GEOTHERMAL TRAINING PROGRAMME
Orkustofnun, Grensásvegur 9,
IS-108 Reykjavík, Iceland

Reports 2003
Number 19

DISTRICT COOLING BY A GEOTHERMAL HEAT SOURCE

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ABSTRACT

The design of an absorption cooling cycle based on a geothermal heat source is proposed for Stargard Szczeciński in West Pomerania, Poland. Engineering Equation Solver (EES) was used to write a model involving modules for each significant physical component of the system. The purpose was to make thermal and fluid-flow simulations to analyse the relationships between internal parameters of a single-stage 10 kW LiBr-water absorption cycle, and the influence on the Coefficient of Performance (COP). A model was designed and then tested for two variants of a 10 kW cooling machine. The results were then taken a step further, and applied to an advanced chiller design for a district cooling system. After a preliminary study of cooling load demand for the city of Stargard Szczeciński, a 500 kW unit is suggested. 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 Sobański, 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 km^2) 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 km^3 (Sokołowski, 1993). Active development of geothermal systems from extensive sedimentary formations in three Polish geothermal

FIGURE 1: Map of Poland showing the town of Stargard Szczeciński and the main geothermal provinces (Kępińska, 2003)

provinces: Polish Lowland, Fore-Carpathians, and Carpathians (Podhale region) is observed (Figure 1). Polish Lowland is the region with most favourable conditions for utilization of geothermal resources. The main tectonic units are: the Szczecin-Mogilno-Łódź Trough, the Fore-Sudetic Monocline, the Pomeranian Anticlinorium, and the Warszawa-Grudziąc Trough. There are big accumulations of Lower Triassic and Lower Cretaceous sandy and muddy complexes with advantageous reservoir properties. Reservoir rocks are recognised as permeable and with high productivity potential. Depth of the occurrence of the porous structures varies between 1000 and 3000 m in extent (Biernat and Parecki, 2002).

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 CO_2 emissions.

The project design calculations presented here consider production of cooling load from geothermal energy in the city of Stargard Szczeciński. 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 absorption 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

2.1 Introduction

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 sorbens in the dehumidifier.

Table 1 introduces current development of cooling technologies, but devices based on Peltier's effect are omitted in this overview. The most descriptive information can be found in a paper written by Rogowska (2002).

TABLE 1: Various known methods of cooling load production
(BHKW Infozentrum Rastatt, 2003)

	Compression chiller	**Absorptive chiller**	**Adsorptive chiller**	**DEC**[1]
Physical cooling -effect	Vaporisation 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 NH_3 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.

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_2O) LiCl	Water (H_2O) Ammonia (NH_3) Water (H_2O)
Soild adsorbents	$CaCl_2$ Active coal Zeolites, silica-gel	Ammonia (NH_3) Ammonia (NH_3) Water (H_2O)

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)

	$LiBr/H_2O$	H_2O/NH_3
Heat source oper. temp. (˚C)	80-110	120-150
Cooling operat. 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. Operating pressure is high; 4. Suitable for using as heat pump due to wide operating range.

The diagrams in Figure 2 and Appendix I present 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.

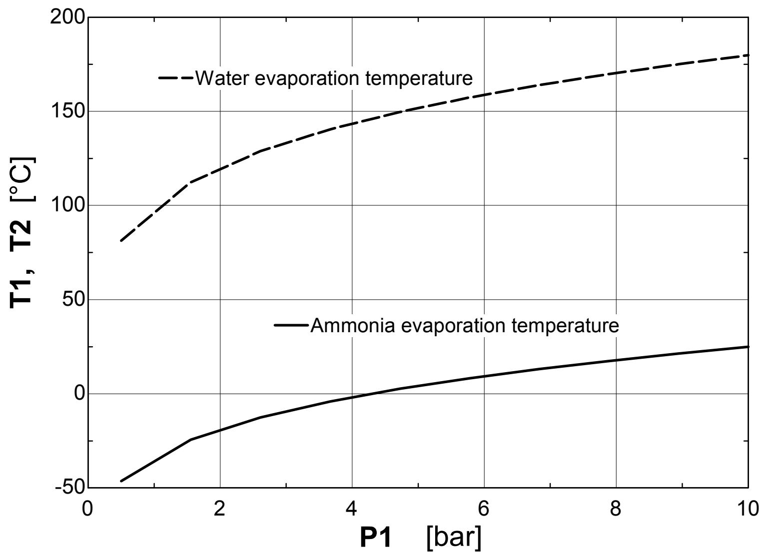

FIGURE 2: Evaporation temperatures for ammonia and water

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.

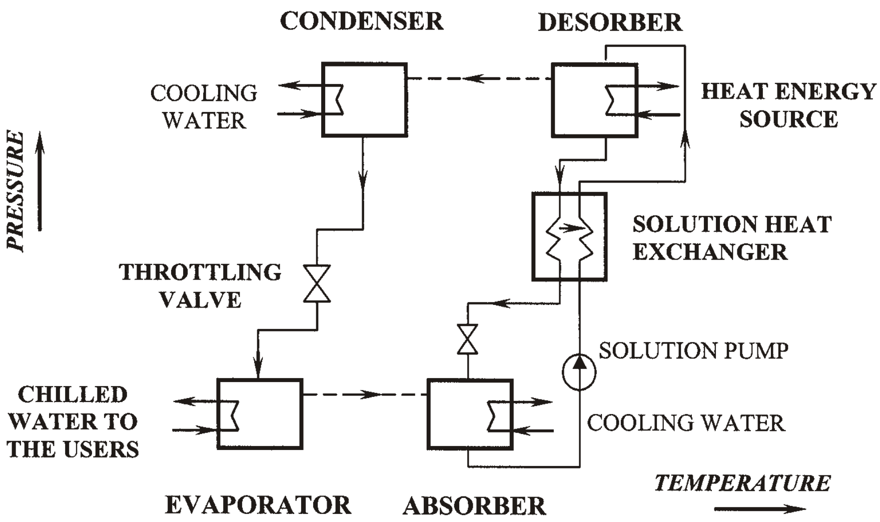

FIGURE 3: Single stage LiBr-water absorption cycle

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 the introduced model for the city of Stargard Szczeciński.

2.3 District cooling

District cooling means the centralised production and distribution of cooling energy. Such systems are under operation in e.g. Japan, USA, Korea, Sweden, and Finland. Chilled water is delivered via an underground pipeline to offices and industrial and residential buildings to cool the indoor air of the buildings. The output of one cooling plant is enough to meet the cooling-energy demand of dozens of buildings. The mission statement can be compared to that of district heating with the exception that in district cooling, the energy produced and supplied to the real estate customer is cooling energy instead of heating energy, e.g. extra heat is removed from the building. Centralised production of cooling energy is more environmentally friendly and cost-effective than distributed building-specific cooling. In addition, centralisation improves operational reliability. More and more buildings require cooling also in the winter, because lighting, computer equipment, and solar heat entering through large windows heat the indoor air. Due to the process cooling required by shop cooling equipment, the demand for district cooling stays even throughout the year.

3. INFORMATION ABOUT STARGARD SZCZECIŃSKI CITY

Stargard Szczeciński is one of the oldest cities in West Pomerania, Poland. In terms of its size, population, and economic potential, Stargard Szczeciński is the third largest city in West Pomerania. The city is spread over 4,810 hectares of land and has a population of 74,000. It is located on the Ina River, 40 km from the Polish-German border, 36 km from Szczecin (the largest city and the capital of West Pomerania), 180 km from Berlin, and 120 km from the Świnoujscie ferry terminal on the Baltic coast. The city lies between two large geographical regions, namely the Szczecin Plain and the Szczecin Lake District, which has a great impact on the various types of landscapes in the surrounding area (City Government Stargard Szczeciński, 2003).

The city enjoys the moderate European climate. The average temperature in the hottest month of the year (July) is 18°C, and the average temperature in the coldest month of the year (January) is -1°C. Most of the days when it is freezing are in December, January, and February. Most of the sunny days are in May, June and September. The number of hot days, when the temperature is above 25˚C, is about 25 a year. The average yearly rainfall is 525-550 mm (City Government Stargard Szczeciński, 2003).

3.1 District heating and plans to develop geothermal energy resources in Stargard Szczeciński

Stargard Szczeciński is one of very few cities that boast a so-called ring district heating system, which in case of a fault in the heat distribution system, enables supply of heat using the longer way round. The district heating company, PEC, whose only shareholder is the city, has a fully automated and computerised central heating plant, and has recently automated its 250 heat distribution centres and installed all the necessary measuring devices in them.

Consistent improvements in the technological processes, extension and modernisation of the district heating system, which is already 40 km long, and strict compliance with the standards laid down in the Energy Act (10 April 1997), all contribute to improving quality parameters and thus have a positive impact on the natural environment. A measurable effect of these activities is the elimination of over 50 heating boilers that used to burn more than twelve thousand tons of solid fuel a year.

In 2001, 45,582 tones of fine coal with an average caloric value of 24.9 MJ/kg, a sulphur content of 0.63%, and a dust content of 16% were burned in the PEC boilers. The yearly emissions were as follows: SO_2 = 466 tons/a, NO_2 = 182 tons/a, CO = 304 tones/a, CO_2 = 98,747 tons/a, B/aP = 36 tons/a, dust 255 tons/a, and soot = 2 tons/a (Kozłowski and Malenta, 2002). 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 (Kozłowski and Malenta, 2002).

3.2 Geothermal conditions and current state of the project

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. The construction of the geothermal heating plant will start in October and is supposed to be finished on 30 April 2004. At that time, technological start-up is being planned. 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 m^3/h. Water temperature is 87°C at its outlet. Construction of the reinjection well GT-2 with directional drilling technology is actually taking place and 70% is already completed. Safety valves have adjustment of 16 bar. Other valuable information about dimensions of the geothermal doublet is shown in Figure 4.

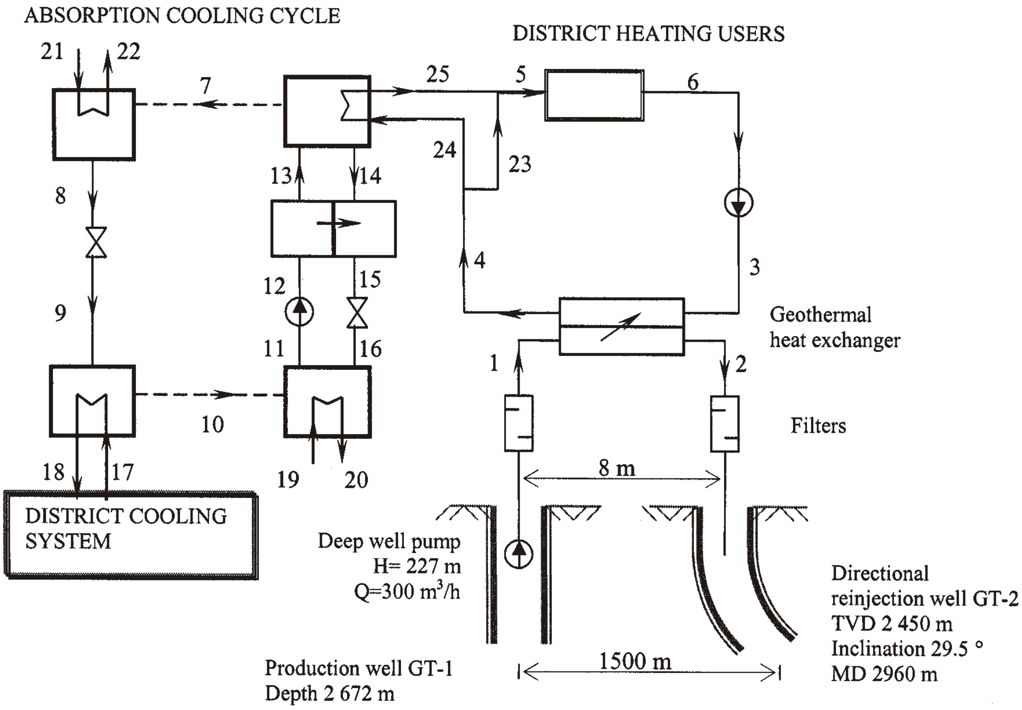

FIGURE 4: Circuit diagram of the LiBr cooling system and the geothermal doublet

The determination of chemical composition and physical analysis were carried out from last stage pumping during the well test. In geothermal water, high content of chlorine, 68,000 mg Cl/dm^3, is observed. Other parameters are dissolved oxygen 3.08 mg O_2/dm^3, specific conductivity 165.6 mS/dm^3, oxidation 26 mg O_2/dm^3, CO_2 aggressive 0 mg CO_2/dm^3, free CO_2 151.8 mg CO_2/dm^3, sulphate 1384 mg SO_4/dm^3 and pH 6.1. The geothermal water from Szczecin Basin is very aggressive and corrosive. This requires mechanistic studies of how water chemistry can effect the corrosion rates and forms of corrosion.

3.3 Heat demand analysis and cooling load requirements

The heating plant supplies heat to a distribution network. Heat demand profile for the city of Stargard Szczeciński is presented in Figure 5, and it is based on data from PEC from the year 2002 (Geotermia Stargard, GS/D/85/2003). The whole annual heating power demand for the users was 10167 MWt. On the basis of the maximum value of volumetric flow (300 m^3/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

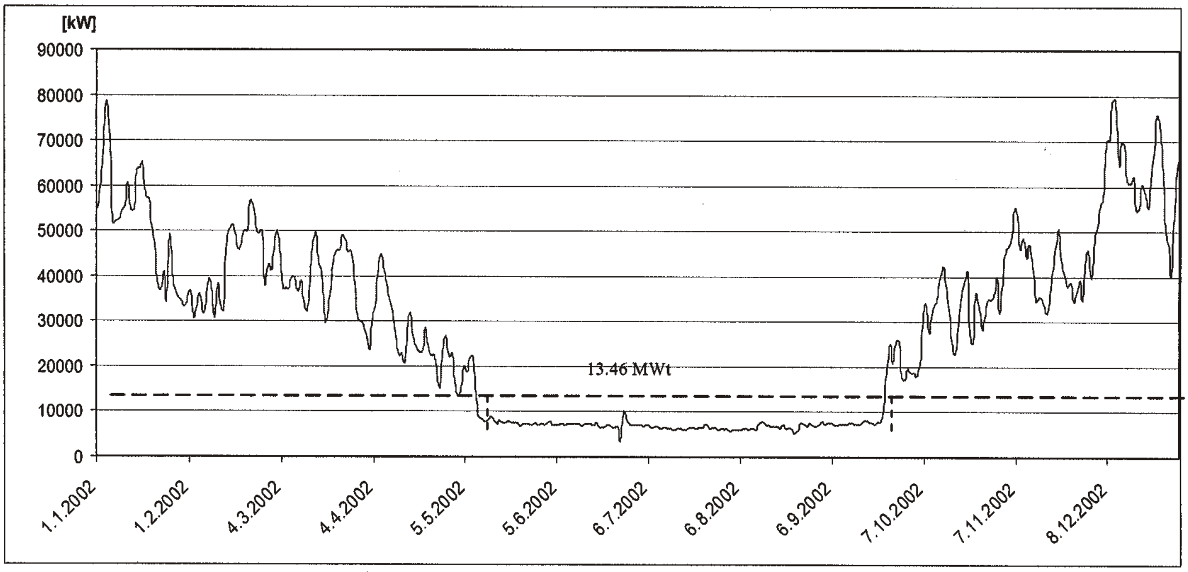

FIGURE 5: Total heat power demand in Stargard Szczeciński in 2002

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 an absorption cooling system.

During the summer time, the heating plant in Stargard Szczeciński operates with one boiler WR10 with the maximum heating power of 10 MW. The capacity of the geothermal heat source would be absolutely sufficient to cover the tap water demand.

The decision on heat management and the development of a district cooling network is strongly dependent on the actual cooling load requirements and future prospects for district cooling based on absorption technology. During the winter, heat must be delivered and in the summer time the heat gains must be removed to ensure thermal comfort. To determine both heating and cooling load amounts, the valid norms in each country must be followed. Calculations are based on energy balance according to ambient conditions (temperature and humidity) given in PN-76/B-03420 and data for internal space in a building or object, depending on its functions and destination from PN-78/B-03421. Stargard Szczeciński is placed in climatic zone II in the winter period, and in zone I during the summer. Mentioned norms are not sufficient for proper determination of cooling loads, because of missing 24 hours temperature distribution. The meteorological data should be prepared like it is done in German DIN 4710 (Zakrzewski, 2002). In the German engineering publication VDI 2078 "Kühllastregeln", the cooling load calculations are presented very exactly. To estimate the cooling load requirements for air conditioning systems, simplified methods according to the following values from Table 4 can be used.

CEN European committee for standardization promotes standardization in relation to environmental issues and strengthens the integration of environmental aspects in the framing of industrial standards to remove technical barriers. Research in Europe will be based on the common norms for main sectors. European standards are currently covered by technical committees (TC). Work programmes (work started, drafts issued) are useful during the design phase of the district cooling system, specially on user's cooling needs (CEN European Committee for Standardization, 2003). The following summarizes the status:

TABLE 4: Cooling load demand in public sector (Schlappmann, 2000)

Destination	Cooling load of the floor area (W/m^2)		Space/person (m^2)		Cooling load/person (W)	
	Min	**Max**	**Min**	**Max**	**Min**	**Max**
Apartments, hotels	41	31	9.2	30	1570	2560
Libraries	94	236	3.7	7.4	420	1400
Big rooms, banks	110	236	3.7	7.4	480	1400
Shopping centres underground	76	122	1.8	2.8	230	440
Shopping centres first floor	81	188	1.5	4.1	280	510
Shopping centres next floors	76	122	3.6	6.7	370	810
Office buildings	72	163	7.5	12	720	1400
Individual offices	105	202	4.5	11.8	700	1630
Dinning rooms	283	490	1.2	1.6	430	790
Conference rooms	185	206	0.56	0.8	185	210

CEN/TC 89 N 602 Buildings - Calculation of cooling load and energy needs for cooling - Part 1: Cooling load calculation; under development;
CEN/TC 89 N 742 Energy calculation - Thermal performance of buildings - Calculation of energy needs for cooling; under development;
prEN ISO 15927-4 Hygrothermal performance of buildings - Calculation and presentation of climatic data - Part 4: Data for assessing the annual energy for heating and cooling (ISO/DIS 15927-4:2003); under approval;
CEN/TC 156 Ventilation for buildings - Cooling load; under development.

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. 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 4, 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.

4.1 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 - H_2O absorption system is presented, and all calculations done in reference to the numbering principle illustrated in Figure 4. 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 4. The main processes and assumptions for the cooling circuit are summarized in Table 5.

TABLE 5: Description of main thermodynamic state points (points refer to Figure 4)

Point	State	Details and assumptions
7	Superheated vapour	Water vapour leaving desorber has no salt content (pure water)
8	Saturated liquid water	Vapour quality set to 0 after condensation
9	Vapour-liquid refrigerant state	Vapour flashes as 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 paper, mass flow, $\dot{m}$ and energy flux, $\dot{Q}$, respectively, were used.

The energy balance in the evaporator can be written as (nomenclature is given at the end of the report):

$$Q_e = m_{10} \cdot h_{10} - m_9 \cdot h_9 \qquad m_{10} = m_9 \;; \text{ where} \tag{1}$$

where Q_e is the cooling load and chilled water will be delivered to the users by the external loop with mass flow $\dot{m}_{17}$

Since the value of m_{10} is known, m_{11} can be calculated from the mass balance in the absorber, and with the assumptions that:

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

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

During the calculations, it is possible to set the lithium bromide mass fractions in the solution, x_{11} and 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 the crystallisation line. If the range of mass fraction of LiBr changes from 50% at the absorber outlet to the solution heat exchanger (Point 11) to 62% at the inlet to the absorber (Points 10 and 16), then the calculated operating conditions avoid crystallisation danger.

The energy balance equation on the absorber is:

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

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

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

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

The energy balance equations for the solution heat exchanger read:

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

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

It is known that heat streams on both sides are equal if an adiabatic shell is assumed.

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

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

$$\varepsilon = \frac{T_{14} - T_{15}}{T_{14} - T_{12}} \tag{8}$$

These parameters influence the COP of the system on a large scale. The higher the effectiveness, the higher the COP. The absence of a solution heat exchanger in the cycle is significant, and causes higher COP, because the heat of the return solution from the desorber can be recovered to heat up the LiBr-water mixture before entering the generator. For this component, the energy balance equation would be:

$$Q_d = m_7 \cdot h_7 + m_{14} \cdot h_{14} - m_{13} \cdot h_{13} \tag{9}$$

Assuming purity of the water vapour leaving the desorber, the following equation is the result:

$$m_{13} \cdot x_{13} = m_{14} \cdot x_{14} \tag{10}$$

Overall, the mass balance on the desorber is expressed as:

$$m_{13} = m_{14} + m_7 \tag{11}$$

Another mass flow parameter, the solution circulation ratio *f,* presents the relationship between the mass flow rate through the solution pump and the vapour flow rate leaving the generator, as follows:

$$f = \frac{m_3}{m_7} \tag{12}$$

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

$$Q_c = m_7 \cdot h_7 - m_8 \cdot h_8 , \qquad \text{and} \qquad m_7 = m_8 \tag{13}$$

A heat transfer model of each heat exchanger was added to the system to provide better understanding and bring the model closer to realistic conditions. All input and output values from the heat exchanger model can be followed, and in the future different and improved heat exchanger models can be considered. In this paper, the *UA* type heat exchanger model was used to specify the size and performance of the heat exchanger as a function of the *UA* value and the logarithmic mean temperature difference, defined as:

$$\Delta T_{lm} = \frac{\left(T_{h,1} - T_{c,1}\right) - \left(T_{h,2} - T_{c,2}\right)}{\ln \dfrac{\left(T_{h,1} - T_{c,1}\right)}{\left(T_{h,2} - T_{c,2}\right)}} \tag{14}$$

where h and c refer to the hot and cold sides of the heat exchanger;
1 and *2* refer to the sides of the heat exchanger; and
ΔT_{lm} describes the potential of heat transfer between the hot and cold sides.

The product of the overall heat transfer coefficient, U, and the heat exchanger area, A, are very useful for heat exchanger calculations, and can be found in the literature from Holman (2002). The amount of the exchanged heat can then be calculated from the formula:

$$\dot{Q} = UA \cdot \Delta T_{lm} \tag{15}$$

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

$$COP = \frac{Q_e}{Q_d} = \frac{m_{17}(h_{17} - h_{18})}{m_{24}(h_{24} - h_{25})} \tag{16}$$

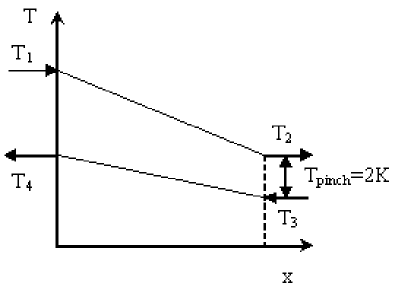

FIGURE 6: Characteristic fluid temperature variation for a counter-flow heat exchanger

This is the main parameter for estimating the efficiency of the operating absorption system. The easiest interpretation formulates it as the ratio of useful energy output (cooling capacity obtained at the 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. The temperature pattern in the counter-flow exchanger is illustrated in Figure 6.

With T_{pinch}= 2 K, the return temperature from the district heating system T_3 can be determined as:

$$T_3 = T_2 - T_{pinch} \tag{17}$$

The heat fluxes on the geothermal and municipal network sides are calculated as:

$$\dot{Q}_{ghx} = \dot{m}_1 \cdot (h_1 - h_2) \tag{18a}$$

$$\dot{Q}_{net} = \dot{m}_4 \cdot (h_4 - h_3) \tag{18b}$$

The value of h_4 is obtained from the energy balance in the heat exchanger. If there is no external work done, there is no heat transfer to the system. The heat exchanger is well insulated and changes in kinetic and potential energy are negligible, so the steady-flow energy equation can be written as:

$$\dot{m}_4 \cdot (h_4 - h_3) = \dot{m}_1 \cdot (h_1 - h_2) = \dot{Q} \tag{19}$$

where $\dot{Q}$ is the heat transferred from the hot stream to the cold stream.

From the known enthalpy h_4, and pressure P_4, the temperature of supplied water to the system T_4 is found. The mean logarithmic temperature difference for the counter-flow heat exchanger was obtained from Equation 14, and is introduced for the given streams 1, 2, 3, 4, as:

$$\Delta T_{ghx} = \frac{(T_1 - T_4) - (T_2 - T_3)}{\ln \dfrac{(T_1 - T_4)}{(T_2 - T_3)}} \tag{20}$$

and used to calculate the UA_{ghx} value of the heat exchanger by using the following relationship:

$$\dot{Q}_{ghx} = UA_{ghx} \cdot \Delta T_{ghx} \tag{21}$$

With the heating load $\dot{Q}_{ghx}$ and the UA_{ghx} product known , the counter-flow geothermal heat exchanger can be selected.

For the purposes of the calculations, the mass flow $\dot{m}_4$ in the municipal network water is used as an input number as well as the mass flow $\dot{m}_{24}$ in the external loop. Connection with the municipal heating network system is available through the external circuit to the generator. To avoid the situation that all the heating medium will have to be delivered to the cooling system, as the first user, a by-pass in point 25 was designed. Such a solution allows for control of the mass flow $\dot{m}_{24}$ in point 5 where mixing of two heat fluxes takes place, with the heat energy guided to the district heating system. The energy balance helps obtain the amount of heat energy delivered to users:

$$Q_{ghx} - Q_d = Q_{user} \tag{22}$$

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

4.2 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-H_2O 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).

4.3 Calculation results

To find suitable operating conditions for specific applications, a sensitivity calculation was performed utilizing EES, and following all assumptions and sequences of equations described in Section 4.1 on thermodynamic equations, and including mathematical correlations for the fluid properties.

The geothermal doublet, heat exchanger, and connection to the cooling system and municipal district heating network, were presented in a simplified way in Figure 5. Following the numerical sequence used in this scheme, results of the calculations are presented in Tables 6 and 7. 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 are introduced:

TABLE 6: Operating parameters of a system with a 10 kW LiBr-water absorption chiller - variant A

	1 h_i [kJ/kg]	2 m_i [kg/s]	3 P_i [kPa]	4 T_i [°C]	5 x_i	6 q_i
[1]	365.4	80	1500	87		
[2]	173	80	1500	41		
[3]	163.9	80	700	39		
[4]	356.4	80	700	84.99		
[5]	356.2	80	700	84.95		
[6]	163.9	80	350	39.07		
[7]	2650	0.004214	5	80.23		
[8]	137.7	0.004214	5	32.88		
[9]	137.7	0.004214	0.9	5.446		0.04616
[10]	2511	0.004214	0.9	5.446		
[11]	81.68	0.03732	0.9	34.21	55	
[12]	81.68	0.03732	5	34.21	55	
[13]	124.7	0.03732	5	55	55	
[14]	202.3	0.03311	5	80.23	62	
[15]	153.9	0.03311	5	54.34	62	
[16]	153.9	0.03311	0.9	47.92	62	
[17]	41.99	0.5	6	10		
[18]	21.99	0.5	6	5.23		
[19]	104.8	0.2515	60	25		
[20]	155	0.2515	60	37		
[21]	104.8	0.2637	6	25		
[22]	144.9	0.2637	6	34.6		
[23]	356.4	79.5	700	84.99		
[24]	356.4	0.5	700	84.99		
[25]	329.9	0.5	700	78.69		

A. Evaporator absorber pressure P_{10}= 0.9 kPa; generator condenser pressure P_{14}= 5 kPa; solution heat exchanger exit temperature T_{13} = 55˚C; generator exit LiBr concentration x_{14} = 62%.

B. Evaporator absorber pressure P_{10}= 0.7 kPa; generator, condenser pressure P_{14}= 7.347 kPa; solution heat exchanger exit temperature T_{13} = 55˚C; generator exit LiBr concentration x_{14} = 60%.

TABLE 7: Operating parameters of a system with a 10 kW LiBr-water absorption chiller - variant B

	1 h_i [kJ/kg]	2 m_i [kg/s]	3 P_i [kPa]	4 T_i [°C]	5 x_i	6 q_i
[1]	365.4	80	1500	87		
[2]	173	80	1500	41		
[3]	163.9	80	700	39		
[4]	356.4	80	700	84.99		
[5]	356.2	80	700	84.95		
[6]	163.9	80	350	39.07		
[7]	2657	0.004279	7.347	83.92		
[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	83.92	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		
[23]	356.4	79	700	84.99		
[24]	356.4	1	700	84.99		
[25]	342	1	700	81.56		

Table 8 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, and return water pump power.

TABLE 8: COP and energy flows at various components in cooling system for both variants

	Variant A	**Variant B**
COP	0.7567	0.6944
Q_d [kW]	13.21	14.4
Q_e [kW]	10	10
Q_a [kW]	12.63]	13.75
Q_c [kW]	10.59	10.65
Q_shx [kW]	1.605	2.635
W_sp [kW]	0.1205	0.2684
Q_ghx [kW]	15395	15395
Q_user [kW]	15382	15381
Wp [kW]	35.26	35.26
ε sxh	0.5625	0.5398

In the cases, A and B, the same cooling load is produced in the evaporator, but in variant A the coefficient of performance is much higher. This means less heating energy is needed to enable a sufficient desorption process in the 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.

4.4 Discussion on the variation in the COP

Figure 7 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.

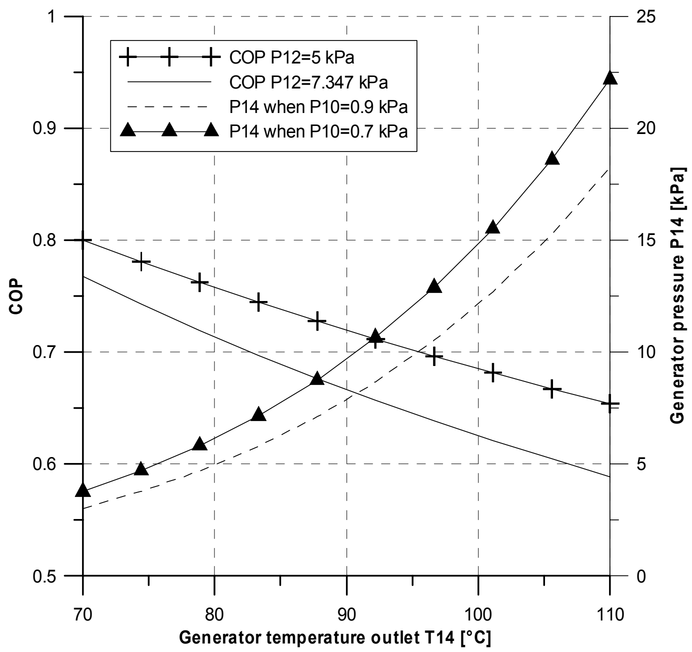

FIGURE 7: Effect of the generator exit temperature T_{14}

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 8, 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.

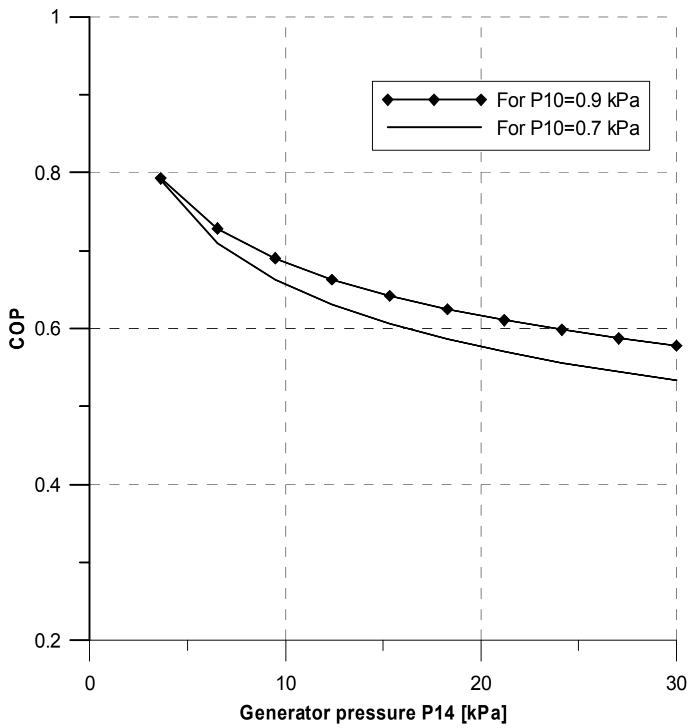

FIGURE 8: Effect of generator pressure, P_{14}

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 9.

Additionally, as seen in Figure 10, 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%.

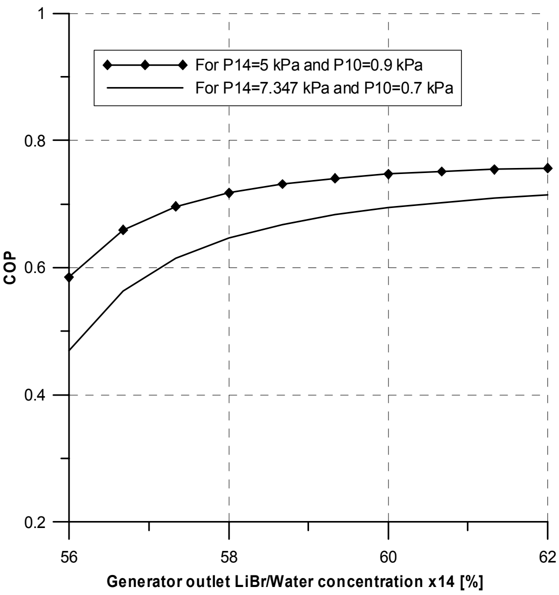

FIGURE 9: Effect of LiBr percentage ratio

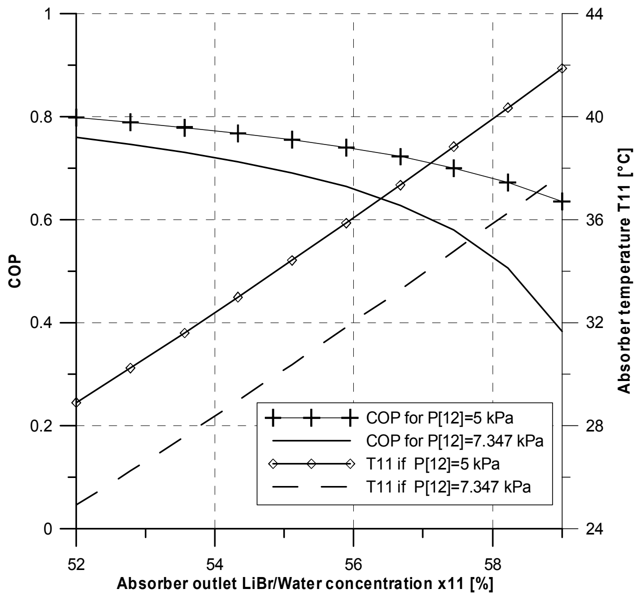

FIGURE 10: Effect of LiBr percentage ratio at absorber exit x_{11}

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.

The generator heat exchanger is a very sensitive point during the designing stage. Thus, the temperature variations at this component cannot be omitted in this study. As can be seen in Figure 11, the higher generator inlet temperature T_{13} improves the COP. 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 12. 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.

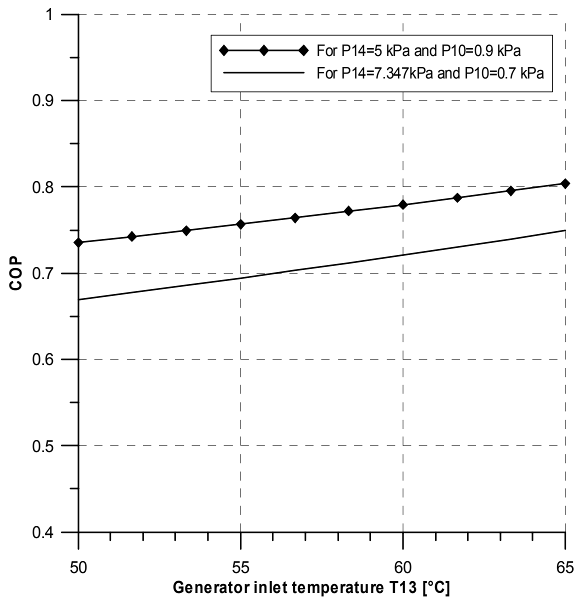

FIGURE 11: Effect of generator inlet temperature T_{13}

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. Figure 11 gives the interpretation of low-temperature cooling water accessibility, as the way to improve the COP value, with COP varying slightly. 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).

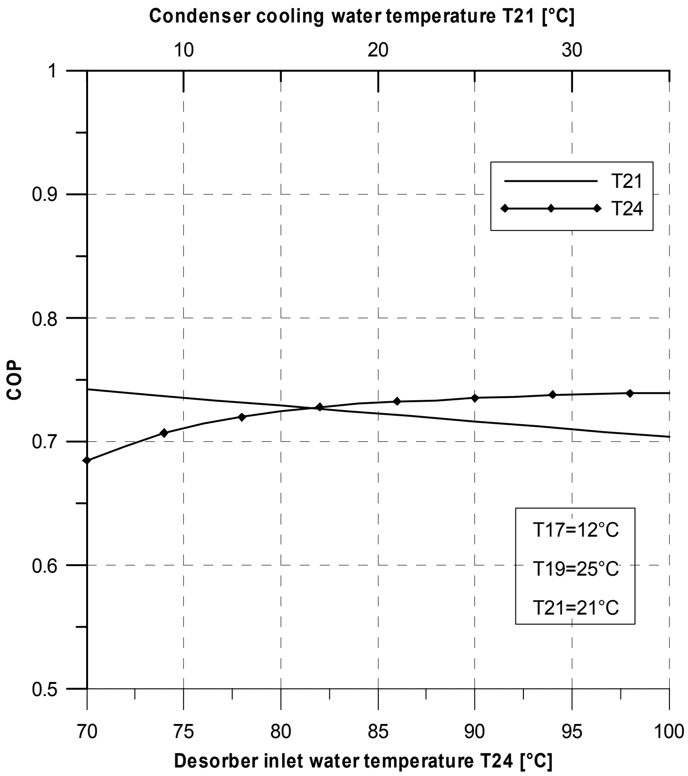

FIGURE 12: Effect of the generator inlet temperature T_{24} and the temperature of cooling source T_2

TABLE 9: Operating parameters of a system with a 500 kW LiBr-water absorption chiller

	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		

4.5 Calculated results for the proposed 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 Szczeciński. Table 9 presents the results of calculations for the main points and follows the numerical sequence in the scheme in Figure 4.

Calculations were based on expected operating conditions. Average mass flow from the production well was assumed to be 70 kg/s (250 m^3/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 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 kg/s and absorber $\dot{m}_{19}$ =13.3 kg/s. Heat fluxes for both components are $\dot{Q}\_c$ = 530.4 kW and $\dot{Q}\_a$ = 667.6 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, ε = 0.5691.

Assumptions made for the 500 kW absorption chiller calculations are introduced in Appendix II, and the results from the EES code for district cooling system are shown in Table 10. They will be used for equipment sizing.

TABLE 10: COP and heat transfer rates for components of the district cooling system with a 500 kW LiBr-water absorption chiller

	Parameters for 500 kW cooling system
COP	0.7164
Q_d [kW]	698
Q_c [kW]	530.4
Q_e [kW]	500
Q_a [kW]	667.6
Q_shx_h [kW]	123.6
Q_ghx [kW]	13471
Q_user [kW]	12773
W_sp [kW]	0.1205
Wp [kW]	30.85
ε_sxh	0.5691

The economic aspects of this system are not evaluated here.

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.

ACKNOWLEDGEMENTS

I would like to thank Dr. Ingvar B. Fridleifsson for giving me the possibility to attend the UNU Geothermal Training Programme. Many warm thanks to Mr. Lúdvík S. Georgsson and Mrs. Gudrún Bjarnadóttir for assistance and help during my stay at Orkustofnun. I am very appreciative to Mr. Páll Valdimarson, for his excellent lectures and sharing his great knowledge. My thanks to Mrs. Fjóla Jónsdóttir for editing my text, Mr. Jón Ágúst Thorsteinsson for an introduction to EES and the critical reading of my report, and Mr. Thorleikur Jóhannesson for efficient consultations.

I wish to extend my warmest gratitude especially to my fiancé Konrad, Mrs. Maria-Victoria Gunnarsson, Dr. Beata Kępińska, and all persons who provided me with help, guidance and advice. I would like to thank Prof. Władysław Nowak and Prof. Władysław Szaflik from the Technical University of Szczecin for giving me permission to participate in this training. Special thanks to Zdzisław Malenta from Eko-Inwest S.A for sharing the data from Geotermia Stargard and support me with information, which were fundamental for my project.

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 / heat transfer rate (kW);
q = Vapour quality;
P = Pressure (kPa);
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 letters
ε = Effectiveness;
ρ = Density (kg/m^3);
ν = 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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APPENDIX I: Properties of common refrigerants

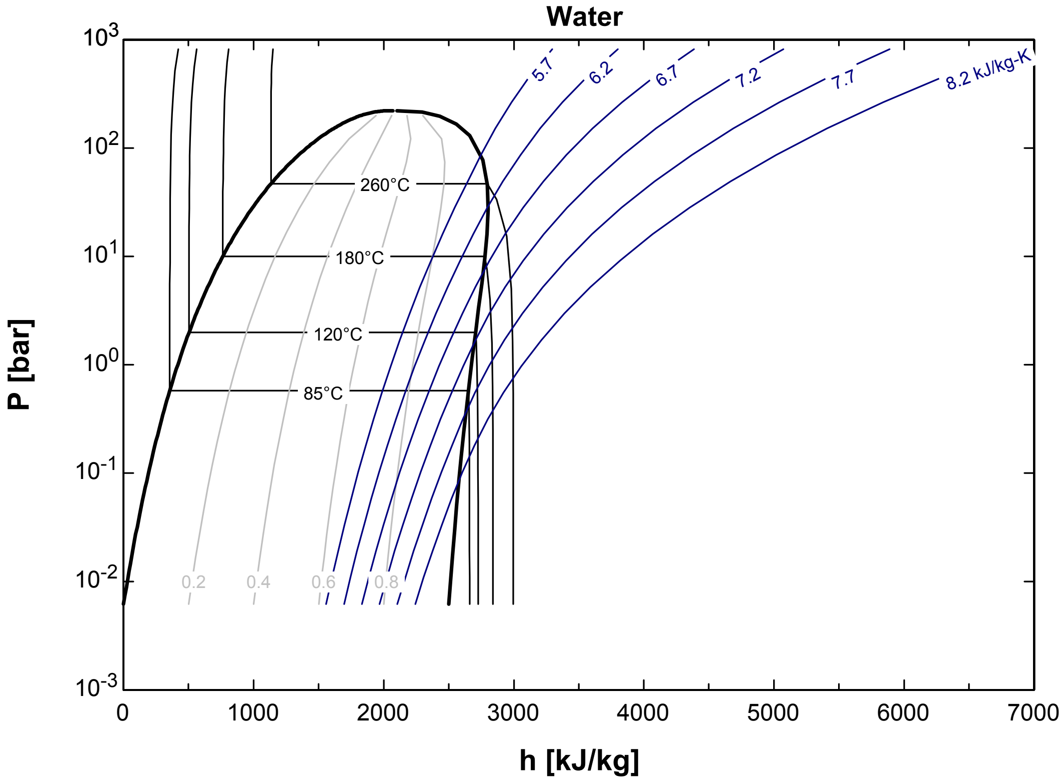

FIGURE 1: Water properties

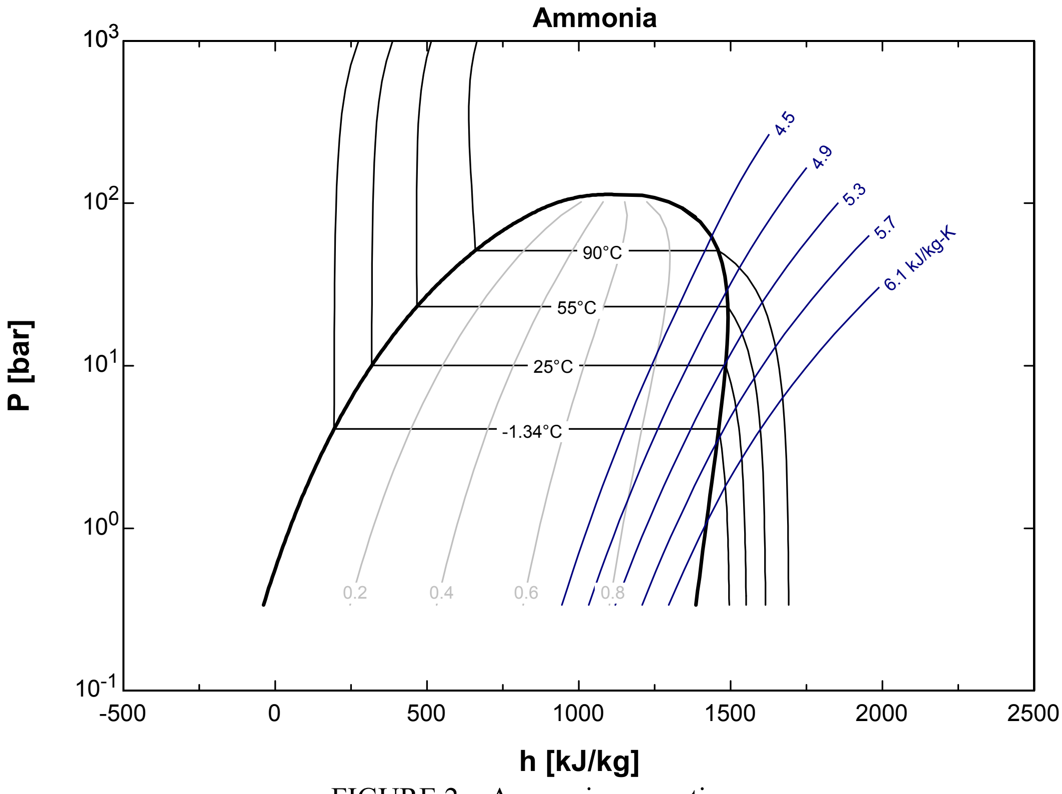

FIGURE 2: Ammonia properties

APPENDIX II: Assumptions for calculations of the district cooling system with 500 kW cooling unit

Assumptions:

"Data for calculations -condenser"

T_pinch_c = 15 [K]
T[21] = 25 [˚C]
T[22] = 35 [˚C]
P[21] = 100 [kPa]

"Data for calculations -evaporator"

T_pinch_e = 3 [K]
P[10] = 0.77 [kPa]
Q_e = 500 [kW]
T[17] = 12 [˚C]
P[17] = 300 [kPa]

"Data for calculations- absorber"

P[19] = 60 [kPa]
T[19] = 25 [°C]
T[20] = 37 [°C]

"Data for calculations -solution pump"

η_p = 0.80
P[12] = 6 [kPa]

"Data for calculations -generator"

P[4] = 700 [kPa]
m[4] = 70 [kg/s]
m[24] = 20 [kg/s]
x[14] = 60 [%LiBr]
T[13] = 55 [°C]

"Data for calculations Users, pump, geothermal heat exchanger"

η_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]