ENERGY COMPARISON OF SOLAR ABSORPTION COOLING WITH CLASSICAL COOLING SYSTEM

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The paper examines the possibilities of applying solar cooling in B&H, since the potential of solar energy is great. The goal is to reduce primary energy consumption, as well as to reduce emissions of harmful gases, primarily CO_2. The feasibility of a single-stage H2O / LiBr absorption cooling device powered by heat from solar collectors for a building in Tuzla was investigated, as well as a comparison with the existing compression cooling device. A solar cooling system was selected for the calculated heat load values. A mathematical model for the absorption single-stage cooling cycle has been developed that includes mass and energy balance equations as well as heat exchange equations. The absorption system is designed for the maximum cooling load of the specified facility.

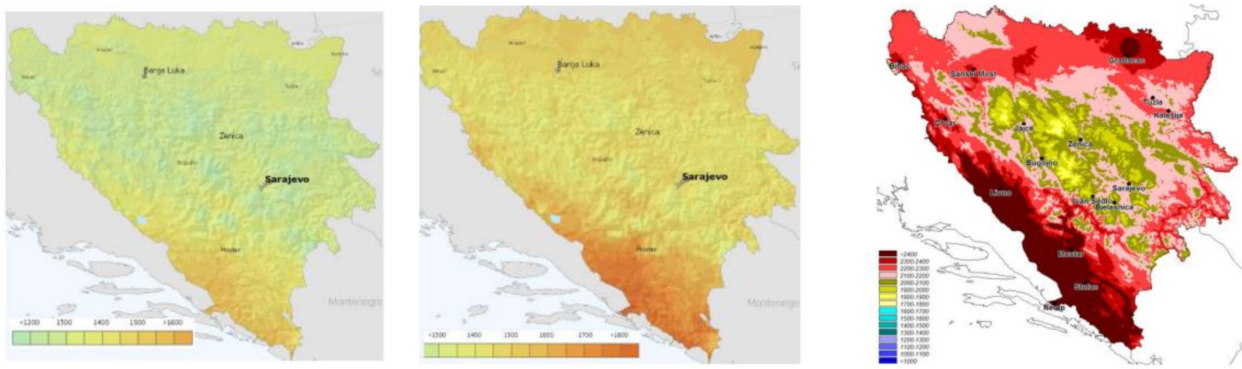
Keywords: solar absorption cooling, compression cooling, comparison, cooling coefficient, energy saving.

1. Introduction

Global climate change has caused the temperature on Earth to rise, and with it the need for cooling. Also, the requirements for cooling energy are increasing due to the increased "waste heat" of a large number of office devices. On the other hand, the increase in energy consumption of compressors in the summer can lead to overload of the electrical network. These are the reasons why renewable energy sources are increasingly used in this area as well. Precisely because of the gradual reduction in the use of coolants with high potential global warming, research is focused on the development of solar cooling technologies. In the period from 1980 to 2000. the total area of air-conditioned space increased from 30 to over 150 million m^2. The annual energy required for cooling in the period from 1990 to 2010 increased from 6 TJ to as much as 160 TJ, and nowdays significantly more [1]. Previous research in the field of solar cooling was mainly focused on optimizing the size of system components, as well as examining the impact of the share of auxiliary heat source on system performance, and comparing these systems with compression. *Vliet and associates [2]* developed in 1982. dynamic computer code for further refrigeration improvement, which was included in the 1997 ASHRAE Funmentals Handbook. *Grossman [1]* described 2001. trends in absorption technology using 1-stage, 2-stage and 3-stage cycles, as well as their costs. The first important EU project *Solar Air Conditioning in Europe (SACE)* [1] was presented in 2002. with the aim of confirming the potential of various heat-driven cooling technologies for the use of solar systems, as well as future needs and estimates of solar cooling in Europe. Of the total number of projects, 40% are in the research phase, 40% in the development phase, and only 20% are in the implementation phase. The conclusion is that in the area of Southern Europe, solar-assisted cooling systems can lead to primary energy savings of 40 to 50%. The SACE project was followed by programs by IEA, IEASHC, Solair, Solera, SolarCombi +, Medisco, Climasol. *G.A. Florides and S.A. Kalogirou [3]* performed in 2007. simulation of the absorption cooling system in Cyprus, with a capacity of 11 kW (using TRNSYS and METEONORM software), and the influence of the field size of solar collectors and tanks on the system performance was investigated. *R. Gomri* [4] performed in 2010. energy and exergy analysis of systems with a cooling capacity of 10 kW and showed that multistage systems have a better COP, and that at higher temperatures it grows approximately linearly. *Mr. Zidianakis, Th. Tsoutsos and N. Zografakis* [5] made in 2007. economic analysis of the cost-effectiveness of a system covering a heat load of 160 kW in Crete for different cases (solar, compression and absorption cooling with a boiler as a heat source). As expected, where the greatest energy savings are achieved, the most expensive is the installation of the system, and therefore the longest and the repayment period of the same. *B. Pavković, B. Delač and V. M. Viola* [6] performed in 2014. economic analysis of the cost-effectiveness of solar cooling at the Thalassotherapia Hospital in Crikvenica. *K.K. Upman, B.L. Gupta, D. Kumar and P. K. Baheti* [7] in 2017. sensitivity analysis found that the repayment period can be shortened to a period shorter than the life of the components (<20 years) only under conditions of increasing the price of electricity by 50% and reducing total investment by 50%, which would be very interesting today. *L. Huang and R. Zheng* [8] showed in 2018 that for China with warm summers and cold winters, a system of lower cooling capacity (65 kW) is not economically viable, and that systems using photovoltaic panels in such areas are cost-effective, with a repayment period of 6 - 7 years. There are 17 large solar systems connected to cooling networks in Europe, which means that compared to the potential of solar energy to generate cooling, the levels of utilization are still very low. Of the total number of existing solar cooling systems, 71% work with absorption cooling devices, 13% on adsorption and 16% on desiccant systems [9]. Various technical solutions are available on the market for the use of solar thermal energy for summer air conditioning of buildings with closed or open systems. Solar cooling systems have a number of advantages over conventional systems, because they do not have harmful refrigerants or a negative impact on the ozone layer, even with a low heating coefficient of solar equipment, as well as higher investment costs.

2. Overview of potentials and use of solar energy in B&H

The most important meteorological data needed for simulation and sizing of solar cooling systems are: temperature and humidity, direct and diffuse solar radiation, as well as microclimate. The research project Study of Possibilities of Using and Promoting the Development of Solar Energy in B&H [10], determined that the area of B&H has favorable conditions for the use of solar energy based on solar irradiation databases Photovoltaic Geographical Information System (PVGIS) (Fig. 1a and 1b). Regarding the northern part of B&H (Fig. 1a and 1b), data on insolation and global radiation, the average insolation from the 5th to the 9th month lasts over 6 hours. The maximum exposure to the sun in July is up to 9 hours. Diagrams of global and diffuse radiation give similar data for the areas of Tuzla and Sarajevo, with over 160 kWh /m^2 from May to August, with a maximum value of about 195 kWh /m^2 in July. These periods of insolation in the southern part of B&H last longer and amount to as much as 220 kWh /m^2 (Fig. 1c) .



a) b) c)

Figure 1. a) Average value annual horizontal irradiation, b) Average optimal global irradiation, c) Distribution of insolation in B&H

It can be stated that the solar potential in B&H is significant, especially in the part of the country that gravitates towards the Adriatic Sea, and that the southern part of B&H is the most suitable for the construction of these plants. However, it is only partially used, both due to the poor economic situation and the ignorance of potential users to use these technologies for heating and cooling purposes. The use of solar energy for cooling purposes in B&H has not yet been launched.

3. Theoretical fundamentals and mathematical model of solar absorption cooling system

The paper analyzes only systems that use thermal energy for propulsion (Fig. 2).

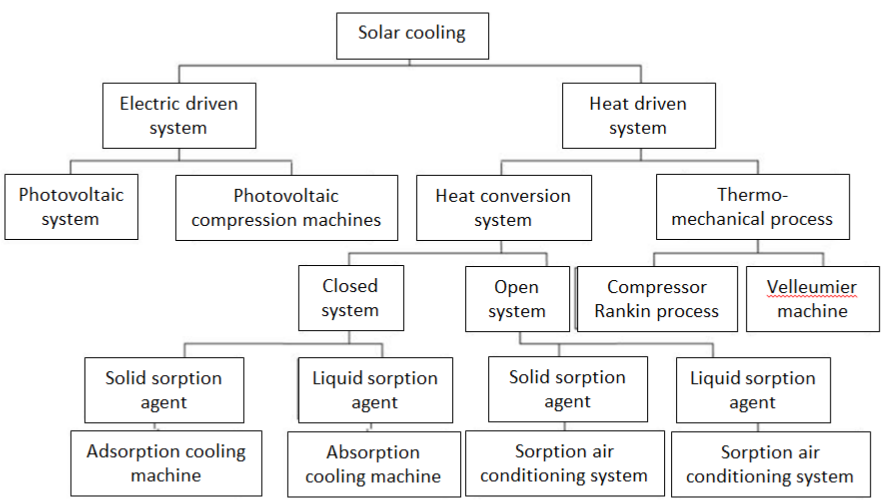


Figure 2. Solar energy conversions [12]

The solar cooling system comprises three main parts: solar energy conversion equipment, the cooling system and the consumer (Fig. 3).

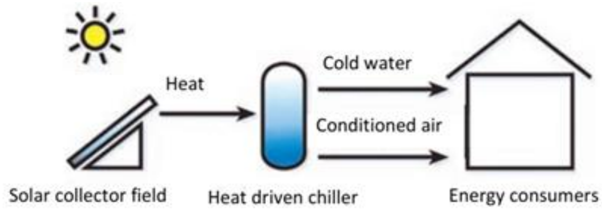


Figure 3. Solar cooling systems [13]

According to the theoretical assumptions of solar cooling, a mathematical model is set in the paper.The components of the system are calculated, and their adoption is performed using commercially available components. Based on energy balances, an analysis of the impact of some of the parameters on system efficiency was performed. Using the MATLAB software, the parameters in individual states (points) of the absorption cooling cycle were calculated. The following assumptions were included in the analysis: the paper compares the energy consumption of the solar cooling system with the existing compression cooling system; a system without an auxiliary heat source is considered here, and the collector surface is selected so that all the required heat is obtained from the collector; during cloudy days, the heat from the hot tank will be used. The comparison of the two cooling systems was performed for the following conditions: the cooling system works 1500 hours a year, in the summer period from the 5th to the 9th month, from 8 am to 6 pm. In order to develop a model for simulating the H_2O /LiBr absorption system, each component is treated as a control volume with its own inputs and outputs. Fig. 4 shows the considered plant with its constituent components.

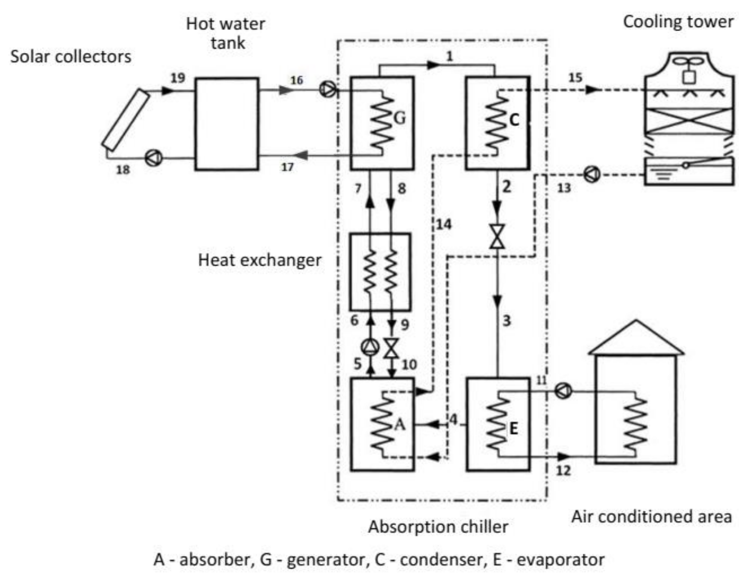


Figure 4. Scheme of a single-stage solar-cooled absorption cooling system

It should be noted that similar systems were the basis for various experiments with solar cooling. The radiated solar energy on the solar collector absorber plates is transferred to the working medium in the collector (18-19). Heat energy is accumulated in the hot water tank which is further used in the generator to evaporate water vapor from the H_2O /LiBr solution. In the period of insufficient sunshine, a hot water tank is provided. For energy analysis of annual system performance, stationary models are sufficiently precise, with simplifications. For each of the components of the absorption refrigeration device, the mass balances apply [16]:

$$\dot{Q}_{Ea} + \dot{Q}_{Ga} + P_P = \dot{Q}_A + \dot{Q}_K \tag{1}$$

Where are: $\dot{Q}_{Ea}$ – actual chilling effect in evaporator (W); $\dot{Q}_{Ga}$ - actual heat input in generator (W); P_P - pump power (W); $\dot{Q}_A$ – heat rejected in apsorber (W); $\dot{Q}_K$ - heat rejected in condenser (W).

Table 1. Mass and heat balances of the absorption cycle components.

Component	Mass balance		Heat balance	
Absorber	$\dot{m}_4 + \dot{m}_{10} = \dot{m}_5$	(2)	$\dot{Q}_A = -\dot{m}_5 h_5 + \dot{m}_4 h_4 + \dot{m}_{10} h_{10}$	(10)
	$\dot{m}_{10} X_{10} = \dot{m}_5 X_5$	(3)	$\dot{Q}_A = \dot{m}_w c_w (T_{14} - T_{13})$	(11)
Generator	$\dot{m}_1 = \dot{m}_7 - \dot{m}_8$	(4)	$\dot{Q}_G = \dot{m}_8 h_8 + \dot{m}_1 h_1 - \dot{m}_7 h_7$	(12)
	$\dot{m}_7 X_7 = \dot{m}_8 X_8$	(5)	$\dot{Q}_G = \dot{m}_G c_w (T_{16} - T_{17})$	(13)
			$\dot{Q}_{Ga} = HIF \cdot FCF \cdot \dot{Q}_G$	(14)
Condenser	$\dot{m}_1 = \dot{m}_2$	(6)	$\dot{Q}_K = \dot{m}_1 (h_1 - h_2)$	(15)
	$\dot{m}_{14} = \dot{m}_{15}$	(7)	$\dot{Q}_K = \dot{m}_w c_w (T_{15} - T_{14})$	(16)
Evaporator	$\dot{m}_3 = \dot{m}_4$	(8)	$\dot{Q}_E = \dot{m}_4 (h_4 - h_3)$	(17)
	$\dot{m}_{11} = \dot{m}_{12}$	(9)	$\dot{Q}_E = \dot{m}_E c_w (T_{11} - T_{12})$	(18)
			$\dot{Q}_{Ea} = CCF \cdot FCF \cdot \dot{Q}_E$	(19)

In the above equations $\dot{m}$ denotes mass flows (kg/s); X concentration of solution; $\dot{Q}$ exchanged heat (W); T temperature (ºC); h enthalpy (kJ/kg); the indexes 1 - 4 refer to refrigerant, 5 - 10 to solution, 11 and 12 is chilled water, 13 - 15 is cooling water, and 16 and 17 is hot water. The enthalpies of the solution in some states depend on the LiBr content and can be determined as:

$$h = \sum A + T \sum B + T^2 \sum C \tag{20}$$

where are : *A*, *B* and *C* coefficients that depend on the concentration of the solution and are calculated according to [16].

The pressures and temperatures of the solution are calculated from:

$$\log(p) = k_1 + \frac{k_2}{T} + \frac{k_3}{T^2} \tag{21}$$

$$T = \frac{-2k_3}{k_2 + \sqrt{k_2^2 - 4k_3 \cdot (k_1 - log(p))}} \tag{22}$$

Where are: k1, k2 , k3 coefficients for the calculation of temperatures and pressures that are calculated according to [14].

An important parameter that describes the efficiency of chillers is the Coefficient of Performance (COP). This is the ratio of the chilled circuit output $\dot{Q}_{Ea}$ and the required drive power $\dot{Q}_{Ga}$:

$$COP = \frac{Cooling\ capacity}{Input\ capacity} = \frac{\dot{Q}_{Ea}}{\dot{Q}_{Ga} + P_P} \approx \frac{\dot{Q}_{Ea}}{\dot{Q}_{Ga}} \tag{23}$$

Thereby the pump power (P_P) is negligible. The energy balance of the solar collector can be written in a form:

$$G_T \cdot A_c = \dot{Q}_{use} + \dot{Q}_{loss} + \dot{Q}_{col} \tag{24}$$

The total irradiated solar energy on the absorber of solar collector $G_T \cdot A_c$ (W), is in one part transmitted to the transmission fluid as the useful heat $\dot{Q}_{use}$ (W), in part it loses to the environment as $\dot{Q}_{loss}$ (W), $\dot{Q}_{col}$ (W) is the energy stored in the collector ($\dot{Q}_{col} = 0$ - if the system state is stationary). The ratio of the useful and total irradiated solar energy represents the efficiency of the solar collector [23]:

$$\eta = \frac{\dot{Q}_{use}}{\dot{Q}_{tot}} \tag{25}$$

The collector heat losses are defined by the collector heat loss coefficients (a_1 and a_2) so the efficiency of the solar collector can also be represented in a form:

$$\eta = \eta_0 - a_1 \frac{t_m - t_a}{G_T} - a_2 \left(\frac{t_m - t_a}{G_T}\right)^2 \tag{26}$$

Where are: η_0 - optical efficiency, *ie.* collector efficiency without heat loss when fluid temperature equals ambient temperature; G_T - solar radiation intensity at collector surface (W/m^2); t_m - mean fluid collector temperature (ºC); t_a - ambient air temperature (ºC); a_1 - coefficient of linear loss of solar collector (W/m^{2}K) and a_2 - coefficient of square loss of solar collector (W/mK). The efficiency of the entire solar cooling system COP_{sol} depends on the coefficient of performace of the absorption chiller COP and the efficiency of the solar collector η:

$$COP_{sol} = COP \cdot \eta \tag{27}$$

4. Analysis of the operation of the absorption cooling device

For the analysis and comparison of two cooling systems, the administrative building (ground floor + 2 floors) in Tuzla, B&H was chosen. The cooling station consists of a cooling machine with plate heat exchanger, circulating pump, tank, closed expansion vessel with control and safety fittings. Based on the required reference cooling capacity of 76 kW, an absorption cooling device with characteristics was selected (Tab. 2), while in Tab. 3 gives the characteristics for the reference compression refrigeration existence.

For this type of cooling device, a typical system design is given (Fig. 5), which includes the connection of the absorption cooling device with the hot and cooling water circuit, as well as the power supply.

The paper analyzes the influence of individual quantities, in order to obtain the operating parameters of the system, since the quantities in different ways affect the altered heat, solution concentrations, etc.

Table 2. Characteristics of absorption refrigeration refrigeration devices [18]

Cooling capacity		kW	105,5
Chilled water	Temperature	°C	12,5/7
	Max operating pressure	kPa	588,1
	Rated water flow	l/s	4,6
	Allowable water flow range	%	80-120%
	Water retention volume	l	73,1
Cooling water	Heat rejection	kW	256,2
	Temperature	°C	31 inlet/35 outlet
	Max operating pressure	kPa	588,1
	Condenser pressure loss	kPa	46,2
	Absorber pressure loss	kPa	46,2
	Rated water flow	l/s	15,3
	Allowable water flow range	%	100-120%
	Water retention volume	l	194,2
Heat medium	Heat input	kW	146,2
	Temperature	°C	88 inlet/ 83 outlet
	Allowable temperature range	°C	70 min. - 95 max.
	Max operating pressure	kPa	588,1
	Generator pressure loss	kPa	60,7
	Rated water flow	l/s	7,2
	Allowable water flow range	%	30-120 %
	Water retention volume	l	84
Electrical	Power supply	W	310
Dimensions	Width	mm	1380
	Depth	mm	1545
	Height	mm	2045
Weight	Operating	kg	1800
	Dry	kg	1450
Noise level		dB	46

Table 3. Characteristics of reference compression devices [17]

Type			**CHA/K 393-P**
Cooling capacity		kW	110
Refrigerant			**R410A**
Chilled water	Temperature	°C	12/7
	Water flow	l/s	5,27
	Pressure drops	kPa	46
	Water connections	"G	$2\,{}^{1}/_{2}$"
Compressors	Type		Spiral
	Quantity		3
	Number of cooling circuits		1
Ventilators	Quantity		2
	Air flow	m^3/s	9,7
Elektrical characteristics	Power supply	kW	40
Unit with pump	Pump nominal power	kW	1,5
	Static pressure	kPa	140
	Water retention volume	l	400
	Expansion vessel	l	12
Dimensions	Width	mm	2350
	Depth	mm	1100
	Height	mm	2220
Weights	Operating	kg	940
	Dry	kg	927
Noise level		dB	79

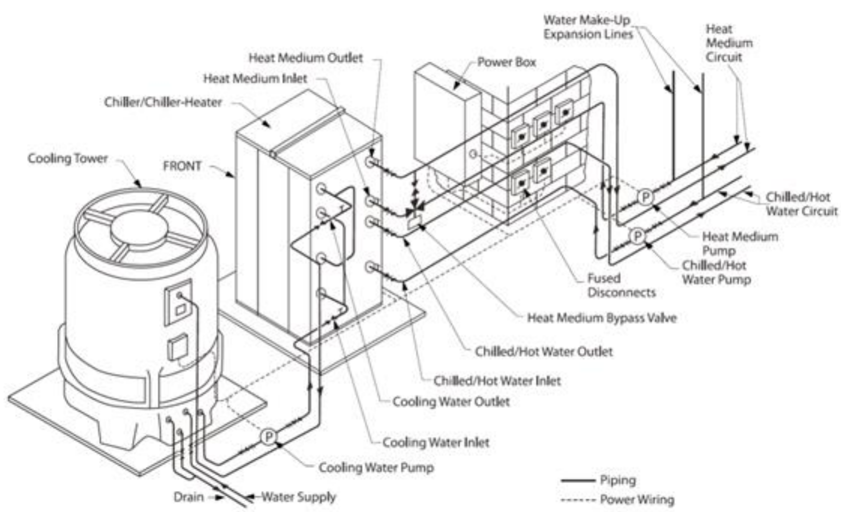


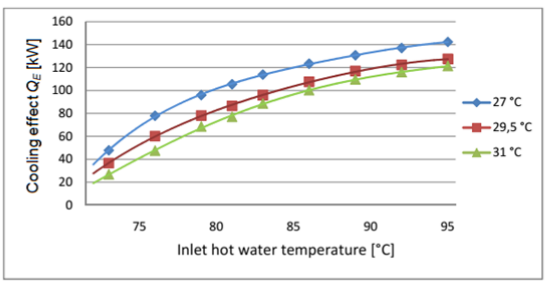
Figure 5. Typical system design [18]

4.1. Analysis of the influence of hot water

temperature at the generator inlet on system performance

The temperature of the hot medium at the inlet to the generator is one of the parameters that have the greatest impact on the behavior of the entire system. Based on expression (14), the cooling effect was obtained depending on the temperature of hot water at the inlet to the generator (Fig. 6a), and from Tab. 2 this temperature must be in the range from 70 to 95°C. Each of the 3 curves shown in Fig. 6a corresponds to the specific inlet temperature of the cooling water used to dissipate heat from the absorber and condenser. Based on Fig. 6a, it can be concluded that the cooling effect increases with increasing hot water inlet temperature on the one hand, and with decreasing cooling water inlet temperature on the other hand. Reason that is, as the inlet temperature of hot water increases, the amount of evaporated clean water in the generator increases, which first goes to the condenser, where it condenses, and then to the evaporator, where it takes heat from cold water which cools the premises. More evaporated water in the generator means a higher mass flow of coolant through the evaporator, and thus a greater cooling effect. The nominal cooling effect is achieved at a hot water inlet temperature of 87.5°C, and at a cooling water inlet temperature of 31°C.

a)



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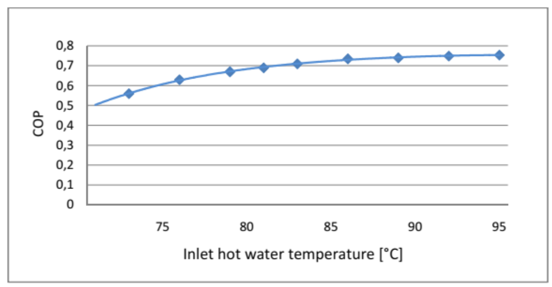


Figure 6. a) Dependence of cooling effect on temperature water at the inlet to the refrigeration unit b) Dependence of COP on operating temperature

An increase in operating temperature has a more significant effect on the increase in cooling efficiency than on the increase in operating heat. According to expression (23) it is indicated that the device shows better performance at higher operating temperatures, ie COP is higher. Fig. 6b also shows that the increase in COP is more pronounced at lower temperatures, and that at certain temperatures the increase in COP becomes insignificant. For temperatures above 90°C, additional heating of the water does not lead to a significant increase in performance, but to unnecessary heat supply to the generator.

4.2. Determination of operating parameters of the absorption cooling cycle

The MATLAB software package was used to determine the parameters of the absorption cooling cycle (Fig. 5). According to subchapter 4.1, the nominal cooling effect is achieved with hot water 87.5° C, for cooling capacity 105.5 kW, and heat supplied to the generator 146.2 kW, data on mass flows of H2O / LiBr solution at individual points of the absorption cycle are obtained , as well as mass flows of coolant - water through the evaporator and condenser, and cycle parameters (temperature and enthalpy) in Tab. 4.

Table 4. Parameters at individual process points

Operating point	t, °C	p, kPa	X, %	$\dot{m}$, kg/s	h, kJ/kg
1	86	8	0	0,0451	2652,41
2	41	8	0	0,0451	170,8
3	5	0,9	0	0,0451	170,8
4	5	0,9	0	0,0451	2510
5	40	0,9	57	1,5863	101,38
6	40	8	57	1,5863	101,38
7	80	8	57	1,5863	181,68
8	86	8	60	1,5412	204,24
9	46	8	60	1,5412	126,34
10	46	0,9	60	1,5412	126,34

When the mass flows and enthalpies of strong and weak solutions are determined, the energy balance is obtained absorption cooling device (Fig.7). Furthermore, it can be stated that of the total heat removed from the system, about 44% is removed from the absorber and about 56% from the condenser (which is according to [19]), so it can be concluded that the heat released by absorption in this device is close to heat released by condensation of the coolant.

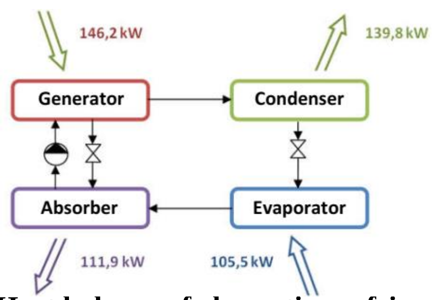
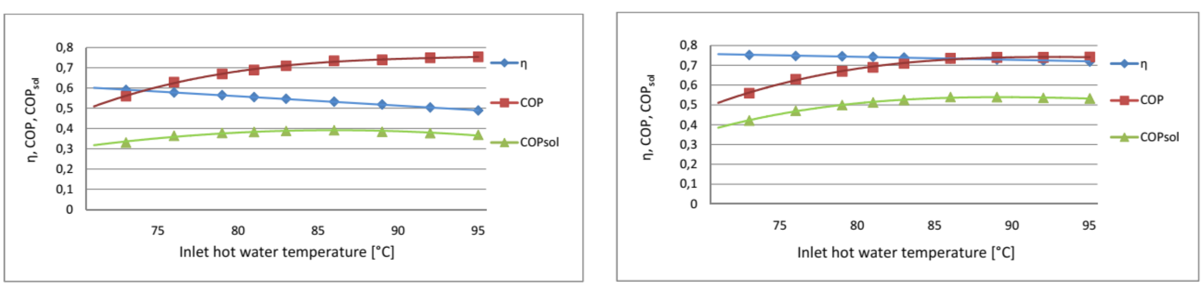


Figure 7. Heat balance of absorption refrigeration device

4.3. Analysis of the operation of solar collectors

The selection of solar collectors (required amount of heat of 146.2 kW in the cooling device generator) is made on the basis of their analysis and performance. The most suitable type of collector is determined by the temperature of the absorption cooling device, and for the area of B&H in terms of costs and performance, one type of vacuum and plate collector was analyzed.



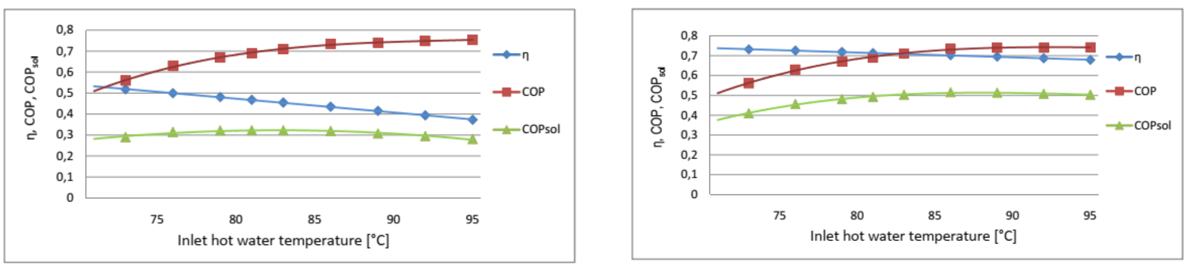
a)

b)

Figure 8. a) Dependence of efficiency of flat plate collector η, COP- absorption cooling device and total COP$_{sol}$ system from warm temperature media; b) Dependence of efficiency of vacuum tube collector η, COP- absorption cooling device and total COP$_{sol}$ from warm medium temperature media

To determine the overall efficiency of the solar cooling system, expression (26) is used. Fig. 8a and 8b show the dependences of the efficiency of the entire system in the case of using flat plate and vacuum tube collectors as well as individual dependence curves η and COP, all on hot medium temperature. For both types of collectors the figures show that the system achieves the best performance at operating temperatures between 85°C and 90°C, and that for temperatures below 83°C, the COPsol is significantly reduced. All diagrams in the figures for the degree of utilization of solar collectors are given for solar irradiation values of 1000 W/m^2 (at noon). Different areas in B&H have different solar irradiations and insolations (Fig. 1), so it can be concluded that the same solar collectors will not achieve the same effect in different areas.

When it comes to solar cooling, the warmest part of the year (from May to September) is the most interesting for the analysis. The levels of plate and vacuum collector utilization, and the total utilization rates of the solar cooling system for different values of operating temperatures on an average irradiation of 700 W/m2 for Tuzla, B&H given on the Fig. 9a and 9b respectively. The obtained data will be used to select the required collector area.



a) **b)**

Figure 9. a) Dependence of efficienecy of flat plate collector η, COP- absorption cooling devices and the total COPsol system from the temperature of the warm hot medium at an average solar irradiation in summer of 700 W / m^2 ; b) Dependence of efficiency of vacuum tube collector η, COP- absorption cooling device and the total COPsol system from the temperature of the warm hot medium at an average solar irradiation in summer of 700 W/m^2.

5. Comparison of solar cooling system with compression cooling system

5.1. Comparison of systems from an energy point of view

By replacing the compression cooling system, with the solar cooling system, primary energy savings are achieved. How much this savings will be depends on the different energy sources used to run the entire system, including all auxiliary sources as well as other energy consuming devices such as cooling towers and pumps.

Primary energy savings according to [20] are:

$$\Delta E_{PE} = E_{PEref} - E_{PEsol} = 24{,}39 \text{ MWh} \tag{28}$$

Where are: E_{PEref} - primary energy consumption of a compression (reference) cooling system (MWh); E_{PEsol} - primary energy consumption in a solar cooling system (MWh).

According to [21], the average energy required for the operation of fans and pumps in the system is 225 W / kW of cooling capacity. In this case, the total power is 23,737.50 W (for 105.5 kW of cooling output). The primary energy consumption of a system that uses a compression refrigeration unit is calculated based on the compressor power and the annual operating time of the system. As the compressor power is 40 kW and the system operating time is 1500 h, it follows that E_{PEref} = 60MWh. On the other hand, the primary energy consumption of the solar cooling system depends on the operating time of all consumers, whose total power is 23.74 kW and at the annual level it is 35.61 MWh.

The relative primary energy savings according to [20] are:

$$\Delta E_{PErel} = \frac{\Delta E_{PE}}{E_{PEref}} = 0{,}4065 = 40.65\ \% \tag{29}$$

5.2. Comparison of systems from an economic point of view

The investment costs of the solar cooling system include all individual costs components, solar collectors, absorption cooling device, hot water tank. The investment costs of the compression refrigeration device are lower and for concrete system, according to [1] they amount to 30.900 EUR. Operating costs are mostly influenced by the cooling capacity and the type of solar collector. Small capacities cooling and monitoring of solar collectors significantly increase the price. Since the analyzed system belongs to a medium capacity plant, slightly higher total costs are to be expected. A comparison of the investment and operating costs for the reference compression cooling system and solar cooling is given in Tab. 5. In addition, these systems achieve the same cooling effect approx. 110 kW.

Table 5. Comparison of investment and operating costs for the analyzed cooling systems

	Compression cooling system (reference)	Solar cooling device with a flat plate collector	Solar cooling device with a vacuum tube collector
*investment costs (EUR)	30.900	139.316,33	174.832,83
**operational costs (EUR/year)	4.972,66	3.977,42	4.332,72

*investment costs refer to valid prices in 2019.

**annual operating costs are calculated according to [14]

To achieve the same cooling capacity, the use of solar systems will achieve certain energy savings. However, for smaller systems, this saving is not enough to compensate for the large investment costs. Only for higher electricity prices and larger capacity systems, the introduction of solar cooling would become justified from an economic point of view.

5.3. Comparison of systems from an environmental point of view

Comparison of systems from an environmental point of view

$$M_{CO_2} = CO_2 factor \cdot \Delta E_{PE} \cdot 0{,}278 \qquad (30)$$

where: M_{CO_2} - annual reduction of CO_2 (tons); $CO_2 factor$ - emission factor for electricity consumption (kgCO_2/kWh) and it is 0,63 [2], ΔE_{PE} - annual electricity savings.

This analysis depends only on the primary energy savings, and based on $\Delta E_{PE} = 24{,}39$ MWh it is obtained that the annual emission reduction is CO_2 4,27 tons.

6. Conclusion

The possibilities of applying solar cooling in B&H were analyzed and presented in the paper, with the aim of reducing primary energy consumption by using solar, instead of conventional compression systems. It has been observed that increasing the temperature of cooled and cooling water significantly increases the COP. Changing the mass flow of hot and cold water does not significantly affect the COP and heat transfer rate of the evaporator. The COP and heat transfer rate of the evaporator improve as the mass flow of cooling water increases. Also, the paper presents comparison of systems with two types of collectors. It has been shown that for a larger flat collector surface it is possible to achieve the same set parameters of hot water, with lower investment costs. The results of this research have shown that solar cooling systems have a number of advantages over compression cooling systems, but their biggest disadvantage is the high initial investment. By replacing compression with solar cooling systems, great savings of primary energy are achieved, peak load is reduced, which could be used in the future to form electricity prices, as well as a significant reduction in CO_2 emissions. On the other hand, when you keep in mind that solar systems do not use CFC compounds, a compound that destroys the ozone layer and is as harmful as CO_2 as much as 12 thousand times, from an environmental point of view, solar single-stage absorption systems provide a good opportunity to reduce electricity consumption.

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Paper submitted: May 17, 2023

Paper revised: January 30, 2024

Paper accepted: May 27, 2024