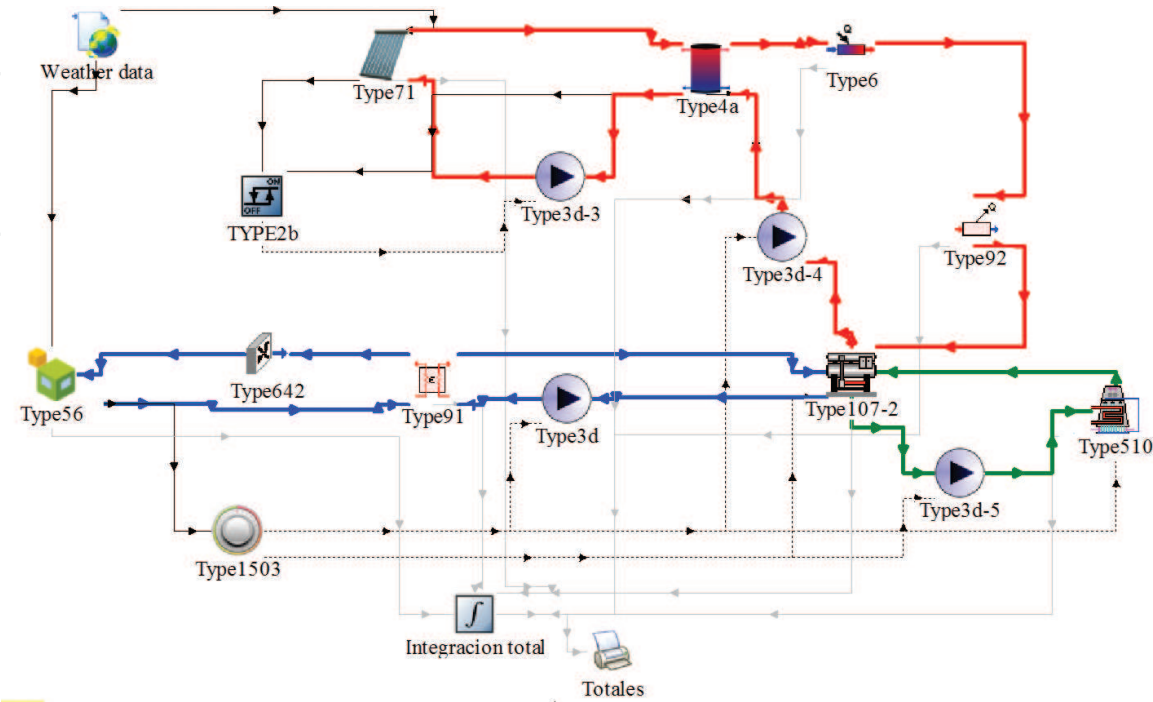


Figure 3.
Single-effect solar absorption cooling system with the auxiliary system in TRNSYS.

The chilled water circuit is used to control the temperature of the building (type 56) using a cooling thermostat (Type 1503) at $25^{\circ}\text{C} \pm 1^{\circ}\text{C}$. The chilled water is pumped (Type 3d-1) from the evaporator from the SC to a heat exchanger (Type 91) and exchanges heat with an airflow rate coming from the building (Type 642) and returns to the SC.

The cooling water circuit is used to extract the heat load from the condenser and absorber of the SC. A water flow rate is pumped (Type 3d-5) from the SC, and it is sent to a cooling tower (Type 510) which decreases its temperature, and it is returned to the chiller.

Figure 4 shows the implementation of the modulating tempering valve (MTV) programmed in Excel (Type62); in this case, the control Type 2b does not turn off

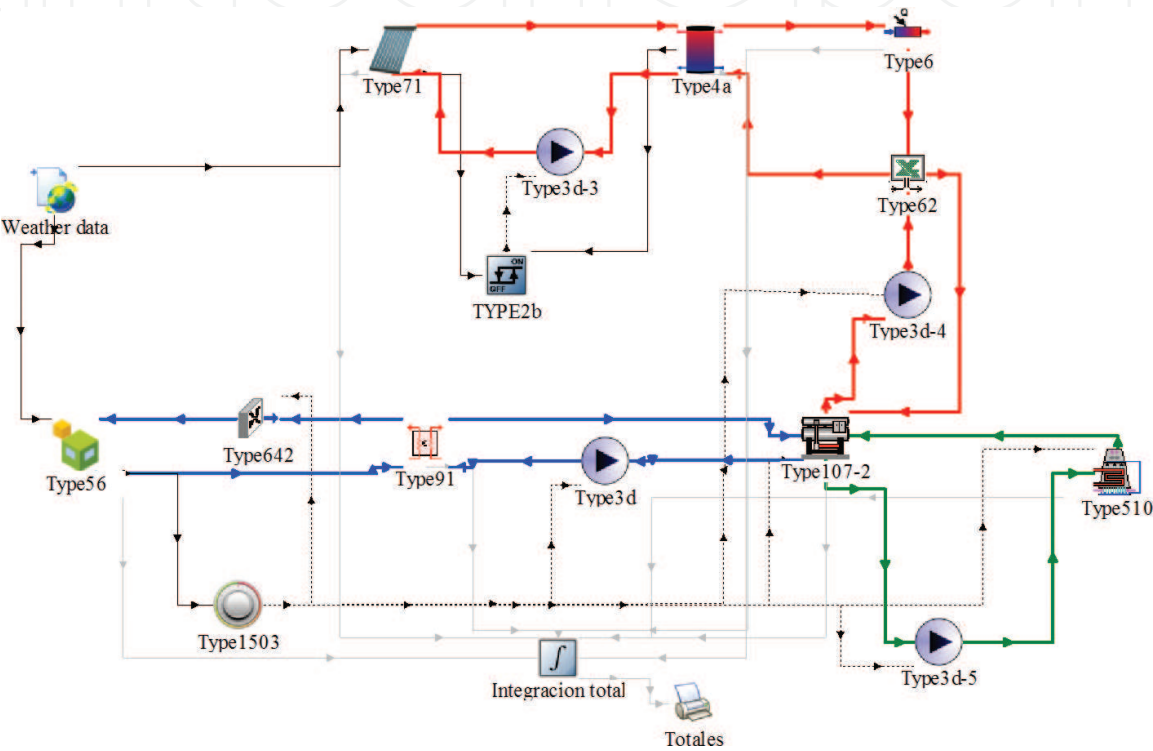


Figure 4.
Single-effect solar adsorption cooling system with MTV in TRNSYS.

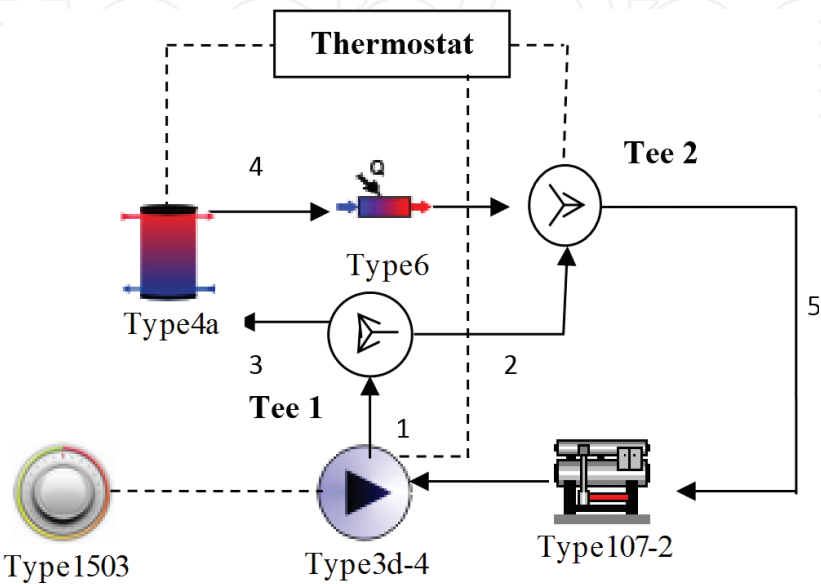


Figure 5.
Connection of the MTV in the solar chiller.

CC	T _{CHW, set}	T _{chilled}	T _{cooling}	T _{hot}
ABS(°C)	6.7	5.5–10.0	26.6–32.2	108.9–116.1
ADS(°C)	6.7	5.0–12.0	10.0–35.0	65.0–95.0

Table 2.
Operation conditions of the CC.

the pump Type 3d-3 when maximum temperature in the generator is reached but when storage tank reaches 130°C.

The MTV consisted of a diverter (Tee1) and a mixer (Tee2) component (**Figure 5**). The objective of the MTV is to keep the input temperature (5) at the maximum range of operation when the storage tank is higher than the limit of operation of CC. The function consists in mixing the high temperature coming from the storage tank (4) with a relative low temperature coming from the chiller (1) changing the flow rates of each stream to keep the maximum temperature (5) at 116°C in the case of absorption system (ABS) and 95°C in the case of adsorption system (ADS) as shown in **Table 2**.

3. Mathematical model of the cooling system

The equations presented in this section describe briefly the main components used for the simulation; a detailed information could be found in [4].

3.1 Assumptions

- The pressure drops and heat losses are negligible into pipelines.
- Fraction pump power (the energy converted to fluid thermal energy, f_{par}) was 0.05.
- Thermophysical properties of the fluids are constant.

3.2 Basic equations

The solar thermal collector efficiency (Type1b) was calculated using a quadratic or linear efficiency equation according to ASHRAE or European standards. The solar collector is an HTP-Evacuated Tube HP-30SC [5] with a net aperture area of 3.86 m². The solar thermal collector efficiency is presented in Eq. (1):

$$\eta = 0.418 - 1.17 \frac{(T_{IN} - T_{ENV})}{Q_{RAD}} \tag{1}$$

where T_{IN} and T_{ENV} are the input temperatures of the solar collector and environment temperature, respectively, and Q_{RAD} is the global radiation incident (W/m² °C).

The building (Type 56) is a simple multi-zone building with four windows and only one zone with some dimensions and main building characteristics are shown in **Table 3**. Four occupants in a rest position and three computers turned on from 08:00 to 18:00 were considered as heat generation [6].

The Type4a is a stratified storage tank. The properties and working conditions supplied were a heat capacity of 2.24 kJ/kg K for the heating fluid, an initial temperature of 70°C, and a loss coefficient = 0.75W/m² K. The characteristics of

Concept	Quantity
North and south wall, m ²	35
Ceiling and floor, m ²	75
West and east wall, m ²	12.5
Thickness of walls (brick), m	0.12
Thickness of insulating (polyurethane), m	0.05
Windows, m ²	2
Air change of ventilation, h ⁻¹	6

Table 3.
Building characteristics.

the thermal insulation are density = 16 kg/m³, thickness = 0.05 m, and thermal conductivity = 0.05 W/m K.

The air conditioning systems used are available in the TRNSYS standard library. This catalog-based component with its own external file predicts the performance of the chiller based on the available range of input data [7]. The file provides values of normalized capacity and COP values as a function of inlet hot water, inlet cooling water, and inlet chilled water temperature.

The absorption (Type107) uses a catalog data lookup approach to predict the performance of a single-effect, hot water-fired absorption chiller. In this design, the heat required to desorb the refrigerant is provided by a stream of hot water. The cooling water stream absorbs the energy dissipated by the absorber and condenser components, and the machine is designed to chill a third fluid stream to a user designated set point temperature. The rated capacity specified was 37,800 kJ/h and a COP rated of 0.70. A thermal fluid was used with a 2.13 kJ/kg °C.

The adsorption chillers (Type 909) cool a liquid stream by evaporating water onto the surface of a solid desiccant matrix. The rated capacity specified was 37,800 kJ/h and a COP rated of 0.53. A thermal fluid was used with a 2.13 kJ/kg °C.

Similar equations are used for both cooling systems. The chilled water, the hot water, and the rejected energy are calculated with Eqs. (2)–(4):

$$m_{CHW} C_{pCHW} (T_{CW,IN} - T_{CHW,SET}) = \text{MIN}(Q_{CHW}, Q_{CAPACITY}) \tag{2}$$

The energy of hot water.

$$Q_{HW} = m_{HW} C_{pHW} (T_{HW,IN} - T_{HW,OUT}) \tag{3}$$

The energy rejected (Q_{CW}) to the cooling water stream.

$$Q_{CW} = Q_{CHW} + Q_{HW} + Q_{AUX} \tag{4}$$

where Q_{HW} and Q_{AUX} are the energy of the hot water and the auxiliary equipment (pumps). $T_{CW,IN}$ and $T_{CHW,SET}$ are the temperature of the input cooling water and the set point of the chiller.

The heater system (Type 6) is used to increase the temperature until the minimum generator temperature of the CC. It was considered to have an efficiency of 100%:

$$Q_{HEATER} = m C_p (T_{OUT} - T_{IN}) \tag{5}$$

Type 510 models a closed circuit cooling tower; a device used to cool a liquid stream by evaporating water from the outside of coils containing the working fluid.

$$Q_{\text{DESIGN}} = m_{\text{WATER}} C_{p\text{WATER}} (T_{\text{IN,DESIGN}} - T_{\text{OUT,DESIGN}}) \quad (6)$$

$$h_{\text{AIR}}(T_{\text{AIR,OUT,DESIGN}}) = h_{\text{AIR}}(T_{\text{AIR,IN,DESIGN}}) + \frac{Q_{\text{WATER,DESIGN}}}{m_{\text{AIR}}} \quad (7)$$

The heat exchanger (Type 91) relies on an effectiveness (ϵ) minimum capacitance approach to model a heat exchanger. Under this assumption, the user is asked to provide the effectiveness of the heat exchanger and inlet conditions:

$$Q_{\text{HX}} = \epsilon Q_{\text{MAX}} \quad (8)$$

where Q_{MAX} is the maximum heat transfer rate across the exchanger.

The pumps compute a mass flow rate using a variable control function ($0 \leq \gamma \leq 1$) and a fixed (user-specified) maximum flow capacity (m_{MAX}).

$$m_{\text{IN}} C_p (T_{\text{OUT}} - T_{\text{IN}}) = P^* f_{\text{par}} \quad (9)$$

$$m_{\text{OUT}} = \gamma m_{\text{MAX}} \quad (10)$$

where f_{par} is the fraction pump power, p is the power (kW), m_{OUT} is the output mass flow rate (kg/s), and C_p is the heat capacity of the fluid (kJ/kg°C). **Table 4** shows the input data supplied to the pumps.

The cooling thermostat (Type 1503) is an N-stage cooling aquastat/simple thermostat modeled to output on/off control functions that can be used to control a fluid cooling system having up to an N-stage heating source. The desired fluid temperature may depend on the time of day or the day of the week. These variations of the cooling set point temperatures are modeled using an optional setup control function and a setup temperature difference [4].

The differential controller (Type 2b) is an on/off differential controller that can have a value of 1 or 0. The value of the control signal is chosen as a function of the difference between the upper (T_{HIGH}) and lower (T_{LOW}) temperatures, and both are compared with two dead-band temperature differences ΔT_{HIGH} and ΔT_{LOW} . The new value of the control function depends on the value of the input control function at the previous time step.

The mass and energy balances of the MTV (type 62) are calculated using equations from 11 to 14, respectively:

$$m_1 = m_5 \quad (11)$$

$$m_3 = m_4 \quad (12)$$

$$m_1 h_1 = m_2 h_2 + m_3 h_3 \quad (13)$$

$$m_5 h_5 = m_2 h_2 + m_4 h_4 \quad (14)$$

where h is the specific enthalpy and m is the flow rate and subscripts corresponding to **Figure 5**.

Auxiliary	Power (kJ/h)	Cp (kJ/kg °C)	m (kg/h)
Pump (chilled water)	1339	4.19	1789
Coil (Type 3d-2)	1339	1.22	2487
Pump (Type 3d-3, solar collector)	2664	2.34	5442
Pump (Type 3d-4, hot water)	1339	2.34	1261
Pump (Type 3d-5, cooling water)	1339	4.19	4230
Cooling Tower (fan)	2013	—	—

Table 4.
Parameter supplied to the pumps of the cooling system.

3.3 Parameters

The energetic performance of the cooling systems can be evaluated using two indicators: solar fraction (SF) and heating fraction (HF). Solar and heating fractions are defined as the amount of energy supplied by solar resources (Q_{COL}) and heater system (Q_{HEATER}), respectively, divided by the total energy supplied ($Q_{COL} + Q_{HEATER}$).

4. Results

Figure 6 shows the behavior of the COP and room and environment temperature for the absorption and adsorption systems during a sunny day (from 2003 to 2016 h). The coefficient of performance is almost keeping constant for ABS (0.71) caused by the short range of operation of T_{HOT} (see **Table 2**) and from 0.46 to 0.48 for ADS. The room temperature was kept from 24 to 26°C; when room temperature is lower than 24°C, the control temperature is turned off (night).

Figure 7 shows the function of the MTV to keep the generator temperature at the maximum range of operation. When the tank temperature (T_4 , from **Figure 5**) remains on the range of operation, all flow rates go to the tank ($m_1 = m_3 = 1261$ kg/h and $m_2 = 0$, from 1476.1 to 1476.8 h); however when it is higher than 95°C (from 1477.0 to 1486.8 h), m_1 splits into m_3 ($m_3 = m_4$) and m_2 to keep T_5 at 95°C, mixing a high temperature coming from stream 4 with stream 2 ($T_2 = T_1$).

Next sections will show the condition to operate the cooling system at improved conditions based on the tilt of the solar collector, activation temperature, and storage tank specific volume (SV) [2, 8, 9]. The period of evaluation was from March to May (1460–3560 h) to operate the cooling system due to the highest ambient temperature in a year in Temixco, Mexico [6], as shown in **Figure 8**, besides there are several days with very poor (2190–2300 h) or null beam radiation.

4.1 Selection of the tilt

Figure 9 shows the energy from the solar collector as a function of the tilt of the solar collector. 7° obtained the maximum solar collector energy (1.265×10^7) during the 3 months evaluated using 30 m² of area solar collector and a storage tank of 3 m³.

4.2 Selection of the activation temperature

Figures 10 and 11 show the COP as a function of the input generator temperature without and with MTV for absorption and adsorption chiller, respectively.

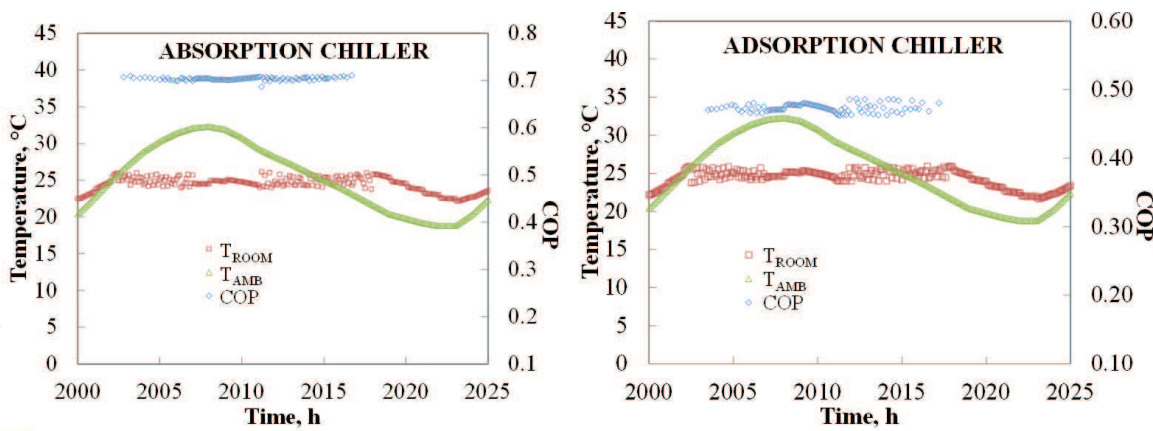


Figure 6.
COP and room and environment temperature behavior within a day.

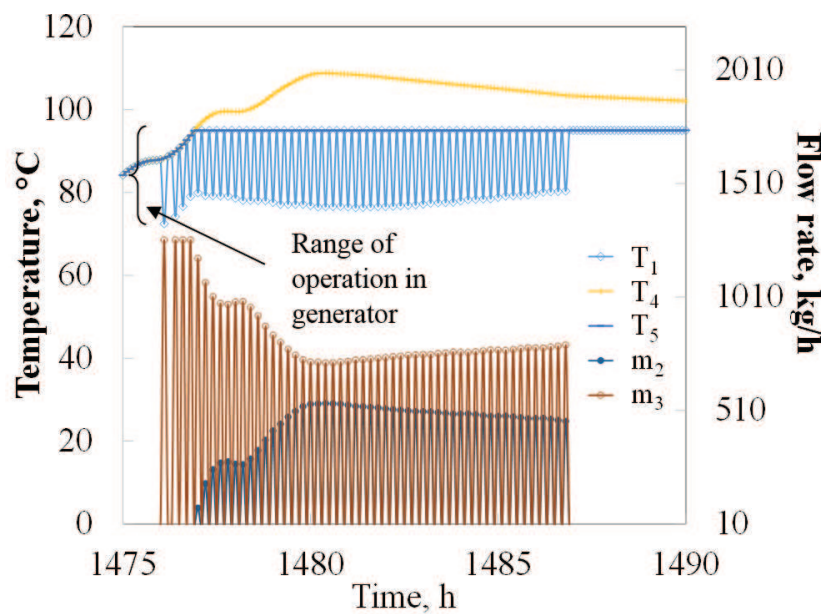


Figure 7.
Behavior of temperatures and flow rates of the modulating tempering valve.

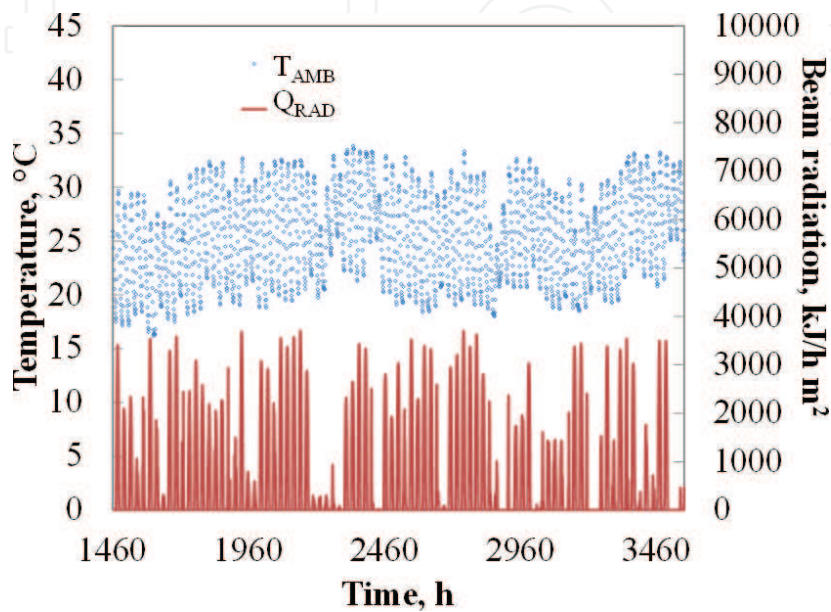


Figure 8.
Environment and beam radiation behavior within 3 months.

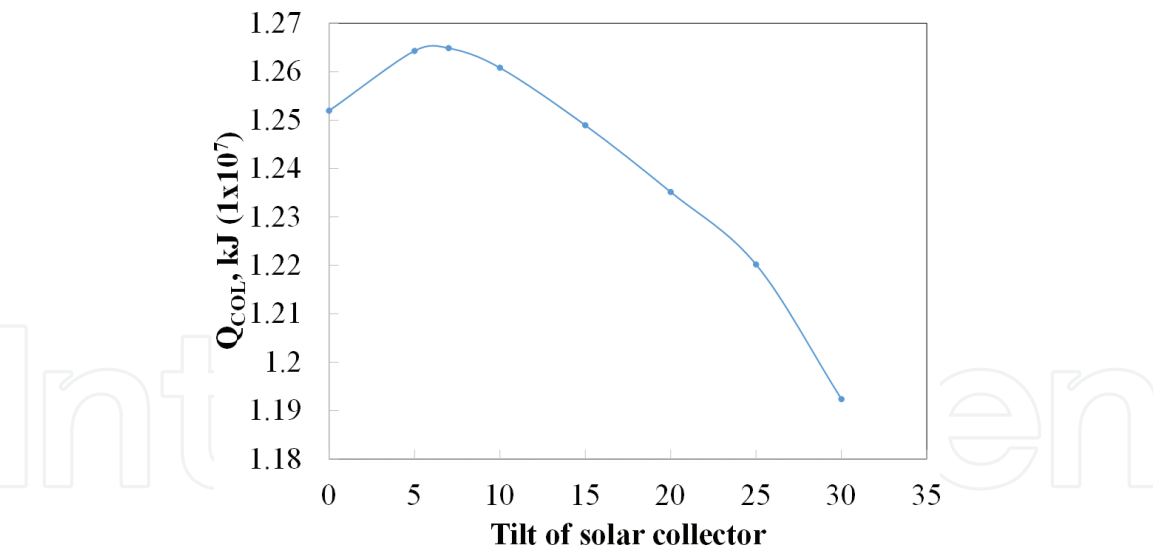


Figure 9.
Behavior of the tilt of the solar collector on solar fraction.

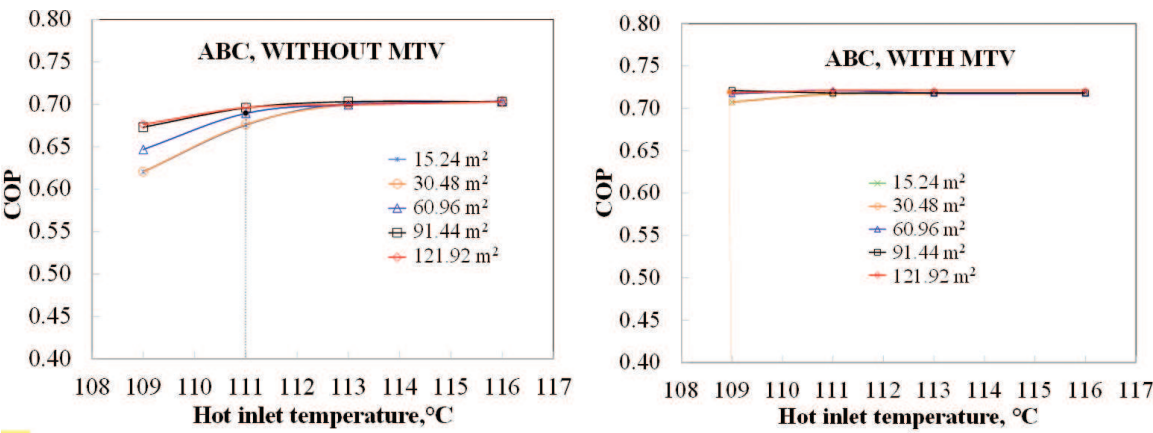


Figure 10.
Effect of the activation temperature on the solar fraction absorption system without and with MTV.

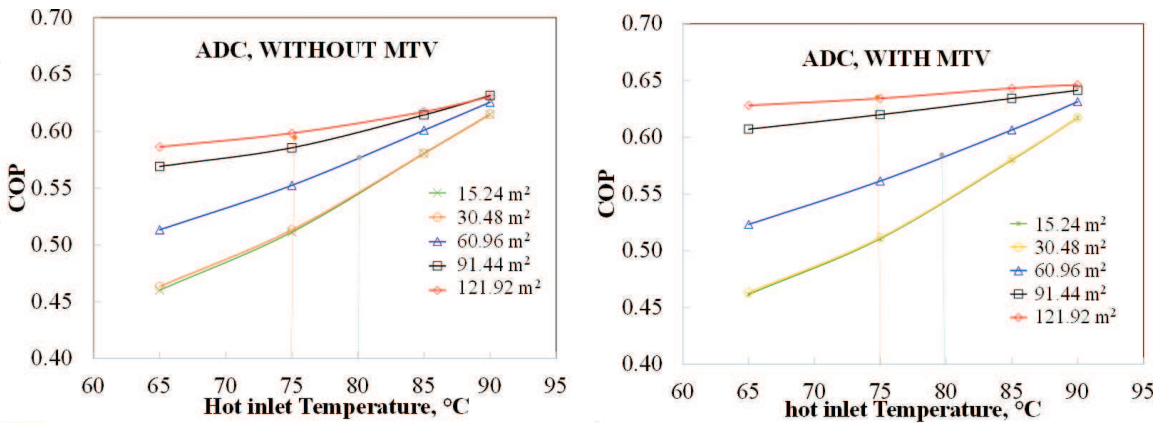


Figure 11.
Effect of the activation temperature on the solar fraction adsorption system without and with MTV.

A temperature of 111 and 109°C was considered a good option to operate the absorption system without and with MTV. While an input temperature generator of 75°C was considered to operate the adsorption system for without and with MTV, 80°C should be used when the area collector is lower than 60.96 m².

4.3 Selection of the number of solar collectors and storage tank

The variation of the solar fraction with respect to the specific volume is presented in **Figures 12** and **13** for absorption and adsorption systems, respectively, at different solar collector areas. These figures show that 35 and 23 l/m² were a good choice for the configuration without and with MTV for absorption system, respectively, while 27 and 22 l/m² were selected for without and with MTV, respectively, for the adsorption system. This represented a 52.2 and 22.7% of reduction for absorption and adsorption systems. A solar collector area of 60.96 m² was selected to obtain high values of the solar fraction. Besides a certain storage volume value, the increment of the solar fraction, is not significant [2].

Figures 14 and **15** show the heater energy as a function of the storage tank at several solar collector areas for absorption and adsorption chillers, respectively. It can be seen that there is an increment of heater energy when the volume is incremented at 30.48 m² of the solar collector area for both chillers because more energy is required at higher storage volume. However, when the solar collector area is higher than 60.96 m², the increment of volume decreases the consumption of energy because the storage tank has a high capacity to keep more energy.

Figures 16 and **17** show the energy of the solar collector (Q_{COL}), heater (Q_{HEATER}), and the total energy supplied (W_{TOTAL}) to the electric equipment

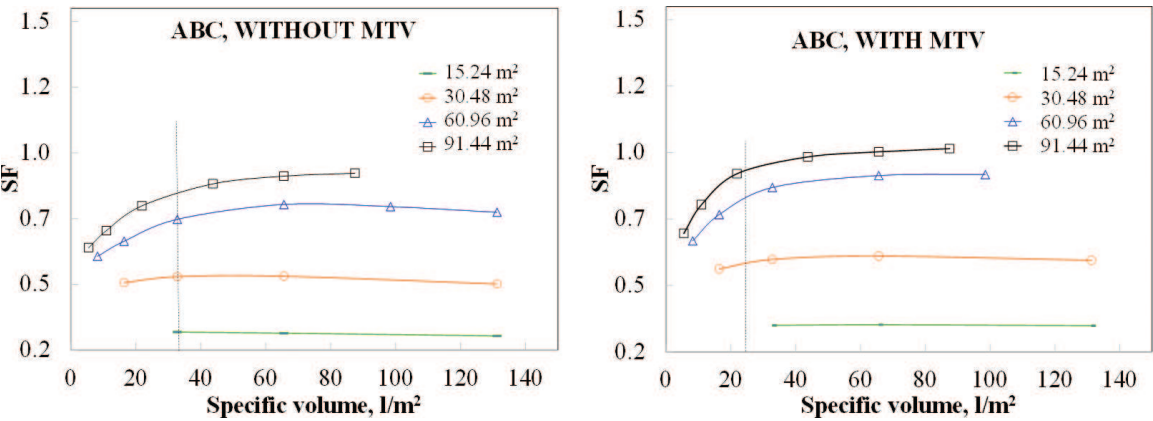


Figure 12.
Solar fraction against storage-specific volume for absorption chiller.

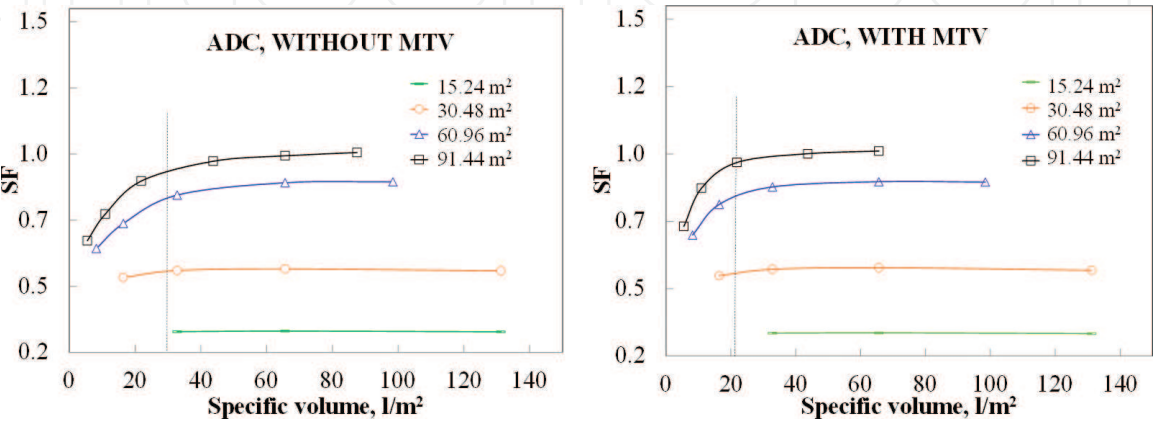


Figure 13.
Solar fraction against storage-specific volume for adsorption chiller.

(pumps, fans, and heater) as a function of the storage tank volume (ST) at 60.96 m² of the solar area. The calculated storage tank volume corresponds to 2.13 and 1.40 m³ without and with MTV for absorption system and 1.64 and 1.34 m³ for adsorption system, respectively. The use of storage volume higher than 3 m³ has a soft effect on the Q_{HEATER} and W_{TOTAL} for absorption chiller with MTV and adsorption chiller.

Table 5 shows the values of the parameter showed in Figures 16 and 17. It was a little decrement of solar collector energy (from 1.77 × 10⁷ to 1.61 × 10⁷ kJ) with and without MTV for absorption chiller because the time of operation of the solar

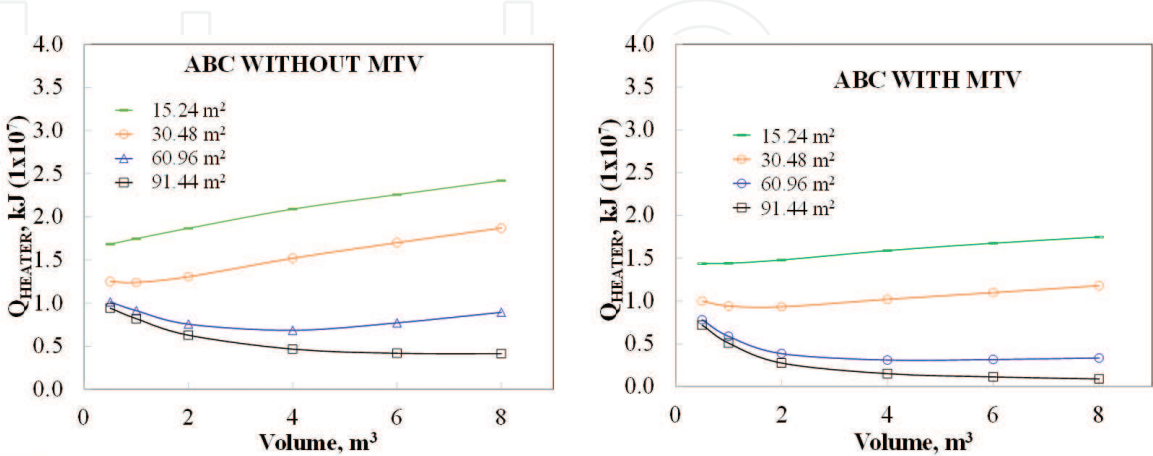


Figure 14.
Energy consumption of the heater against the storage volume for absorption chiller.

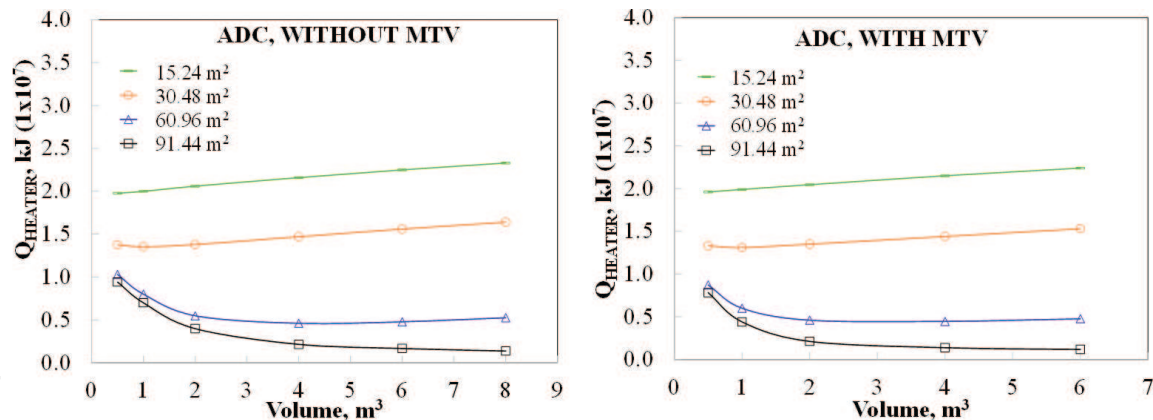


Figure 15.
Energy consumption of the heater against the storage volume for adsorption chiller.

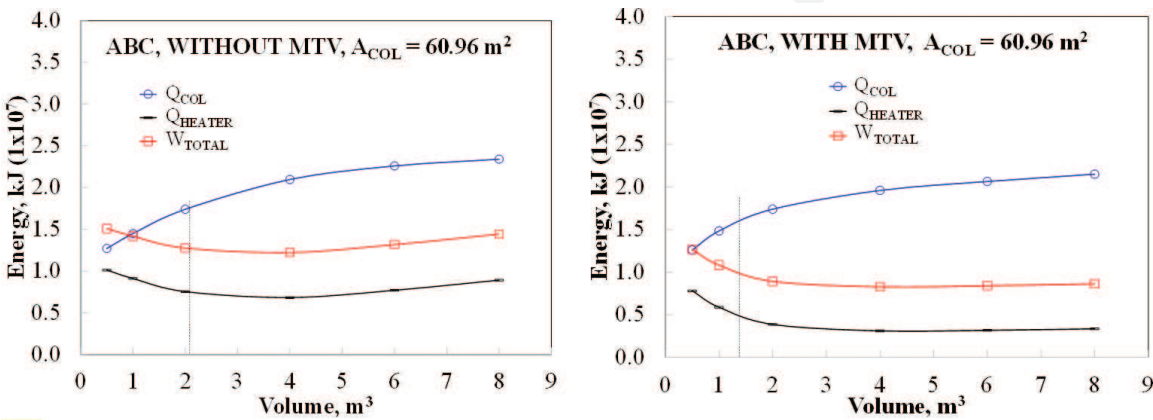


Figure 16.
The energy of the equipment against the storage volume for absorption chiller.

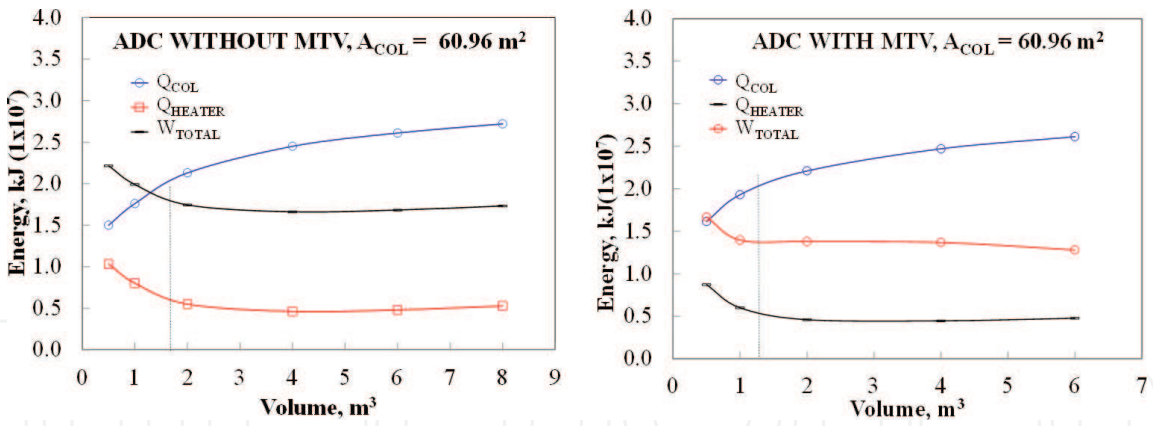


Figure 17.
The energy of the equipment against the storage volume for adsorption chiller.

	SV, l/m ²	A _{COL} , m ²	ST, m ³	Q _{COL} , kJ (1 × 10 ⁷)	Q _{HEATER} , kJ (1 × 10 ⁷)	W _{TOTAL} , kJ (1 × 10 ⁷)
ABS	35	60.96	2.13	1.77	0.74	1.27
ABS, MTV	23	60.96	1.40	1.61	0.48	0.95
ADS	27	60.96	1.64	2.02	0.62	1.69
ADS, MTV	22	60.96	1.34	2.05	0.52	1.44

Table 5.
Results of the selected conditions for cooling chillers.

collector was more using the MTV, then the temperature of the storage tank was higher and this reduces the efficiency of the solar collector, however it was a significant reduction on the Q_{HEATER} and W_{TOTAL} for both chillers, mainly for absorption chiller, this represents almost a 54.16 and 33.68%, respectively, while 19.23 and 17.36% of reduction was for adsorption chiller.

The reduction of the Q_{HEATER} is lower for adsorption than absorption chiller because the range of operation of the generator temperature is shorter for absorption than adsorption; however, W_{TOTAL} energy is higher for adsorption chiller than absorption chiller (1.27 kJ for absorption and 1.69 kJ for adsorption without MTV), because it has lower COP (around 0.53) than absorption chiller (around 0.70) and electric equipment have more time of operation.

5. Conclusion

This chapter presented a thermal analysis of the absorption and adsorption chillers in a dynamic condition for conditioning a building located in Temixco, Mexico, using TRNSYS and Microsoft Excel software from March to May. In this study, both chillers with and without MTV to increase the operation time using evacuated solar collectors were compared. The following conclusions are presented:

- The maximum solar collector energy was obtained with an angle of tilt of 7°.
- The COP has the higher values when using the input generator temperature. It selected a minimum temperature of working of 111 and 109°C for absorption chiller without and with MTV, and 75°C for adsorption chiller.

- The selection of 35 and 23 l/m² was a good choice for the configuration without and with MTV for absorption chiller, respectively, while 27 and 22 l/m² were selected for without and with MTV, respectively, for adsorption chiller.
- The working condition selected was 2.13 and 1.40 m³ storage tank without and with MTV for absorption chiller and 1.64 and 1.34 m³ without and with MTV for adsorption chiller using 60.96 m³. The results showed a significant reduction of the Q_{HEATER} and W_{TOTAL} energy for both chillers, mainly for absorption chiller; this represented almost 54.16 and 33.68%, respectively, while 19.23 and 17.36% of reduction was for adsorption chiller, because adsorption has a lower COP (around 0.53) than absorption chiller (around 0.70) and electric equipment have more time of operation.

Nomenclature

Q	energy, kJ
m	mass flow rate, kg/s
Cp	heat capacity, kJ/kg C
T	temperature, °C
h	enthalpy, kJ/kg
COP	coefficient of performance
ST	store tank
SV	specific volume, l/m2
TW	total working
MTV	modulating tempering valve
<i>Subscript</i>	
CHW	chilled water
CW	cooling water
HW	hot water
AUX	Auxiliary system
HX	heat exchanger equipment
COL	solar collector

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