

Cooling Load Production in Sorption Cycles Supplied by a Geothermal Heat Source for Air Conditioning Systems

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ABSTRACT

Direct integration of geothermal system to the absorption or adsorption chillers would expand the spectrum of applications, giving the opportunity to create geothermal heat utilisation more complex. Thermodynamic, chemical and physical properties study is required to describe sorption cycles, with the aim to obtain the most effective operating conditions and reliable co-operation with air conditioning system.

This work gives an overview of sorption technology for cold production. Chillers utilise low temperature geothermal heat as the driven energy directly, minimizing electrical energy consumption in air conditioning system. Authors focused the investigations on absorption cooling cycle based on geothermal heat source.

Engineering Equation Solver (EES) was used to write a code involving modules for each significant physical component of the system. The purpose was to make thermal and fluid-flow calculations and analyse the relationships between internal parameters of a single-stage 10 kW LiBr-water absorption cycle, and the influence on COP. A model was designed and then tested for two variants of a 10 kW cooling machine. The results were applied to an advanced 500 kW chiller unit design for a district cooling system. Calculations are based on the geothermal conditions and data from Geotermia Stargard.

1. INTRODUCTION

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. Many scientific analyses have been carried out to determine the official forecasts (Ministry of Environment, 2000) for geothermal energy production (Zapalowicz et al., 2002; Kabat and Sobanski, 2002). The technical potential of geothermal energy utilization is estimated to be 1512 PJ per year. So the heat energy stored in these reservoirs is of great significance and could be used to decrease the contribution of "conventional" coal-fired power plants in the global energy sector, and thus have positive environmental influence.

Geological explorations have proven that Poland is rich in low-enthalpy geothermal waters, where about 60% (250,000 km²) of the territory has temperatures varying from 30 to 130/C at depths of 1-4 km. Flow tests show suitable conditions from several l/s to 150 l/s. The total

volume of the reservoirs, within the above defined temperature interval, is estimated approximately at 6,500 km³ (Sokolowski, 1993). Active development of geothermal systems from extensive sedimentary formations in three Polish geothermal provinces: Polish Lowland, Fore-Carpathians, and Carpathians (Podhale region) is observed. Polish Lowland is the region with most favourable conditions for utilization of geothermal resources. The main tectonic units are: the Szczecin-Mogilno-Lodz Trough, the Fore-Sudetic Monocline, the Pomeranian Anticlinorium, and the Warszawa-Grudziac Trough. There are big accumulations of Lower Triassic and Lower Cretaceous sandy and muddy complexes with advantageous reservoir properties.

Geothermal low-temperature heat utilization is mainly for space heating purposes, but there is also a long tradition for using warm springs for balneo-therapy and recreation. Five geothermal heating plants are currently being operated in Poland. Systems are designed assuming adaptation to current local conditions. Along with the construction of the geothermal power stations, modernisation of the existing heat distribution network, heat distribution centres as well as internal networks in buildings has been necessary. A doublet geothermal installation principle is being applied in all technical solutions. Different exploitation parameters such as temperature, pressure, heat energy and total dissolved solids, requires individual approaches. Peak loads are covered by gas or oil boilers, and because of low temperatures in reservoirs, heat pump application is recommended. Heat gains from sunlight, lighting, computers, and other electrical appliances raise the indoor temperature of buildings, such that cooling is required sometimes throughout the year. All buildings equipped with air conditioning and cooling systems have high consumption of electrical energy. In many countries, approximately 5-10% of electricity production covers air conditioning needs (Zakrzewski, 2002). Finally, it is important to consider that the bigger electrical energy market will cause higher CO₂ emissions.

The project design calculations presented here consider production of cooling load from geothermal energy in the city of Stargard Szczecinski. To avoid energy losses during energy conversion from heat into electricity, according to the second law of thermodynamics, the best way is to use the energy flux directly. Therefore the emphasis of this study is on the sorption technology for cooling purposes, and the possibility of its operation in a heating mode when there is no demand for cooling.

2. OVERVIEW OF COOLING TECHNOLOGIES

Cooling for comfort and for preservation of food and medicine, has been supplied for most of this century by the vapour compression cycles. The compression cycle requires electrical energy supply, but in an absorption circuit the so-called thermal compression is being used, where a

minimum amount of work is needed. In thermal compression, the binary solution works in the refrigeration cycle and the driving energy is only in the form of heat. 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 (zeolite–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 utilizing 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. The most descriptive information can be found in a paper written by Rogowska (2002).

Absorption chillers operate on one of the earliest known principles of refrigeration. The cycle uses a refrigerant (known as the primary fluid), and an absorbent (known as the secondary fluid). The most common fluids creating binary solution are introduced in Table 2. The refrigerant is chemically and physically absorbed by the absorbent for the purpose of transferring heat. The evaporation of the primary fluid removes heat, thus providing the refrigeration effect (Herold et al., 1996).

Table 2: Various pairs of fluids used in cooling systems

	Absorption agent	Refrigeration agent
Liquid working fluids	Lithium bromide (LiBr)	Water (H ₂ O)
	Water (H ₂ O)	Ammonia (NH ₃)
	LiCl	Water (H ₂ O)
Solid adsorbents	CaCl ₂	Ammonia (NH ₃)
	Active carbon	Ammonia (NH ₃) Methanol
	Zeolite, Silica gel	Water (H ₂ O)

2.1 Coefficient of Performance

COP helps to estimate the operation of heat driven machines. It is defined as the amount of cooling achieved by a refrigeration chiller (down source of heat Q_{down}) per unit of heat supplied (upper source of heat Q_{up}). Some part of energy must be released to surrounding (Q_{amb}).

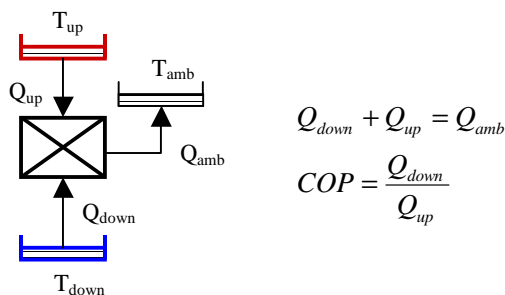


Figure 1: Heat driven chiller, energy balance and COP equation.

The absolute limit for this performance is determined by the second law of thermodynamics. This relation was determined after assumption of adiabatic boundary for the system and than analyzing the entropies increment.

$$COP = \frac{T_{amb} - T_{up}}{T_{up}} \cdot \frac{T_{down}}{T_{down} - T_{amb}} \tag{1}$$

2.2 Basic Principles of Absorption Technology

Absorption cooling is the first and oldest form of air conditioning and refrigeration. An absorption heat pump or chiller does not use an electric compressor to mechanically pressurize the refrigerant. Instead, the absorption device uses a heat source, to evaporate the already-pressurized refrigerant from an absorbent/refrigerant mixture. This takes place in a device called the vapour generator. Although absorption coolers require electricity for pumping the refrigerant, the amount is small compared to that consumed by a compressor in a conventional electric air conditioner or refrigerator. The absorption cycle requires a cooling water supply to enable processes in the absorber and the condenser.

Two basic configurations of absorption technology are available in commercial applications nowadays, their characteristics are presented in Table 3. For cold water temperatures above 0°C (mostly air conditioning for buildings), cycles with lithium bromide as the absorbent and water as refrigerant are designed. For industrial refrigeration and ice production, an ammonia-water technology is employed.

Table 3: Comparison of single-effect vapour absorption technology (Srikhirin et al., 2001)

	LiBr/ H ₂ O	H ₂ O/ NH ₃
Heat source operat. temp. (°C)	80-110	120-150
Chilled fluid temp. (°C)	5-10	<0
Cooling capacity (ton)	10-100	3-25
COP	0.5-0.7	0.5
Current status	Large water chiller	Commercial
Remarks	1.Simplest and widely used; 2.Using water as refrigerant, cooling temperature is > 0°C 3.Negative system pressure; 4.Water cooled absorber is required to prevent crystallization at high concentration.	1. Rectification of refrigerant is required; 2.Working solution is environmentally friendly; 3. High operating pressure; 4.Suitable for using as heat pump due to wide operating range.

The diagram in Figure 2 presents property curves for ammonia and water. Following the evaporation lines, the differences for both fluids can be observed.

Ammonia as the refrigerant, with its physical and chemical properties, evaporates at lower temperatures and at higher pressures. For water, the pressure of evaporation could be lower but temperature will be higher.

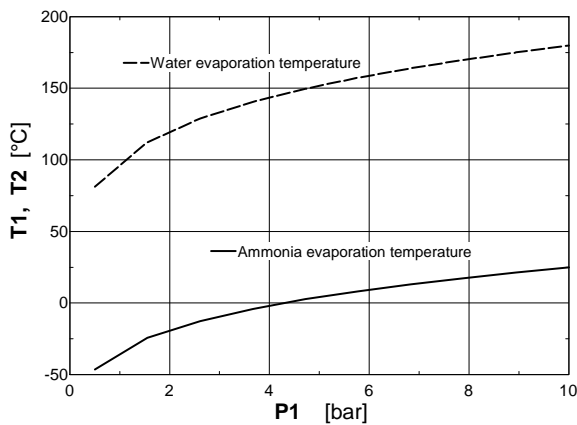


Figure 2: Evaporation temperatures for ammonia and water

In places where electrical power is expensive or unavailable, or where there is waste, gas, geothermal or solar heat available, absorption machines provide reliable and quiet cooling. More advantages of absorption systems against the vapour compression cycle are presented below:

- Required heat energy for generator - could be waste heat or a renewable energy source;
- Lower demand for electrical energy;
- Dependability (reliability)- few moving parts, less replacement parts;
- Low-noise and vibration-free;
- Long life time and low operating costs compared to compressor cooling system;
- Environmentally friendly refrigeration fluids.

The absorption cycle can be compared to the more familiar mechanical vapour compression cycle in that both cycles evaporate and condense a refrigerant liquid at two or more pressures within the unit. The absorption cycle uses a heat-operated generator, a heat-rejecting absorber and a liquid solution pump as presented in Figure 3. Chilled water to the users is cooled down because it gives the heat to the refrigerant (water) in the evaporator causing the change of its thermodynamic state into the vapour phase. Then the refrigerant vapour enters the absorber and it is absorbed by the second fluid (LiBr). The absorption process requires heat removal, so the cooling water accessibility is of great importance. A weak solution is pumped by the solution pump into a solution heat exchanger (SHX) where it is heated up before entering the generator (desorber). The purpose of installing the SHX is the increase in efficiency of the process. A desorption process will start at the moment when heat energy is supplied to the generator to start boiling. Therefore, refrigerant vapour of high enthalpy and high pressure will be produced; the rich hot solution flows back through the SHX to the absorber. It reaches the lower level of pressure after isenthalpic throttling in the valve. The cycle of absorption can start again. Water vapour enters the condenser, the heat of condensation of the refrigerant vapour is rejected to the cooling circuit, so that the quality (vapour fraction) reaches 0. After the expansion valve, the liquid flows to the evaporator.

Exact description and the parameters can be obtained in introduced model.

2.3 Basic Principles of Adsorption Technology

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 (Anyanwu E, 2003).

Practically solid adsorption refrigeration cycles may be classified according to the adsorbent/refrigerant combination used.

It could be seen in Figure 3 and Figure 4 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 low 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).

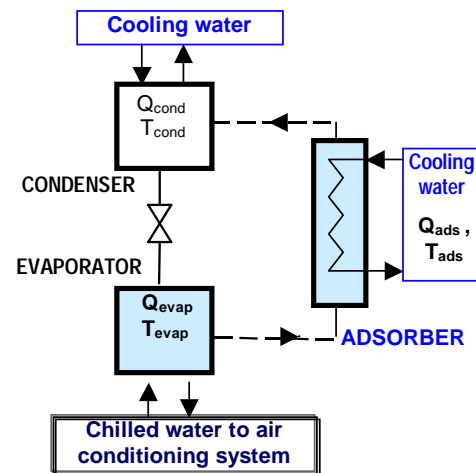


Figure 4a: Operation of adsorption cooling cycle.

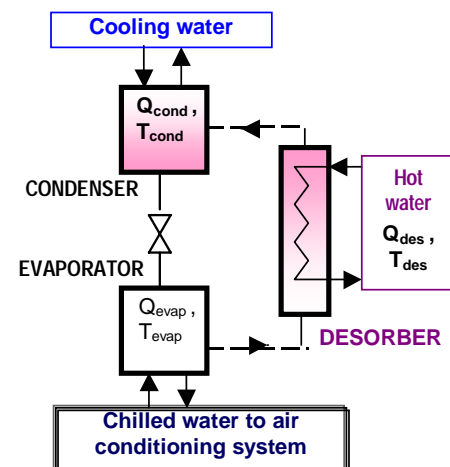


Figure 4b: Operation of adsorption cooling cycle.

In a such adsorption cooling system we can separate two basic phases. The first one is introduced in Figure 4 a, when after heat removal from chilled water the refrigerant vapour leaves evaporator. Then it enters the adsorber space where the refrigerant vapour is adsorbed in the adsorbent bed. This process requires intensive cooling, because adsorption is an exothermic phenomenon.

The second phase (Figure 4b) is desorption process as the result of heating up the adsorbent bed to release adsorbate (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, than the switching of the chambers takes place.

An idealized basic adsorption cycle consists of four thermodynamic steps (Points 1-4), which can be well represented with the Clapeyron diagram, as shown in Fig.5 . The pressure–temperature–concentration (P–T–X) diagram illustrates the above described processes and their typical operating temperatures and pressures(Sumathy et al., 2003).

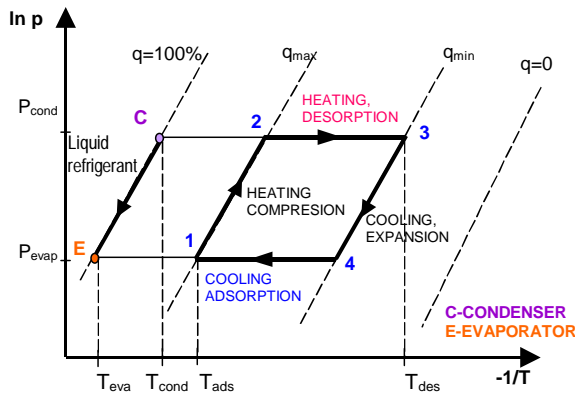


Figure 5: Basic thermodynamic adsorption cycle (P,T,q)

The cycle begins at a point (point 1) where the adsorbent is at low temperature T_{ads} and at low-pressure P_{evap} (evaporating pressure). 1–2 represents the izosteric heating of adsorbent which contains specified amount of adsorbate (refrigerant vapour). The progressive heating of the adsorbent from 2 to 3 causes some adsorbate to be desorbed and its vapor to be condensed (point C). When the adsorbent reaches its maximum temperature T_{des} ; desorption ends. Then the liquid adsorbate is transferred into the evaporator from C to E. The decrease in temperature T_{des} to T_4 induces the decrease in pressure from P_{Cond} to P_{Evap} : Then adsorption and evaporation occur while the adsorbent is cooled from T_4 to T_{ads} .

All solid sorption cycles operate between two levels of temperature (T_{ads} and T_{des}) and pressure (P_{evap} , P_{cond}). Additionally concentration of refrigerant (adsorbate) changes during the cycle and is dependent on temperature under constant pressure conditions. Thus, adsorption and desorption of the refrigerant occur simply by varying the temperature of the combination in a system. This is clearly demonstrated in Fig.6 and Fig. 7 , where the quantity of water adsorbed by zeolite or active carbon is strongly dependent on temperature and only weakly dependent on vapour pressure.

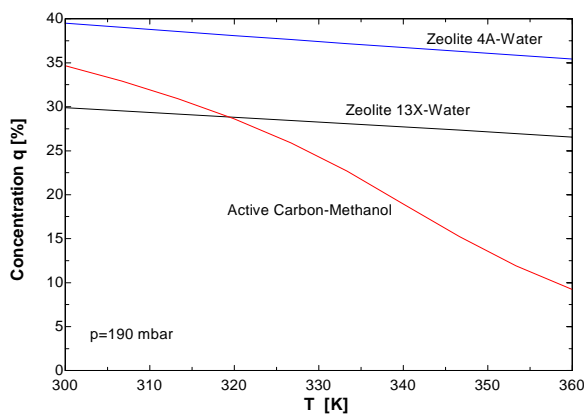


Figure 6: Concentration changes with the temperature

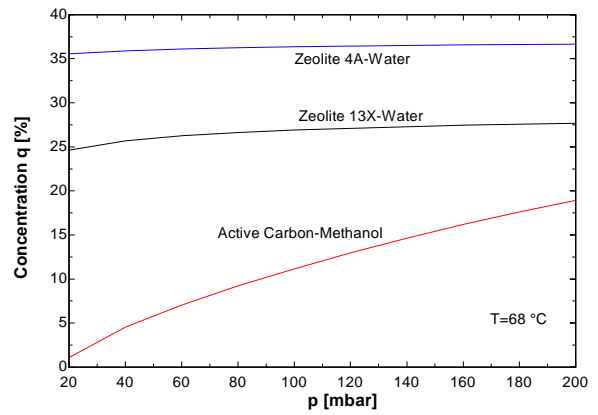


Figure 7: Concentration changes with the pressure

Adsorption characteristics of adsorbents are determined by the adsorption isotherms, for the amount a substance adsorbed.

Theoretical adsorption equilibrium models that provide significant insight into the adsorption properties and act as state equations for simulation of adsorption refrigerators will be introduced in a further work.

2.4 Comparison

Waste heat, solar or geothermal energy driven absorption (liquid– vapor) and adsorption (solid–vapor) 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 HFC’s or natural fluids.

Analyzing literature and realized projects, we noticed that the research in the field of chemical heat pumps including solid/vapor adsorption systems was quite intensive in the last few years.

In comparison with single stage LiBr/H₂O absorption chiller, adsorption chillers are noticeable competitor. First of all LiBr solution is corrosive, so the expenses for the construction of chiller from resistant alloy steel were quite high. We had the risk of crystallization when the desorption temperature was below 61°C in combination with 31°C cooling medium. By silica gel- water, zeolite- water adsorption chillers such problems will not occur.

In a Figure 8 COP variability with driving source temperature was introduced. Curves describing the changes of COP for absorption and adsorption cooling devices, have different course. As mentioned before 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. 8). Adsorption cycles using silica gel–water, zeolite- water active carbon–ammonia and active carbon–methanol as the adsorbent–refrigerant pairs 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 and also great possibility for utilization of low temperature geothermal resources.

Adsorption cycles have been considered for use in heat pump systems, as have many different absorption cycles.

Table 4 introduces more available details about current cooling machines, their parameters and producers.

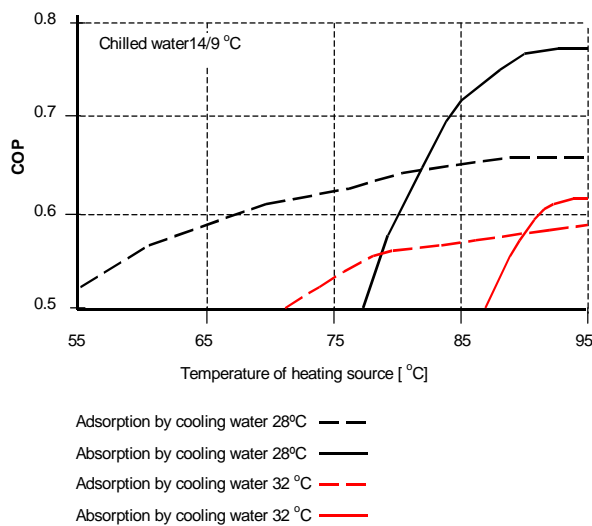


Figure 8: COP for absorption and adsorption 1 stage refrigerators

3. GEOTHERMAL CONDITIONS IN STARGARD SZCZECINSKI

Engineering calculations presented in this paper were carried out for one of the oldest cities in West Pomerania, Stargard Szczecinski.

The city's heating system modernisation plan for the period until the year 2010 provides also the possibility to use large geothermal deposits located right under the city. This opportunity has been used by a company called Geotermia Stargard, whose natural partner in this project is the District Heating Company PEC. According to the agreement that has already been signed, the heat from the geothermal resources will supply the heating system. The expected annual heat production is estimated at 290,000-310,000 GJ that will be delivered to users for a period of 25 years (Kozlowski and Malenta, 2002).

Information presented in this chapter was shared by Geotermia Stargard in the document GS/D/85/2003.

The main components of the heating system on the side of Geotermia Stargard are production and reinjection geothermal wells, a geothermal-heat exchanger, and a return water circulation feed pump. Production well GT-1 is completed; it collects water from the Lower Liassic water-bearing strata - the Mechów Beds and the Radów Beds (Biernat and Parecki, 2002). A pumping test from the existing production was carried out, which proved the well capacity to be 300 m³/h. Water temperature is 87°C at its outlet. Construction of the reinjection well GT-2 with directional drilling technology is completed. Safety valves have adjustment of 16 bar. Other valuable information about dimensions of the geothermal doublet is shown in Figure 3.

On the basis of the maximum value of volumetric flow (300 m³/h), temperature inlet (87°C) and outlet (41°C) from the geothermal heat exchanger, and properties of the geothermal fluid the potential of heat extraction is estimated to be 13460 kW. In the time period from May to the end of September 2002 there is a surplus of geothermal heat energy because only tap water has to be prepared. The heating load does not exceed 8000 kW. Theoretical consideration allows formulation of the following scenario:

Possibility of utilizing the geothermal surplus energy during the summer time up to the maximum amount of 13,460 - 9,000 kW = 4,460 kW for operation of a sorption cooling system.

4. SYSTEM DESIGN

The purpose of this work is to make the thermal and fluid-flow calculations for a proposed cooling system based on a geothermal heat source. For geothermal conditions in Stargard Szczecinski authors decided to choose for calculations an absorption cycle, giving in this case the highest COP values.

The ammonia-water pair is not suitable for application because of the high temperature requirements for the generator (125-170°C). Therefore, the single stage (single effect) LiBr - water cooling technology was chosen as the most appropriate match to available temperatures. In this case, the range of driven temperatures for the desorption process from 75 up to 120°C provides the value of coefficient of performance (COP) about 0.7 (Florides et al., 2003).

The use of geothermal heat energy or other sources of energy, like for example waste energy from industrial processes, can be interesting for the Polish climate with the need for space heating during the wintertime, and space cooling during the summertime. The geothermal water will be utilized as a heat source for the cooling system. The surplus will be a supplement for the heat energy supply for the district heating network featuring different operational requirements.

The cooling system considered here, shown in Figure 3, is designed to deliver chilled water to the balneology centre, office buildings with air conditioning, and storage, as these are possible users of refrigeration. It is especially important to create a distribution network that combines possible connection of new users and a variable cooling load demand during the year.

To give a better overview in this work we investigated more exactly an absorption cooling cycle, analyzing the influence of internal parameters on the efficiency and cooling load production.

4.1 Modelling Work with the Engineering Equation Solver (EES)

Modelling of thermal systems presents many advantages. The most important is the elimination of the expenses for solutions which could bring big risks during future operation. Modelling involves the knowledge about the processes in the system, introduced as the equations describing each state point. The next step is the validation of the model with respect to expected conditions to determine e.g. necessary input and output energy, and to predict temperature variations. Finally, by applying optimization techniques the suitable solution can be found. If time allows, simulations on the behaviour of the complete system can be repeated for corrections.

In this study, the computer software EES (F-Chart Software, 2003) was used to investigate the performance of the absorption cooling system and its behaviour in configuration with a geothermal heat source. Each component of the system is 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.

The EES program was written as a set of modules for each physical component of the system. This structure enables adding, replacing, or removing the modules; and also modules can be employed several times, with different parameter inputs. This process happens without changing the other components. Such principles present many advantages. It is suitable to explore the effect of design alternatives considering system configuration and new models of the used components.

The property equations including relationships of enthalpy, temperature, concentration, and vapour pressure of LiBr-H₂O solution were used from the external library. These functions use correlations from the ASHRAE Handbook of Fundamentals (ASHRAE, 1989). Thermodynamic properties of water have been implemented using the thermodynamic property correlation (Harr et al., 1984).

4.2 Heat and Mass Transfer Equations and Basic Assumptions for a Cooling System

The absorption cooling cycle consists of coupled mass and heat transfer balances in the evaporator, absorber, desorber and condenser without the solution heat exchanger. A single stage LiBr - H₂O absorption system is presented, and all calculations done in reference to the numbering principle illustrated in Figure 3. 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;
- The pump is isentropic;
- The throttling processes in valves are isenthalpic;
- Heat source supplies pressurized hot water to desorber.

It is necessary to properly understand the thermodynamic state of each point in the diagram. The external energy transfers to the absorption system are shown on the schematic in Figure 3. The main processes and assumptions for the cooling circuit are summarized in Table 5.

Table 5: Thermodynamical state points for an absorption cycle

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 - 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 paper, mass flow and respectively energy flux were used.

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 them 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.

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

$$W = \frac{\dot{m}_{12} \cdot v_{12} (p_{12} - p_{11})}{\eta_{sp}} \quad (2)$$

As mentioned in point 2, the value of the electrical energy for this pumping is very small and in comparison with heat energy used in this system, can be neglected.

The effectiveness of the heat exchanger ε is a useful parameter to describe the performance of a heat exchanger. Definition says that it is the ratio of the actual heat transfer to the maximum possible heat transfer for given inlet conditions (Herold at al., 1996). In calculations the minimum heat capacity occurs on the colder side of solution heat exchanger, and then the effectiveness would be written as:

$$\varepsilon = \frac{T_{14} - T_{15}}{T_{14} - T_{12}} \quad (3)$$

These parameter influences COP of the system in large scale. The higher 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.

Another mass flow parameter is so called solution circulation ratio, f presenting the relationship between mass flow rate through solution pump and vapour flow rate leaving generator, as follows:

$$f = \frac{m_3}{m_7} \quad (4)$$

A heat transfer models of each heat exchanger were added to the system to provide better understanding 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:

$$\Delta T_{lm} = \frac{(T_{h,1} - T_{c,1}) - (T_{h,2} - T_{c,2})}{\ln \left(\frac{T_{h,1} - T_{c,1}}{T_{h,2} - T_{c,2}} \right)} \quad (5)$$

where h and c refer to the hot and cold sides of the heat exchanger, 1 and 2 mean 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 very useful for heat exchanger calculations and can be found in the literature (Holman, 2002). The amount of the exchanged heat can be then calculated from the formula:

$$\dot{Q} = UA \cdot \Delta T_{lm} \quad (6)$$

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

$$COP = \frac{Q_e}{Q_d} = \frac{m_{17}(h_{17} - h_{18})}{m_{24}(h_{24} - h_{25})} \quad (7)$$

This is the main parameter to estimate the efficiency of operating absorption system, the easiest interpretation could be formulated 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 in an analysis.

The main geothermal multi-plate counter-flow heat exchanger was calculated using the assumption that the heat capacities of both fluids are equal, and the pinch method was applied to determine the temperature difference between them.

For the purposes of the calculations, the mass flow \dot{m}_4 in the municipal network water was an input number as well as the mass flow \dot{m}_{24} in external loop.

Connection with the municipal heating network system is available through external circuit to the generator. To avoid situation that all heating medium will have to be delivered to the cooling system, as the first user, a by-pass in point 25 was designed. Such solution allows controlling of the mass flow \dot{m}_{24} , in point 5 mixing of two heat fluxes takes place. Then the heat energy is guided to district heating system. Energy balance helps to obtain amount of heat energy delivered to users:

$$Q_{ghx} - Q_d = Q_{user} \quad (8)$$

Return water pressure is expected as 350 kPa (point 6), pressure after the circulation pump 700 kPa (point 3). External water loop at the evaporator side delivers chilled water to the consumers. During calculations T_{17} , \dot{m}_{17} were assumed.

4.3 Calculation Results

To find suitable operating conditions for specific applications, a sensitivity calculation was performed utilizing a computer program EES, which follows all assumptions, sequence thermodynamic equations and involving mathematical correlations for the fluid properties.

Geothermal doublet, heat exchanger, connection to cooling system and municipal district heating network, were presented in simplified way in Figure 3. In each state point enthalpy, mass flow, pressure, temperature, LiBr concentration and quality of water after throttling, were investigated.

To illustrate the relationships between parameters in the absorption cooling system with the same evaporator cooling capacity of 10 kW, two variants will be introduced:

- Evaporator, absorber pressure $P_{10}=0.9$ kPa; generator, condenser pressure $P_{14}=5$ kPa, solution heat exchanger exit temperature $T_{13}=55^\circ\text{C}$; Generator exit LiBr concentration $x_{14}=62\%$,
- Evaporator, absorber pressure $P_{10}=0.7$ kPa; generator, condenser pressure $P_{14}=7.347$ kPa; solution heat exchanger exit temperature $T_{13}=55^\circ\text{C}$; Generator exit LiBr concentration $x_{14}=60\%$.

Table 6 presents a summary of energy fluxes in main components like desorber, evaporator, absorber, condenser, solution heat exchanger, power needed for the solution pump, capacity of geothermal heat exchanger, heating load delivered to the district heating users, return water pump power.

Table 6: COP and energy flows at various components in cooling system for both variants

Variant A	Variant B
COP=0.7567	COP=0.6944
$Q_d=13.21$ [kW]	$Q_d=14.4$ [kW]
$Q_e=10$ [kW]	$Q_e=10$ [kW]
$Q_a=12.63$ [kW]	$Q_a=13.75$ [kW]
$Q_c=10.59$ [kW]	$Q_c=10.65$ [kW]
$Q_{shx}=1.605$ [kW]	$Q_{shx}=2.635$ [kW]
$W_{sp}=0.1205$ [kW]	$W_{sp}=0.2684$ [kW]
$Q_{ghx}=15395$ [kW]	$Q_{ghx}=15395$ [kW]
$Q_{user}=15382$ [kW]	$Q_{user}=15381$ [kW]
$W_p=35.26$	$W_p=35.26$
$\mathcal{E}_{sxx}=0.5625$	$\mathcal{E}_{shx}=0.5398$

In introduced two cases A and B, the same cooling load is produced in evaporator, but in the variant A the coefficient of performance is much higher. It means less heating energy is needed to enable a sufficient desorption process in generator. In addition, the size of other heat exchangers is smaller. To analyse the reason of the higher COP, a sensitivity analysis is carried out in the next section.

4.3.1 Discussion on COP Variation

Figure 9 shows the effect of the generator outlet temperature T_{14} on the variation of the coefficient of performance on the two variants introduced in Section 4.3. The cooling COP of the absorption cooling system is higher for scenario A, with the pressure in generator $P_{12} = 5$ kPa. Another conclusion is that the higher the T_{14} , the lower the COP value. This behaviour may be explained by the fact that although the high temperature of the heat source tends to increase the cooling COP, it also increases the average temperatures in the condenser and absorber, which results in the decrease in the COP. This negative effect reduces the benefits of high-temperature heat source utilization and decreases COP, but at the same time it confirms that absorption technology with a low-temperature geothermal source gives acceptable results in cooling load extraction.

The influence of the generator temperature T_{14} on the desorber pressure P_{14} was also investigated. It depends on the properties of the lithium bromide-water solution. With the concentration x_{14} known, the pressure in the generator can easily be calculated.

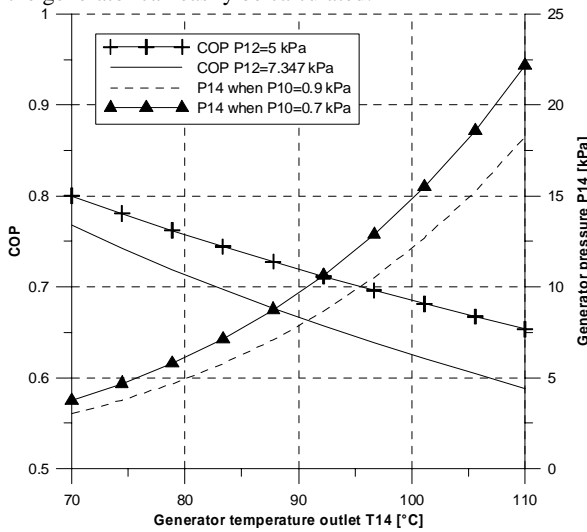


Figure 9: Effect of generator exit temperature T_{14}

An increase of the generator exit temperature T_{14} , causes lowering of the COP and increase of desorption pressure. These two parameters, COP and generator pressure P_{14} , are strongly connected with each other. The mentioned relationship is shown in Figure 10, also taking into account assumed conditions in the two variants A and B, with the different evaporator pressures of 0.7 kPa and 0.9 kPa, and, respectively, different LiBr concentrations of 62% and 60% at the inlet to the absorber. It can be seen that higher desorption pressure in the generator influences COP in a negative way.

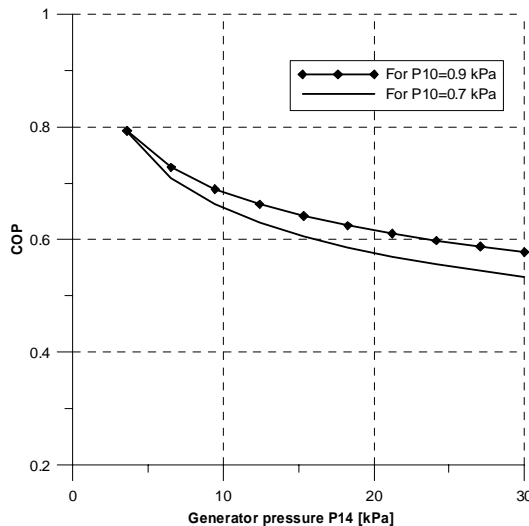


Figure 10: Effect of generator pressure

When LiBr is dissolved in water, there exists a specific minimum solution temperature for any given salt concentration. It is important to operate the system within such conditions to keep a distance from the crystallization line. Since the generator exit LiBr percentage ratio x_{14} is kept fairly constant at 60%, the COP value reaches almost maximum, as shown in Figure 11. Additionally, as seen in Figure 12, COP increases with lower concentrations of LiBr in the working solution at the absorber exit side (x_{11}). With

the considered variants, the LiBr percentage ratio was chosen as 55%.

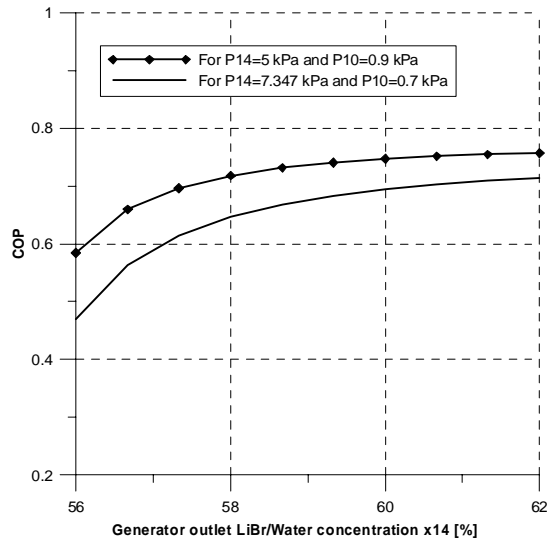


Figure 11: Effect of LiBr percentage ratio

As mentioned before, there is a strong relationship between LiBr concentration and temperature of the solution. To ensure proper operation of the absorption cycle, the temperature at the outlet from absorber T_{11} has to be kept at a lower level. It requires the supply of low temperature cooling water for the absorber heat exchanger. In this paper, the temperatures of the cooling medium were assumed as 25°C at the inlet, and around 35-37°C at the exit.

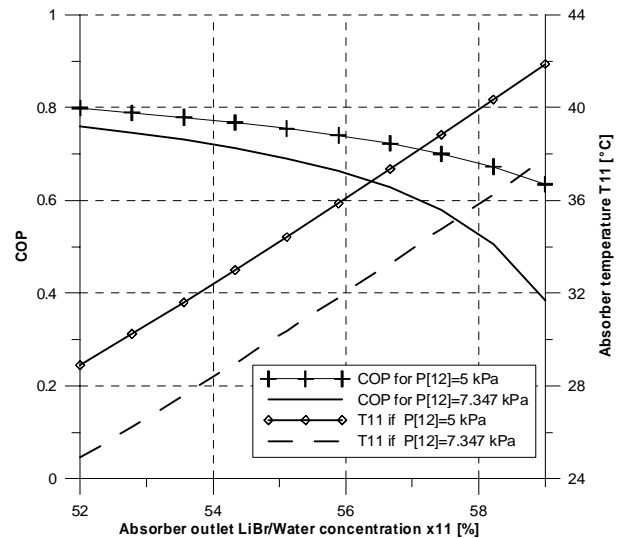


Figure 12: Effect of LiBr percentage ratio at absorber exit x_{11}

The most important effect, which is discussed in literature by Herold et al. (1996), is the COP increase with higher temperature of the medium delivered to the generator (state point 24).

Increasing the heat source temperature contributes to improvement in the COP for the cooling circuit as can be seen in Figure 13. However, there exists a certain limit of COP, and higher temperatures for the generator will not overcome it. So there is no COP benefit to operate single effect machines with high temperatures; to get higher COP, double effect devices are available.

The low temperature of the heat source has an influence on the bigger size of the heat exchange area in the desorber

and other components of the absorption cooling cycle, and it will be visible as a higher cost of the system.

For an absorption cooling system, heat from the absorber and condenser must be rejected at the cooling tower, or another source of cooling must be found, e.g. sea, river water. The cooling streams to the condenser, and the absorber can be connected in parallel or serial flow arrangements. In the considered cases, the temperature of the cooling fluid to both components was identical (25°C).

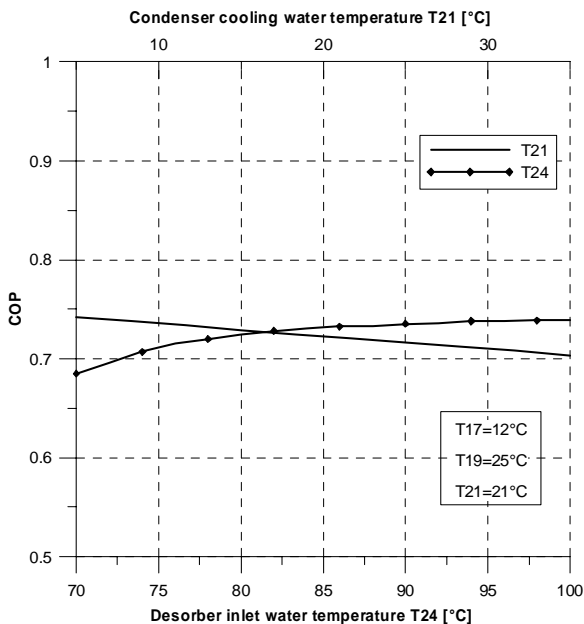


Figure 13: Effect of generator inlet temperature T_{24} and temperature of cooling source T_{24}

4.4 Calculated Results for 500 kW Cooling Unit

The system in this study consists of a district heating network and the connection with an absorption cooling unit. Evaluation of this kind of a system is related to the demands for cooling and heat supply for the district heating users. The initial cooling load demand analysis proved that 500 kW cooling power would be sufficient for the requirements of the city of Stargard Szczecinski. Table 7 presents the results of calculations for the main points and follows the numerical sequence in the scheme in Figure 3.

Calculations were based on expected operating conditions. Average mass flow from the production well was assumed to be 70 kg/s (250 m³/h), the pressure 1500 kPa, and the transferred heat by the geothermal fluid is 13,471 kW. Enthalpy, mass flow, pressure, temperature, LiBr concentration and quality of water were investigated in each state point. Mass flows and temperatures in the introduced system, were controlled at the external heating controlled at the external heating loop to the desorber, so that the district heating users will get sufficient amount of hot water with the right parameters. The desorber will be supplied with 20 kg/s of 85°C hot water. Under these conditions, to obtain 500 kW chilled water at the evaporator, the exit temperature from the desorber needs to be 76.67°C. After mixing with stream 23, heat is transferred to the district heating system in the amount of 12,773 kW. Design of a single-stage absorption chiller was possible after taking into account previous considerations regarding COP variations. The pressure level chosen for the evaporator and absorber was 0.77 kPa, and for the desorber and condenser at 6 kPa. Coefficient of performance equals 0.7164 for the introduced 500 kW unit. Heating power delivered to the desorber was

698 kW and the solution heat exchanger power of 123.6 kW determines the choice of equipment.

There is no doubt that the cooling water demand for this size of the cooling unit plays a significant role. For this case, a 25°C cooling water source was assumed. The results show that a high flow of the cooling medium is needed for the condenser, $\dot{m}_{21} = 12.68 \text{ kg/s}$ and absorber $\dot{m}_{19} = 13.3 \text{ kg/s}$. Heat fluxes for both components are $\dot{Q}_{-c} = 530.4 \text{ kW}$ and $\dot{Q}_{-a} = 667.6 \text{ kW}$, respectively.

There are two pumps in the system, - power needed for the solution pump is 0.1205 kW, and for the return water pump it is 30.85 kW. The effectiveness of the solution heat exchanger, $\varepsilon = 0.5691$. Assumptions made for the 500 kW absorption chiller calculations are introduced in Table 8, and the results from the EES code for district cooling system are shown in Table 9. They will be used for equipment sizing.

Table 9: COP and heat transfer rates for components of the system with 500 kW LiBr- water absorption chiller

Parameters for 500 kW cooling system	
COP	=0.7164
Q_d	=698 [kW]
Q_c	=530.4 [kW]
Q_e	=500 [kW]
Q_a	=667.6 [kW]
Q_{shx_h}	=123.6 [kW]
Q_{ghx}	=13471 [kW]
Q_{user}	=12773 [kW]
W_{sp}	=0.1205 [kW]
W_p	=30.85 [kW]
ε_{shx}	=0.5691

5. CONCLUSIONS AND REMARKS

• Geothermal conditions in Stargard Szczeciński are suitable as a heat source for an absorption cooling

chiller, the 85°C supply water gives COP 0.71.

• During summer, there is a surplus of geothermal heat energy; this could be used for cooling purposes.

• Internal parameters have a significant influence on the COP and the reliability of the cooling system.

• Results of calculations allow the proper choice of equipment.

• High cooling water demand for the condenser and absorber could be obstacles for implementation of the absorption technology.

• The EES program enables the creation of various internal conditions for a chosen refrigerant-absorbent solution, as well as considering the different parameters of the geothermal heat source.

Further work will include validation of the model with expected conditions to determine amounts of input, output energy, and prediction of temperature variations. Finally, applying optimization techniques including economics, should give suitable solution. If time allows, simulations can be carried out to represent the behaviour of a complete system under different operating conditions such as a different heat source, cooling water, chilled water, and hot water temperatures. Further studies are recommended to

investigate the operation of an absorption system as a heat pump during the winter time.

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NOMENCLATURE

A	= Area (m ²);
COP	= Coefficient of Performance;
EES	= Engineering Equation Solver;
f	= Solution circulation ratio;
h	= Enthalpy (kJ/kg);
m	= Mass (kg);
\dot{m}	= Mass flow (kg/s);
Q	= Energy (kJ);
\dot{Q}	= Heat demand (kW); Heat transfer rate (kW);
q	= Vapour quality;
P	= Pressure (kPa);
SHX	= Solution heat exchanger;
T	= Temperature (°C);
ΔT_{lm}	= 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 letters

ε	= Effectiveness;
ρ	= Density (kg/m ³);
v	= Specific volume (m ³ /kg);
η	= Efficiency

Subscripts

shx	= Solution heat exchanger
ghx	= Geothermal heat exchanger
net	= District heating network
user	= District heating users
e	= Evaporator
d	= Desorber
c	= Condenser
a	= Absorber
p	= Pump
sp	= Solution pump

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Table 1: Known various methods of cooling load production (BHKW Infozentrum Rastatt, 2003)

	Compression chiller	Absorptive chiller	Adsorptive chiller	DEC ¹
Physical Cooling - Effect	Vaporise of refrigerant			Evaporation of refrigerant
Kind 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 NH3 as absorption agent	Water with solid as adsorption agent (SILICA-Gel)	Water
Coefficient of performance ²	1.3-1.65	0.6-1.0	0.4-0.6	0.3

1 Desiccative and evaporative cooling

2 Coefficient of performance= ratio of received cooling load to employed heating load. 0.6-1.0 by absorptive chillers means that 1 kWh heat provides 0.6-1.0 kWh cold.

Table.4 Overview of current cooling technologies (BINE Informationsdienst, 2004)

	Absorption chiller			Adsorption chiller
No of stages	1-stage	2-stage	1-stage	1-stage
Working pairs	LiBr-Water	LiBr-Water	Water-Ammonia	Silica gel-water
Driving temperature [°C]	80-110	140-160	80-120	60-95
COP	0.6-0.8	0.9-1.2	0.3-0.7	0.4-0.7
Available capacity on the market	Few Producers >20kW Many Producers >100kW	Few Producers >50kW More Producers >100kW	Small units directly fired Big units especially designed for customers	50-350 (Mayekawa) 250-500 (Nishiyodo)
Producers	York, Yazaki, EAW, Trane, Carrier, Broad, Ebara, LG Maschinery, Sanyo- McQuay, Sulzer-Escher Wyss, Entropie, Century		Directly fired: Robur, Colobri, ATW, Mates; Hot water, steam: ABB, Colibri, Mattes	Mayekawa, Nishiyodo

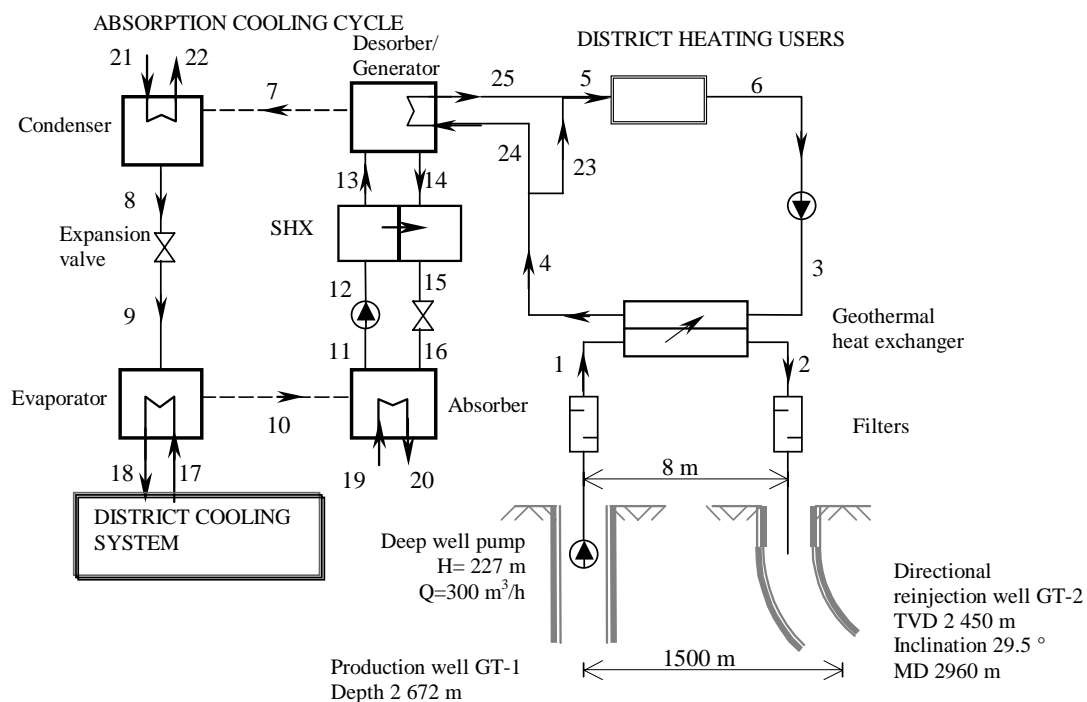


Figure 3: Circuit diagram with LiBr cooling

Table 7: Operating parameters of system with 500 kW LiBr- water absorption chiller

	1	2	3	4	5	6
	h_i [kJ/kg]	m_i [kg/s]	P_i [kPa]	T_i [°C]	x_i	q_i
[1]	365.4	70	1500	87		
[2]	173	70	1500	41		
[3]	163.9	70	700	39		
[4]	356.4	70	700	84.99		
[5]	346.4	70	700	82.61		
[6]	163.9	70	350	39.07		
[7]	2649	0.2138	6	79.56		
[8]	167.5	0.2138	6	40		
[9]	167.5	0.2138	0.77	3.221		0.06176
[10]	2506	0.2138	0.77	3.221		
[11]	76.49	2.566	0.77	31.7	55	
[12]	76.49	2.566	6	31.7	55	
[13]	124.7	2.566	6	55	55	
[14]	192	2.352	6	79.56	60	
[15]	139.5	2.352	6	52.32	60	
[16]	139.5	2.352	0.77	41.29	60	
[17]	50.65	20.65	300	12		
[18]	26.44	20.65	300	6.221		
[19]	104.8	13.3	60	25		
[20]	155	13.3	60	37		
[21]	104.8	12.68	100	25		
[22]	146.7	12.68	100	35		
[23]	356.4	50	700	84.99		
[24]	356.4	20	700	84.99		
[25]	321.5	20	700	76.67		

Table 8: Assumptions for calculations of the district cooling system with 500 kW cooling unit.

Modules	Data for calculations
Condenser	$T_{pinch_c}=15$ [K] $T[21]=25$ [°C] $T[22]=35$ [°C] $P[21]=100$ [kPa]
Evaporator	$T_{pinch_e}=3$ [K] $P[10]=0.77$ [kPa] $Q_e=500$ [kW] $T[17]=12$ [°C] $P[17]=300$ [kPa]
Absorber	$P[19]=60$ [kPa] $T[19]=25$ [°C] $T[20]=37$ [°C]
Solution pump	$\eta_p=0.80$ $P[12]=6$ [kPa]
Generator	$P[4]=700$ [kPa] $m[4]=70$ [kg/s] $m[24]=20$ [kg/s] $x[14]=60$ [%LiBr] $T[13]=55$ [°C]
Users, pump, geothermal heat exchanger	$\eta_{p_net}=0.80$ $T_{pinch_ghx}=2$ [K]
Parameters from the well	$m[1]=70$ [kg/s] $T[1]=87$ [°C] $T[2]=41$ [°C] $P[1]=1500$ [kPa]