Solar energy to drive absorption cooling systems suitable for small building applications

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ABSTRACT

Air conditioning systems have a major impact on energy demand. With fossil fuels fast depleting, it is imperative to look for cooling systems that require less high-grade energy for their operation. In this context, absorption cooling systems have become increasingly popular in recent years from the viewpoints of energy and environment. Two types of the absorption chillers, the single effect and the half-effect systems, can operate using low temperature hot water.

This paper presents the simulation results and an overview of the performance of low capacity single stage and half-effect absorption cooling systems, suitable for residential and small building applications. The primary heat source is solar energy supplied from flat plate collectors. The complete systems (solar collectors and absorption cooling system) were simulated using a developed software program. The energy and exergy analysis is carried out for each component of the two systems. When evaporator temperature is maintained constant at 5°C and the condenser temperature is fixed at 28°C, 32°C and 36°C respectively the percentage of the used energy covered by solar collectors and the percentage of auxiliary heating load were calculated versus time of day.

INTRODUCTION

Nowadays world is marked with an intensive use of energy, cooling and air conditioning remains having the large share of energy consumption. Besides the energy consumption, climatic changes have become an international issue that was treated in many international summits (the last one was the United Nations Climate Change Conference Copenhagen – December 2009). In this context, solar chilled water production through absorption cycles may be considered one of the most desirable applications to reduce energy consumption and CO_2 gas emissions. The use of solar energy to drive cooling cycles for air conditioning is an attractive concept because of the coincidence of peak cooling loads with the available solar power.

Most of Arab countries, enjoys of solar radiations. During the summer, the demand for electricity greatly increases because of the use of vapour compression air conditioning system, which increases the peak electric load, and major problems in the country's electric supply arise. Solar energy can be used for cooling as it can assist in providing the heating energy for absorption cooling systems.

Two types of the absorption chillers, the single effect and half-effect cycles, can operate using low temperature hot water. Theoretical and experimental works on single effect absorption cooling systems have been extensively reported [Gomri 2009, Gomri 2010, Kaynakli 2007, Sencan. 2005, Sosen A. 2001]. The principle of the half-effect cycle is that it has two lifts. The term lift is used to represent a concentration difference between the generator and absorber. This concentration difference is what drives or gives the potential for mass to flow into the absorber. With the single effect there is only one lift.

Many researchers have reported works on half-effect cycles. Ma and Deng [Ma 1996] have reported preliminary results of an experimental investigation on a 6 kW vapour absorption system working on half-effect cycle with H_2O-LiBr as the working fluids. With hot water temperature requirement of around 85°C, a chilled water temperature of 7°C has been reached in their experiment. Sumathy et al. [Sumathy 2000] have tested solar cooling and heating system with a 100 kW half-effect absorption chiller working on the same cycle. The system has been used successfully with generation temperatures in the range of 65–75°C to achieve a chilled water temperature of 9°C. Arivazhagan et al. [Arivazhagan 2005] presented a simulation studies conducted on a half-effect vapour absorption cycle using R134a-DMAC as the refrigerant-absorbent pair with low temperature heat sources for cold storage applications. Arivazhagan et al. [Arivazhagan 2006] presented experimental studies on the performance of a two-stage half eeffect vapour absorption cooling system. The prototype was designed for 1 kW cooling capacity using HFC based working fluids (R134a as refrigerant and DMAC as absorbent).

In the paper presently considered, it is planed to use solar energy to drive half-effect and single effect vapour absorption cooling machines to produce chilled water for air conditioning. The systems are proposed to provide cooling in summer (June, July August and September) for a building located in

Constantine, east of Algeria (longitude 6.62 °E, latitude 36.28°N and altitude of 689m). The results presented in this paper were for 1-day period operation (21st of July).

SIMULATION OF THE PERFORMANCE

Solar collector system

The various relations that are required in order to determine the useful energy collected and interaction of the various constructural parameters on the performance of a collector are taken from [Frederick 1976, Kalogirou 2004, Akhtara 2007, Sartori 2006, Karatasou 2006].

The energy balance equation of the solar collector can be written as follows [Frederick 1976]:

$$I_G . A_c = Q_u + Q_{loss} + Q_{stg} \qquad (1)$$

Where I_G is the instantaneous solar radiation incident on the collector per unit area, A_C is the collector surface area, Q_{loss} is the heat loss from the collector and Q_u is the useful energy transferred from the absorber to the fluid flowing through the tubes of the collector. Q_{stg} is the energy stored in the collector (Q_{stg}=0: the solar thermal system is considered at steady state conditions).

The useful energy gain of the flat plate collectors is calculated by:

$$Q_u = A_c . F_R . \left[(\tau . \alpha) . I_G - U_L . (T_{fi} - T_a) \right] \qquad (2)$$

Where A_C is the collector area, F_R is the collector heat removal factor, $(\tau.\alpha)$ is the transmittance-absorptance products, U_L is the collector overall loss coefficient, T_a is the ambient air temperature and T_{fi} is the fluid temperature at the inlet to the collector.

The Collector heat removal factor (F_R) is the ratio of useful heat obtained in collector to the heat collected by collector when the absorber surface temperature is equal to fluid entire temperature on every point of the collector surface.

$$F_R = \frac{m.C_p}{A_c.U_L} \left[1 - e^{-\frac{(A_c.U_L.F')}{m.C_p}} \right] \qquad (3)$$

Where m is the mass flow rate of water, C_p is the specific heat of water and F' is the collector efficiency factor. F' represents the ratio of the actual useful energy gain to the useful energy gain that would result if the collector absorbing surface had been at local fluid temperature.

The collector overall heat loss coefficient (U_L) is the sum of the top (U_T, bottom U_B and edge U_E heat loss coefficient. It means that:

$$U_L = U_T + U_B + U_E \qquad (4)$$

The thermal efficiency of the solar collectors is the ratio of useful energy obtained in collector to solar radiation incoming to collector. It can be formulated as:

$$\eta_{th\text{-}FPC} = \frac{Q_u}{I_G . A_c} \qquad (5)$$

The prediction of collector performance requires knowledge of the absorbed solar energy by collector absorber plate. The solar energy incident on a tilted collector consists of three different distributions: beam radiation, diffuse radiation, and ground – reflected radiation. The details of the calculation depend on which diffuse sky model is used. For estimating sky diffuse solar radiations several models have been developed [Bugler 1979, Klucher 1979, Reindl 1990, Perez 1990]. They vary mainly in the way that treat the three components of the sky diffuse radiation, i.e. the isotropic, circumsolar and horizon radiation streams. In this study the absorbed radiation on the absorber plate is calculated by Perez's model [Elminir 2006].

Single effect and Half-effect absorption cooling systems

As shown in figure 1 the main components of a single effect absorption cooling system are the generator (g), the absorber (ab), the condenser (cd), the evaporator (ev) , the pump (P1), the expansion valve (V1), the reducing valve (V2) and the solution heat exchanger (HEX-1).

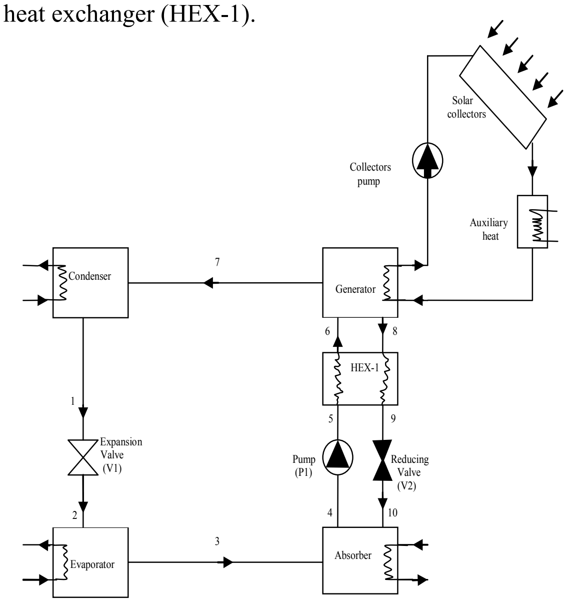

Figure 1. The schematic illustration of solar single effect absorption cooling system

The half-effect absorption refrigeration system as shown in figure 2 consists of condenser, evaporator, two generators, two absorbers, two pumps, two solution heat exchangers, two solution reducing valves and a refrigerant expansion valve. In the system operation, the evaporator and low pressure absorber operate at low pressure (evaporation pressure P_{ev}). The low pressure generator (LPG) and high absorber operate at intermediate pressure (P_i) and the high pressure generator (HPG) and the condenser operate at high pressure (condenser pressure, P_{cd}). Both generators (LPG and HPG) can be supplied with heat at the same temperature.

The refrigerant vapour from the evaporator is absorbed by the strong solution in the low absorber. The weak solution from the low absorber is pumped to the low generator through the low solution heat exchanger. The strong solution in the low generator is returned to the low absorber through the low solution heat exchanger. The refrigerant vapour from the low generator is absorbed by the strong solution in the high absorber. The weak solution from the high absorber is pumped to the high generator through the high solution heat exchanger. The strong solution in the high generator is returned to the high absorber through the high solution heat exchanger. In both heat exchangers, the weak solution from the absorber is heated by the strong solution from the generator. The refrigerant is boiled out of the solution in the high generator and circulated to the condenser. The liquid refrigerant from the condenser is returned to the evaporator through an expansion valve.

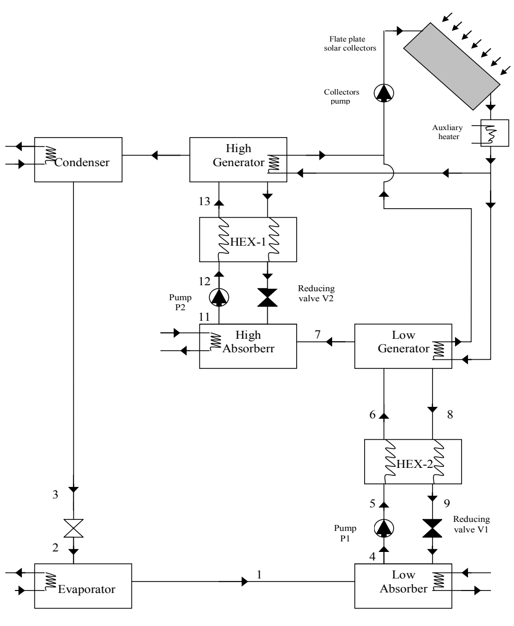

Figure 2. The schematic illustration of solar half-effect absorption cooling system

The two systems are simulated assuming the following conditions:

- The analysis is made under steady conditions.
- There is no departure of chemical substances from the cycle to the environment.
- The kinetic and potential energy effects are neglected.
- The refrigerant (water) at the outlet of the condenser is saturated liquid.
- The refrigerant (water) at the outlet of the evaporator is saturated vapour.
- The Lithium bromide solution at the absorber outlet is a strong solution and it is at the absorber temperature
- The outlet temperatures from the absorber and from generators correspond to equilibrium conditions of the mixing and separation respectively.
- Pressure losses in the pipelines and all heat exchangers are negligible.
- Heat exchange between the system and surroundings, other than in that prescribed by heat transfer at the HPG, evaporator, condenser and absorber, does not occur
- The reference environmental state for the system is water at an environment temperature T_0 of 25°C and pressure (P_0) of 1atm.
- The system produce chilled water, and generator is driven by hot water.
- The system rejects heat to cooling water at the condenser and absorbers.
- The refrigerant flow rate leaving the low temperature generator is equal to the refrigerant flow rate leaving the high temperature generator.
- For half-effect system both generators are supplied with heat at the same temperature.

Simulations are carried out for a constant refrigeration capacity Q_{ev}=10kW, pump efficiency η_P =85%, heat exchangers effectiveness $\varepsilon_I=\varepsilon_{II}$=70%, condensation temperature is equal to the absorber temperature $T_{cd}=T_{ab}$. Condensation temperature is varied in the following range: T_{cd}=28°C to 36°C. The outlet temperature of cooling water has been assumed at T_{cd}-3 and the inlet temperature of cooling water has been assumed at T_{cd}-8. Evaporation temperature is maintained at T_{ev}=5°C. The outlet temperature of chilled water has been assumed at T_{ev}+3 and the inlet temperature of chilled water has been assumed at T_{ev}+8. For half-effect system the generation temperature "T_g" is varied from 42°C to 62°C. The outlet temperature of hot water has been assumed at T_g+8 and the inlet temperature of hot water has been assumed at T_g+21.

For single effect system the generation temperature "T_g" is varied from 54°C to 83°C. The outlet temperature of hot water has been assumed at T_g+8 and the inlet temperature of hot water has been assumed at T_g+15.

In this analysis, the thermal-physical properties of the working fluids have to be known as analytic functions. A set of computationally efficient formulations of thermodynamic properties of lithium bromide/water solution and liquid water developed by Patek and Klomfar [Patek 2006] are used in this work. The equations for the thermal properties of steam are obtained from correlation provided by Patek and Klomfar [Patek 2009].

For the thermodynamic analysis of the absorption system the principles of mass conservation and first law of thermodynamic are applied to each component of the systems (see figures 1 and 2).

Mass Conservation

Mass conservation includes the mass balance of total mass and each material of the solution. The governing equations of mass and type of material conservation for a steady state and steady flow system are:

$$\sum m_i - \sum m_0 = 0 \tag{6}$$

$$\sum m_i . x_i - \sum m_0 . x_0 = 0 \tag{7}$$

Where m is the mass flow rate and x is de mass fraction of LiBr in the solution. The mass fraction of the mixture at different points of the systems (figures 1 and 2) is calculated using the corresponding temperature and pressure data.
The mass flow rate of refrigerant is obtained by energy balance at evaporator and is given as,

$$m_3 = \frac{Q_{ev}}{(h_2 - h_3)} \tag{8}$$

Energy analysis

The first law of thermodynamics yields the energy balance of each component of the absorption cooling system as follows:

$$\left(\sum m_i . h_i - \sum m_0 . h_0\right) + \left(\sum Q_i - \sum Q_0\right) + W = 0 \tag{9}$$

The thermal efficiency (coefficient of performance) of the absorption cooling system is obtained by
Single effect system:

$$COP = \frac{Q_{ev}}{(Q_g + W_{p1})} \tag{10}$$

Half-effect system:

$$COP = \frac{Q_{ev}}{(Q_{HPG} + Q_{LPG} + W_{p1} + W_{p2})} \tag{11}$$

Exergy analysis

Exergy analysis is the combination of the first and second law of thermodynamics and is defined as the maximum amount of work, which can be produced by a stream or system as it is brought into equilibrium with a reference environment and can be thought of as a measure of the usefulness or quality of energy [Kotas 1987]. According to Bejan et al. [Bejan 1996] the exergetic balance applied to a fixed control volume is given by the following equation:

$$E_{xd} = \left(\sum_i m_i . ex_i\right)_{in} - \left(\sum_i m_i . ex_i\right)_{out} - E_{heat\,j} - W \tag{12}$$

Where E_{xd} is rate of exergy destruction. E_{heat} is the net exergy transfer by heat at temperature T, which is given by

$$E_{heat} = \sum_j \left(1 - \frac{T_0}{T_j}\right) . Q_i \tag{13}$$

W is the mechanical work transfer to or from the system.
The specific exergy of flow is:

$$ex = (h - h_0) - T_0 (s - s_0) \tag{14}$$

m is the mass flow rate of the fluid stream, h is the enthalpy, s is the entropy and the subscripts 0 stands for the restricted dead state.

The exergetic efficiency can be calculated as the ratio between the net exergy produced by the evaporator (exergy desired output) and the input exergy to the generator (exergy used) plus the mechanical work of the solution pumps.

Single effect system:

$$\eta_{exergy} = \frac{Q_{ev} . \left(1 - \frac{T_0}{T_b}\right)}{\left[Q_g . \left(1 - \frac{T_0}{T_h}\right) + W_{p1}\right]} \tag{15}$$

Half-effect system:

$$\eta_{exergy} = \frac{Q_{ev} . \left(1 - \frac{T_0}{T_b}\right)}{\left[(Q_{HPG} + Q_{LPG}) . \left(1 - \frac{T_0}{T_h}\right) + (W_{P1} + W_{P2})\right]} \tag{16}$$

T_b and T_h are the mean temperature of the cold source (in the evaporator) and the hot source (in the generator) respectively.
Qg or Q_{gH} is the heat supplied to the generator or HPG and Q_{gL} is the heat supplied to the LPG.

Mathematical model for the global system

For this part, the mathematical model is based on the following assumptions:

- The solar collector loop is chosen without a storage tank.
- An auxiliary heat source is provided, so that the hot water is supplied to generators when solar energy is not sufficient to heat the water to the required temperature level needed by the generators

The overall thermal efficiency of the complete solar absorption cooling system $\eta_{overall}$ is the product of COP and solar efficiency $\eta_{th\text{-}FPC}$ [Ortega 2008]:

$$\eta_{overall} = COP.\eta_{th\text{-}FPC} \tag{17}$$

Single effect system:

$$Q_g = Q_u + Q_{aux} \tag{18}$$

Half-effect system:

$$Q_{gH} + Q_{gL} = Q_u + Q_{aux} \tag{19}$$

Q_{aux} is the auxiliary heating load.

RESULTS AND DISCUSSION

The energy and exergy analysis was carried out on 21st of July in Constantine (East of Algeria; Latitude 36.28°N, Longitude 6.62°E). A computer program was written for thermodynamic analysis. The program was based on the energy balance, exergy balance and thermodynamic properties for each reference point. The initial conditions are given into the program including the ambient conditions, the solar energy collector parameter specification, the component temperatures, pumps efficiencies, effectiveness of heat exchangers and evaporator load. With the given parameters, the thermodynamic properties at all reference points in the system were calculated. The results obtained from the present study may be presented as follows.

The effects of the generators, absorbers and condenser temperature on the coefficient of performance are shown in figure 3.

The high COP values are obtained at low condenser temperatures. For a given evaporator, absorber and condenser temperature, there is a minimum generator temperature which corresponds to a maximum COP. In this study and when the condenser temperature is varied from 28°C to 36°C the maximum COP values of the single effect cooling systems are in the range of 0.75–0.82 and for half-effect cooling systems are in the range of 0.41–0.44. It should be noted that the COP initially exhibits a significant increase when the generator temperature increases, and then the slope of the COP curves become almost flat. In other words, increasing the generator temperature higher than a certain value does not provide much improvement for the COP.

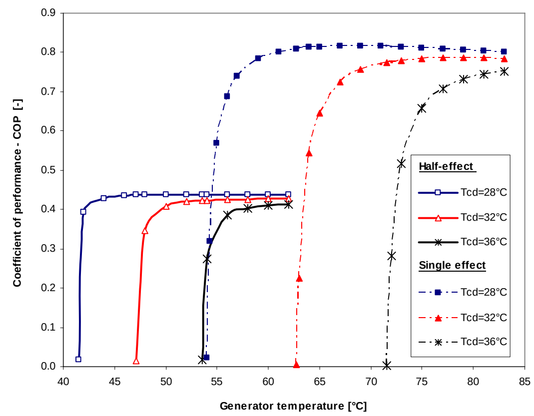

Figure 3. Coefficient of performance versus generator temperature 'Tg' and condenser temperature 'Tcd' (Tev=5°C)

The variation of exergy efficiency with generator temperature for single effect and half-effect cooling systems at different condenser temperatures is shown in figure 4.

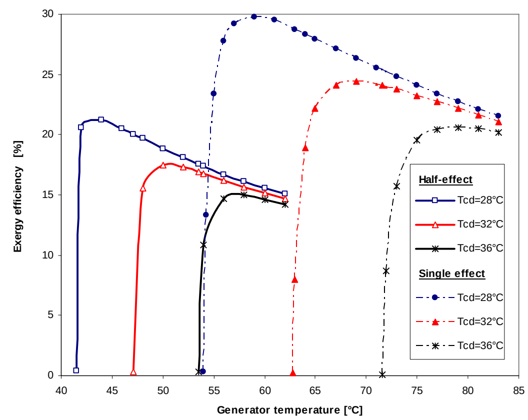

Figure 4. Exergy efficiency versus generator temperature 'Tg' and condenser temperature 'Tcd' (Tev=5°C)

Exergy efficiency increase with an increase in the generator temperature for both cases (single effect and half-effect cooling systems) up to a certain generator temperature (for a given evaporator, absorber and condenser temperatures, there is a minimum generator temperature which corresponds to a maximum exergy efficiency) and then decrease.

In this study and when the evaporation temperature is maintained at 5°C and condenser and absorber temperatures are varied from 28°C to 36°C the maximum exergy efficiency values of the single effect cooling systems are in the range of 20.6%–29.7% and for half-effect cooling systems are in the range of 14.6%– 19.9%.

As the efficiencies of the first and second laws are examined here, it is seen that exergy efficiency increases up to a certain generator temperature (variable with condensation temperature). Furthermore, COP value also increases up to about this temperature. At the higher generator temperatures, while COP remains approximately constant, exergy efficiency decreases gradually. The reason for this is that the increase in the generator temperature negatively influences the exergy efficiency value as seen from Eqs. (15) and (16).

Figure 5 shows the variation of the number of flat plate solar collector (N_{FPC}) during the peak solar gain hour on 21st of July versus the generator temperature for different value of condenser temperature. For a given condenser temperature there is an optimum generator temperature for which the N_{FPC} is minimum. This optimum generator temperature corresponds to the generator temperature giving the maximum COP and the maximum exergetic efficiency of the absorption cooling systems.

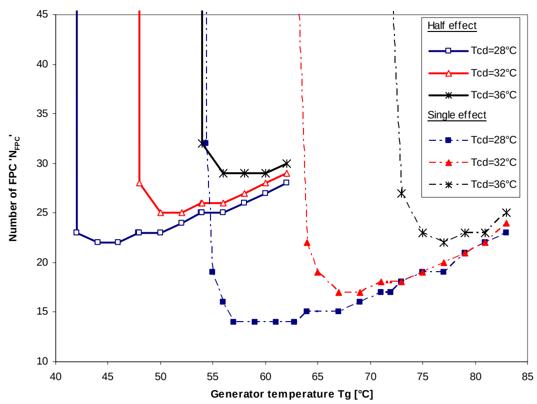

Figure 5. Variation of the number of FPC 'N_{FPC}' versus generator temperature 'Tg' and condenser temperature (Ac=2.03m^2)

Figure 6 shows the variation of energy efficiency of solar flat plate collectors (solar collector thermal efficiency, $\eta_{th\text{-}FPC}$), against time of day for different generator temperature (generator temperature giving the maximum COP and exergy efficiency) and condenser temperatures.

It can be seen that the instantaneous collector efficiency is slightly higher after 12h and then decreases gradually at the shut off of operation hour. We note the existence of a maximum value of roughly 60% corresponding to the maximum incident radiation (also a minimum generator temperature of 43°C and a minimum condenser temperature of 28°C). Solar collectors are more efficient with half-effect system than that with single effect system; this can be explained by the fact that single effect system need higher generator temperature than that required by half-effect system.

The COP of the absorption cooling systems is a constant value during all the day then and as can be seen from Eq.17 the thermal efficiency of the global systems will decrease with an increase of the generator temperature and condenser temperature

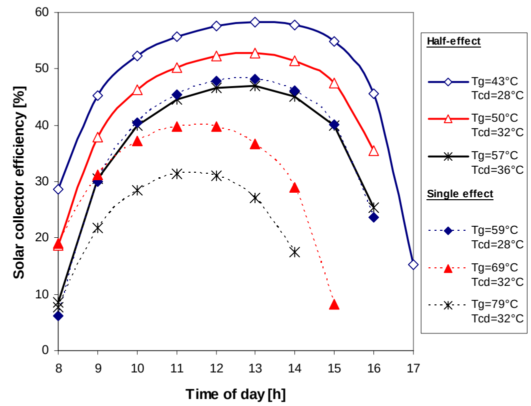

Fig.6. Variation solar collectors thermal efficiency versus time of day and generator temperature 'Tg' and condenser temperature 'Tcd'

Figures 7, 8 and 9 show the variation of energy delivered to the generators (LPG and HPG for half-effect system and to the generator for single effect system) and the contribution percentage of solar energy and auxiliary heating load. When the condenser temperature is fixed at 28°C, 32°C and 36°C and generators temperature giving the maximum COP and exergy efficiency are considered For single effect system it can be seen that between time of day 10 and 14 for single effect system the solar collectors provide about 89%, 87% and 84% of heat energy required respectively with a cover of about 95% between time of day 11 and 13 which correspond to the maximum of solar radiation. The daily cover (between time of day 8 and 17) is about 58%, 53% and 48% respectively. Between time of day 10 and 14 the solar collectors provide about 93%, 92% and 91% of heat energy required respectively for the half-effect system with a cover of about 98% between time of day 11 and 13. The daily cover for the half-effect system is about 65%, 62% and 59% respectively. It is clear that the high values of condenser temperatures influence negatively the rate of solar energy provided. The low solar radiation values early in the morning and later in the afternoon hours cause a significant increase in the required auxiliary heating. From the view point of the percentage of heat load covered by solar energy the half effect absorption cooling system is more efficient than the single effect system.

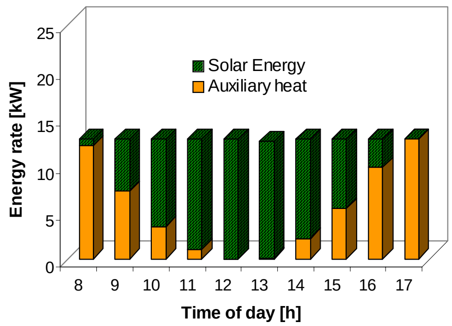

Figure 7a. Single effect system. Contribution percentage of solar energy and auxiliary heating load (Tcd=28°C, Tg=59°C)

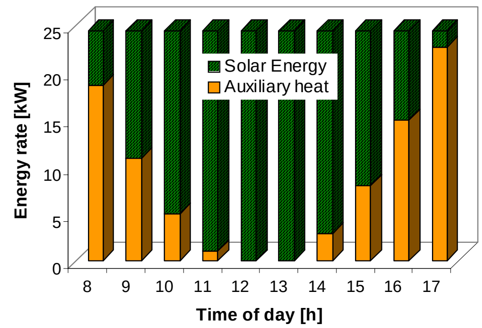

Figure 7b. Half-effect system. Contribution percentage of solar energy and auxiliary heating load (Tcd=28°C, Tg=43°C)

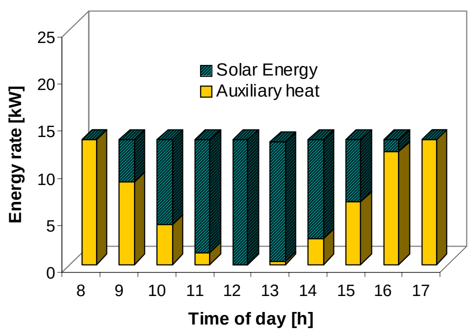

Figure 8a. Single effect system. Contribution percentage of solar energy and auxiliary heating load (Tcd=32°C, Tg=69°C)

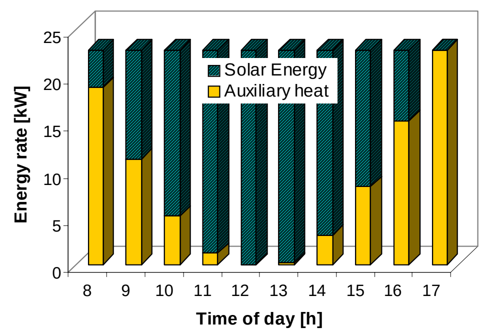

Figure 8b. Half-effect system. Contribution percentage of solar energy and auxiliary heating load (Tcd=32°C, Tg=50°C)

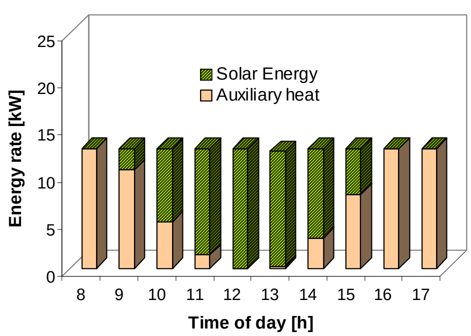

Figure 9a. Single effect system. Contribution percentage of solar energy and auxiliary heating load (Tcd=36°C, Tg=79°C)

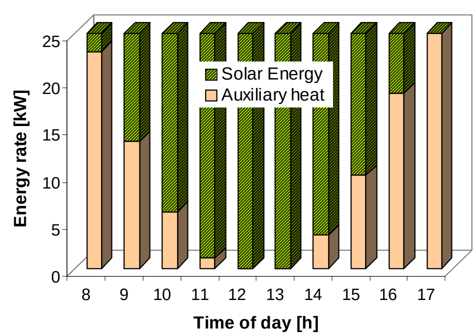

Figure 9b. Half-effect system. Contribution percentage of solar energy and auxiliary heating load (Tcd=36°C, Tg=57°C)

CONCLUSION

In this paper, an attempt has been made to study the combination: flat plate solar collectors and single effect or half effect absorption cooling systems. The main results obtained are concluded below:

- By using a half-effect LiBr absorption chiller instead of single effect chiller, it is possible to decrease the required generator temperature of the chiller and this allowed the use of simple flat plate collectors to produce hot water to be used as the hot source for absorption chiller.
- When the evaporator temperature is maintained constant at 5°C and condenser temperature is varied from 28°C to 36°C and generators temperatures are varied from 42 to 83 °C, for single effect absorption cooling systems the maximum COP is 0.82 and the maximum exergetic efficiency is about 30%. For half-effect systems the maximum

COP is 0.44 and the maximum exergetic efficiency is about 20%

- For a given condenser temperature there is an optimum generator temperature for which the N_{FPC} is minimum. This optimum generator temperature corresponds to the generator temperature giving the maximum COP and exergetic efficiency of the absorption cooling system.
- We note the existence of a maximum value of solar collector thermal efficiency, $\eta_{th\text{-}FPC}$ of roughly 60% obtained with half-effect system (about 45% for single effect system).
- The thermal efficiency of the global system will decrease with an increase of the generator temperature and condenser temperature
- From the view point of the percentage of heat load covered by solar energy the half-effect absorption cooling system is more efficient than the single effect system.

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