Absorption and adsorption chillers applied to air conditioning systems

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Abstract   This work presents an application possibility of sorption refrigerators driven by low temperature fluid for air conditioning of buildings. Thermodynamic models were formulated and absorption LiBr-water chiller with 10 kW cooling power as well as adsorption chiller with silica gel bed were investigated. Both of them are using water for desorption process with temperature $T_{des} = 80\,^{\circ}$C. Coefficient of performance (COP) for both cooling cycles was analyzed in the same conditions of the driving heat source, cooling water $T_c = 25\,^{\circ}$C and temperature in evaporator $T_{evap} = 5\,^{\circ}$C. In this study, the computer software EES was used to investigate the performance of absorption heat pump system and its behaviour in configuration with geothermal heat source.

Keywords: Chiller; Conditioning systems; Heat exchangers

Nomenclature

A	–	area, m^2
a	–	concentration, water uptake, kg/kg
COP	–	coefficient of performance
f	–	solution circulation ratio
m	–	mass, kg
$\dot{m}$	–	mass flow, kg/s
h	–	enthalpy, kJ/(kgK)
K, n	–	coefficients in Freundlich equation

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P	–	pressure, Pa
R	–	radius, m
q	–	vapour quality
Q	–	energy, kJ
$\dot{Q}$	–	heat flux, kW
SHX	–	solution heat exchanger
T	–	temperature, °C
ΔT_m	–	logarithmic mean temperature difference, °C
T_{pinch}	–	smallest temperature difference, K
U	–	overall heat transfer coefficient, W/m^{2}K
$\dot{V}$	–	volumetric flow rate, m^3/h,
W	–	power, kW
x	–	mass fraction, %

Greek symbols

ρ	–	density, kg/m^3
ν	–	specific volume, m^3/kg
δ	–	fin thicknes

Subscripts

c	–	chilled water
a, ads	–	adsorber
d, des	–	desorber
$e, evap$	–	evaporator
$c, cond$	–	condenser
eq	–	equilibrium state
s	–	saturation
w	–	water
a	–	adsorbent
shx	–	solution heat exchanger
ghx	–	geothermal heat exchanger
net	–	district heating network
$user$	–	district heating users
p	–	pump
sp	–	solution pump

1   Introduction

Current development of technology is strongly connected with rational management of natural resources and environment protection. The goal of the Kyoto Protocol ratification is to enhance renewable energy utilization, with the main aim to reduce the products of coal combustion which are the main pollutants for air, water and soil. Furthermore, the current integration process with the European Union will require the adjustment of the power engineering sector of Polish industry.

Growing requirements related to thermal comfort are observed. Sunlight heat gains, lighting, computers, and other electrical appliances raise the indoor temperature of buildings, causing necessity of air conditioning installation. In that way the parameters inside of objects are kept on the constant required level. All buildings equipped in air conditioning and cooling systems have high consumption of electrical energy. In rich countries approximately 5–10% of electricity production covers the demand for air conditioning [10]. It is important to consider that bigger electrical energy market will cause higher CO_2 emission. There is an increased interest in the development and use of sorption chillers (absorption and adsorption) due to their various economic and impressive environmental benefits, enabling low temperature energy to be used directly for driving of a cycle, avoiding energy losses for heat energy conversion into electrical energy.

Processes occurring inside of adsorber and desorber take over the function of classical refrigerating compressor contributing to lower electrical energy consumption. Waste heat, solar or geothermal energy driven absorption (liquid-vapour) and adsorption (solid–vapour) systems have the advantage of being environmentally benign: both factors ODP (Ozone Depletion Potential) and GWP (Global Warming Potential) are zero, and that's why are an alternative to vapour compression chiller even using fluorohydrocarbon (HFC) or natural fluids [4] as working fluids.

Sorption cooling systems are characterized by easy regulation system depending on the cooling power demand, by partial capacity the heat consumption is proportional keeing the working parameters on the required level. Advantages of sorption cycles are simple construction, reliability, long life time, low operating and conservation costs, low-noise, vibration-free and finally less replacement parts. Sorption chillers are used in air conditioning systems, if the aim is to obtain chilled water (+2 °C to 10 °C) or in industrial refrigeration (-20 °C to 0 °C), if lower temperatures are needed.

Adsorption chillers supplied by hot water from sun collectors were investigated. In publication [8] authors presented basics of operation, advanced scientific and experimental works describing adsorption cooling from sun energy for air conditioning and refrigeration purposes. They proved that the proper selection of adsorbent-adsorbat are of the fundamental importance for effective operation of such system. Solar energy, waste heat from technological processes and other heat sources in low temperature range 50–100 °C [7] are valuable source of energy for driving adsorption cooling units.

2   Overview of sorption cooling technologies

There is an increased interest in the development and use of adsorption chillers due to their various economic and impressive environmental benefits, enabling solar energy or waste heat to be used for applications such as district networks and cogeneration plants. Compared to adsorption systems that require heat sources with temperatures above 100 °C (zeolit-water systems, activated carbon-methanol systems), a silica gel/water adsorption refrigerator uses waste heat with temperatures below 100 °C. This creates new possibilities for utilization of low-temperature energy.

Desiccant cooling systems combine sorptive dehumidification, heat recovery, evaporation, and heating to create a cooling process. Desiccant and evaporative cooling (DEC) devices are especially used in air conditioning systems; heat energy can be used for the required regeneration of the sorbents in the dehumidifier.

Table 1 introduces current development of cooling technologies.

Table 1. Known various methods of cooling load production [2].

	Compression chiller	Absorptive chiller	Adsorptive chiller	DEC
Physical Cooling Effect	Vaporise of refrigerant			Evaporation of refrigerant
Type of compression	Mechanical compression	Thermal, absorption loop	Thermal, adsorption of water steam	Sorptive drying
Power source	Electrical energy	Heat energy 85–180 °C	Heat energy 55–95 °C	Heat energy 50–100 °C
Refrigerant agent	Chlorinated CHC or chlor free hydrocarbons	Water with LiBr or NH$_3$ as absorption agent	Water with solid as adsorption agent (SILICA-Gel)	Water
Coefficient of Performance	1.3–1.65	0.6–1.0	0.4–0.6	0.3

In Fig.1 coefficient of performance (COP = chiller load/heat input) variability with driving source temperature was introduced. Curves describing the changes of COP for absorption and adsorption cooling devices, have different courses. Absorption cycles may reach relatively high performance depending on driven temperature but with strong limitation to the lowest possible driven temperature of the heat source. Temperatures below 80 °C cause rapid decrease of efficiency (Fig. 1).

Adsorption cycles using silica gel-water, zeolite-water active carbon-ammonia and active carbon-methanol as the adsorbent–refrigerant couples are not so temperature sensitive, the main advantage over other systems in their ability to be driven by heat of relatively low temperatures. This enables highly desirable waste heat recovery.
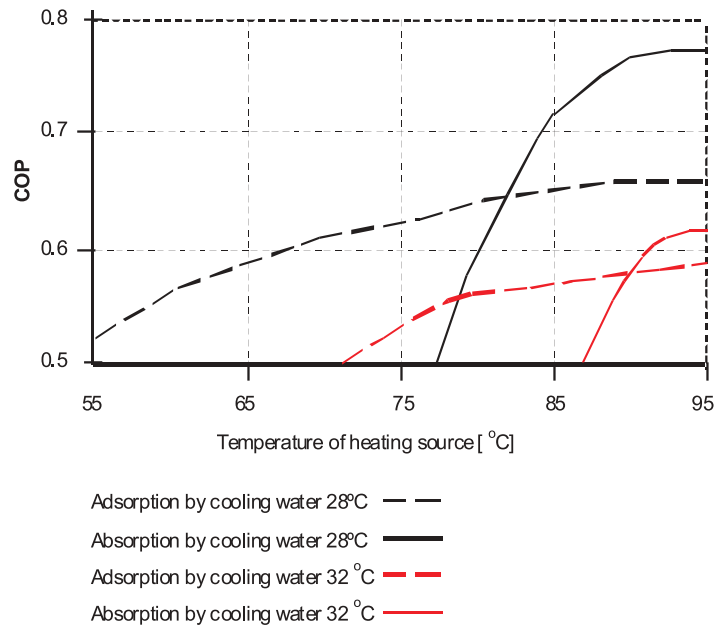

Figure 1. COP for absorption and adsorption 1 stage refrigerators [5].

3 Sorptive cooling models

3.1 Absorption chiller

The absorption cooling cycle consists of coupled mass and heat transfer balances in the evaporator, absorber, desorber and condenser. A single stage LiBr-H$_2$O absorption system is presented, and all calculations done in reference to the numbering principle illustrated in Fig. 2. The following basic assumptions were made to simplify the modelling of the system:

- heat losses and heat gains between the system and its environment are neglected,

- the steady state of the refrigerant is pure water,

- there are no friction or pressure losses in pipes and components,

- pump is isentropic,

- throttling processes in valves are isenthalpic,

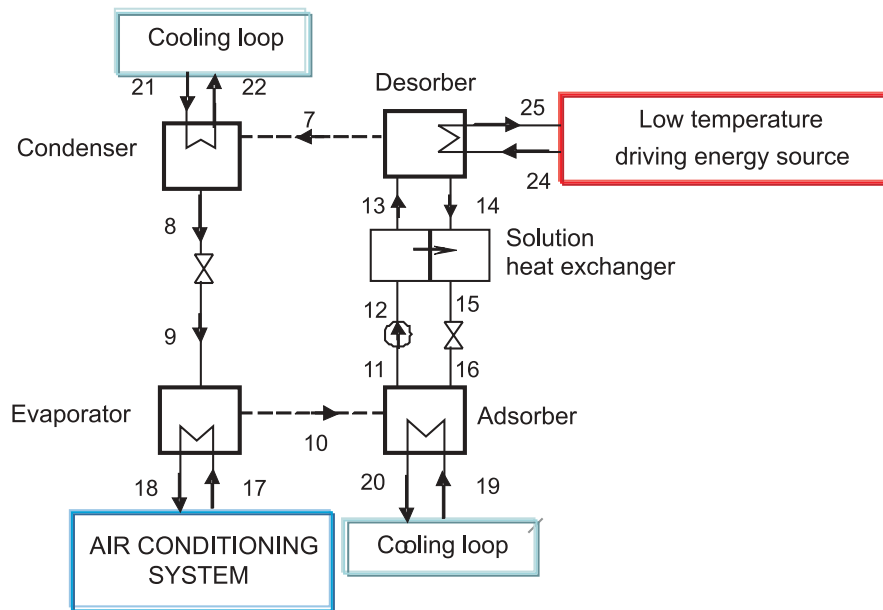- heat source supplies pressurized hot water to desorber.

Figure 2. Circuit diagram with LiBr-water absorption cooling cycle.

In this study, the computer software EES [3] was used to investigate the performance of the absorption cooling system and its behaviour in configuration with a low temperature heat source. Each component of the system has been treated as an independent module with a certain number of input values, required mass and energy balance equations, and involved relationships between parameters with the aim to calculate the output variables.

It is necessary to properly understand the thermodynamic state of each point on the diagram. The external energy transfers to the absorption system were showed in the Fig. 2. Main processes and their assumptions for cooling circuit are summarised in Tab. 2.

Table 2. Description of main thermodynamic state points.

Point	State	Details and assumptions
7	Superheated vapour	Water vapour leaving desorber has zero salt content (pure water)
8	Saturated liquid water	Vapour quality set to 0 after condensation
9	Vapour-liquid refrigerant state	Vapour flashes and liquid passes through expansion valve
10	Saturated vapour	Vapour quality set to 1 after evaporation
11	Saturated liquid solution	Weak solution of water and LiBr
12	Subcooled liquid solution	Calculations based on isenthalpic pump model
13	Subcooled liquid solution	Calculations based on heat exchanger model
14	Saturated liquid solution	Rich solution of water and LiBr
15	Subcooled liquid solution	Calculations based on heat exchanger model
16	Vapour-liquid solution state	Adiabatic expansion in the valve

To perform equipment sizing and performance evaluation of a single stage LiBr water absorption chiller, after taking into account the basic assumptions, energy balance for each component must be considered. For calculations in this work mass flow $\dot{m}$ and respectively energy flux $\dot{Q}$ were used.

The energy balance on evaporator can be written as:

$$Q_e = m_{10} \cdot h_{10} - m_9 \cdot h_9, \quad \text{where} \quad m_{10} = m_9 . \tag{1}$$

The value $\dot{Q}_e$ is the cooling load and chilled water will be delivered to the users by external loop with mass flow $\dot{m}_{17}$.

Since the value of m_{10} is known, we can calculate m_{11} from the mass balance in absorber and the assumption that:

$$m_{11} \cdot x_{11} = m_{16} \cdot x_{16} , \tag{2}$$

$$m_{11} = m_{10} + m_{16} . \tag{3}$$

During the calculations, is possible to set the lithium bromide mass fraction in the solution x_{11}, x_{16} as the input value, or calculate it from the known relationships. Point 15, the outlet from the solution heat exchanger is the closest approach to crystallisation line. If the range of mass fraction LiBr changes from 50% at absorber outlet to the solution heat exchanger (point 11) to 62% at the inlet to absorber (Point 1 and 16) so the calculated operating conditions avoid crystallisation danger.

An energy balance on the absorber:

$$Q_a = m_{10} \cdot h_{10} + m_{16} \cdot h_{16} - m_{11} \cdot h_{11} . \tag{4}$$

The minimum work input to the solution pump can be obtained from equation:

$$W = \dot{m}_{12} \cdot v_{12}(p_{12} - p_{11}) . \tag{5}$$

An energy balance for solution heat exchanger:

$$Q_{shx\_cold} = m_{14} \cdot h_{14} - m_{15} \cdot h_{15} \tag{6a}$$

$$Q_{shx\_hot} = m_{13} \cdot h_{13} - m_{12} \cdot h_{12} \tag{6b}$$

It is known that heat streams on both sides are equal, if the adiabatic shell will be assumed.

$$\dot{Q}_{shx\_cold} = \dot{Q}_{shx\_hot} . \tag{7}$$

These parameter influences COP of the system in large scale. The higher the effectiveness the COP increases. The absence of solution heat exchanger in the cycle is significant, and causes higher COP, because the return solution heat from desorber can be recovered to heat up the LiBr-water mixture before entering the generator. Generator energy balance would be:

$$Q_d = m_7 \cdot h_7 + m_{14} \cdot h_{14} - m_{13} \cdot h_{13} . \tag{8}$$

Assumption about purity of water vapour leaving desorber can be done, and following equation is the result:

$$m_{13} \cdot x_{13} = m_{14} \cdot x_{14} . \tag{9}$$

Overall mass balance on the desorber is expressed as:

$$m_{13} = m_{14} + m_7 . \tag{10}$$

Finally, the condenser heat can be determined from an energy balance, which gives:

$$Q_c = m_7 \cdot h_7 - m_8 \cdot h_8, \text{ where } m_7 = m_8 . \tag{11}$$

Heat transfer models of each heat exchanger were added to the system to show better understanding of phenomena taking place there and bring closer to realistic conditions. All inputs and outputs values from heat exchanger

model can be followed, and in the future better consideration of different heat exchanger models could be carried out.

In this paper the UA type heat exchanger model was applied to specify the size and performance of heat exchanger as the function of UA value and logarithmic mean temperature difference defined as.

$$\Delta T_{lm} = \frac{(T_{h,1} - T_{c,1}) - (T_{h,2} - T_{c,2})}{\ln \frac{(T_{h,1} - T_{c,1})}{(T_{h,2} - T_{c,2})}} \ , \tag{12}$$

where h and c refer to the hot and cold sides of hat exchanger. Points 1 and 2 denote the side of heat exchanger. ΔT_{lm} describes the potential of heat transfer between hot and cold side. The product of overall heat transfer coefficient U and the heat exchanger area A is a convenient way to specify the size of a heat exchanger in a single parameter (i.e. the UA value). This parameter is very useful for calculations and can be found in literature [6].

The amount of the exchanged heat can be than calculated from the formula:

$$\dot{Q} = UA \cdot \Delta T_{lm} \ . \tag{13}$$

The main parameter to estimate the efficiency of operating absorption system is the coefficient of performance, COP. As it was mentioned COP is defined as the ratio of useful energy output (cooling capacity obtained at evaporator) to the primary energy input (heat input for the generator plus work input for the pump. The work input for the pump is negligible relative to the heat input at the generator, therefore, the pump work is often neglected for the purposes of analysis.

Finally the coefficient of performance for cooling cycle is defined as:

$$\mathrm{COP} = \frac{Q_{evap}}{Q_{des}} = \frac{m_{17}(h_{17} - h_{18})}{m_{24}(h_{24} - h_{25})} \ . \tag{14}$$

External water loop at the evaporator side delivers chilled water to the consumers.

According to the numbers in Fig. 2, results of calculation were presented in Tab. 3. In each state point enthalpy, mass flow, pressure, temperature, LiBr concentration and quality of water after throttling, were investigated. Assumed operation conditions are: evaporator, absorber pressure $P10 = 0.7$ kPa; generator, condenser pressure $P14 = 7.347$ kPa; solution heat exchanger exit temperature $T13 = 55$ °C; Generator exit LiBr concentration $x14 = 60\%$.

Table 3. Operating parameters of 10 kW LiBr- water absorption chiller.

Point	h	m	P	T	x	q
	[kJ/kg]	[kg/s]	[kPa]	[°C]		
7	2657	0.004279	7.347	78		
8	167.1	0.004279	7.347	39.91		
9	167.1	0.004279	0.7	1.881		0.06379
10	2504	0.004279	0.7	1.881		
11	73.35	0.05134	0.7	30.18	55	
12	73.35	0.05134	7.347	30.18	55	
13	124.7	0.05134	7.347	55	55	
14	200.4	0.04706	7.347	78	60	
15	144.5	0.04706	7.347	54.91	60	
16	144.5	0.04706	0.7	39.74	60	
17	41.99	0.5	6	10		
18	21.99	0.5	6	5.23		
19	104.8	0.2739	60	25		
20	155	0.2739	60	37		
21	104.8	0.2653	6	25		
22	144.9	0.2653	6	34.6		
24	356.4	1	700	80		
25	342	1	700	76.5		

3.2   Adsorption chiller

Solid adsorbents, such as silica gel, zeolites, activated carbon in combination with suitable adsorbats (refrigerants) are used in refrigeration systems to produce chilled water or manufacture ice [1]. Practically solid adsorption refrigeration cycles may be classified according to the used adsorbent/refrigerant combination. It could be seen in Fig. 3 that adsorption cooling cycle contains analogous components (evaporator, condenser) to the absorption one. The difference refers to the adsorber/desorber chamber and its construction. Due to small heat transfer coefficient in adsorbent bed, special design must be applied to ensure proper heat exchange between solid adsorbent and heat transfer medium (cooling water for adsorption or hot water for desorption process).

In an adsorption cooling system we can separate two basic phases. The first phase when after heat removal from chilled water the refrigerant vapour

leaves evaporator. Then it enters the adsorber where the refrigerant vapour is adsorbed in the adsorbent bed. This process requires intensive cooling, because adsorption is an exothermic phenomenon. The second phase is desorption process as the result of heating up the adsorbent bed to release adsorbat (refrigerant) from it. Vapour moves to condenser and after throttling liquid refrigerant flows to evaporator. To ensure continuous operation of adsorption chiller two beds are applied, in which adsorption and desorption processes occur in the same phase as than the switching of the chambers.
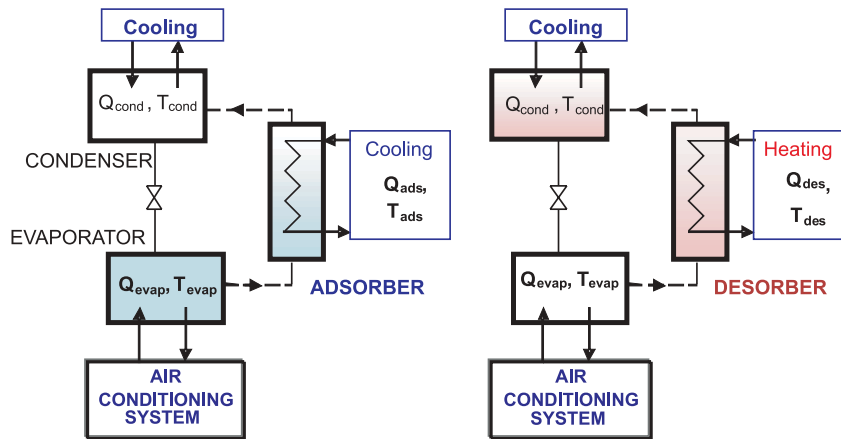
Figure 3. Two phases – adsorption and desorption of adsorption cycle.

A novel type of two-dimensional adsorber-desorber model inside of adsorption cooling system was formulated. An advanced type of silica gel-water adsorption chiller supplied from low temperature heat source produces a cooling load for air conditioning system. To increase the heat transfer and in this way also the COP a finned surface was implemented inside the adsorber.

A two dimensional adsorber- desorber model was developed in cylindrical coordinates, as the component of adsorption chiller driven from low temperature heat source. Following assumptions were set:

- heat and mass transfer are time dependent,

- adsorption equlibrium is described with the following form of equation $a = f(p, T)$,

- accumulation of heat and mass in a gas phase are neglected because of low gas density,

- no pressure drop in adsorbent bed,

- no heat losses to environment,

- adsorbent properties and structure are uniform in the whole volume,

- refrigerant pressure during adsorption and desorption phases is constant in whole volume of an adsorbent,

- adsorber construction is symetrical in x dimension,

- convection heat transfer is considered between fluid and metal wall as well as between wall and adsorbent bed,

- conduction was assumed in x (axial) and r (radial) direction,

- properties of heat transfer fluid, metall wall and adsorbent bed are independent of temperature.

In developed adsorber model with finned surface, both temperature and concentration are considered in two geometrical dimensions (axial and radial) and are functions of time. Adsorber scheme with finned pipe was presented in Fig. 4. Physical dimensions of the model were placed in Tab. 4.
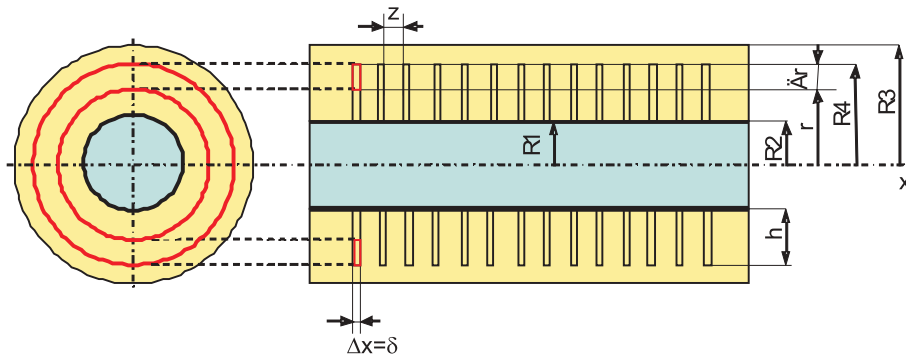

Figure 4. Adsorber with fins.

All compounds were divided into controll volumes (Fig. 5) having the shape of a ring. Each cell has cylindrical dimensions Δx and Δr. Fluid, metal wall and adsorbent bed were considered in calculations as composed of many layers in radial and axial dimensions.

Table 4. Dimensions of an adsorber and fins.

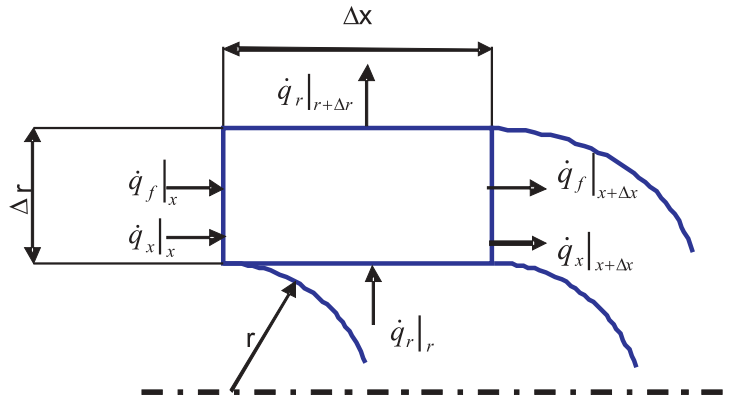| Radius and lenght [m]	Fin [m]
R1 = 0.005	
R2 = 0.006	$h = 0.012$
R3 = 0.026	$\delta = 0.003$
R4 = 0.018	$z = 0.025$
L = 0.460	

Figure 5. Control volume.

General energy balance for each type of cell in an adsorber was written. For fluid energy balance has a following form:

$$\frac{\partial T_f}{\partial t} = \frac{\lambda_f}{c_{pf}\rho_f}\frac{\partial^2 T_f}{\partial x^2} + \frac{\lambda_f}{c_{pf}\rho_f}\left(\frac{1}{r}\frac{\partial T_f}{\partial r} + \frac{\partial^2 T_f}{\partial r^2}\right) - v_f \cdot \frac{\partial T_f}{\partial x} \ . \qquad (15)$$

Energy balance for metal wall:

$$\frac{\partial T_m}{\partial t} = \frac{\lambda_m}{c_{pm}\rho_m}\frac{\partial^2 T_m}{\partial x^2} + \frac{\lambda_m}{c_{pm}\rho_m}\left(\frac{1}{r}\frac{\partial T_m}{\partial r} + \frac{\partial^2 T_m}{\partial r^2}\right) \ . \qquad (16)$$

Energy balance for adsorbent bed:

$$\frac{\partial T_a}{\partial t} = \frac{\lambda_a}{c_{pa}\rho_a}\frac{\partial^2 T_a}{\partial x^2} + \frac{\lambda_a}{c_{pa}\rho_a}\left(\frac{1}{r}\frac{\partial T_a}{\partial r} + \frac{\partial^2 T_a}{\partial r^2}\right) + \frac{\Delta H}{c_{pa}}\frac{\partial a}{\partial t} \ . \qquad (17)$$

Additional equation describing a mass transfer inside of adsorbent bed is an mass balance. It was written by assumption of LDF (Linear Driving Force) model.

Difference between the amount of adsorbed refrigerant a and maximium adsorbed amount a_{eq} in equilibrium state of a given pressure is described by the differential equation:

$$\frac{\partial a}{\partial t} = k_m \cdot (a_{eq} - a) \ .$$

(18)

The resistance in mass transport was written in the following form:

$$k_m = \frac{15 \cdot D_{so}}{R_p^2} \cdot \exp\left(\frac{-E_a}{8.314 \cdot T_a}\right) \ .$$

(19)

Equilibrium water uptake a_{eq} is expressed by the adsorption-equilibrium equations. In this context, Freundlich equation with coefficients $K = 0.346$, $n = 1.6$ [9] for silica gel-water is used:

$$a_{eq} = K \cdot \left(\frac{P_s \ (T_w)}{P_s \ (T_a)}\right)^{\frac{1}{n}} \ .$$

(20)

Here a_{eq} is the amount adsorbed in the equilibrium state, where $P_s(T_w)$ and $P_s(T_a)$ are the saturation vapour pressure at temperatures T_w (water vapor) and T_a (silica gel), respectively.

Mathematical model of adsorber requires appropriate boundary and initial conditions. Convection between two different cells takes place at the inner side of the tube (metal-fluid circulating in a pipe) and outer side of metal wall (metal-adsorbent). Boundary conditions for fluid and metal wall were written as follows:

$$\alpha_{am} \cdot (T_a - T_m) = -\lambda_m \cdot \left.\frac{\partial T_m}{\partial r}\right|_{r=R2}$$

(21)

and

$$\alpha_{fm} \cdot (T_f - T_m) = -\lambda_m \cdot \left.\frac{\partial T_m}{\partial r}\right|_{r=R1} \ .$$

(22)

There is no heat flux to the fluid at the outlet from adsorber, as well as in the central x-axis.

$$\left.\frac{\partial T_f}{\partial x}\right|_{x=L} = 0 \text{ and } \left.\frac{\partial T_f}{\partial r}\right|_{r=0} = 0 \ .$$

(23)

In adsorption and desorption phase following conditions were assumed:

$$T_f|_{x=o} = T'_{ads} \text{ and } T_f|_{x=o} = T'_{des} \ .$$

(24)

For metal wall:

$$\left.\frac{\partial T_m}{\partial x}\right|_{x=0} = \left.\frac{\partial T_m}{\partial x}\right|_{x=L} = 0 \ . \tag{25}$$

For adsorbent layer mentioned below equations were written:

$$\left.\frac{\partial T_a}{\partial x}\right|_{x=0} = \left.\frac{\partial T_a}{\partial x}\right|_{x=L} = 0 \tag{26}$$

and

$$\left.\frac{\partial T_a}{\partial r}\right|_{r=R3} = 0 \ . \tag{27}$$

Initial conditions for the first adsorption cycle:

$$T_f|_{t=0} = T_m|_{t=0} = T_a|_{t=0} = T|_{t=0} = T_0 \ \text{and} \ \ a|_{t=0} = a_0 \ . \tag{28}$$

Boundary conditions were also formulated for the fins. Between fin and adsorbent a convection heat transfer was assumed, and it is described by the boundary condition of the third order for both lateral sides of fin and fins's top surface. Fin's material is assumed to be the same as the metal tube.

Evaporator and condenser were formulated as steady-state models. Their capacity requires adjustment to the amounts of adsorbed refrigerant in a bed (for an evaporator) and condenser is calculated according to the desorbed amount of refrigerant.

Initial parameters and operating conditions are as follows:

$$T_{des} = 303 - 403 \ [\text{K}] \ \ \text{every} \ \ 10 \ [\text{K}] \ . \tag{29}$$

Formulated dynamic model of an adsorber inside of adsorption cooling cycle consists of the set of partial differential equations, boundary conditions and initial parameters. Numerical solution was based on the 4th order Runge-Kutta algorithm for ordinary differential equations. Sensitivity of COP calculated from equation (14) was analyzed, depending strongly on the heat source temperature, cooling temperature, evaporation temperature, two levels of pressure in the refrigerator.

4   Calculation results of 10 kW cooling units

Absorption cooling cycle was calculated in Engineering Equation Solver (EES) [3] software as a steady state model. Modelling involves the knowledge about the processes in the system, introduced as the equations describing each state point (1 to 11).

Calculations of an adsorption cooling system were made using the AdsSpace.Net program. The software was specially developed for the purpose of a novel mathematical model. The mathematical model was simulating 120 different simulations for different temperature conditions. The temperature values of occurring for both chillers were summarized in Tab. 5. Desorption temperature was assumed as 80 °C and cooling power reached 10 kW from both units. Temperature of chilled water delivered to air conditioning system was 5–8 °C.

Table 5. Main temperature values during calculations.

LiBr-water	Silica gel-water
$T_{des} = 80$ °C	$T_{des} = 80$ °C
$T_{cond} = 25$ °C	$T_{cond} = 25$ °C
$T_{evap} = 2$ °C	$T_{evap} = 5$ °C
$T_c = $ from 10 to 5 °C	$T_c = $ from 12 to 8 °C

Results of simulations allowed obtaining the capacities of heat exchangers inside the cycle and reached values of COP (Tab. 6). Additionally for an adsorption chiller Specific Cooling Power (SCP) was calculated. SCP defines the cooling power in relation to amount of used adsorbent.

Table 6. COP and energy flows at various components in cooling systems.

LiBr-water	Silica gel-water
COP=0.67	COP=0.39
Q_des=14.9 [kW]	Q_des=25.6 [kW]
Q_evap=10 [kW]	Q_evap=10 [kW]
Q_ads=13.75 [kW]	Q_ads=25.6 [kW]
Q_cond=10.65 [kW]	Q_cond=10.61 [kW]
	SCP=0.311 [kW/kg]

5   Conclusions

The aim of this study was to analyse the efficiency of the single stage 10 kW LiBr-water absorption and silica gel-water adsorption cycle in similar operation conditions. The analysis of the COP calculation results introduced in Tab. 2 allows to formulate following conclusions:

1. Absorption chillers can obtain higher maximal value of COP than adsorption chillers.

2. COP characteristics of absorption chillers have higher variability and sensitivity according to heating source temperature and cooling water temperature.

3. Adsorption chillers reaches lower COP values than absorption chillers and are less sensitive to heating source temperature and cooling water temperature.

4. Heat exchangers as adsorber and desorber need to have higher power for an adsorption chiller to reach the same 10 kW cooling power. Condenser and evaporator have almost the same capacities.

In installation of the sorption cooling system for the air-conditioning of the building there should be considered variation of the heating source temperature as well as the thermal comfort conditions. Final selection of the chiller type should be a choice of lower heat energy consumption for supplying the chiller during a year, which takes into account all possible variable parameters.

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