EVALUATION OF SOLAR SORPTION REFRIGERATION SYSTEM PERFORMANCE IN LATVIA

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Abstract. There are many well known researches and commercial products for solar air-conditioning (SAC) units of small and medium power (up to 100 kW). Solar energy application for refrigeration by using of the sorption cycle for Latvian climate conditions was considered in the presented paper. Advantages of lithium-bromide and ammonia-water fluids were compared taking in account the difference between real and ideal performance of the sorption systems. Since heat rejection in thermally driven chillers is bigger than in conventional vapor compression refrigeration, the climate conditions have big influence on the total system performance. As the solar radiation level in Latvia is less than 400 $W \cdot m^{-2}$ dual sources of driving heat (biofuel combustion and solar energy) for the sorption cycle were evaluated. Computer modeling of sorption chillers results in development of a new temperature threshold criterion for efficient use of solar cooling systems at different energy costs and various temperature levels.

Keywords: solar energy, air-conditioning, refrigeration, absorption, adsorption.

Introduction

Solar energy from the sun heat collector can provide the thermal driving power for different sorption refrigeration cycles.

In case of absorption process refrigerant vapor after evaporator (cooling duty) becomes incorporated in liquid mixture forming strong solution in the absorber (exothermic process) and in the desorber refrigerant steam left solution by endothermic reaction is supported by external heat from the solar collector. In the aqua ammonia (NH_3/H_2O) refrigeration cycle water performs as an absorbent and ammonia is used as the refrigerant agent. This mixture can provide refrigeration to negative temperature up to -60 ºC. The second common pair is water-lithium bromide solution($H_2O/LiBr$), where water servesas the refrigerant agent and strong mixture ($H_2O/LiBr$) performs as absorber.

The sorption cyclescan be realized also by the adsorption process, when gas (the refrigerant) interacts with a solid body (the adsorbent) by physical or chemical forces. The mostly used pairs are represented by silica gel-waterandzeolite-water.

The absorption and adsorption systems are mainly used for close cooling cycles. Desiccant cooling, which consists of combination of dehumidification and evaporation processes, is used in case of the open sorption cycle. In this case, fresh and warm air is directly involved in the process. There are solid and liquid desiccations used in cooling.

Market research shows that the absorption cycle takes about 71 % share of all solar sorption chillers market, the desiccant cooling share is 16 % and the rest13 % belong to the adsorption system.

For efficiency indicators SF (solar fraction), *COP* (coefficient of performance) and seasonal S*COP* or solar thermal COP_{th} represented by the ratio of the cooling energy and heat from the solar collector during the cooling season are used. Electrical energy consumption described by COP_{el}representsthe ratio of cooling energy per unit of the consumed electrical energy (for pump and fan operation). For refrigeration cycle evaluation *EER* (energy efficiencyratio)equal to the cooling energy divided by the consumed electrical (or thermal) energyis used. Similarly seasonal *EER* or S*EER* are used.

Air conditioners with vapor compression are driven by electrical power operation with S*EER* up to 9 and 9.5 will challenge SAC economical feedback.

According to the "Solar Air Conditioningin Europe" Evaluation report [13]in 2003 average thermal *COP* for absorption unit is around 0.5 up 0.7 and even 1.3 for the double cycle, but for the adsorption system *COP* is smaller around 0.6. However, the advantage of the adsorption system is lower temperature +55 to +85 ºC for driving heat than in case of absorption operation. COP_{el} or consumption of electrical energy for thermal driven sorption cycle reach 5 to 6, which is crucial for the systemoverall efficiency. Depending on the requested cooling temperature level and available solar radiation intensity various solar thermal collector designs can be applied. The flat plate collector can

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efficiently generate heat with temperature up to 100 ºC and provide water cooling up to +5 ºC to +10 ºC. The evacuated collector can heat up to 120 ºC and more, what resulted in cooling down to 0 ºC. The parabolic or linear Fresnel collector application will increase the heat level up to 200 ºC and higher, therefore refrigeration to -15 ºC to -20 ºC becomes possible.

The main components ofSAC including solar collectors, absorption and adsorption units technically mature are available in the market with wide range of solar power and cooling parameters. The energy efficiency becomes the key issue for solar sorption system implementation.

In this article we will use COP_{th} (1) for thermal efficiency and *EER* (2) for electrical efficiency of SAC.

Solar radiation in higher latitudes provides less energy than for southern countries. The first adsorption cooling system driven by solar energy were developed and tested during 2013 in Riga (Latvia) [1; 2] with decimal latitude of 56.9^0. Experimental results presented by P.Shipkovs et al. give that for 100 % input energy (consisting of 92 % solar energyand 8 % of electrical energy) output of the cooling energy gain is 38 %. This performance with COP_{th} = 0.41 and *EER* = 4.75 proves feasibility of solar cooling in low solar intensity conditions.

$$COP_{th} = \frac{Q_{ac}}{Q_{sol}}, \tag{1}$$

where Q_{ac} – absorption chiller cooling power (evaporation heat of item 4 in Fig. 2), W;
Q_{sol} – power from solar collector (7 in Fig. 2), W.

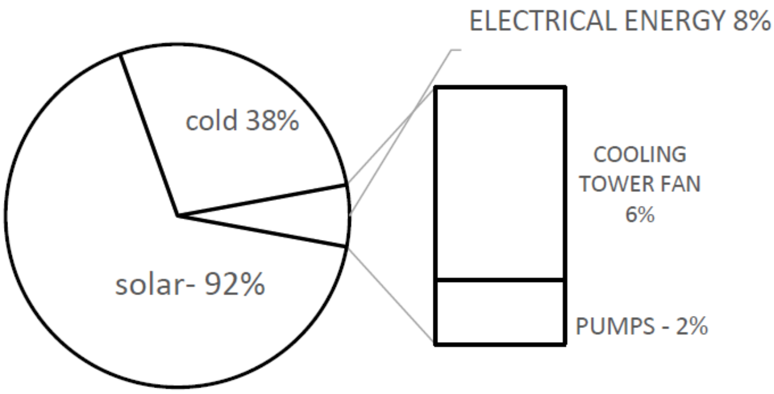

Fig. 1. **Energy balance for SAC adsorption solar cooling system according to [2]**

Fig. 1 shows that the heat rejection system (dry or wet cooling tower) has to remove the total amount of heatof 138 % from the absober and condenser (cooling duty plus solar energy) together with the heat from kinetical energy of flow (or electrical energy consumed by the pump and fun)converted in heat. It is important that the main user of electrical energy is the fan of the cooling tower – around 80 % of total electrical consumption, the rest 20 % are consumed by circulation pumps.

$$EER = \frac{Q_{ac}}{W_P + W_F}, \tag{2}$$

where W_P – electrical power of circulation pumps, W;
W_F – electrical power of air fan of re-cooling tower, W.

The SAC experimental study with location in Tampere, north latitude 61.29º in Finland is reported in [3]. The absorption chiller was employed with rated capacity of 12 kW and rated *COP* 0.825. A lot of efforts were devoted to various configurations of the system elements and storage tank sizing etc. In addition to the solar collector another thermal energy source – district heating network was involved. The electrical seasonal efficiency *EER* was 11 and solar thermal COP_{th} was measured between 0.75 and 0.85. Direct connection between the solar collectors and the chiller performs better than connection through the hot storage tank (in series).

Careful selection of the system configuration and proper component sizing can provide competitive performance withthe electricly driven vapor compression chiller even for countries with smaller solar irradiation. On the other hand, improvement of solar cooling performance by adding heat from CHP (combined heat and power plant) will create the synergyeffect for versatilerenewable energy sources. Last, but not the least, the cooling tower operation inrelatively cold climate will be more efficient than in southern areas. This fact will balance the lack of solar radiation intensity and enhance emerge of SAC in high latitude areas.

Materials and methods

The sorption SAC system contains a number of main components – solar collector, cold and hot tanks, absorber, circulation pumps and cooling tower with fan. This paper presents modeling of component alignment with the aim to increase *EER* for cooling duty. The performance of the absorption chiller governed by a set of the main parameters consists of T_s the solar collectoroutput temperature in case of direct connection equal to the desorber temperature and T_c the outlet temperature of the cooling tower or inlet temperature of refrigerant condenser. The target temperature for the cooling process is represented by T_{AC}.

The mathematical model represented in [11] for H_20/LiBr operation predicts $COP_{th} = 0.77$ for $T_s = 90$ ºC, $T_c = 40$ ºC and $T_{AC} = 7$ ºC, this set of conditions repeats the data of the experimental measurements in Finland [3] with very similar thermal *COP* considered above. However, this model does not cover electrical energy consumption expressed by *ESF* or *EER*.

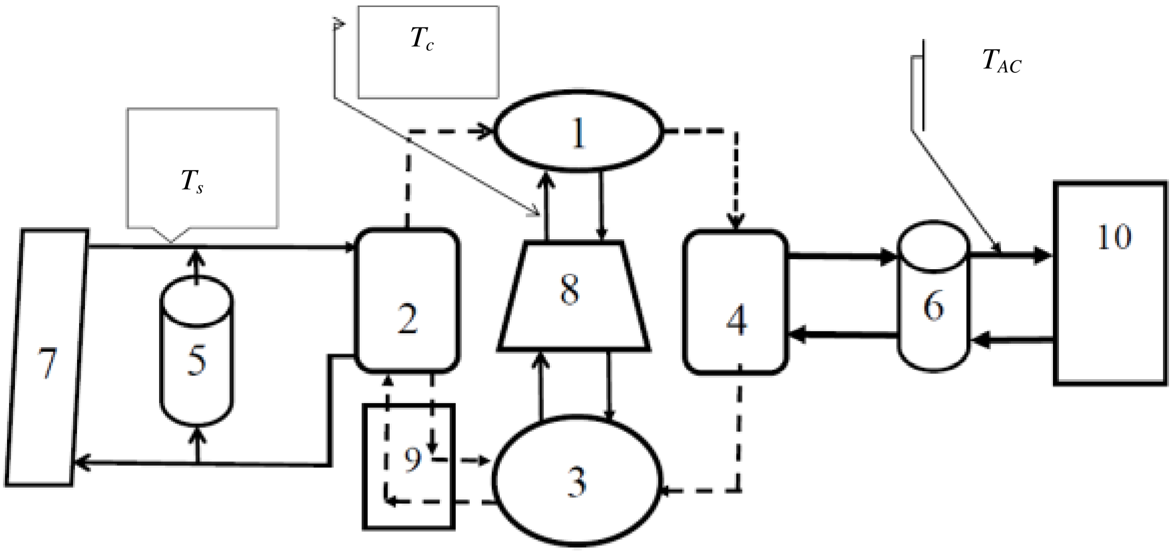

Fig. 2. **Main parts of solar absorption cooling system (pumps, fan and valves not shown):** 1 – absorber condenser; 2 – desorber (refrigerant steam generator); 3 – absorber; 4 – refigerant evaporator; 5 – hot water tank; 6 – cold water storage; 7 – solar collector; 8 – cooling tower (wet or dry); 9 – heatexchanger between strong and weak solution in absorber; 10 – cooling duty; T_{AC} – temperature of direct flow to cooling duty

With purpose to take in consideration COP_{th} and *EER* together we suggest to start from the description of the re-cooling tower, which had to reject heat from the chiller according to the following expression

$$Q_{CT} = Q_{AC} + Q_{SOL} + W_P + W_F, \tag{3}$$

where Q_{CT} – rejected heat power from re-cooling tower, W.

Accordingly, the re-cooling tower of dry type heat balance will look as

$$G_{air}\,\Delta TC_a = Q_{CT} = Q_{AC} + Q_{SOL} + W_P + W_F, \tag{4}$$

where G_{air} – air mass flow in cooling tower, $kg \cdot s^{-1}$;
C_a – air heat capacity, $kJ \cdot (kgK)^{-1}$;
ΔT – air temperature increase in cooling tower, ºC.

The air fan of the re-cooling tower electrical power is equal to

$$W_F = G_{air} \Delta H \frac{g}{\eta}, \tag{5}$$

where W_f – air fan electrical power, W;
ΔH – re-cooling tower air flow resistance, m;
η – fan efficiency(unit less);
g – gravity acceleration, $m \cdot s^{-2}$.

As far as the air flow in the re-cooling tower will be in turbulent mode, the following equation for the tower resistance should be applied

$$\Delta H = K G_{air}^{\ 2}, \tag{6}$$

where K – proportional coefficient, calculated from the re-cooling tower data sheet, $m \cdot s^2 \cdot kg^{-2}$.

Then fan power will be calculated as

$$W_F = G_{air}^3 K g \eta^{-1} = Kg \frac{\left(Q_{AC} + Q_{SOL} + W_P + W_F\right)^3}{\Delta T^3 Ca^3}, \tag{7}$$

The cooling power of the chiller (absorption equipment output) can be given by COP_{th}

$$Q_{AC} = COP_{th} \left(Q_{SOL} + W_P + W_F\right), \tag{8}$$

The air fan power consumption can be represented as

$$W_F = \frac{Kg Q_{AC}^3}{\Delta T^3 Ca^3} \left(1 + \frac{1}{COP_{th}}\right), \tag{9}$$

Because of electrical power consumption mainly consists of the fan power, it is possible to find the relationship between EER and COP_{th} by the following relationship

$$EER = \frac{\Delta T^3 Ca^3}{Q_{AC}^2 \frac{Kg}{\eta} \left(1 + \frac{1}{COP_{th}}\right)}, \tag{10}$$

This equation (10) gives the relation between the main operational parameters of SAC. However, the value of cooling duty Q_{ac} and COP_{th} depends on the main temperatures of the process (solar collectors and cooling tower). For further consideration, we will assume T_{ac} equal to 10 ºC, what is valid for SAC performance.

Thanks to the results presented in [3], we can create empirical equations for the absorption chiller power Q_{ac} in kW and thermal performance COP_{th}

$$Q_{AC} = \frac{T_s - 2.15 T_c + 1.9}{0.0125 T_c + 0.79}, \tag{11}$$

where Ts – temperature of flow from solar collector, ºC;
Tc – temperature of re-cooling liquid flow from cooling tower, ºC.

Equation (11) is valid, if T_{AC} temperature is fixed at the inlet 10 ºC and outlet temperature 15 ºC for cooling duty. The solar collector outlet temperature in the range from 60 ºC to 90 ºC and the cooling flow temperature from the tower 8 (Fig. 2) difference is equal to 8 ºC and the outlet flow from the tower is in the range 27 ºC to 37 ºC.

It is possible to describe the coefficient of thermal performance COP_{th} by using empirical data [3]

$$COP_{th} = 1.045 - 0.0067\ T_c \ (12)$$

This equation is valid for the range of COP_{th} from 0.8 to 0.87 for 12 kW rated power absorber [3] for the power range from 9 kW to 12 kW. It is obvious that COP_{th} increases when T_s is growing. Therefore, it is important to select when possible direct connection of the solar collector to the absorption chiller desorber (steam generator). However, high temperature of the collector T_{mean} will decrease the output of the solar collector, accordingly (13).

The solar collector energy output per unit of area Q_{sol}in W·m^{-2}can be described by equation

$$Q_{sol}= A_0 I\eta_{col} - A_1(T_{mean}-T_{amb}) - A_2(T_{mean}-T_{amb})^2, \quad (13)$$

where A_0, A_1 and $A2$ – collector constants from data sheet [11];
I – solar radiation in plane of collector array, W·m^{-2};
η_{col} – solar collector efficiency (unitless);
T_{mean} –mean temperature of flow in solar collector, ºC;
T_{amb} – ambient air temperature near collector array, ºC.

It is known that the average solar radiation intensity I in Latvia by taking into account that the solar radiation time is 1800 h per year and the solar collector energy gain is 700 kW·h·m^{-2} per year. Thus, the average value for Latvia I = 388 W·m^{-2}.

If we substitute the empirical expression for COP_{th} (12)and absorption chiller power (11) in equation for EER (10), it becomes possible to find the numerical relation between EER and the solar collector temperature and brine after the re-cooling tower. The influence of ambient temperature will be represented by empirical equation for the solar collector (13) and the cooling tower (6) operation.

Results and discussion

The suggested descriptions offered in this article (11) and (12) are valid for limited interval of operation. Outside the nominal range (70-100 % of rated value) COP_{th} will sharply drop. To eliminate this problem it is necessary to avoid usage of oversize power output for the absorber regarding the cooling duty for refrigeration. Balancing the absorption chiller output and the cooling power consumption is possible by using the cold water storage 6 in Fig.2. The calculation results according to equations (10-12) and the experimental data [3] are presented in Fig. 3.

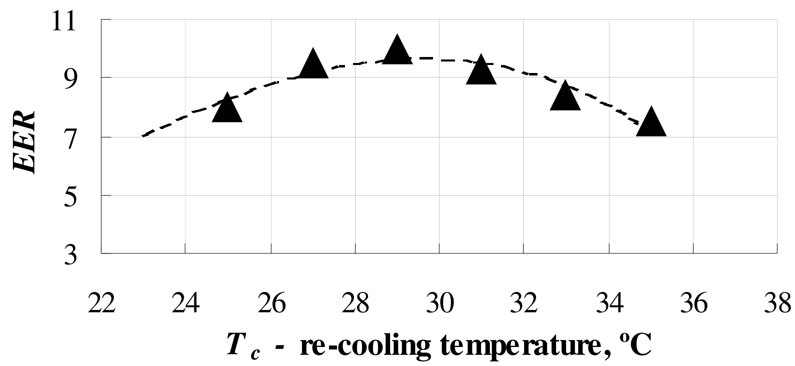

Fig. 3. **Temperature Tc threshold criterion for efficient performance of SAC**

Re-cooling temperature T_c had the threshold value to ensure the EER target value around 10 to keep the solar sorption chiller competitive with vapor compression refrigeration. The threshold value depends on the combination of the operational parameters – cooling, ambient and solar collector temperatures, absorption chiller power, solar radiation and empirical constants for the solar collector and re-cooling tower. The trend line represented in Fig.3 confirms that expenses to get low T_c closer to ambient temperature will increase the fan electrical consumption in the re-cooling tower. Therefore, EER will drop. When T_c is growing up by slowing down the fan operation, COP_{th} will decrease and it enlarges the demand of heat rejection. This again results in the fan duty increase. Implementation of the threshold criterion for T_c will open a way to find the parameter selection to ensure the best possible value of EER – efficiency of use of electrical energy for cooling by the absorption chiller.

The numerical solution of equations(10-12) with purpose to reach the highest possible EER should be based on the main equipment performance description – that will provide background for component proper selection and recommendation for energy efficient operation modes of SAC systems.

Conclusions

1. Experimental measurements presented in a number of articles prove that the absorption and adsorption chillers for SAC can be competitive even for high latitude countries with low level of solar radiation intensity like Latvia and Finland.
2. There is large selection of highly mature components for SAC implementation (solar collector, absorber and absorber units, cooling towers etc.). In this situation, the component proper sizing and operational parameter alignment become crucial to provide the best possible electrical efficiency *EER* for SAC.
3. Lots of articles are devoted to simulation of the absorber thermal operation. However, it is not possible to overcome the technology limitation for COP_{th} based on LiBr crystallization at high temperatures. Therefore, in most of the cases the re-cooling temperature T_c in the condenser and absorber becomes the threshold value, which depends on the cooling tower performance and ambient temperature.
4. Sorption chillers need to reject more heat per unit of produced cold than vapor compression units. Therefore, the influence of the cooling tower on the electrical efficiency of sorption SACis very high.

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