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## LiBr-H<sub>2</sub>O absorption cycle design for whole year use in medium climate conditions

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### Abstract

In the paper a new LiBr-H<sub>2</sub>O sorption system for the whole year use in medium European conditions has been presented. Thermodynamics of the system has been simulated for the steady state conditions, using realistic efficiency factors for each exchanger. The simulation has been done for each characteristic period of the year. Also the heat exchangers surfaces have been estimated using cautiously selected formulas. Finally the economic of the designed system has been calculated as well as the environmental effects. The conclusion is that the use of such a system has to be encouraged by tax system since the environmental effects are very promising, but the investment costs are high.

**Keywords:** LiBr-H<sub>2</sub>O absorption cycle; Solar energy

### Nomenclature

$c_p$	–	specific heat capacity at constant pressure, kJ/(kg·K)
$COP$	–	Coefficient of Performance of the system,
$h$	–	specific enthalpy, kJ/kg
$p$	–	pressure, Pa
$Q$	–	heat/heat transfer rate, kW
$r$	–	heat of condensation, latent heat, kJ/kg
$T, t$	–	temperature, K, °C

### Greek symbols

$\xi$	–	concentration of lithium bromide water solution, %
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$\lambda$	-	thermal conductivity, W/(m·K)
$\mu$	-	dynamic viscosity, Pa·s
$\nu$	-	kinematic viscosity, m <sup>2</sup> /s
$\rho$	-	density, kg/m <sup>3</sup>
$\sigma$	-	surface tension, N/m

### Subscripts

$A$	-	absorber
$C$	-	cooling, condenser
$E$	-	evaporation, evaporator
$G$	-	generator
$H$	-	heating, heating source
$s$	-	solution
$v$	-	vapour

## 1 Introduction

One of the possibilities for decreasing energy consumption for small or average buildings is the use of sorption system for cooling purpose, powered by solar, thermal or other renewable heat sources. However this possibility is rather expensive, so only in hot climates this approach may lead to economical success. In the medium climate conditions it seems that the use of sorption system during twelve months, as a part of “intelligent building” system, may lead to economically reasonable system. In this paper such a system is designed and static simulation of its work for different thermal conditions computed.

## 2 Basic climate data

The climate in Poland is a medium European one. There are lots of influences of mild oceanic climate from the west and a dry continental climate from the east. Winter lasts about three months with snowfalls. Spring usually lasts three months with maximum temperatures about 30°C. Typical temperatures data has been shown on Fig. 1. It is possible to operate solar heated sorption systems from March to October. The design of the whole year system is based on the shown temperatures.

The most frequently used heat source for absorption systems is gas or steam. It is beneficial to use absorption systems together with renewable energy sources such as geothermal sources or solar energy. In this work solar energy is the main heat source. The solar radiation intensity for South Poland (Kraków) is shown in the Fig. 1.

The new sorption system designed here should assure thermal comfort during the whole year. It means that during summer the cooling energy for air conditioning has to be provided as well as heating energy in winter. During transitional

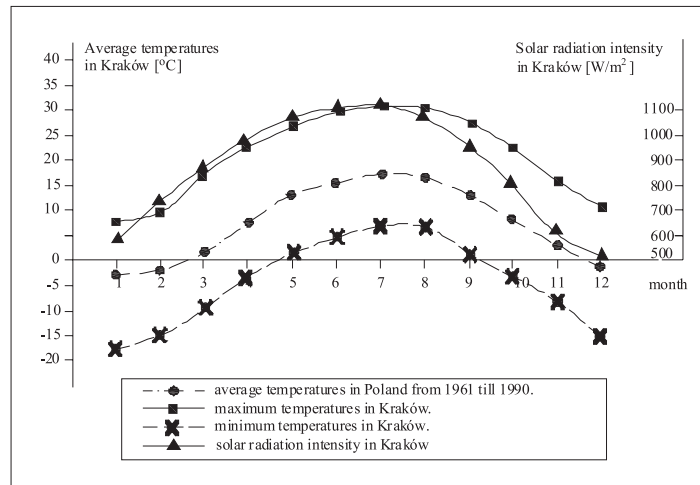


Figure 1. Average temperatures and solar radiation intensity in Poland/Kraków.

periods sorption system can be used for preparation of hot domestic water. The system has been designed for small 140 m<sup>2</sup> house where the heat demand in December is 14 kW (Fig. 2) and cooling power demand for air conditioning in summer is 8-9 kW. Average hot domestic water demand for the four person family is about 300 liters per day.

### 3 The new cycle set up

In Fig. 3 the design of a new system for whole year use has been shown. Main heat source used here is the solar energy. Due to the moderate values of solar radiation in our region and due to heat losses which occur in indirect systems it may be the advantage to use generators heated directly by solar energy. The new system is composed of 3 component sorption systems, i.e. half effect double pump, as well as single effect and double effect in parallel mode.

The outline of main components of the proposed system is given below:

1. additional heat exchanger,
2. high temperature generator 2,
3. regenerative heat exchanger RHE,
4. absorber 2,
5. regenerative heat exchanger RHE,
6. high temperature generator 1,

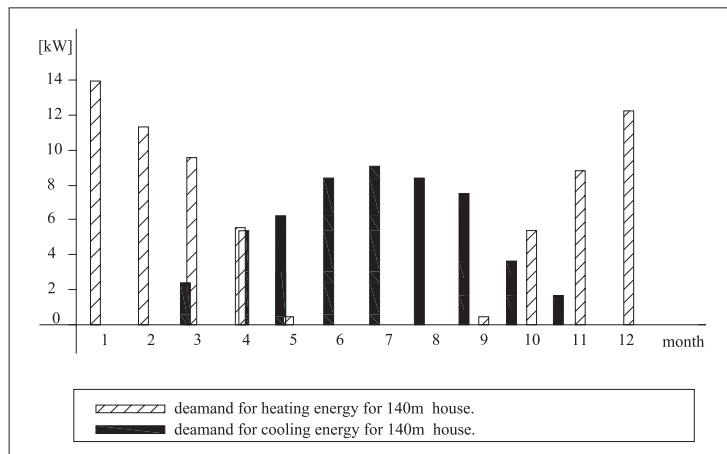


Figure 2. Average heat and cooling energy demand changes during the year for 4 person family for 140m<sup>2</sup> house.

7. regenerative heat exchanger RHE,
8. absorber 1,
9. ground heat exchanger,
10. evaporator,
11. cold storage tank,
12. heat storage tank,
13. hot domestic water,
14. fan coil,
15. dry cooling tower,
16. condenser,
- A. expansion valve.

To operate the system in individual mode the following valves should be opened and following pumps should operate:

Single effect:

- opened valves: 9, 17, 21, 36, 28, 29, 30, 31, 32, 33, 37, 38, 47,
- working pumps: 0, V, VI and VII.

Half effect:

- opened valves: 3, 7, 10, 16, 6, 21, 36, 17, 18, 47,
- working pumps: 0, I, VI and VII.

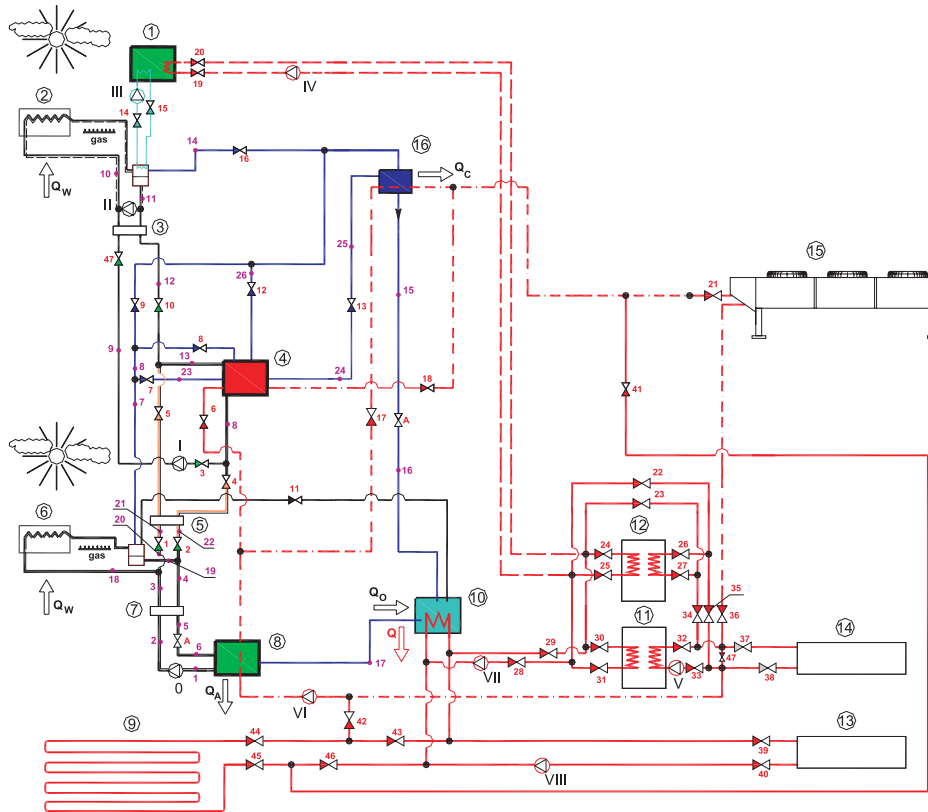


Figure 3. Designed new thermodynamic sorption system setup.

Double effect:

- opened valves: 1, 2, 4, 5, 7, 12, 13, 17, 21, 36, 47,
- working pumps: 0, VI and VII.

Heating mode – variant 1 (using additional exchanger):

- opened valves: 14, 15, 19, 20, 24, 25, 26, 27, 34, 35, 37, 38,
- working pumps: II, III, IV and V.

Heating mode or hot domestic water (HDW) mode – variant 2 (using evaporator as a condenser):

- open valve (heating): 11, 28, 29, 24, 25 (or 22,23), 26, 27,
- working pumps: 0, V, VII,
- open valve (HDW): 11, 39, 40,

- working pumps: 0, VIII.

When subsystems work as a heat pump, in cooling water and ground heat exchanger loop next valves can be opened: 6,18, 17 – depending on the system and 21, 37, 38, 43, 44, 45,46. All these systems work with different driving temperatures and with different  $COP_H$  and  $COP_C$  coefficients depending on characteristic driving temperatures (Fig. 4). The picture contains values of coefficient of performance for cooling. For heating  $COP_H \approx COP_C + 1$ .

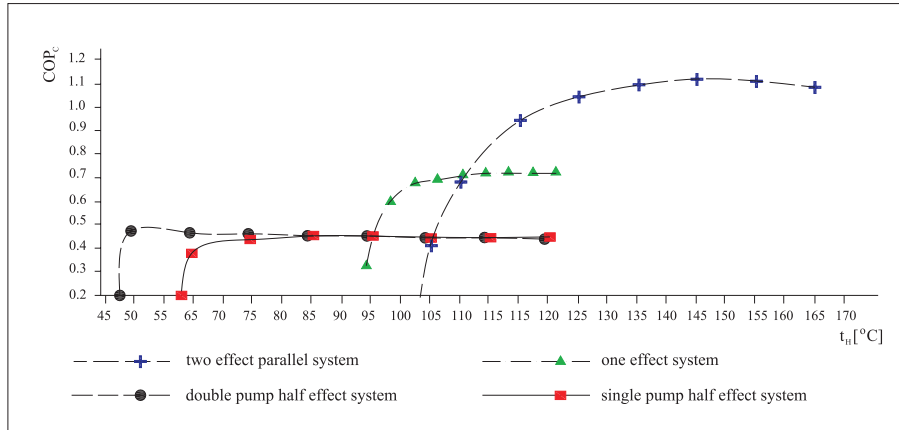


Figure 4. Cooling coefficients of performance for sorption systems.

Half effect systems can be used from March to October and the single effect systems can be used from March to September. Double effect systems can be applied from April till August but their efficiency is too low so it is more economic to use the single effect system instead. It is better to use double effect systems in winter with direct heating by additional backup boiler.

During the designing process the economy of the system shall be considered. Multieffect systems are more expensive than single effect ones but they are more efficient, so there is an advantage to use multieffect cycle. The multieffect system proposed here is a double effect parallel one. It is more complicated than the double effect series system because of the solution distribution cycle, but is safer to use since there is less possibility of crystallization compared to series cycle.

The single effect cycle is the basis of the system. It can be used during the whole summer. A single effect LiBr/H<sub>2</sub>O absorption system, with water as a refrigerant and lithium bromide as absorbent can operate with a heat source temperature from 90 to 120°C, and a coefficient of performance ( $COP$ ) about 0.7 (with condenser cooling water temperature 30°C and the chilled water supply temperature 7°C).

The single effect consists of four main heat exchangers, one regenerative heat exchanger, expansion valve and a liquid pump. There are two main circuits in the cycle. The first one is the circuit of solution and the second one is circuit of refrigerant. Solution circulates between high temperature generator 1 and absorber 1 through solution heat exchanger RHE. The heat exchanger improves the cycle efficiency. Water circulates between condenser and evaporator through a thermal compressor. Low pressure water vapour from the evaporator flows to absorber where it is absorbed by the strong solution. Diluted (weak) solution from absorber 1 is delivered by pump 0 to generator 1. The pump elevates the pressure of the weak solution from low level to high level which is in the generator. In the generator the solution is heated by solar radiation. This causes evaporation of water from solution and solution becomes strong. The liquid strong solution returns to the absorber 1 through a throttling valve providing a pressure drop to maintain the pressure difference between the generator and the absorber. Released in a generator 1 refrigerant vapour flows to air condenser or ground exchanger. In condenser vapour condenses into liquid and than enters to evaporator through expansion valve. In evaporator water boils due to heat delivered with cooled water. All other components which don't participate in the single effect cycle are cut off by ball valves. The detailed system steady state work simulation has been completed, and the results are shown in Tab. 1.

Table 1. Table of results of single effect cycle analysis.

Point number	Temperature [°C]	Pressure [kPa]	Mass fraction LiBr [kg/kg]	Mass flow rate [kg/s]	Enthalpy [kJ/kgK]
1	35	0.87	0.575	0.041	97.7
2	35	5.94	0.575	0.041	97.7
3	43	5.94	0.575	0.041	114.7
4	84	5.94	0.626	0.038	216.4
5	74	5.94	0.626	0.038	197.8
6	74	0.87	0.626	0.038	197.8
7	64	5.94	–	0.003	2620
15	36	5.94	–	0.003	150.8
16	5	0.87	–	0.003	150.8
17	5	0.87	–	0.003	251.0
$t_H = 70\text{ °C}$	$t_C = 30\text{ °C}$	$t_E = 12\text{ °C}$	$Q_E = 8\text{ kW}$	$Q_A = 11.96\text{ kW}$	$Q_C = 8.37\text{ kW}$
$Q_G = 12.4\text{ kW}$	$COP_C = 0.648$				

When the solar radiation is too low what results in too low driving temperatures it is necessary to use half effect cycle. The half effect systems own its name thanks to achieving values of  $COP$ . It is a half of single effect units ( $COP \sim 0.38$  with  $t_C = 32\text{ °C}$  and  $t_E = 9\text{ °C}$ ). There are two kinds of half effect systems: the first one called the Double Pump Half Effect – DPHE containing two absorbers

and two generators and the second one called the Single Pump Half Effect – SPHE and containing two generators but only one absorber.

Proposed system uses DPHE cycle which contains two single effect circuits. The driving temperature is in the range from 50–100°C. A solution from absorber 1 is pumped through a solution heat exchanger to generator 1 where it boils. A strong solution (low H<sub>2</sub>O) from generator 1 is pumped to absorber 1 and the whole circuit is repeated. Released in generator 1 water vapour is absorbed in absorber 2 by a strong solution flowing from high temperature generator 2. Here, in high temperature and high pressure circuit, circulation of solution is repeated and proceeds similarly as in circuit 1. In contrary to first circuit vapour from generator 2 flows to condenser and then to evaporator. Due the exothermic nature of the absorption process the heat has to be removed with cooling water. A concentration of solution after the absorber is a function of pressure and temperature of cooling water. Results from heat and mass transfer balance are listed in Tab. 2.

Table 2. Table of results of DPHE cycle simulation.

Point number	Temperature [°C]	Pressure [kPa]	Mass fraction LiBr [kg/kg]	Mass flow rate [kg/s]	Enthalpy [kJ/kgK]
1	35	0.87	0.575	0.053	68.6
2	35	0.87	0.575	0.053	68.6
3	44	2.22	0.575	0.053	86.2
4	60	2.22	0.614	0.049	165.2
5	50	0.87	0.614	0.049	146.3
6	50	0.87	0.614	0.049	146.3
7	52	2.22	–	0.003	2598.1
8	35	2.22	0.462	0.023	71.2
9	35	5.94	0.462	0.023	71.2
10	43	5.94	0.462	0.023	89.0
11	67	5.94	0.542	0.020	151.6
12	57	5.94	0.542	0.020	130.9
13	57	2.22	0.542	0.020	130.8
14	55	5.94	–	0.003	2602.9
15	36	5.94	–	0.003	150.8
16	5	0.87	–	0.003	150.8
17	5	0.87	–	0.003	2523.2
$t_H = 70\text{ °C}$	$t_C = 30\text{ °C}$	$t_E = 12\text{ °C}$	$Q_E = 8\text{ kW}$	$Q_{A1} = 12.1\text{ kW}$	$Q_{A2} = 9.7\text{ kW}$
$Q_C = 8.3\text{ kW}$	$Q_{G1} = 9.7\text{ kW}$	$Q_{G2} = 12.3\text{ kW}$	$COP_C = 0.362$		

Single effect and half effect cycles can operate using solar energy during a summer or spring and autumn. To use the system in winter direct energy from backup boiler is necessary. This is a high temperature energy source which requires two effect systems.

As mentioned above parallel two effect system is used in this design here. Refrigerant and the solution are circulating between absorber, two generators,



evaporator, condenser and regenerative heat exchanger. Water vapour from evaporator enters the absorber 1 where is absorbed by the strong solution. Weak solution from the absorber is pumped by solution pump 0 to both generators parallelly. The solution exchanges heat with a hot solution in the regenerative heat exchanger. Generator 1 is heated by gas to boiling temperature. Superheated water vapour from high temperature generator 1 becomes a heating source of low temperature generator (absorber 2). In low temperature generator steam returns heat and condenses. Condensation process is a heat source for another part of the solution flowing from absorber 1 and another part of water evaporates. Properties of solution leaving both generators are average properties of both streams. Vapour from low temperature generator (absorber 2) enters condenser (or ground exchanger) and the condensation heat is removed. The absorber and the condenser would reject heat to the same sink. Results from heat and mass transfer balance are listed in Tab. 3.

Table 3. Table of results from double effect cycle simulation.

Point number	Temperature [°C]	Pressure [kPa]	Mass fraction LiBr [kg/kg]	Mass flow rate [kg/s]	Enthalpy [kJ/kgK]
1	37	0.87	0.589	0.03	109.6
2	37	90.65	0.589	0.03	109.6
3	46	90.65	0.589	0.03	126.6
4	158	7.78	0.648	0.028	364.1
5	148	7.78	0.648	0.028	345.4
6	148	0.87	0.648	0.028	345.4
7	120	90.65	-	0.001	2725.0
8	106	7.78	0.693	0.013	294.3
13	53	7.78	0.589	0.015	141.4
15	41	7.78	-	0.003	171.8
16	5	0.87	-	0.003	171.8
17	5	0.87	-	0.003	2510.0
18	46	90.65	0.589	0.015	126.6
19	170	90.65	0.61	0.015	440.9
20	46	90.65	0.589	0.015	126.6
21	46	7.786	0.589	0.015	126.6
22	96	7.786	0.693	0.013	277.0
23	120	90.65	-	0.001	2725.0
24	109	90.65	-	0.001	459.6
25	109	7.786	-	0.001	459.6
26	80	7.786	-	0.002	2650.0
$t_H = 180\text{ °C}$ $Q_{G1} = 5.9\text{ kW}$	$t_C = 30\text{ °C}$ $Q_{G2} = 6.2\text{ kW}$	$t_E = 12\text{ °C}$ $COP_C = 1.08$	$Q_E = 8\text{ kW}$	$Q_A = 13.1\text{ kW}$	$Q_C = 5.8\text{ kW}$

The additional heat exchanger or adopted evaporator as a condenser can be used to produce hot domestic water. The first system requires additional external exchanger and the second one uses evaporator to condense water vapour and to recover heat.

The first cycle works as a boiler. Solution is boiled in generator 2 causing its boiling. Vapour transfers heat to hot water in additional exchanger and leaves it as a fluid. In generator no 2 water is mixing with remaining strong solution and the cycle is repeated. The remaining part of the machine is cut off by ball valves. Temperature of heated water is 95°C and pressure is 94 kPa. When the corresponding temperature in pipes is 65°C whereas pressure is 32 kPa.

In the second cycle evaporator works as a condenser. Water vapor from generator 1 flows to the evaporator which works as a condenser. Heat of condensation and from superheated vapour returns to the heated water flowing inside the tubes from heating system. From condenser (evaporator) condensed refrigerant flows to absorber 1 and then is pumped to generator 1 where the cycle is repeated. Complimentary to the system are cold and heat storage tanks which will store heat energy for refrigerant boiling or directly for cooling system and hot domestic water. Heat and mass exchanger – absorber 2 works as a low temperature generator or absorber depending on the used system. Cooling water heat is removed in air condenser or ground heat exchanger. The backup system is a gas boiler cooperating with the double effect cycle.

In Tab. 4 the *COP* values for actually manufactured sorption energy transformers are shown. Our system is within the same range of *COP*. The ultimate in present technology – three effect system has higher *COP* values, but requires higher temperatures has more operating problems due to crystallization. Besides the investment costs are much higher.

Table 4. Comparison of cooling demand and *COP* coefficient for chillers available on the market.

Manufacturer	Cooling power $Q_E$ [kW]	Electric demand [kW]	<i>COP</i>
Rotartica	4.5	1.8	0.8
Yazaki	35	0.03	0.7
Thermax	134	3.7	1.06
Yazaki	350	2.5	1.00
Broad	350	6.0	1.16
Dunham-Bush	840	5.5	1.0
Mitsubishi	1002	3.5	1.12
Trane	1176	10.4	0.97–1.04
York	1540	15.2	1.19
Carrier	1750	13.2	0.97
McQuay	1750	7.10	1.20
Voltas	1750	7.6	—

## 4 System design and simulation

For modeling designed sorption system the equations for medium properties were chosen. In Tab. 5 approximation equations for thermodynamic properties of water and saturated water vapour calculations were put together.

Table 5. Approximation equations for thermodynamic properties of H<sub>2</sub>O calculations.

Properties	Proposed equation
1. Water density [4]	$\rho = 0.000015451 \cdot t^3 - 0.0059003 \cdot t^2 - 0.019075 \cdot t + 1002.3052$ [kg/m <sup>3</sup> ]
2. Specific heat of water [4]	$c_p = 0.000003216145833 \cdot t^4 - 0.000798668982 \cdot t^3 + 0.0780295139 \cdot t^2 - 3.0481614 \cdot t + 4217.7377$ [J/kg·K]
3. Thermal conductivity of water [4]	$\lambda = -6.5104167D - 10 \cdot t^4 + 0.00000018923611 \cdot t^3 - 2.671875E - 05 \cdot t^2 + 0.0027103175 \cdot t + 0.5520119$ [W/m·K]
4. Dynamic viscosity of water [4]	$\mu = -0.000001(0.000031538716146 \cdot t^4 - 8.913055428199999D - 03 \cdot t^3 + 0.9795876934 \cdot t^2 - 55.4567974 \cdot t + 1791.74424)$ [N s/m <sup>2</sup> ][kg/m·s]
5. Surface tension of water [4]	$\sigma = (235.8(1 - (T/647.096))^{1.256}(1 - 0.625(1 - (T/647.096))))10^{-3}$ [N/m]
6. Thermal conductivity of saturated water vapour [10] 273 < T < 815 K	$\lambda = 0.0167 + 6.79 \cdot 10^{-5} \cdot T + 8.0621 \cdot 10^{-8} \cdot T^2 - 3.512 \cdot 10^{-11} \cdot T^3$ [kW/m·K]
7. Heat of condensation H <sub>2</sub> O [4]	$r = -0.00132635 \cdot t^2 - 2.29983657 \cdot t + 2500.43063$ [kJ/kg]
8. Dynamic viscosity of saturated water vapour [6]	$a_1 = -13.02888482. a_2 = 0.006364014326. a_3 = -4.658585822 \cdot 10^{-06}$ $\ln \mu_v = \sum_{i=1}^3 a_i \cdot T^{i-1}$ [Pa·s]
9. Kinematic viscosity of water [4]	$\nu = -0.0001(3.1770833333 \cdot (10^{-8}) \cdot t^4 - 0.0000089652777778 \cdot t^3 + 0.00098270833333 \cdot t^2 - 0.055322222222 \cdot t + 1.7876666667)$ [m <sup>2</sup> /s]
10. Density of saturated water vapour [1] 273 < T < 815 K	$\rho_v = 4.192 \cdot 10^{-12} \cdot T^4 - 1.25128 \cdot 10^{-8} \cdot T^3 + 1.45079 \cdot 10^{-5} \cdot T^2 - 8.12253 \cdot 10^{-3} \cdot T + 2.17634$ [kg/m <sup>3</sup> ]

The main properties of lithium bromide solution used for heat and mass transfer balance are enthalpy and pressure. For lithium bromide solution enthalpy the Patersen equation (1) [9] has been applied. It can be used in range  $0 < \xi < 70$  [%] and  $0 < t < 180$  [°C]:

$$h = \sum_{i=0}^5 \sum_{j=0}^2 A_{ij} \cdot \xi^i \cdot t_s^j,$$

$$c_p = \sum_{i=0}^5 (A_{i1} \cdot \xi^i + 2A_{i2} \cdot \xi^i \cdot t_s), \quad (1)$$

where:

$$\begin{aligned} A_{00} &= 1.134123 \cdot 10^0, & A_{01} &= 4.124891 \cdot 10^0, & A_{02} &= 5.743693 \cdot 10^{-4}, \\ A_{10} &= -4.80045 \cdot 10^{-1}, & A_{11} &= -7.643903 \cdot 10^{-2}, & A_{12} &= 5.870921 \cdot 10^{-4}, \\ A_{20} &= -2.161438 \cdot 10^{-3}, & A_{21} &= 2.589577 \cdot 10^{-3}, & A_{22} &= -7.375319 \cdot 10^{-6}, \\ A_{30} &= 2.336235 \cdot 10^{-4}, & A_{31} &= -9.500522 \cdot 10^{-5}, & A_{32} &= 3.277592 \cdot 10^{-7}, \\ A_{40} &= -1.188679 \cdot 10^{-5}, & A_{41} &= 1.708026 \cdot 10^{-6}, & A_{42} &= -6.062304 \cdot 10^{-9}, \\ A_{50} &= 2.291532 \cdot 10^{-7}, & A_{51} &= -1.102363 \cdot 10^{-8}, & A_{52} &= 3.901897 \cdot 10^{-11}. \end{aligned}$$

For the case of solution pressure following equation is valid (Kaita [5]) for  $40 < \xi < 65\%$  and  $20 < t < 210^\circ\text{C}$ :

$$\log p = k_0 + k_1/(TD + 273.15) + k_2/(TD + 273.15)^2, \quad (2)$$

$$TD = \sum_{i=0}^2 \sum_{j=0}^3 A_{ij} (\xi - 40)^j \cdot t_s^i$$

where:

$$\begin{aligned} k_0 &= 7.05, & k_1 &= -16035.04, & k_2 &= -104095.5 \\ A_{00} &= -9.133128 \cdot 10^0, & A_{01} &= -4.759724 \cdot 10^{-1}, & A_{02} &= -5.638171 \cdot 10^{-2}, \\ A_{03} &= 1.108418 \cdot 10^{-3}, & A_{10} &= 9.439697 \cdot 10^{-1}, & A_{11} &= -2.882015 \cdot 10^{-3}, \\ A_{12} &= -1.345453 \cdot 10^{-4}, & A_{13} &= 5.852133 \cdot 10^{-7}, & A_{20} &= -7.324352 \cdot 10^{-5}, \\ A_{21} &= -1.556533 \cdot 10^{-5}, & A_{22} &= 1.992657 \cdot 10^{-6}, & A_{23} &= -3.924205 \cdot 10^{-8}. \end{aligned}$$

The solution concentration is calculated on the basis of the Rozenfeld Eq. (3) [11]:

$$\xi = A + \frac{B}{p + C} + D \cdot p^2 + \frac{E + F \cdot \log(p)}{G + H \cdot t} [\%] \quad (3)$$

where:

$$A = 147, B = -66.2, C = 2, D = -139 \cdot 10^{-7}, E = -70.35, F = -49.83, G = 1, H = 0.01.$$

The remaining solution properties can be calculated from equations in Tab. 6.

Table 6. Transport properties of LiBr-H<sub>2</sub>O solution depending on concentration and temperature.

Author	Range	Equation	Constans
Density of LiBr-H <sub>2</sub> O [kg/m <sup>3</sup> ]			
Florides [4]	20% < $\xi$ < 60% 0°C < $t$ < 200°C	$\xi_o = \xi/100$ $\rho = A + B \cdot \xi_o + C \cdot \xi_o - (D + E \cdot \xi_o) \cdot (273 + t)$	$A = 1145.36$ , $B = 470.84$ , $C = 1374.79$ , $D = 0.333393$ , $E = 0.571749$
Murakami [8]	65% < $\xi$ < 70% 333K < $T$ < 430K	$\rho = A_1 \cdot T + A_2$	$A_1 = -0.014396 \cdot \xi + 0.19405$ $A_2 = 30.723 \cdot \xi + 51.718$
Dynamic viscosity of LiBr-H <sub>2</sub> O [kg / m · s]			
Florides [4]	45% < $\xi$ < 65% T [K]	$B = A_1 + (A_2/T) + A_3 \cdot \ln(T)$ $\mu = e^{B/1000}$	$A_1 = -494.122 + 16.3967 \cdot \xi - 0.14511 \cdot \xi^2$ $A_2 = 28606.4 - 934.568 \cdot \xi + 8.52755 \cdot \xi^2$ $A_3 = 70.3848 - 2.35014 \cdot \xi + 0.0207809 \cdot \xi^2$
Dembecki [2]	0% < $\xi$ < 65% 273K < $T$ < 373 K	$b = A + B \cdot \xi + C/T + D \cdot \xi/T + E \cdot \xi^2 + F \cdot \xi^2/T + G/T^2 + H \cdot \xi/T^2 + I \cdot \xi^3$ $\mu = 10^b$ [Ns/m <sup>2</sup> ]	$A = -3.04992473591$ , $B = 1.871635445 \cdot 10^{-2}$ , $C = -9.004581595102$ , $D = 5.237691321$ , $E = -7.879187234 \cdot 10^{-4}$ , $F = 1.617696615 \cdot 10^{-1}$ , $G = 2.682895769 \cdot 10^5$ , $H = -2.365619773 \cdot 10^3$ , $I = 5.611360670 \cdot 10^{-6}$
Kinematic viscosity of LiBr-H <sub>2</sub> O [m <sup>2</sup> / s]			
Dembecki [2]	0% < $\xi$ < 65% 273K < $T$ < 373 K	$b = A + B \cdot \xi + C/T + D \cdot \xi/T + E \cdot \xi^2 + F \cdot \xi^2/T + G/T^2 + H \cdot \xi/T^2 + I \cdot \xi^3$ $\nu = 10^b$	$A = 5.7582877186$ , $B = 1.334933430 \cdot 10^{-2}$ , $C = -1.056897182 \cdot 10^3$ , $D = 6.215138411$ , $E = -7.722896821 \cdot 10^{-4}$ , $F = 1.515134060 \cdot 10^{-1}$ , $G = 2.889152151 \cdot 10^5$ , $H = -2.411303539 \cdot 10^3$ , $I = 5.580141595 \cdot 10^{-6}$
Thermal conductivity of LiBr-H <sub>2</sub> O [W/m·K]			
Kim [K]	T [K]	$\lambda = \sum_{i=1}^3 \sum_{j=1}^4 a_{ij} \cdot \xi^{j-1} \cdot T^{i-1}$	$a_{11} = -0.3863624126$ , $a_{12} = -0.3122938151$ , $a_{13} = 17.75694663$ , $a_{14} = -41.62113683$ , $a_{21} = 0.005245122201$ , $a_{22} = -0.006413302194$ , $a_{23} = 0.0800954908$ , $a_{24} = 0.2130478667$ , $a_{31} = -6.398936707 \cdot 10^{-006}$ , $a_{32} = 1.013622815 \cdot 10^{-005}$ , $a_{33} = 0.0001002925362$ , $a_{34} = 0.0002814495731$
Surface tension of LiBr-H <sub>2</sub> O [N/m]			
Kim [6]	0% < $\xi$ < 60% 273 K < $T$ < 500K	$\sigma = \sum_{i=1}^3 \sum_{j=1}^4 a_{ij} \cdot (\xi \cdot 100)^{j-1} \cdot T^{i-1}$	$a_{11} = 100.0572325$ , $a_{12} = -38.32451999$ , $a_{13} = 72.41610217$ , $a_{21} = -0.02833253108$ , $a_{22} = 0.2991780643$ , $a_{23} = -0.3763508519$ , $a_{31} = -0.0002198127185$ , $a_{32} = -0.0004491657076$ , $a_{33} = 0.0009148617576$

Heat and mass transfer balance calculations for each system component have been used to find all masses, temperatures, pressures and enthalpies in the system. Components and process efficiencies have been estimated on the basis of the comparison of the simulation results to the real LiBr-H<sub>2</sub>O system measurements. Estimated efficiencies are shown in Tab. 7:

Table 7. Exchangers and process efficiencies in proposed system.

	Condenser	Evaporator	Absorber 1	Absorber 2	High temperature desorber 1	High temperature desorber 2	Regenerative heat exchanger
Exchange efficiency	92%	92%	96%	96%	96%	96%	98%
Process efficiency	–	–	90%	90%	90%	90%	–

## 5 System simulation

To calculate heat and mass transfer fluxes in each exchanger above shown equations have been used. To estimate surfaces of exchangers standard widely used formulas have been applied as a first guess. Then the obtained surfaces as well as heat and mass transfer coefficients have been checked up against the values of real working system.

In Tab. 8 heat exchanger surfaces of proposed system for cooling mode are given. The system works in Poland/Kraków weather conditions and cooling demand is equal to data in Fig. 2.

When the system is working as a heat pump it has to cooperate with the low temperature heating system (like water floor heating system). For heating purpose all required heat is from cooling water from condenser and absorber. Maximum achieved temperatures are 35-40°C and higher temperatures can cause crystallization in regenerative heat exchanger. In Tab. 9 we can find heat exchanger surfaces of the proposed system for heating mode (working as a heat pump). The system works in Poland/Kraków weather conditions and heating demand is also equal to data in Fig. 2.

When the system will be used for preparation of domestic hot water we can use one of the described (two) systems. In the case of four persons family the daily demand is about 300 dm<sup>3</sup>/day. For such demand the heat exchanger capacity is 4KW and water temperature on entrance to exchanger is 70°C and 45°C on exit. The exchanger surfaces are: 2.3 m<sup>2</sup> – condenser and 3.1m<sup>2</sup> – additional exchanger (boiler).

Taking all results into consideration heat exchanger areas of each subsystem were estimated and listed in Tab. 10. Due to the fact that operation of exchangers can be lowered by corrosion, sediments or pollution heat exchanger areas have been increased by 20%.

Table 8. Heat exchanger area for proposed system working as a chiller.

month	to tH		QE [kW]	Single effect system [m <sup>2</sup> ]			Half effect system [m <sup>2</sup> ]			Double effect system [m <sup>2</sup> ]									
	[°C]	[°C]		Cond.	Evap.	Abs.	Gen	Cond	Evap.	Abs 1	Abs 2	Gen 1	Gen 2	Cond	Evap	Abs	Gen 1	Gen 2	
January	7	55																	
February	10	80																	
March	17	90	2.1	2.3	2.3	4.1	2.5	3.5	1.5	1.1	4.4	3.8	3.5	4.1	1.3	2.7	4.1	1.3	
April	22.7	102	5	2.7	3.9	3.9	7.2	2.8	3.8	2.9	2.0	7.7	6.50	2.2	5.5	1.7	4.1	1.4	
May	25.4	104	6	2.9	4.1	4.4	8.2	2.9	4.5	3.3	2.3	8.8	7.5	2.2	5.6	1.9	5.1	1.8	
June	29.2	109	8	3.2	4.7	5.4	9.9	3.3	5.8	4.2	2.8	10.6	9.2	2.4	6.4	2.4	6.2	2.5	
July	30.5	108	8.3	3.2	4.7	5.5	10.4	3.3	5.8	4.3	2.9	11.2	9.7	2.4	6.6	2.5	7.1	2.8	
August	30.3	100	7.8	3.1	4.5	5.5	11.8	3.2	5.6	4.0	2.7	12.6	10.7	2.3	6.4	2.4	6.7	2.7	
September	27.6	95	7	3.0	4.3	5.2	11.0	3.1	5.2	3.6	2.5	12.8	10.8	2.3	6.1	2.2	6.1	2.6	
October	22.5	80	3	2.4	3.2	3.3	8.5	2.5	7.3	1.8	1.4	8.90	7.5	2.4	4.6	1.2	3.4	1.3	
November	15.5	56	1.7					2.6	6.0	1.2	1.0	18.0	22.1	2.5	4.1	1.1	2.0	0.9	
December	10.6	45																	

\* - driving temperature  $t_H = 170^\circ\text{C}$ 

Table 9. Heat exchanger area for proposed system working as an heat pump.

month	to tH		Q C [kW]	Single effect system [m <sup>2</sup> ]			Half effect system [m <sup>2</sup> ]			Double effect system [m <sup>2</sup> ]									
	[°C]	[°C]		Cond.	Evap.	Abs.	Gen	Cond	Evap.	Abs 1	Abs 2	Gen 1	Gen 2	Cond	Evap	Abs	Gen 1	Gen 2	
January	-18	55	14																
February	-15	80	11	3.2	5.1	9.9	150	3.5	8.5	4.6	3.1	32	26.8	2.9	8.1	4.5	16	1.4	
March	-10	90	9	2.9	4.8	5.6	85	3.2	8.0	4.1	2.9	18.6	14.9	2.8	8.1	3.9	13.2	1.3	
April	-3	102	5	2.4	3.9	3.6	45	2.7	7.8	2.8	1.9	7.1	6.3	2.4	6.6	2.53	7.2	1.1	
May	3	104	1	2.5	2.4	1.3	3.6	2.8	4.9	1.2	1.1	2.1	1.9	1.2	3.3	1.3	1.7	1.0	
June	5	109																	
July	8	108																	
August	9	100																	
September	3	95	1	2.5	2.4	1.3	6	2.9	4.8	1.2	1.1	2.2	2.00	1.2	3.3	1.3	1.7	1.3	
October	-3	80	4.5	2.3	3.7	6.6	128	2.7	7.9	2.4	1.7	11.3	9.10	2.3	6.2	2.3	6.6	1.0	
November	-8	56	8.3					3.1	8.2	3.5	2.6	86.0	79.8	2.8	7.9	3.6	12.2	1.2	
December	-16	45	12																

\* - driving temperature  $t_H = 170^\circ\text{C}$

Table 8. Heat exchanger area of used subsystems.

Single effect system [m <sup>2</sup> ]				Half effect system [m <sup>2</sup> ]						Double effect system [m <sup>2</sup> ]				
CON	EVAP	ABS	GEN	CON	EVAP	ABS1	ABS2	GEN1	GEN2	CON	EVAP	ABS	GEN1	GEN2
3.9	6.1	6.7	14.1	4.2	10.2	5.6	3.7	21.8	17.8	4.2	7.8	5.4	8.2	4.5

The final surfaces of exchangers are listed in Tab. 11. All tubes are smooth and made of Cu, Ni covered and its length is 1500 mm with internal diameter 17 mm. The final areas are equal to the largest calculated areas of exchangers. For example resulting evaporator area is 12.7m<sup>2</sup>, what is the highest number from 7.6 m<sup>2</sup> (single effect); 12.7 m<sup>2</sup> (half effect) and 9.7 m<sup>2</sup> (double effect).

Table 9. Final heat exchanger surface of proposed whole year working system.

Exchanger type	CON	EVAP	ABS 1	ABS 2	GEN 1	GEN 2
Exchange surface [m <sup>2</sup> ]	4.2	10.2	6.7	4.5	21.8	17.8

## 6 Environmental analysis of proposed whole year absorption working cycle

The absorption systems are considered to be environmental friendly. Examination of the global warming impact of these machines requires consideration of both direct and indirect effects. The direct effects relate to greenhouse gases, and the indirect effects relates to carbon dioxide production. All these influences on environmental system are described by the total equivalent warming impact (*TEWI*). One of the most important part of *TEWI* is global warming potential (*GWP*). The indirect effect relates to the emission of CO<sub>2</sub> due to power plant working.

*TEWI* index value can be calculated from Eq. 4:

$$TEWI = GWP \cdot L \cdot n + n \cdot E_a \cdot \beta_{CO_2} [\text{kgCO}_2], \quad (4)$$

where:

- $GWP \cdot L \cdot n$  – allows for equivalent CO<sub>2</sub> emission caused by refrigerant leakage,
- $n \cdot E_a \cdot \beta$  – allows for CO<sub>2</sub> emission connected with energy production for device powering,
- $GWP$  – global warming potential of refrigerant (for R134a=1800 – average value for whole year),
- $L$  – yearly refrigerant loses due to leakages [kg/year],
- $n$  – time of device operating, [year]



- $E_a$  – yearly energy consumption for device powering (driving energy),  
 $\beta_{CO_2}$  – CO<sub>2</sub> emission index for production of 1 kWh of driving energy (for Poland 0.94), [kg CO<sub>2</sub>/kWh].

Comparing the absorption heat pump or system with the compressor and taking into considerations assumptions mentioned earlier (such as operating time equals 30 years, efficiency, energy consumption) we get:

$$TEWI_{COMP} = 561\,340 \text{ kg CO}_2/30 \text{ year,}$$

$$TEWI_{ABS} = 150768 \text{ kg CO}_2/30 \text{ year.}$$

As we can see in terms of CO<sub>2</sub> emission absorption systems are found to be superior to the compressor one. In the light of tighten up environmental laws and standards the absorption systems are very good alternative for other systems nowadays used. The investment cost of absorption system is higher comparing to compressor system. The cost of compressor system together with a ground heat exchanger is 13–16 000 EUR for the cooling, heating and domestic water demands. Total cost of proposed absorption system estimated on the basis of heat exchangers surface is about 32.000-35.000 EUR but is recompensed by environmental benefits.

## 7 Conclusions

Absorption systems are very promising due to the environmental friendly operation. The main disadvantage of the system is the investment cost, which exceeds the conventional compressor system by factor of 2 (more or less). To balance investment cost the operation costs of the system have to be calculated for the whole year use. To achieve that a new system making possible the whole year operation has been designed. The system is composed of several simpler subsystems, using solar energy and ground heat exchangers, with gas back up for winter time and heat source for the heat pump.

The system has been optimized for medium Europe climate conditions for the whole year operation. The heat exchangers have been calculated using standard formulas. They can be reduced using rotary systems as patented in Rotartica LiBr-H<sub>2</sub>O heat transformer. This would reduce investment costs. The simulation of the system has been calculated using efficiency coefficient for each element taken from the realistic experimental data of conventional LiBr-H<sub>2</sub>O heat transformer. The steady state operation of the system has been computed for each specific time of the year. As a result of this simulation it has been proved that the system is efficient and may work during the whole year. The environmental analysis has also been shown, proving that the TEWI values are four times lower comparing to the compressor system. Since the investment costs are high the encouragement for the system financing should go from the political authorities and the tax system.

The emphasis for the system use should be in connection with the intelligent building programs, where other heat sources may be used.

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