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ORIGINAL ARTICLE

The Verification of the Payback Time for a Solar Driven Absorption Cooling System Depending on Technological Development and Design Data

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ABSTRACT

In this study a solar driven absorption cooling system is offered for a hotel. Solar energy is only used in absorption chiller. The payback time of the offered system with different scenarios are investigated. Solar collectors and absorption chiller price are the main contributors to the total system price. The effects of the unit prices of the absorption chiller and solar collectors depend on the technology development. However, electricity prices increases day by day due to the increased fuel prices. Also, storage option is taken into account to increase the load factor of the offered system. The effect of the carbon prices are also investigated for the offered system for different ton prices of the carbon. The best and worst scenarios are given with respect to the variation of the prices and load factor. The results show that, the payback time changes 2.18 to 20.3 years. With 2010 prices with 10 $ per ton carbon credit the payback time is calculated 10.1 years. However, the cost of electricity and carbon increases and the offered system payback time could be logical in less than ten years.

Key Words: *Absorption chiller, solar energy, carbon credit, payback time.*

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1. INTRODUCTION

In summer seasons, electricity consumption increases due to the increased cooling demand for covering the thermal comfort conditions in hotels. To provide the cooling demand, vapor compression air conditioning systems are widely used because of their low first investment costs and user friendly operation characteristics when compared to other alternative air conditioners. The compressor of vapor compression devices consumes great deal of electricity. Instead of vapor compression, chemical compression devices for instance, absorption cooling, might be an alternative to decrease the electricity demand in summer season. Absorption cooling devices consume heat, instead of electricity and can be adapted instead of vapor compression systems when considering energy economy and sustainability. Sustainable development depends on environmental sustainability, energy sustainability and energy economy [1]. Environment, energy and economy become highly important and connected together tightly to satisfy the sustainability. Every system designer must consider these issues before designing a system. Global warming is one of the biggest problems which made all countries try to find a method to decrease their green house gas emissions under the limit values. Increased electricity demand means increased gas emissions due to the combustion of fossil fuels. As an alternative cooling method, absorption cooling devices can be commissioned instead of vapor compression devices. Absorption cooling systems are environment friendly systems and have no harmful effects when the systems run by the waste heat or renewable sources. Absorption cooling systems (ACS) have a growing trend in the cooling market. However, the first investment and auxiliary equipment costs are still nearly five times bigger than vapor compression air conditioners.

Absorption cooling devices are able to be run directly or indirectly [2]. Indirect ACS utilizes heat sources from renewable energies such as solar and geothermal energy to produce the cooling effect. Direct ACS produces heat from burning natural gas then utilizes heat in ACS units. Directly operated ACS option depends on the natural gas price when compared to electricity selling price. Economic and engineering calculations become very important to decide the using ACS instead of vapor compression systems (VCS). Indirect fired ACS decreases electricity consumption and due to the using of renewable energy or waste heat, energy efficiency and sustainability can be satisfied. The waste heat utilization is the ultimate purpose of increasing the energy efficiency of a system. Solar and geothermal energy usage in an ACS is difficult due to the source temperature for geothermal energy and discontinuity of solar energy. Solar radiation and cooling demand of humankind increases simultaneously, but the solar radiation has a discontinuous characteristic and energy storage systems become more important when an absorption cooling device is driven by solar energy. Parabolic trough type solar collectors are used to obtain 140-150°C steam temperature which is enough to drive a double effect ACS. Geothermal energy sources are more convenient to drive a single effect absorption cooling device due to its low heat capacity. The coefficient of performance (COP) of ACS is to be increased to be able to cope with vapor compression systems.

Two main working fluid pairs are widely used in absorption cooling systems, $LiBr-H_2O$ and H_2O-NH_3 [3]. The first component is named as absorbent and the second as coolant. The working fluid pair selection is very important for satisfying the cooling demand. Below 4°C, LiBr becomes crystal and crystallization badly affects the performance of the system. For H_2O-NH_3 working fluid pair, crystallization is not a problem, however, the investment cost is higher because of the separation column of NH_3 and controlling the system becomes more important. $LiBr-H_2O$ working fluid pair is suitable for air conditioning applications due to the crystallization characteristic of LiBr. Some other working fluid pairs are also available or under research. In an absorption chiller, a solution pump is commissioned which pumps the solution to the generator and pump is the only part of an ACS which consumes electricity. This electricity consumption is neglected in thermodynamic calculations when compared to heat loads of other components. However, in vapor compression system, compressor consumes large amounts of electricity to compress the vapor. When two cooling systems are compared a generator and an absorber is added instead of the compressor in VCS. ACS is driven by heat source instead of electricity. Chemical separation of absorbent and coolant is driven by heat source, fed into generator in ACS.

Solar energy, geothermal energy or waste heat has lower exergy when compared to fossil fuels. The source temperature and the mass flow rate of source are significant to decide on the type of an ACS. If the source temperature is around 100°C, using a single effect ACS is more applicable or the source temperature is around 150°C a double effect ACS is preferred due to its higher COP. A heat recovery option can be applied and COP can be increased by using low temperature generators in double effect ACS. The COP value for single effect and double effect ACS is around 0.7 and 1.2 respectively at full load conditions [2,19]. Florides et al. [5] presented a method to evaluate the characteristics and performance of a single effect $LiBr-H_2O$ absorption chiller. Single pass vertical tube heat exchangers were used for absorber and evaporator. The solution heat exchanger was designed single pass annular heat exchanger. Condenser and generator were considered horizontal tube heat exchangers. Overall heat transfer coefficients were given for the selected type heat exchanger at working conditions. Coefficient of performance was found 0.704, for 10 kW evaporator (cooling) load. Wang et al. [4] performed thermodynamic analysis of gas fired air cooled adiabatic absorption refrigeration system. In the analysis 16 kW small sized absorption chiller was simulated and the effects of ambient temperature to the solution temperature and COP was investigated. As a result, COP decreases while the ambient air temperature rises. Sencan et al. [6] performed exergy analysis of a single effect lithium bromide/water absorption cooling and heating system. Exergy losses of each component was calculated and as a result exergy loss of absorber and generator greater than other components due to the mixing and circulation. The effect of heat source temperature was also carried out and at increasing heat source temperatures COP also increases but exergetic efficiency decreases with increasing source temperatures. Liu and Wang [7] investigated performance of a solar/gas driving double effect LiBr-H_2O absorption system. In the system a solar collector and a natural gas burner connected to a water tank. Heat supply to the generators is taken from both systems. Economic comparison of the other alternatives was performed and the solar/gas driven double effect absorption chiller with a 13.6 m^2 solar collector area was found best. Won and Lee [8] made a comparison study between water-LiCl and water-LiBr. The simulation is performed in a computer program for a double effect ACS which was based on mass and heat balances. The variation of COP and flow ratios at different operation temperatures was investigated. According to the results water-LiCl working fluid pair showed better performance which COP is higher and flow ratio is lower than water-LiBr. Won and Kang [9], compared four different working fluid pairs which are water-LiCl, water-LiBr-LiSCN, water-LiCl-$CaCl_2$-$Zn(NO_3)_2$ and water LiBr. As a result, new mixtures showed better performance than LiCl and LiBr aqueous solutions. Xu and Dai [10] made a theoretical analysis and optimization of a parallel flow type double effect absorption chiller. The design parameters such as heat recovery ratio, solution circulation ratio are performed. Misra et al. [11] performed thermoeconomic optimization of a double effect ACS. Izquierdo et al. [12] made exergetic analysis of a double effect ACS driven by low grade heat. Entropy generation for air cooled and water cooled thermal compressor was given. As a result, exergy destruction in air cooled systems is higher than water cooled. Gomri and Hakimi [13] performed the second law analysis of double effect ACS. According to the results, the performance of the system increases with increasing low pressure generator temperature but decreases with increasing the temperature of high pressure generator. The highest exergy loss was found in the absorber. Gomri [14] investigated the effects of the generator temperature on COP, exergy efficiency and heat exchange. Arora and Kaushik [15] compared the performances of a single effect ACS and double effect series flow type ACS. The effects of generator temperature, absorber temperature and dead state temperature were also investigated on COP and exergy efficiency. Gebreslassie et.al. [16] analyzed ACS from half to triple effect. The effect of heat source temperature on exergy destruction was investigated. The COP values for double effect ACS were found 1.6 which is higher than the literature.

In this study, economic analysis of solar driven double effect absorption cooling systems is investigated. The effects of the unit prices, load factor of the system and carbon credit on payback time is found for the offered system. The best and worst price scenarios are given for the payback time.

2. OFFERED SYSTEM STRUCTURE

For a series flow type, LiBr-H2O (LiBr is absorbent and H2O is coolant) working fluid pair, the connection layout is given in Fig. 1. Double effect absorption chiller has three pressure zones in which generators are located in high pressure zone, condenser is placed in medium pressure zone and evaporator and absorber are considered in the low pressure zone.

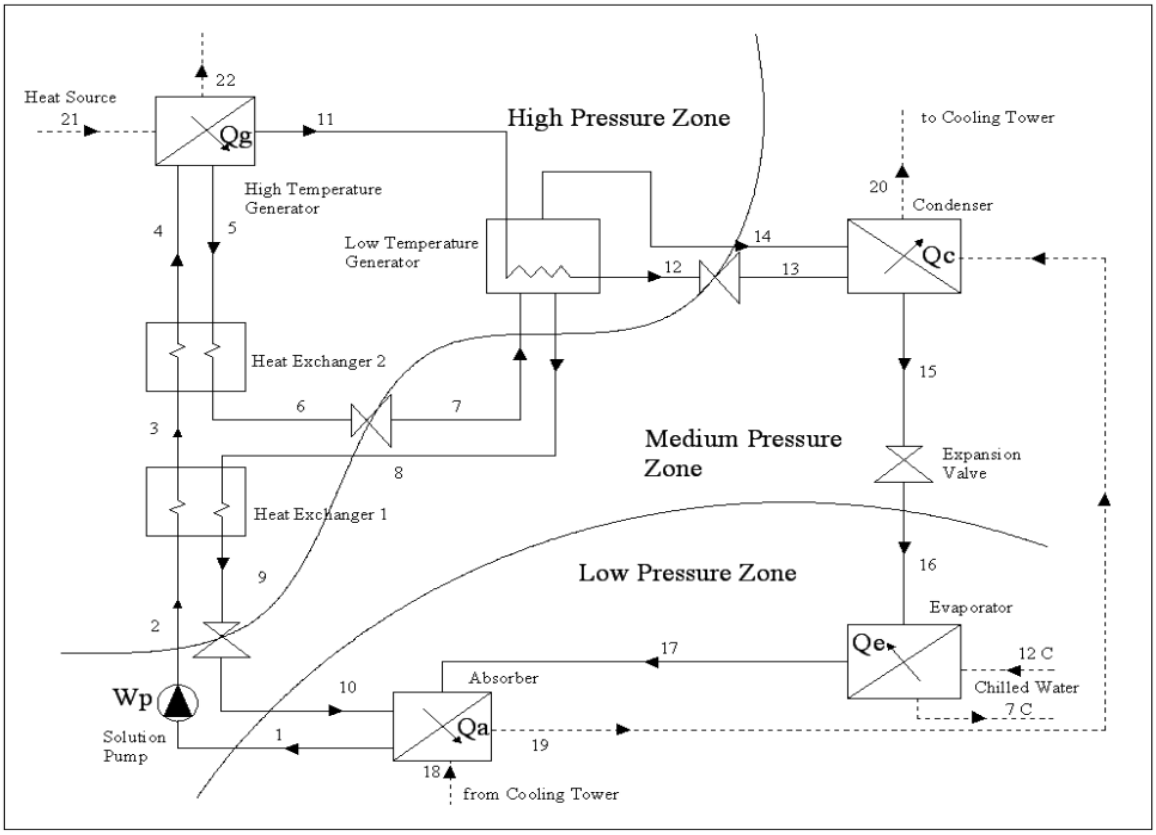

Figure 1. The connection layout of a series flow type double effect absorption chiller [17].

The working principle of a series flow type double effect absorption chiller is explained below. Steam, coming from the evaporator, is absorbed by LiBr solution and pumped to the high temperature generator through the heat exchangers by a solution pump located after the absorber. The heat source, fed to the high temperature generator, separates some quantity of H_2O from LiBr-H_2O solution. Superheated steam leaves the high temperature generator and then enters the low temperature generator to evaporate water from the solution. At the outlet of low temperature generator superheated steam becomes saturated water. The evaporated vapor mixes and condenses in the condenser and cooling water takes the condensing heat, coming from the absorber. The condensed water passes through the expansion valve and enters the low pressure zone. Due to the low pressure, water boils very low temperatures. The required heat is supplied from the chilled water which generally enters the evaporator 12°C and exits 7°C. This temperature difference supplies the cooling effect. The coolant water (steam) then enters the absorber to be absorbed by LiBr solution. On the other hand, the rich solution (the concentration of LiBr increased due to evaporation in HTG and LTG) passes through a heat exchanger and an expansion valve and both cycles (water side and solution side) are completed. In Fig. 1, stream number 21 is designated as heat source which is chosen as saturated steam for economical reasons. It can be produced either renewable or waste heat according to the availability of the source.

In Fig. 2 the connection layout of a solar driven absorption cooling system in Thermoflex software is given. Solar collector field is considered as parabolic trough type collectors to obtain 150°C steam. Solar collector field is considered as a closed loop and the heat transfer fluid is chosen as water for economic considerations. However, in the second part of this study the type of heat transfer fluid is also investigated.

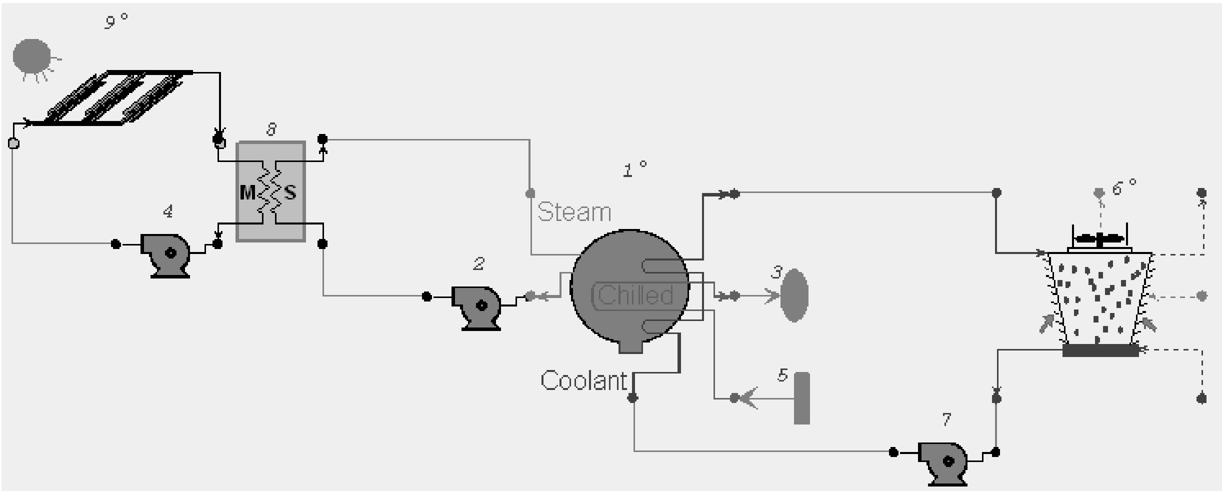

Figure 2. Solar driven absorption cooling system.

(1: absorption chiller, 2: pump, 3: chilled water outlet, 4: pump, 5: chilled water inlet, 6: cooling tower, 7: pump, 8: heat exchanger for steam production, 9: solar collector field)

In the solar collector field the pressurized water (heat transfer fluid) enters the collectors at 12.3 bar pressure and 155°C temperature with a mass flow rate of 1.4 kg per second. At the exit of the solar field the pressure and the temperature values of the heat transfer fluid are 12 bar and 180°C, respectively. In the analysis, normal direct irradiance to the collectors is accepted as 900 W/m^2. The heat transfer fluid enters a heat exchanger to evaporate water. At the exit of the heat exchanger saturated steam with a pressure of 4 bar is obtained. Saturated steam is fed to the generator of absorption cooling system and after heat exchange in the generator water is pumped to the heat exchanger again to become saturated steam. Heat is removed from absorber and condenser by cooling water which a cooling tower supplies the cooling water. Chilled water enters the evaporator at 12°C and leaves at 7°C. Stream 3 enters the fan coils, located in rooms or different places in the hotel.

Parabolic trough type collectors are used in solar field design. Irradiance is focused on an absorber tube at the focal point of the parabolic shape collector. Coaxial two pipes, a glass tube and an absorber tube, is used. The air, between two tubes, is vacuumed to prevent convection heat losses. A sun tracking system on east-west direction is considered. Useful heat transfered to the heat transfer fluid and efficiency of the solar collector can be found from Eq. 1 and 2.

$$\dot{Q}_{HTF} = F_R A_{ra} \left[\dot{q}_G - \frac{A_{ata}}{A_{ra}} U_L (T_i - T_a) \right] \quad (1)$$

$$\eta = \frac{\dot{Q}_{HTF}}{A_{ra} I_d} \quad (2)$$

In Eq. 1, F_R, A_{ra}, U_L, T_i and T_a indicate collector heat gain factor, reflector area, heat transfer coefficient, inlet temperature of heat transfer fluid to the collector and ambient temperature, respectively. In Eq. 2, I_d indicates the direct irradiance to the reflector area [18]. There are also other factors which affect the performance of a collector, such as optical efficiency, collector cleanness factor and so on. Optical efficiency and cleanness factor are accepted as 0.75 and 0.95 in this study. Other design specific formulations for parabolic collectors can be found on the literature.

In Table 1 some design and operational oriented input values of solar collector field, heat exchanger, absorption cooling system and cooling tower is given. According to this input values solar collector field design and total cooling load can be calculated.

Table 1. Some input values of the components based on the designed solar cooling system.

Solar collector field	**Receiver tube outside diameter [mm]**	**38**
	Receiver tube wall thickness [mm]	**2.7**
	Reflector aperture width [m]	**5**
	Reflector geometric concentration ratio [-]	**131.6**
	Reflector rim angle	**80°**
	Receiver emissivity	**0.15**
	Glass envelope emissivity	**0.9**
Steam supply heat exchanger	**HTF exit temperature [°C]**	**155**
	Steam quality [-]	**1**
Absorption chiller	**Steam condensate subcooling [°C]**	**2**
	Chilled water exit temperature [°C]	**7**
	COP	**1.1**
Cooling tower	**Temperature difference water side [°C]**	**10**
	Temperature difference air side [°C]	**7**

3. RESULTS

According to the given input values some of the results are summarized in Table 2. In Table 2 two different efficiency terms are given for solar field. Collector efficiency indicates the ratio of energy absorbed by heat transfer fluid to energy impinging active collector area. Field efficiency is the ratio of energy absorbed by heat transfer fluid to energy impinging field area. In the simulations ambient temperature and relative humidity are accepted as 30°C and 60%, respectively.

Table 2. Simulation results of solar driven double effect absorption cooling system.

Solar collector field	Reflector aperture area [m^2]	256.8
	Collector field required land area [m^2]	1177
	Estimated cost [$]	77,040
	Collector efficiency [%]	66
	Field efficiency [%]	14
Steam supply heat exchanger	Heat transfer [kW]	151.2
	Estimated cost [$]	2,420
Absorption chiller	Cooling capacity [kW]	140
	COP (%100 load)	0.927
	Chilled water mass flow rate [kg/s]	8.33
	Estimated cost [$]	119,980
Cooling tower	Exit air temperature [°C]	37.3
	Exit air relative humidity [%]	100
	Fan electricity consumption [kW]	1.055
	Estimated cost [$]	13,720

Estimated cost of the system components is calculated to be 213,160 $. Other costs such as piping, transportation, engineering, construction, labor etc. is taken 10 % of the estimated component cost. Total cost of the system is found to be 234,476 $ for the solar driven double effect absorption chiller. Total electricity consumption of the system is found to be 3.175 kW, including solar field pump, heat exchanger pump, and absorption chiller pump and cooling tower pump.

For the same ambient air conditions a water cooled electric chiller is also analyzed. Chilled water inlet and outlet temperatures are 12 and 7°C. COP of the electric chiller is taken 5. Cooling water temperature rise is set to 10°C. At this conditions electricity consumption of the electric chiller is found to be 29.92 kW.

Simple payback method can be used to determine such a project in a hotel. As shown in Table 2 total field requirement is 1177 m^2. This area has to be provided by hotel management. Therefore, in this study, land cost is not included the total cost of the system. The formulation of the simple payback time is given in Eq. 3. In Eq. 3, total investment cost indicates the total cost of equipments and other costs. Annual benefit indicates the decrement in electricity bill for the hotel. In Turkey, the cost of electricity is 0.2152 $ per kWh electricity. According to the given conditions the payback time is found to be 10.4 years.

$$Paybacktime[year] = \frac{Total \cdot investment \cdot \cos t[\$]}{Annual \cdot benefit[\$ / year]} \quad (3)$$

However, the solar collector cost, absorption chiller cost and electricity cost directly effects the payback time. In Fig. 3 the effect of unit collector area cost on payback time is given. The effect of the cost of absorption chiller per kW cooling on payback time is given in Fig. 4. The effect of the cost of electricity on payback time is shown in Fig. 5. The cost of solar collector field and the absorption chiller depends on the technology improvement and low cost materials usage on the production. As seen from the graphs the payback time is shortened 1.5 years for the best scenario. However, the cost of electricity strongly affects the payback time. In the best scenario (highest electricity price) the payback time shortens roughly half of the today's electricity prices.

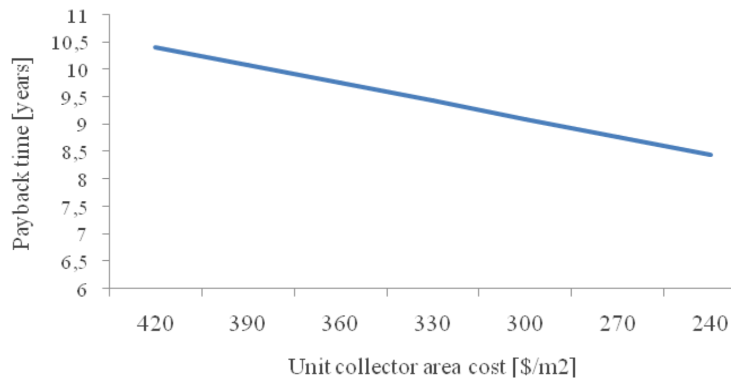

Figure 3. The effect of unit collector area cost on payback time.

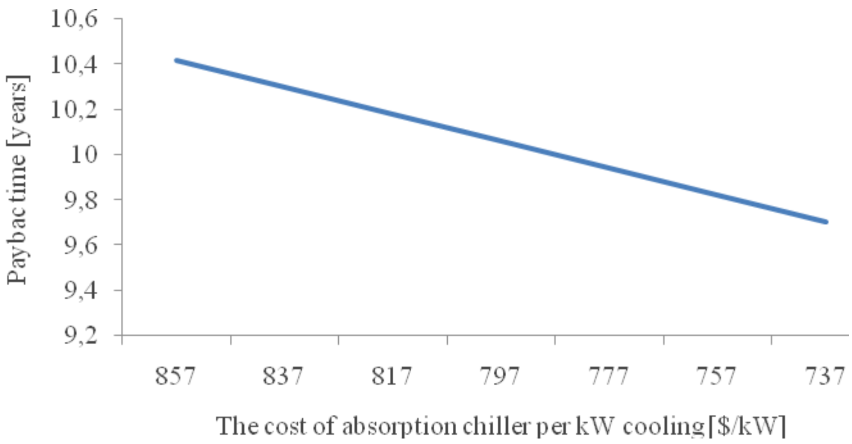

Figure 4. The effect of the cost of absorption chiller on payback time.

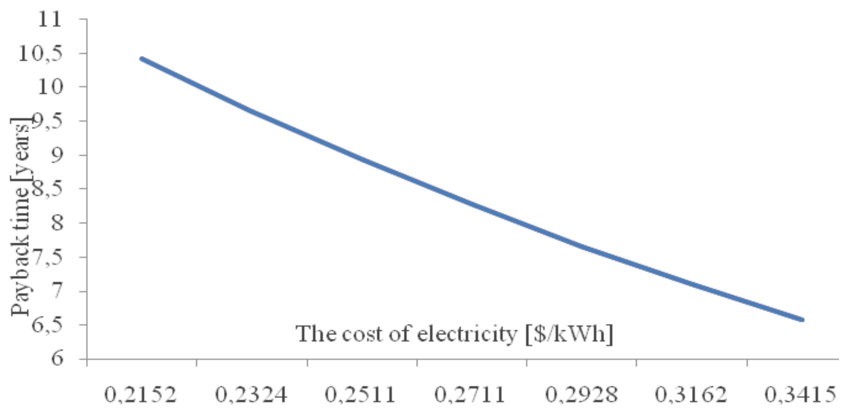

Figure 5. The variation of the cost of electricity and the effects of payback time.

Solar energy can be stored different ways, hot heat transfer fluid storage and cold chilled water storage. Heat or cold storage affects the load factor of the system. Cold storage is more applicable in HVAC applications due to the low temperature difference between storage media and ambient. Load factor of the designed system can be increased by storage. Fig. 6 shows the effect of the load factor on payback time. When compared to other scenarios the effect of the yearly operation hours of the system is more than the others. Therefore, this system with storage has to be applied in a hotel which works at least 6000 hours in a year.

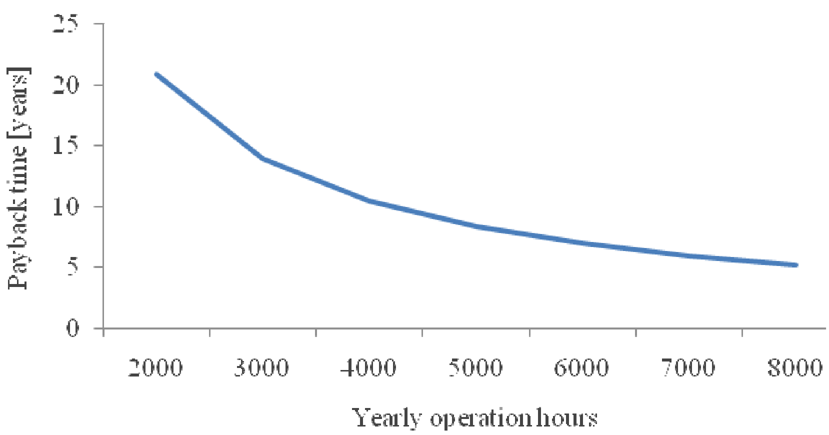

Figure 6. The effect of the heat storage on payback time.

The offered system for a hotel is a renewable energy process. Therefore, carbon credit has to be taken into account when payback time is calculated. In Fig. 7 the effect of the variation of carbon price on payback time is given.

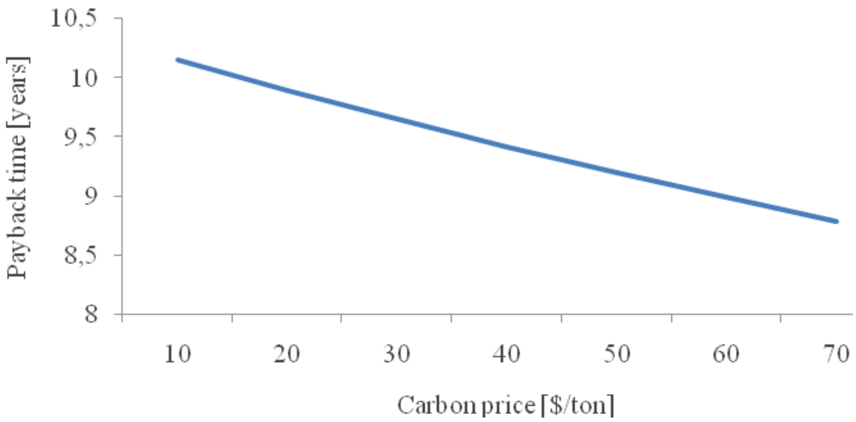

Figure 7. The effect of carbon price variation on payback time.

According to the results given above different scenarios are investigated. In Table 3 total effects of the cases are given. Case 1 and Case 7 reflects the worst and best scenarios of the offered system.

As seen from Table 3 payback time changes 2 to 20 years according to the unit prices and load factor of the system. With 2010 prices and 4000 hours operation in a year the payback time of the system is found 10.4 years without carbon credit. The payback time with 2010 prices seems to be higher for an energy conversion system. However, the cost of electricity and carbon increases year by year and the offered system price could be logical in less than ten years.

Table 3. Different scenarios of the offered system.

	Case 1	Case 2	Case 3	Case 4	Case 5	Case 6	Case 7
Unit collector price [$/m2]	420	390	360	330	300	270	240
Absorption chiller unit price [$/kW]	857	837	817	797	777	757	737
Electricity cost [$/kWh]	0,2152	0,2324	0,2511	0,2711	0,2928	0,3162	0,3415
Load Factor [h/a]	2000	3000	4000	5000	6000	7000	8000
Carbon price [$/ton]	10	20	30	40	50	60	70
Payback time [years]	20,30	11,73	7,64	5,31	3,85	2,87	2,18

CONCLUSIONS

In this study a solar driven absorption cooling system is offered for a hotel. Solar energy is only used in absorption chiller. The payback time of the offered system with different scenarios are investigated. Solar collectors and absorption chiller price are the main contributors to the total system price. The effects of the unit prices of the absorption chiller and solar collectors depend on the technology development. However, electricity prices increases day by day due to the increased fuel prices. Also, storage option is taken into account to increase the load factor of the offered system. The effect of the carbon prices are also investigated for the offered system for different ton prices of the carbon. The best and worst scenarios are given with respect to the variation of the prices and load factor. The results show that, the payback time changes 2.18 to 20.3 years. With 2010 prices with 10 $ per ton carbon credit the payback time is calculated 10.1 years. However, the cost of electricity and carbon increases and the offered system payback time could be logical in less than ten years.

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