A New Geothermal Cooling – Heating System for Buildings

Geothermal Cooling – Heating for Hot – Humid Climates

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A NEW GEOTHERMAL COOLING – HEATING SYSTEM FOR BUILDINGS

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A 30 credit units Master´s thesis

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ABSTRACT

Single stage absorption chillers (water/lithium bromide) can operate using low grade heat, thus single stage absorption chillers can operate using low temperature geothermal resources. An EES computer model was written for a single stage absorption chiller based on overall heat transfer coefficients, mass flows, salt balances, energy balances and other assumptions. The single stage model was calibrated with well known and available data from US and Japanese manufacturers. The system was considered to work in hot and humid climates like those in Abu Dhabi and Shanghai. The results show that when cooling water is 28 °C hot the initial and annual total cost is high. However, as cooling water temperature decreases, the initial and annual total cost decreases sharply (when cooling water is at 20°C the reduction is around 35% of the total annual cost). The results show a great influence of cold and hot water on the performance and total annual cost of absorption chillers.

The second part of the computer model contains a Total Equivalent Warming Impact (TEWI) analysis. TEWI considers both the direct refrigerant effects and the primary energy impact on equivalent carbon dioxide (CO_2) emissions. Competing technologies are vapour–compression and gas cycles. The results show some recommendations for absorption system diversification to reduce the Total Equivalent Warming Impact. The results also show that absorption hot water chillers have the lowest contribution to Global Warming (TEWI).

PREFACE

Geothermal is a renewable source of energy which holds abundant energy. Single stage absorption chillers (water/lithium bromide) can operate using low grade heat, thus single stage absorption chillers can operate using low temperature geothermal resources. The results show the great influence of cold and hot water on the performance and total annual cost of absorption chillers. The results show some recommendations for absorption system diversification to reduce the Total Equivalent Warming Impact. The results also show that absorption hot water chillers have the lowest contribution to Global Warming (TEWI).

"...If you can wait and not be tired by waiting,
Or being lied about, don't deal in lies,
Or being hated, don't give way to hating,
And yet don't look too good, nor talk too wise:

If you can dream - and not make dreams your master;
If you can think - and not make thoughts your aim;
If you can meet with Triumph and Disaster
And treat those two impostors just the same;..."

"If…" by Rudyard Kipling, it is only part of the whole poem, however even this small part has the potential to be my inspiration and driving force, thus I would like to thank the person who showed me that poem.

Everything should be as simple as it is,
but not simpler.
—Albert Einstein (1879-1955).

A mathematical theory is not to be considered complete
until you have made it so clear that you can explain it
to the first man whom you meet on the street.

—David Hilbert (1862-1943)

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LIST OF SYMBOLS

LiBr	Lithium Bromide
ABS	Absorption System
TEWI	Total Equivalent Warming Impact
COP	Coefficient of Performance [-]
COP_R	Coefficient of Performance (in terms of Resources) [-]
IPLV	Integrated Part Load Value [-]
APLV	Applied Part Load Value [-]
1...20	State points in the ABS model
P	Pressure [kPa]
P_{low}	Low pressure level in the ABS [kPa]
P_{high}	High pressure level in the ABS [kPa]
P_{atm}	Atmospheric pressure [kPa]
$P_{con} = P_{15}$	Inlet pressure of cooling water (condenser) [kPa]
$P_{abs} = P_{13}$	Inlet pressure of cooling water (absorber) [kPa]
$P_{gen} = P_{11}$	Inlet pressure of hot water (generator) [kPa]
$P_{evap} = P_{17}$	Inlet (return) pressure of chilled water (evaporator) [kPa]
$\dot{m}_1 ... \dot{m}_{20}$	Mass flow in state points 1 – 20 in ABS model [kg/s]
$T_1 ... T_{20}$	Temperature in state points 1 – 20 in ABS model [°C]
$h_1 ... h_{20}$	Specific enthalpy in state points 1 – 20 in ABS model [kJ/kg]
q_{wLiBr}	Weak water/lithium bromide solution [%]
q_{sLiBr}	Strong water/lithium bromide solution [%]
η_{asb}	Efficiency of the absorber [-]
η_{shx}	Efficiency of the solution heat exchanger [-]
η_{gen}	Efficiency of the generator [-]
η_{evap}	Efficiency of the evaporator [-]
η_{cond}	Efficiency of the condenser [-]

η_{Carnot}	Thermal efficiency [-]
c_P	Specific heat [kJ/kg°C]
$HVAC$	Heating, Ventilation and Air – Conditioning
$HVAC/R$	Heating, Ventilation and Air – Conditioning / Refrigeration
$ASHRAE$	American Society of Heating, Refrigerating and Air – Conditioning Engineers
DOE	Department of Energy
RHE	Refrigerant Heat Exchanger
GWP	Global Warming Potential
$eflh$	Equivalent full load hours [h]
poh	Plant on hours [h]
D_{GWP}	Direct Global Warming Potential
I_{GWP}	Indirect Global Warming Potential
BTU	British Thermal Unit
CFC	ChloroFluoroCarbon
LNG	Liquefied Natural Gas
LPG	Liquefied Petroleum Gas
$U.S\,RT$	United States Refrigeration Ton
TCE	Ton Coal Equivalent
HFC	HydroFluoroCarbon
$HCFC$	HydroChloroFluoroCarbons
HDR	Heat Dissipation Ratio
POH	Plant On Hours
$EFLH$	Equivalent Full Load Hours
IIR	International Institute of Refrigeration
PLC	Programmable Logic Controller
EES	Engineering Equation Solver

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1 INTRODUCTION

The purpose of this preliminary project is to examine a current single effect absorption chiller unit and determine the circumstances under which the absorption technology is most applicable and most environmentally friendly (beneficial to the local and global environment).

The first phase of the project involves designing a single effect absorption chiller unit. The performance characteristics of the single effect cycle are examined by changing (optimizing) the low enthalpy hot water flow, the cooling water flow, temperature and some flow agreements of the cooling and heat sources. Low temperature water which drives the heat comes from a geothermal production well.

The second phase of the project involves computing a total equivalent warming impact (TEWI). To emphasize great influence on TEWI, the absorption technology is compared to the classical cooling technologies, such as vapour compression cycle, which are widely available.

1.1 Why Absorption Technology

Space cooling may be accomplished from geothermal energy by an absorption cycle. The cooling effect is achieved through the use of two working fluids under two pressure levels, and some certain quantity of heat input, instead of electrical input as in the widespread and more familiar vapour compression cycle. Each of these technologies accomplish the removal of heat through the evaporation of a refrigerant (working fluid) at low pressure and the rejection of gained heat through the condensation of the refrigerant at a higher pressure. The major difference between the absorption and vapour compression cycle is the way in which pressure levels are created. The vapour compression cycle employs a mechanical compressor (electricity is the driving force) to create the pressure differences necessary to circulate the refrigerant. In the absorption system, a secondary fluid or absorbent is used to circulate the refrigerant. In the ABS machines the mechanical compressor is replaced by a thermal compressor (absorber, solution heat exchanger, and generator).

The standard vapour compression cycle contains a condenser, evaporator, throttling (expansion) valve, and a compressor. Figure 1, shows components and flow directions in the standard vapour compression cycle.
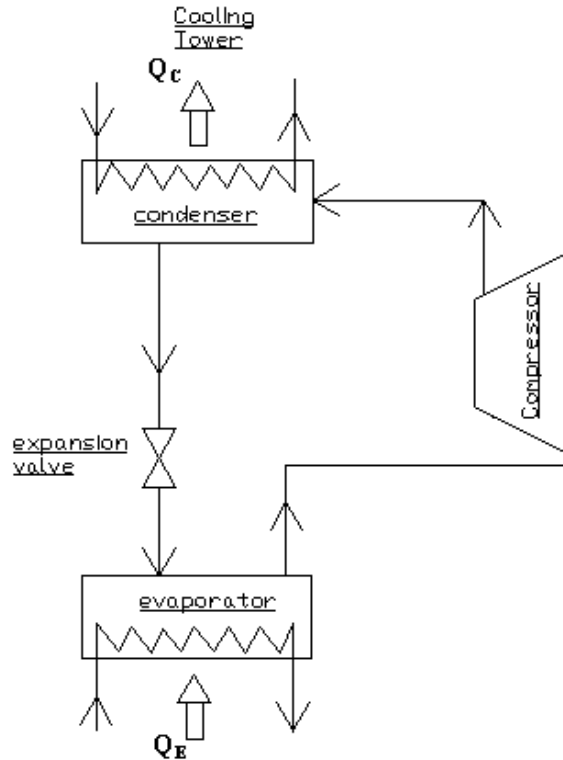
Figure. 1 Vapour - Compression Cooling (Refrigeration) Cycle.

An absorption cooling (refrigeration) cycle contains a generator, an absorber, a solution heat exchanger, an expansion valve and a pump instead of a mechanical compressor.

The generator might be a shell and tube heat exchanger, which sprays ($\underline{3}$) a weak LiBr – H_2O solution above the heating tubes. The main aim of this procedure is to boil off secondary fluid (working fluid - water). The weak LiBr – H_2O solution surrounds a bundle of tubes which carry either steam or hot water (geothermal brine in this case). Hot water is required to boil off the weak LiBr – H_2O solution. The solution boils, sending the refrigerant vapour upward into the condenser ($\underline{7}$) and leaving behind a concentrated lithium bromide solution ($\underline{4}$). The concentrated LiBr – H_2O strong solution moves down to the solution heat exchanger ($\underline{4}$), where it is cooled by the weak solution ($\underline{2} – \underline{3}$). The refrigerant vapour reaches the condenser ($\underline{7}$); the vapour migrates through mist eliminators to the condenser tube bundle. The refrigerant vapour condenses on the tubes. The heat is removed by cooling water ($\underline{15} – \underline{16}$) which moves through the inside of the tubes. As the refrigerant condenses, it collects at the bottom of the condenser ($\underline{8}$). The refrigerant liquid moves down to the evaporator by passing through ($\underline{8}$) the refrigerant expansion valve ($\underline{9}$). The expansion valve is the boundary between the high pressure level (condenser side) and low pressure level (evaporator side). At point $\underline{9}$ the water stream is split into streams $\underline{9a}$ (water) and stream $\underline{9b}$ (steam). Stream $\underline{9}$ is almost sub – cooled water, however it has great influence on the COP of the chiller unit, due to smaller mass flow to the evaporator. In the evaporator ($\underline{9a}$) refrigerant is sprayed over the evaporator tube bundle. Due to the extreme vacuum of the evaporator (e.g. P_{low} = 0,9 kPa, absolute pressure), the refrigerant liquid boils at low temperatures depending on pressure (5,4 °C in the case of 0,9 kPa_{abs}), creating the refrigerant effect. As the refrigerant vapour migrates to the absorber from the evaporator ($\underline{10}$), the strong lithium bromide solution from the solution heat exchanger ($\underline{6}$) is sprayed over the top of the absorber tube bundle. The strong LiBr – H_2O solution plus

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refrigerant vapour, creates the extreme vacuum in the absorber (low pressure side). During the mixing some quantity of heat is generated, heat is removed by the cooling water. In most cases the cooling water comes from a cooling tower. Now diluted – weak LiBr – H_2O solution flows down to the solution pump (1). The cycle is now completed and the whole process begins once again. The figure below shows components and flow directions in the lithium bromide absorption cycle.
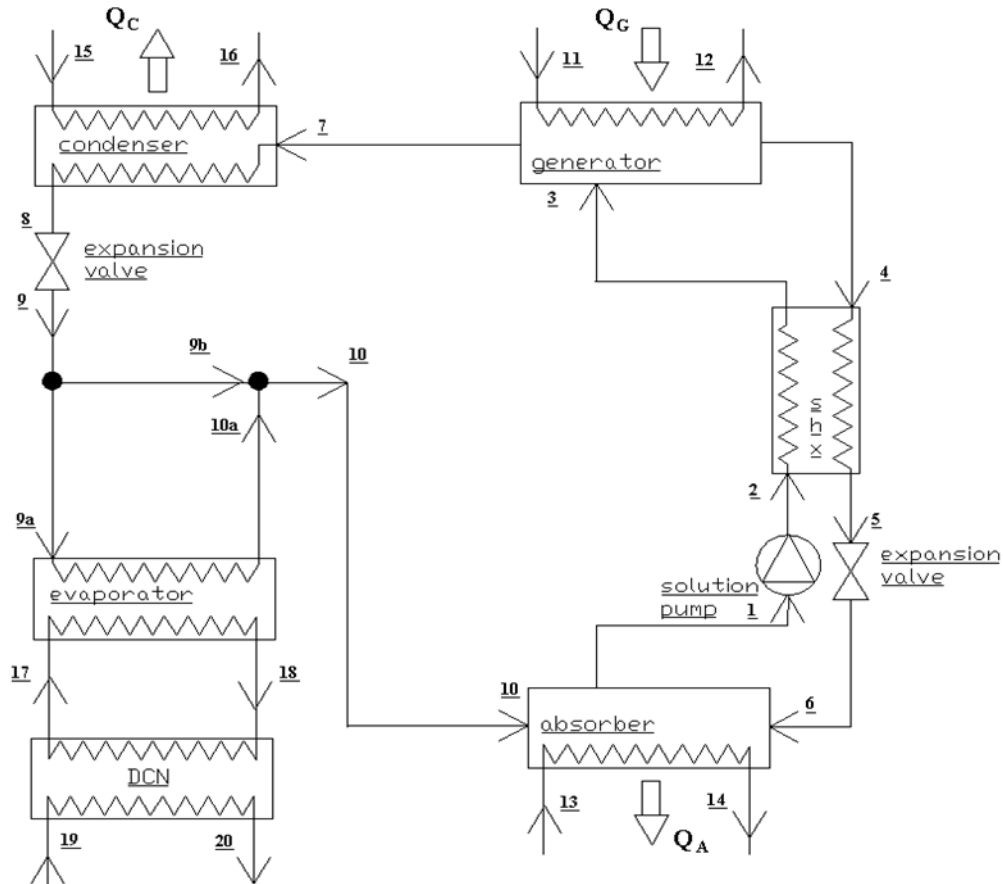
Figure 2 Scheme of the single - stage absorption chiller.

The ability of the absorption unit to work depends on the type of working fluid which is used. The desirable properties of ideal working fluids are high affinity between the absorbent and the refrigerant (LiBr and H_2O), and a low heat of mixing (in the absorber). Lithium bromide has an advantage as an absorbent because it is essentially non – volatile, resulting in the cycle design that avoids the need for a rectifier. Water has an advantage as an absorbent because it does not crystallize.

In an absorption cycle, water vapour from the evaporator (10) is absorbed by the LiBr – H_2O solution in the absorber. Due to that mixing it is possible to pump weak LiBr – H_2O liquid solution instead of vapour (which enters the absorber from the evaporator). The pumping of a liquid solution is much less energy – consuming than the compression of vapour. Absorption technology uses the heat (generator, 11 – 12) as a driving force instead of electrical power. The strong LiBr – H_2O solution under high pressure is sent to the generator (desorber, 3) to boil off the water vapour that was absorbed by the strong LiBr – H_2O solution in the absorber. The figure below shows the simplified scheme of a single – effect absorption chiller.
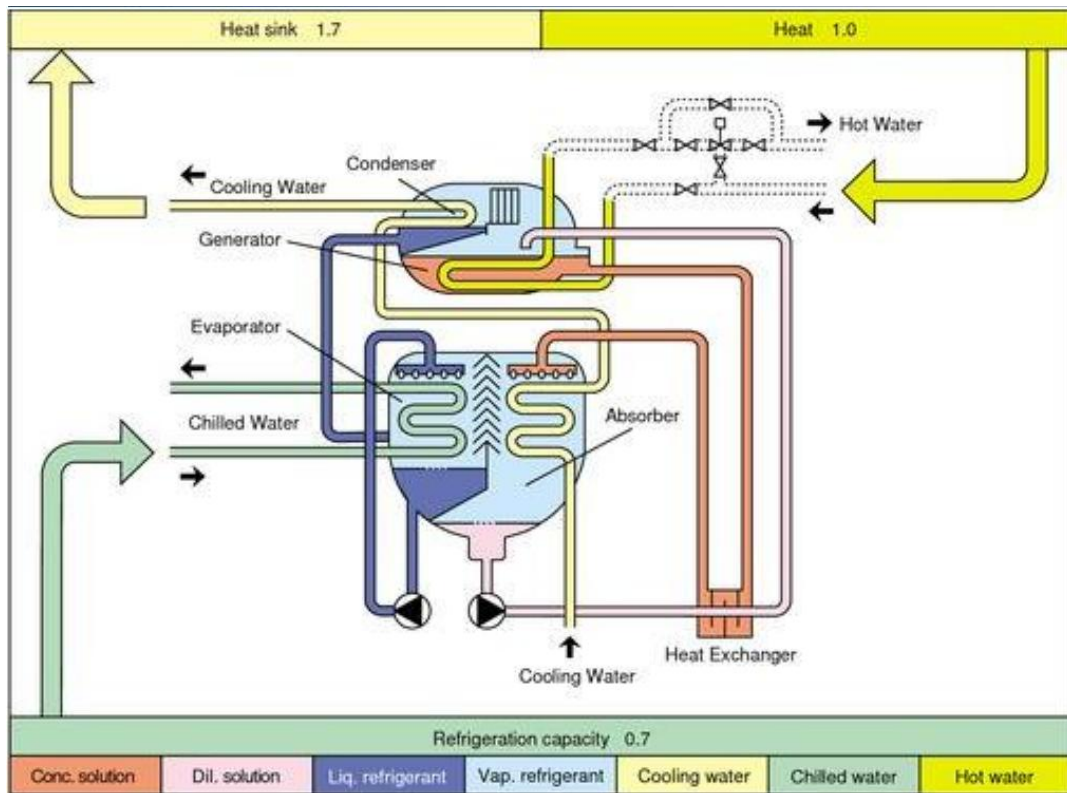
Figure 3 Cooling (Chilling) Cycle Schematic(SANYO Electrical Air Conditioning CO., Ltd,).

The chiller unit is divided into two regions based on pressure level. The high pressure region contains the generator, condenser, and solution heat exchanger. The low pressure area contains an evaporator and absorber. The refrigeration expansion valve, pump and solution expansion valve are boundaries in the chiller unit. The solution heat exchanger has a significant role in the reduction of the quantity of heat input at the generator.

A refrigeration heat exchanger can be used as well, but due to small increases in COP and high increases of initial cost (cost of heat exchanger) this solution is not widely introduced.
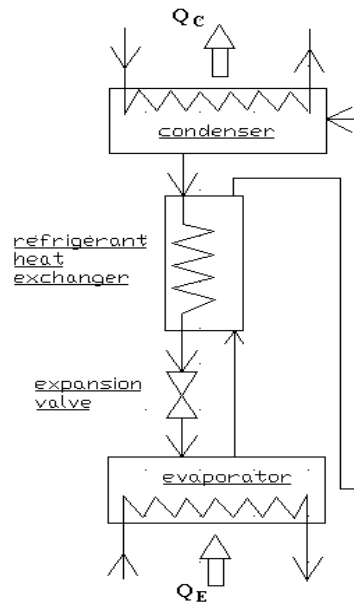
Figure 4 ABS modification – RHE.

Absorption units are widely available for commercial purposes in two basic configurations. For applications above 0 °C, the cycle uses lithium bromide as the absorbent and water as the refrigerant (mostly air conditioning applications). For applications below 0 °C (water – ammonia cycle) ammonia is the refrigerant and water is the absorbent.

The figure below describes the main difference in COP values between single and double stage LiBr absorption chillers.
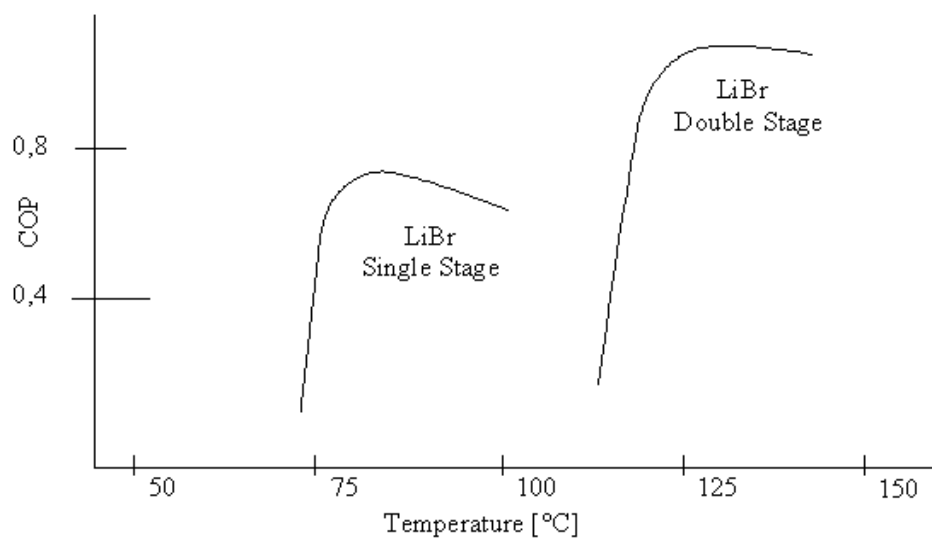
Figure 5 LiBr-H$_2$O single and double effect cycle performance(Mary H. Dickson 2005).

1.2   History and Status of Absorption Technology

Absorption technology with water/lithium bromide solution has been widely used since the 1950s when the technology was developed by several manufacturers in the United States. Absorption machines based on a water/lithium bromide solution are typically configured as water chillers for air-conditioning systems in large buildings. Absorption units are

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available in sizes ranging from 10 (35, 2 kW) to 1500 ton (5, 28 MW). In this case, 1 ton is a unit of refrigeration capacity and is equal to 12 000 BTU/hr = 3,517 kW.

Before the 1970s fossil fuels were cheap and widely available, thus electricity was also inexpensive and energy consumption was not a concern when operating a chiller. During this period an equal number of absorption and electric centrifugal chillers were sold (Plazak, 1996).

In the 1970s, a steady increase in energy costs caused the absorption chiller industry to examine the efficiency of their units. The double and triple effect units were developed to increase the COP, and therefore their share in the market increased. The oil embargo increased the price of fossil fuels more dramatically than electric energy prices and thus overnight killed the domestic absorption market. The absorption market has recovered to its pre 1973 levels (Plazak, 1996). However, the increase of fossil fuel prices is positive from a geothermal energy standpoint because then geothermal energy becomes competitive.
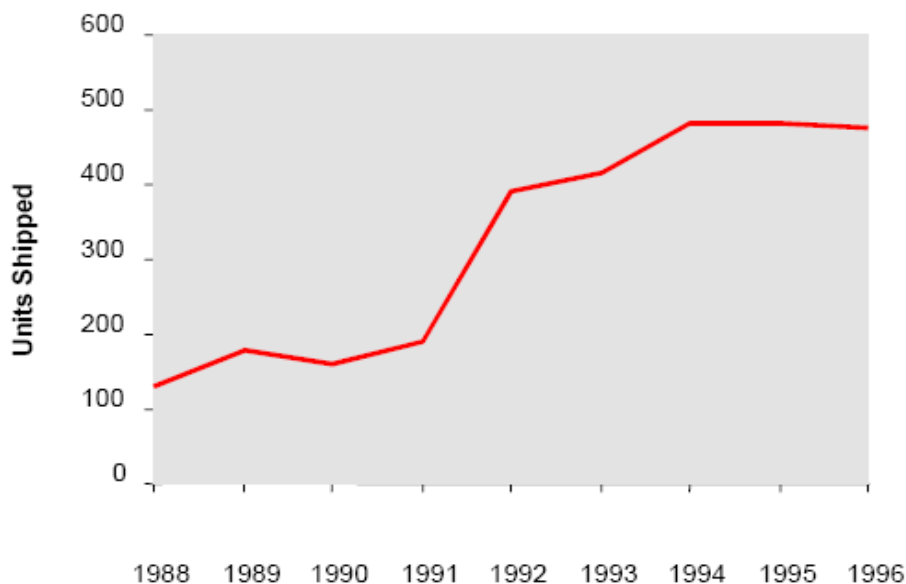
Figure 6 Production of ABS units 1988 – 1996 (ABS guideline, New Building Institute, 1998)

The 1980s saw decreasing prices of natural gas and a steady increase in electrical rates. This resulted in the growth of the newly developed single and multi stage direct-fired absorption units in the middle to late 1980s. A direct-fired unit is one in which natural gas is combusted in the generator instead of using the gas to create steam or hot water. The growth was also a result of the gas and electric utility initiatives, which supported programs for gas cooling in the summer to keep the supply of natural gas constant throughout the whole year. The phase out of CFC based refrigerants used in electric centrifugal chillers also helped boost the direct-fired market (Plazak, 1996).

The Montreal Protocol on Substances that Deplete the Ozone Layer classifies Freon – 11 and Freon – 12 as Annex A substances and bans their production and consumption as of 1996. Although CFC's use is now forbidden, modern compression chillers contribute to Global Warming much more than absorption chillers powered by geothermal brine. Geothermal energy is an environmentally friendly and sustainable energy source.

Absorption technology is currently experiencing a resurgence of interest from end users, who need an independent and clean technology. The current single effect market is relatively small but stable and is used in applications where low grade heat is available. The current market for single stage absorption chillers is around half of the total absorption chillers market. The current size of the absorption market is the result of a decline which was driven by political decisions in the early 1970s (as was mentioned before) to limit usage of natural gas due to expected supply shortfalls, which never happened, but the absorption industry was damaged extensively. The three manufacturers of absorption equipment saw their business dry up and production of absorption machines was reduced to less than 10% of its earlier production levels. Companies were forced to reassign staff and relocate resources. In the meantime, the Japanese absorption industry was developing extremely well due to local domestic demand and product innovation (Keith E. Herold, 1995).

1.3   Applications

Ideal candidates for absorption applications are those where the (electric) peak demand charge is high. Since cooling is generally the primary cause of sharp spikes (peaks) in an industry and building's electric load profile, it seems to be beneficial to investigate alternatives that may reduce this peak. Absorption cooling units powered by geothermal heat may reduce or flatten the electric peaks in a building's electric load characteristic. The absorption cooling application should be used to reduce the electric load into the building.

A hybrid system seems to be an extremely beneficial solution. Hybrid means that most of the cooling demand is covered by an absorption unit, and only some small (sharp) part of cooling demand is covered by an electric centrifugal chiller. This type of application may greatly reduce the total electric load into the cooling facilities. The profitability of such applications strongly depends on the local electricity, heat (geothermal water), and cooling (condenser) water prices. In summary, good applications for absorption chillers have the following characteristics:
-    High energy demand,
-    Simultaneous need for air conditioning and heating (heating and/or tap water),
-    High energy flow into the building,

From an environmental and economical standpoint the beneficial applications for absorption units are when prices of electricity are relatively high, electrical energy load is high and a geothermal (or other renewable) heat source is available. The most suitable places for absorption chillers are:
-    District cooling and heating networks,
-    Schools, hospitals and other public utilities,
-    High residential buildings, hotels

Currently there seems to be a need to develop absorption machines which will be able to work with hot water temperatures in the range of 65 °C – 90 °C (low grade heat sources), therefore the most suitable absorption machines are single and half stage units.

1.4 Standard and Ratings

Currently, (2008) there is no state/federal standard or norm that regulates absorption cooling systems. Therefore, there are several metrics, which are used to define absorption chiller efficiency, like:
- COP
- IPLV
- APLV

Coefficient of Performance – COP

COP is widely used in many types of comparisons, which sometimes might give distorted facts, data or point of view. Therefore it is beneficial to distinguish site COP and resource COP.

Chiller Type	Site COP	Source – to – Site factor	Resource COP$_R$
Electric	2,00 – 6,10	0,27	0,54 – 1,65
Absorption	0,65 – 1,20	0,91	0,59 – 1,10

Table 1 Effectiveness of Chiller Units (ABS guideline, New Building Institute, 1998).

$$COP_{ABS} = \frac{Q_E}{Q_G + P_{work}}$$

(1.4.1)

$$COP_{EL} = \frac{Q_E}{P_{comp}}$$

(1.4.2)

COP$_R$ is equal to the COP of chiller type, multiplied by (Source–to–Site factor) the efficiency of the conversion of primary energy to the secondary energy form (e.g. coal to electricity). Due to greater "Source–to–Site factor value" in the case of the absorption chiller, there is no significant difference in COP$_R$'s of electric and absorption chiller units.

In fact absorption cooling machines do not have high COP values, nevertheless these machines have relatively high Resource COP.

An important number in the case of absorption machines is the coefficient of performance (COP). COP of these units is defined as refrigeration (cooling, chilling) capacity divided by the driving heat input and power required for running the solution pump. The typical range of COP in absorption machines is between 0,7 (single effect) < COP < 1,2 (double stage).

Integrated Part Load Value – IPLV

Integrated Part Load Value (IPLV) is another measurement of chiller efficiency. IPLC is an industry standard for calculation of an annual COP based on a typical load profile and the part load characteristics of chillers. It was originally conceived as a part of ANSI/ASHRAE Standard 90.1 (Standard for Energy Efficient Design of New Non residential and High – Rise Residential Buildings) (ABS guideline, New Building Institute, 1998) in response to a need for directly comparing manufacturers' part load data. The

method assumes that the chiller operates at a specific part load for a specific number of hours during the year. According to the following equation:

$$IPLV = \frac{1}{\dfrac{0,17}{A} + \dfrac{0,39}{B} + \dfrac{0,33}{C} + \dfrac{0,11}{D}} \tag{1.4.3}$$

The figure below provides the proper assumptions and appropriate values for the equation (1.4.3)

Chiller Load (Load factor) [%]	Chiller Water Return Temperature [°C]	Symbol from the equation	Part Load Hours [%]
100	29,44 (85,00 °F)	A	17
75	26,03 (78,85 °F)	B	39
50	22,50 (72,50 °F)	C	33
25	19,03 (66,25 °F)	D	11

Table 2 IPLV Calculation Assumptions (ABS guideline, New Building Institute, 1998).

COP ratings A, B, C and D at each part load condition are obtained from the chiller manufacturer and should be delivered from the actual chiller tests and surveys. Note that the calculation allows for a 1, 36 °C – 1, 41 °C (2, 5 °F) drop in the entering cooling water temperature every 10% reduction in the cooling load. A lower entering cooling water temperature corresponds to part load cooling demand that results from a drop in ambient temperature.

Although IPLV is a useful way to compare different manufacturers' chiller machines, it probably does not represent whole and variable operating conditions. Thus for applications where cooling load is not significantly affected by ambient temperature conditions this estimation should provide reliable results. Chiller performance should be modelled to actual building load profiles tailored to site–specific ambient conditions.

Load factor

Load factor is the ratio of the average load over a designated period of time to the peak load occurring during that period. Ambient temperature and air humidity have great influence on load factor.

The load factor value increases with increasing ambient temperature values. Therefore as the load factor becomes greater, the more economically feasible is the absorption technology application.

Applied Part Load Value – APLV

The Applied Part Load Value, APLV is calculated using the same IPLV formula, except that actual chilled and condenser water temperatures and flow rates are used. The advantage of using the APLV over the IPLV is that this rating gives more accurate approximation of actual operating conditions. The disadvantage is the additional performance data that needs to be collected.

1.5 Manufacturers

Cention

Cention Corporation is a manufacturer of absorption chillers located in Korea with US representation. For over a quarter of a century since its foundation, Cention has been a market leader of the air conditioning equipment industry in Korea by introducing high technology and consistent research & development.

The Cention absorption chiller/heater is driven by LNG, LPG or natural gas, so it gives off no sulfur dioxide or smoke, and very little NOx. The unit can function as both a cooler and a heater, so it takes up less installation area than other separate chilling and heating units. Moreover, the Cention absorption chiller/heater uses about 7% less energy compared to conventional machines, as it uses parallel flow technology, is compact in size, and has a high temperature generator located at the bottom of the unit. The Cention absorption chiller/heater can be adapted to geothermal hot water or steam. However, Cention absorption chiller/heater machines are beyond the scope of this study due to their high temperature generator.

Sanyo

Sanyo is one of the leading brands in the absorption cooling technology. The main focus of the company in business is based on the theme of "Clean Energy". As an example of contributing to the protection of the global environment, SANYO's full line-up of absorption chillers is suited to a wide range of space applications and also creates clean and comfortable living environments. SANYO has successfully developed absorption chillers with capacities of up to 6 600 U.S RT (Refrigeration ton) while at the same time minimizing growth in weight and volume of the unit, thus realizing improvements in energy savings and space efficiency.
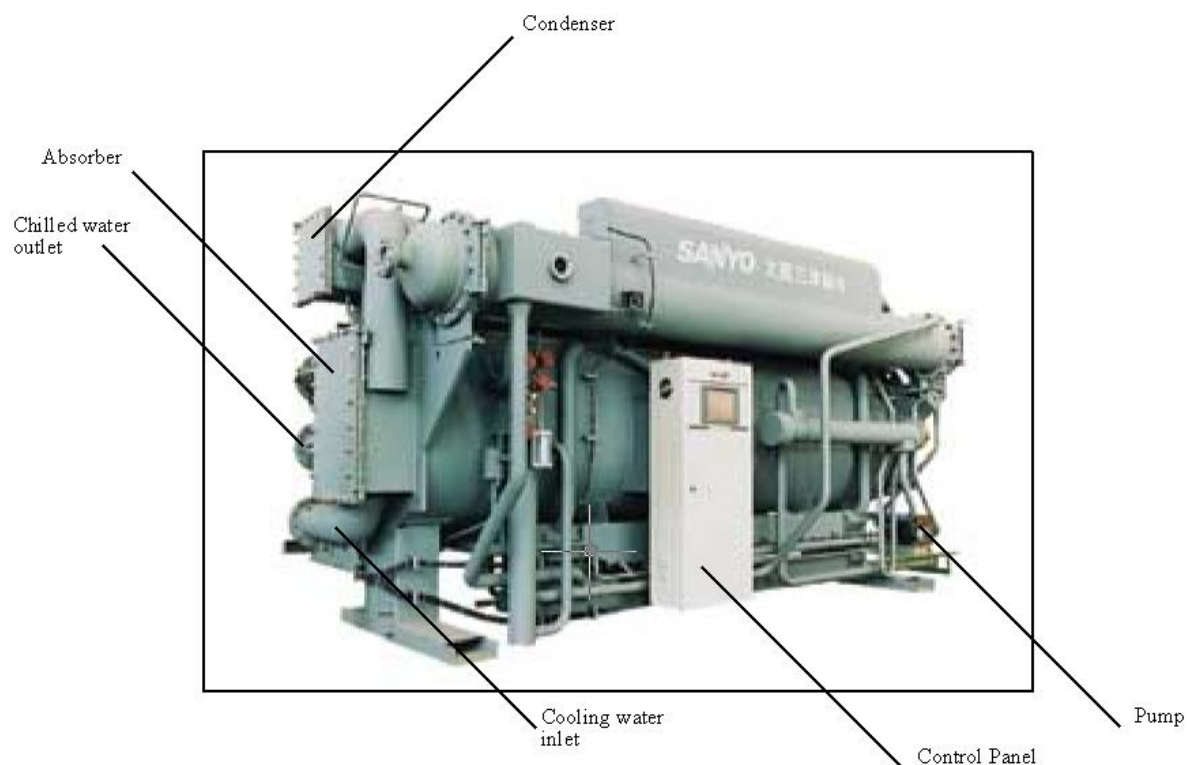
Figure 7 SANYO, single–stage hot water chiller unit (SANYO Electrical Air Conditioning CO., Ltd.)

Yazaki

A Yazaki absorption chiller/heater, using water as the refrigerant, is today's best choice in air conditioning for protecting the environment and reducing the cost of energy. Double-effect cycles and advanced technology ensure high performance and long term reliability. With over 100 000 units operating worldwide, Yazaki is a leading supplier of non-CFC based space cooling. Capacities of 30 through 100 tons are available to either cool or heat installations such as schools, offices, hospitals, industrial facilities, and hotels.

Chillers manufactured by Yakazi are available for purchase globally, but they will only warranty the product in Japan (Rober A. Zogg, 2005). Yakazi ships 2 to 3 units per month within Japan. They have received inquiries from many other countries, especially Europe. They use a new LiBr working solution, which may be their patented $LiBr/LiCl/LiNO_3$ solution.

York

Formerly known as Hitachi, the line is now manufactured in the United States by York International in Huston, Texas. This is the only large tonnage manufacturing plant in the US since 1992. Units up to 600 tons are completely pre–packaged and shipped ready to work. Large units require some on–site assembly. Single stage absorber units are the older design but are smaller and cheaper than two stages. If the heat source is relatively inexpensive and available the single stage makes sense. Single stage units that are used in co–generation and waste heat applications, also might be applicable in geothermal solutions. The size range is 120 – 1 377 tons. York units can cool down chilled water to 7, 22 °C.

2 GEOTHERMAL ENERGY AS A DRIVING FORCE

2.1 Introduction, World Energy Consumption

With the increase in world population, industrialization and improvement in the standard of living, there has been a continuous increase in the consumption of energy. Thus the main aim became to find a cost–effective form of energy that neither affects global warming nor threatens national energy security, therefore geothermal energy has become a significant player.

In recent years, the increase in energy consumption has been at least as intensive as the population explosion. The figure below shows an estimate of the world's energy consumption, growth of population and energy production over past years.
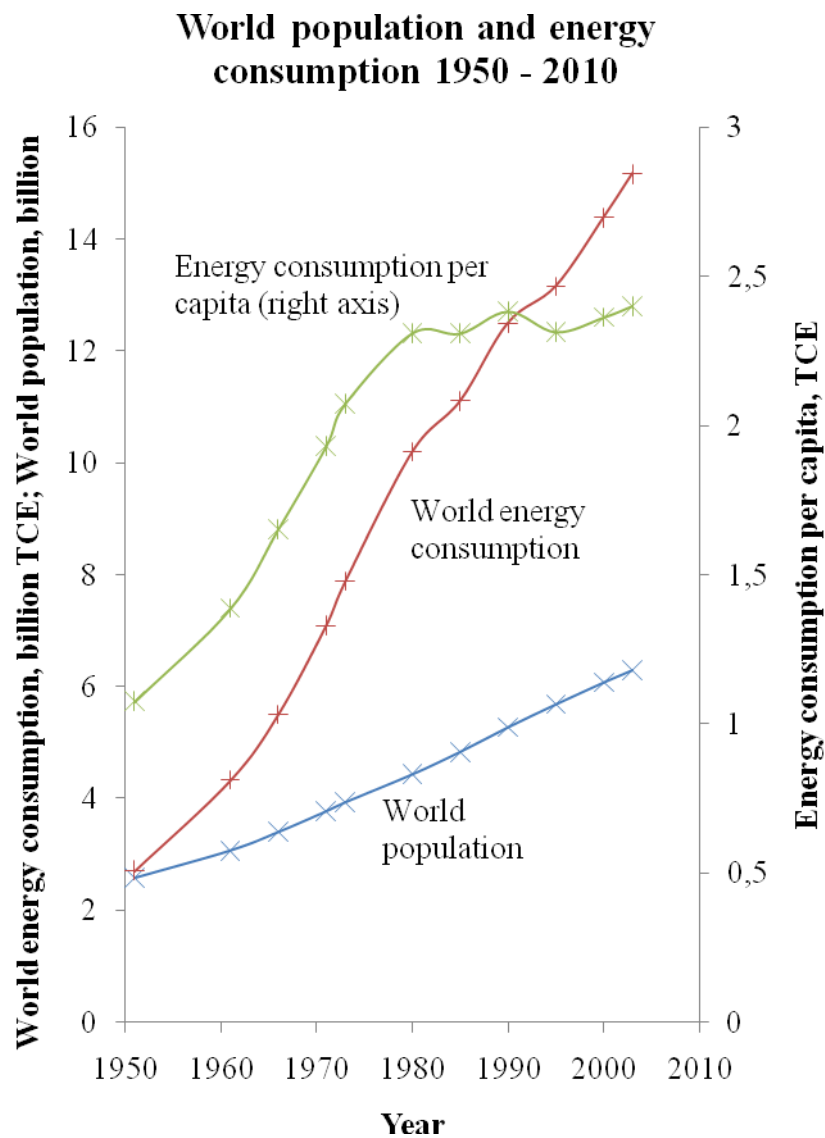
Figure 8 World population and energy consumption (H. Gupta, S. Roy, 2007).

There was a tremendous 50% jump in world energy consumption between the years 1980 – 2003, mostly due to around a 40% increase in world population, as well as due to worldwide improvements to the standard of living in both developing and developed countries. According to papers published by the United States Census Bureau the world population is currently around 6 500 000 000 and it is expected to reach around 10 000 000 000 in 2055(H. Gupta, S. Roy, 2007). A ton of coal equivalent (TCE), is equal to 29, 3076 GJ.

2.2 Geothermal Energy – an Alternative

Geothermal energy, unlike fossil fuels, is considered to be a relatively clean and renewable energy source. It is a relatively diffuse resource – the amount of heat which flows through the earth's surface equals 10^{21} joules/year. For a comparison, amount of heat which reaches the earth from the sun is equal to $5,4\ 10^{24}$ joules/year.

The use of the Earth's heat as an energy source began in the twentieth century when electricity was generated for the first time from geothermal steam at Larderello, Italy in 1904. The technology spread rather slowly around the world during the first half of twentieth century. At the beginning of the second half of twentieth century, interest was developing in other parts of the world, especially in New Zealand, the United States of America, Japan and Iceland, where geothermal power plants were built in 1958, 1960, 1961 and 1969. Electricity production is not the only application for geothermal energy. Hot geothermal water began to be used for large–scale district heating systems in Iceland in 1930. The development of geothermal energy resulted in a worldwide maximum growth rate of ~ 22, 5 % between 1980 and 1990, and ~ 16, 7 % in next decade. Many development efforts in utilizing geothermal heat sources have been made in recent years (H. Gupta, S. Roy, 2007).

Geothermal resources vary widely from one location to another, depending on the temperature and depth of the reservoir, the type of rock and the chemistry and abundance of ground water. Geothermal resources are usually classified into three categories:

- high enthalpy resources (liquid and vapour reservoirs at temp. above 180 – 200 °C),
- medium enthalpy resources (at temperatures around 100 – 180 °C),
- low enthalpy resources (at temperatures below 100 °C),

The term enthalpy is used due to the nature of water, because temperature alone does not describe sufficiently the useful energy content of water and/or a steam.

The wide spectrum of geothermal energy applications is given on the Lindal diagram (Figure 9) below.
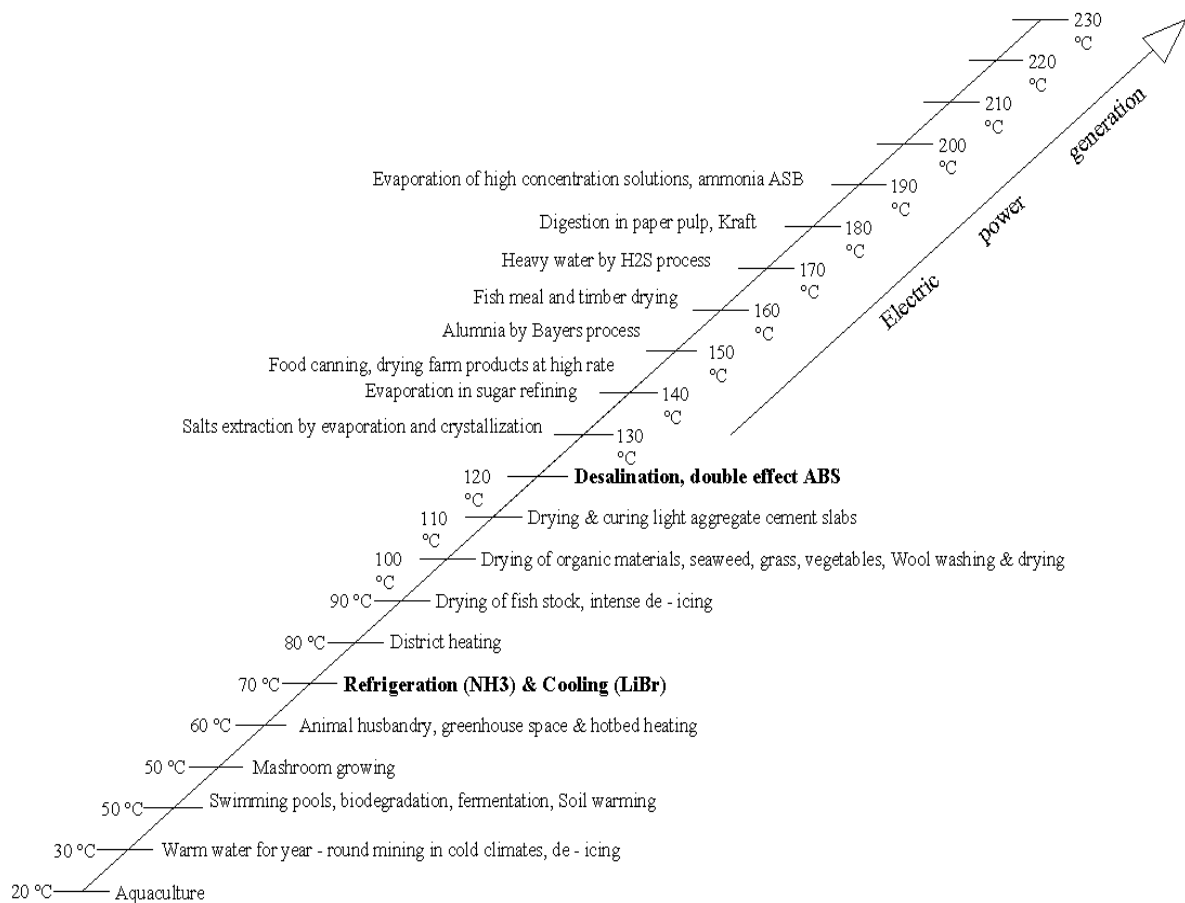
Figure 9   Modified Lindal diagram (R. DiPippo, 2008).

From here on thesis will focus on a low–enthalpy application where hot geothermal brine can be utilized as a driving force in absorption (water/lithium bromide) machines to produce a cooling effect by using a low grade heat source. Moreover, absorption machines do not utilize the whole enthalpy carried by hot water, thus water can be used in district heating systems.

2.3   Types of Geothermal Systems

A geothermal heat source could be simply defined as a reservoir beneath the surface of the Earth. Energy extracted from such a geothermal reservoir can have a competitive price compared with other forms of energy, and could also be useful for electricity generation or other applications. A geothermal reservoir can contain heat in solid rock as well as in the brine that fills the fractures and pore spaces within the rock. The estimation of a geothermal reservoir is made on the basis of geological and geophysical data such as:

- Depth, thickness and extent of geothermal aquifers
- Properties of rock formations
- Salinity and geochemistry of geothermal brine
- Temperature, porosity and permeability of rock formation (Rummel and Kappelmeyer, 1993)

Natural aquifers are often porous rocks that can store brine. Porosity refers to the cavities present in the rock, whereas the ability to transmit water is known as permeability. A

14

geothermal aquifer must be able to sustain a flow of geothermal fluid, so even highly porous rock will only be suitable as a geothermal aquifer if the pores are interconnected.

A geothermal resource is distinct from a geothermal reserve, which refers to the part of a resource that can be extracted economically at present price level. Reserves are confirmed on the basis of detailed reservoir datasets obtained by deep drilling into potential resource areas. Therefore, the main factors in estimating reserves are the costs of drilling and the quality of available data on subsurface rock formations (Harsh Gupta, 2007).

Radioactive decay of long-lived isotopes, particularly those of potassium, uranium and thorium, continuously generates heat within the Earth. The amount of radioactive elements present in the rocks releases enough heat to account for a major portion (typically, 60% for continental crust) of the total heat flow observed on the Earth's surface.

Heat is lost from the Earth's surface at an average rate of $\sim 80mW/m^2$. In most areas, this heat reaches the Earth's surface in a diffuse state, making it uneconomical to exploit this vast heat resource. It is believed that heat transfer below the lithosphere is mostly by convection and in the lithosphere by conduction. Rocks are relatively poor conductors of heat (Harsh Gupta, 2007).

In some specified geological circumstances heat may be locally transferred within a few kilometres of the Earth's surface through the process of convection by magma or molten rocks. The magma has temperatures in the neighbourhood of 1000 °C and interacts with the near-surface rocks, causing surface manifestations of geothermal activity such as the hot springs, geysers and fumaroles. Under certain suitable geological conditions, the heat becomes trapped, forming heat reservoirs. In such areas, after drilling a few hundred meters, temperatures of the order of 200 – 300 °C are found and the regions could be suitable for harnessing the geothermal energy. The major producing geothermal fields of the world exploit such situations.

Geothermal systems can be divided according to the following types:

- vapour dominated,
- hot water,
- geopressured,
- hot dry rock (HDR),
- magma,

Due to the scope of this project only hot water geothermal system will be briefly described.

2.3.1 Hot Water Geothermal Systems

In hot water geothermal fields, water – convection currents carry the heat from the deep source to the shallow reservoir. The bottom of the convective cell may be heated through conduction from hot rocks. The geology of hot geothermal fields is quite similar to that of an ordinary ground water system. Typically hot water geothermal reservoirs vary from 60 – 100 °C and they occur at depths ranging from 1500 – 3000 m.

Within the reservoir, convection currents of hot water are set up providing a good heat exchange, and the temperature difference between the top and bottom of the reservoir is not very significant. A variety of rocks have been found to constitute good reservoirs. At Wairakei (New Zealand), it is pumiceous and breccia and tuff, and at Cerro Prieto (Mexico), it is deltaic sands. A good reservoir could also be formed at geological unconformities and formation boundaries.

An impermeable cap rock or a cap rock with low permeability, overlying the reservoir is necessary to prevent the escape of the hot reservoir fluids through convection. The heat loss through conduction is not prevented by the cap rock. However, the amount of heat conducted is significantly smaller than the amount of heat which could be lost by convection heat and mass transfer.

Depending upon the temperature, chemistry and structure of the reservoir, hot water systems have been divided into several subtypes, which are briefly described below.

- System characterized by low (50 °C) to medium temperatures (150 °C) and producing water with a chemical composition to the regional surface and shallow ground waters,
- Systems characterized by the presence of partly non–meteoric water. Such systems usually occur in deep sedimentary basins,
- Systems characterized by the presence of brine of very high salinity. The chemistry can vary considerably from one field to another,
- Systems characterized by the presence of natural cap rocks. Geothermal fields at Cerro Prieto and Salton Sea (California) have cap rocks constituted by fine–grained, low–permeability sediments,
- Systems characterized by the creation of their own self–sealing cap rocks. These cap rocks are formed through the chemical alteration and deposition of sediments near the surface where the temperature decreases suddenly. Wairakei and Yellowstone Park are typical examples (H. Gupta, S. Roy, 2007),

Figure 10 is the schematic representation of a hot–water field and the reference curve represents the variation of the boiling point of the pure water with depth.
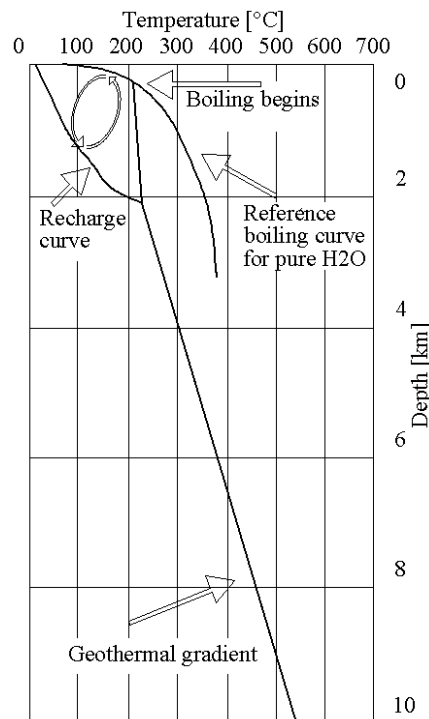
Figure 10 Temperature variations in hot – water geothermal system (H. Gupta, S. Roy, 2007).

2.4   Direct Use of Geothermal Energy

In the direct use sector, the potential of geothermal energy is very large for space cooling and heating, and water heating. In developed countries, around 35% of total primary energy consumption is used in buildings. The European Union's commitment to reduce green house gas emissions by 20% by the year 2020 opens a huge potential for geothermal applications. However, the largest potential happens to be in China. Owing to geological conditions, there are widespread low–temperature geothermal resources in many provinces of China. Geothermal resources are already widely used in China for space heating and cooling.

Country	China	Sweden	US	Turkey	Iceland	Japan	World Total
Direct Use (2005)	12 604,6	10 000,8	8 678,2	6 900,5	6 806,1	2 861,6	75 924,9
Unit	GWh Thermal						

Table 3 Direct use of geothermal energy (China Renewable Energy and Sustainable Development Report, 2008)

In early January 2008 the 3rd Geothermal Heat Pump Technology Utilization Conference was held in Beijing. At the conference it was learned that through September 2007 the Beijing area already had constructed 497 water source geothermal heat pump projects, which cumulatively have an installed coverage area of 10,52 million square meters of heating capacity, moreover these projects are growing at a pace of 150%/year (China Renewable Energy and Sustainable Development Report, 2008).

2.5   Electricity Production

Following the necessary exploratory works, production wells are drilled into a geothermal reservoir. Typically, in a large scale commercial operation, a production well could be as deep as 3,000 m or more. The production well enables the highly pressurised hot water to rise up from the reservoir under its own force (in many cases water has to be pumped to the surface). Depending on the type of power station, steam and/or hot water is then piped from the production well directly to the power station. The temperature and conditions of the geothermal reservoir determine which type of electricity generation technology is used. There are three common types of geothermal electricity generation technology:

Dry steam power plants are fed directly from geothermal steam reservoirs with temperatures of more than 220 °C. In this system, steam that has risen to the surface is piped straight to the power plant where it is then directed into a turbine, spinning the generator and therefore producing electricity.

Flash steam power plants use a geothermal reservoir with water of temperatures greater than 180 °C. This very hot water flows up through wells in the ground under its own pressure. As the water flows upward the pressure decreases and some of the hot water flashes (i.e. boils) to steam. The flashed steam is then separated from the remaining water and fed directly into a turbine that is connected via a shaft to the generator to generate

electricity. It is also possible to flash the remaining hot water to produce low pressure steam (dual flash) which may also be used in electricity production or process heating.

Binary cycle power plants can operate on water and/or steam over a wide range of temperatures (typically 110 to 180 °C) and geothermal reservoir conditions. In some binary cycle stations hot water delivered via its production well(s) is first flashed into steam to rotate a primary turbine/generator unit. Following this initial generation process, the heat from the separated hot water and/or low pressure steam is used to boil a secondary fluid, which is usually a hydrocarbon fluid (e.g. liquid pentane) with a lower boiling point. The secondary fluid is vaporised in a heat exchanger and the vapour is used to turn a secondary (binary) turbine to generate electricity. The vapour is then condensed and pumped back to the heat exchanger in a closed cycle.

Typically, the cooled geothermal water that is not evaporated through the cooling towers from each power station type is either re-injected back into the geothermal reservoir from which it was extracted, or discharged to nearby surface water (and in some cases onto land) after being treated. In some cases, prior to re-injection or discharge of the cooled water, there is the opportunity to re-use this water for direct use purposes.

2.6 Possible Hurdles

Exploitation of any type of energy brings different sets of problems. Water which has been in contact with rock for thousands or even millions of years contains dissolved minerals. Hot water tends to be more reactive and aggressive than cold water. Geothermal waters may often contain around one percent of dissolved solids. Usually these could be carbonates, sulphates, chlorides and/or dissolved silica, the amount of which becomes significant where waters have been in contact with rock at high temperatures (around 200 °C and above). Thus one can often meet brines, which is the name for geothermal waters. Dissolved gases are common as well, especially at higher temperatures and liquid/vapour reservoirs or vapour dominated reservoirs. Currently available techniques can significantly lessen the negative impact of minerals carried by the brines; however this has to be taken into consideration during the design stage of the power plant, district heating system or any other application which uses the geothermal brine. With correct research and design, these contaminations can be handled and disposed of without operational or environmental difficulty. If such threats are ignored during the research and design stage, the entire (piping, heat exchangers) system can easily fail within a matter of months.

There are at least a few objectives that a geothermal exploration program must accomplish in order to be successful:

- Locate exactly the geothermal reservoir beneath the surface,

- Define the chemical composition, origin and properties of the geothermal fluid,

- Estimate the volume of the reservoir and its type, temperature, flow and pressure of the fluid, permeability of the formation,

- Predict the nature of the geothermal fluid in the reservoir – is it going to be a one-phase or two-phase mixture,

- Forecast (estimate) the *deliverability* of the geothermal reservoir,

If some of the above mentioned research was neglected or misapplied it could result in some serious consequences like:

- The geothermal reservoir can be depleted when more hot water or steam is withdrawn than can be naturally recharged (natural water inflow) or artificially re–injected by a reinjection well. Thus it is important to predict and exploit the geothermal reservoir in sustainable way, because once it is fully depleted, it may take centuries to recharge such a reservoir,

- Gas emissions can be unpleasant or even dangerous. Hydrogen sulphide in high concentration is dangerous, and carbon dioxide or other toxic emissions can occur during the exploitation of geothermal wells,

- Geothermal brines often have a high content of dissolved solids, minerals or toxic substances, thus harmful the impact of brine and its minerals on the piping system, heat exchangers and cooling tower ought to be properly estimated during the research and design stage (H. Gupta, S. Roy, 2007),

2.7 The Benefits of Geothermal Energy

When properly developed and managed, geothermal energy systems are clean, abundant, and reliable sources of renewable energy. The use of geothermal energy sources for electricity generation of direct use eliminates the need for the exploitation and utilization of non–renewable and more harmful resources. Installed geothermal electricity generation capacity world–wide is equivalent to the output of around ten nuclear power plants (57 TWh/a of electricity (Bertani, 2005), and 76 TWh/a for direct use (Lund et al., 2005)). The utilization of geothermal energy does not require any type of non–renewable fuel; it is a self–sufficient technology. Moreover geothermal energy is not dependent on climatic variations. Geothermal energy as a renewable energy source helps to reduce the reliance on imported crude oil or other fossil fuels.

The use of geothermal energy conserves non–renewable fossil fuels and mitigates the carbon dioxide emissions, which would have been produced by a conventional power plant producing an equivalent amount of electricity. The utilization of geothermal resources is not land intensive; it is quite the contrary when compared to conventional power plants. Geothermal power plants and district systems are reliable, as they can operate 24h/day compared to wind, solar or tidal power plants (stations). Small sized geothermal power plants, heat pumps, absorption chillers, and district systems powered by geothermal energy can be installed in remote areas, thus providing electricity, heat and/or cold (thermal comfort).

3 ONGOING MARKETS

3.1 Overview

This chapter is partially based on the "Residential HVAC Market Research" published (July 14, 2004) by National Energy Management Institute.

Recently tight and unstable oil and natural gas markets have generated extremely high (40 −50 \$/barrel, Jan 2009 or 140 − 150\$/barrel Jul 2008) levels of crude oil and petroleum product prices. Continued increases in prices and decreasing resources of fossil fuels are expected, reflecting tightness in supplies for the next years. Electricity prices are rather less volatile than natural gas or crude oil prices; regardless, electricity prices are going to rise during the next years. Therefore, now is the proper time to develop and spread absorption cooling technology powered by geothermal heat, as a cost, energy efficient and environmentally friendly method for providing thermal comfort to people's lives.

The history of water/lithium bromide absorption chiller development suggests that developing such a product for light–commercial (high–residential buildings) applications are technically and economically feasible (sometimes it requires specified conditions).

The key risks lie in whether prominent and capable manufacturers will consider the market potential to be sufficient to justify development costs and whether product costs can be low enough to appeal to the market.

3.2 Current Markets

United States

The United States is a unique and valuable example of developing HVAC market. In the US there are an estimated 120 million single–family homes. Almost 20 million, or 17%, of these homes were added just in the past 10 years. As the housing stock grows, so does the number of installed HVAC units. As the number of HVAC units grows, so does the number of absorption chiller units.

Demand in the US for heating, ventilation and air conditioning (HVAC) equipment is forecast to increase 3.2 percent per year to \$16.8 billion in 2011. Advances will be primarily driven by strong gains in non residential construction, along with ongoing growth in the residential replacement market. Although gains will be restrained by projected declines in new single unit housing completions between 2006 and 2011, nearly three quarters of HVAC demand is attributable to replacements. Growth in the replacement sector will benefit from rising interest in more energy efficient building systems driven by rising energy prices, spurring the replacement of older HVAC equipment with newer models. Changing regulations regarding minimum efficiency requirements for many of these systems will also affect sales of HVAC equipment.

In 2006, electric powered HVAC systems accounted for 82 percent of US sales value and natural gas-burning systems accounted for 16 percent of sales. Systems operating on geothermal energy will post the strongest gains, albeit from a very small base.

Unitary air conditioners will remain the largest segment in the cooling equipment industry, accounting for more than 70 percent of total demand in 2011. Absorptive liquid chilling systems are expected to post the strongest gains through 2011 as industrial and commercial construction rebounds over the forecast period (The Freedonia Group, 2007).

China

In 2000, the percentage of the population living in cities and towns was 36,1% in China, which means that about 457 million people lived in cities and towns. Shanghai ranked first, with more than 88% of its citizens living in the urban area. In 2000, the population of Shanghai had reached 16,7 million but the land area of Shanghai is only 6340,5 km^2. Therefore, the population density in the downtown area is very high – up to 53 326 people/km^2. The per–capita gross living area was 15,5 m^2 (in 2003), which was still at a low level if one considers global standards, thus it was predicted that about 5,7 billion m^2 of residential buildings will need to be constructed by 2008. That means dramatically expansion of HVAC market.
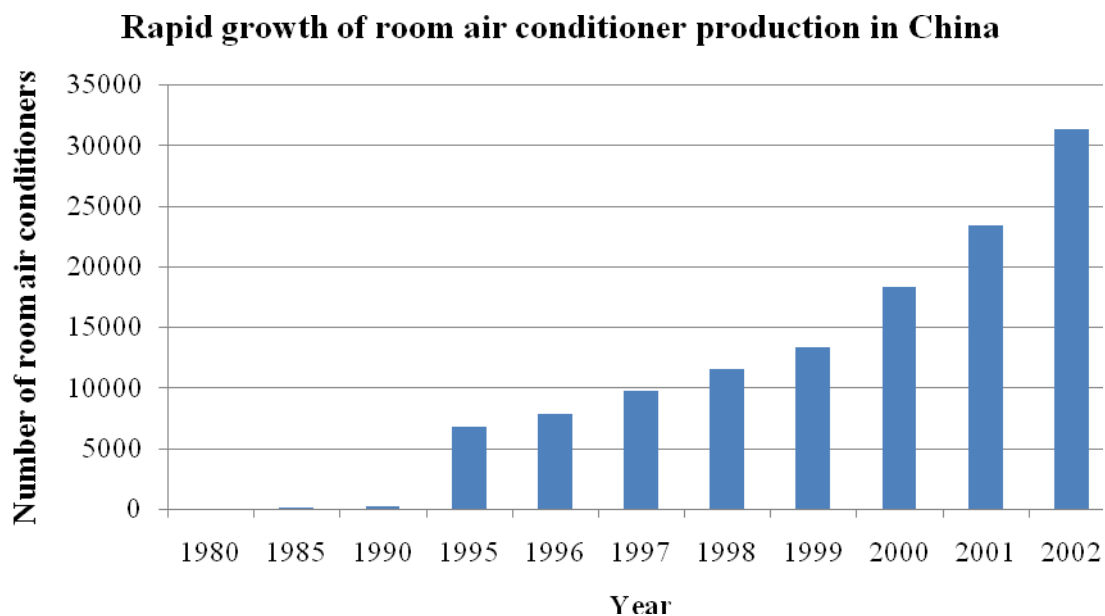
Figure 11 Rapid growth of room air conditioner production in |China (Based on: China Statistical Yearbook, 2003)

China has been the fastest growing major nation for the past 25 years. Rapid growth of the Chinese economy, industry and living standards caused an increase in the HVAC market in China. The HVAC market has been growing the fastest in the areas of the Yangtze River and the Yellow River, mostly due to very hot summers and very cold winters. Some of the large cities in China, like Beijing and Shanghai have begun to build district heating and cooling systems to cope with the city's energy crisis and pollution problems.

3.3   Existing Models and Applications

Most of the world's LiBr absorption manufacturing capacity is currently in Asia (Japan and China) and the US, as is much of the LiBr absorption chiller development work. Published LiBr absorption development efforts have taken place in the US, Japan and Europe. An Existing application of double–effect absorption chillers can be found at a hospital in Brooklyn, New York. Absorption chillers provide 280 tons of cooling capacity (985, 6 kW). The chiller operates on steam created by natural gas–fired boilers. The Thermax unit has a high 1,25 coefficient of performance (COP) value. The unit generates chilled water to air building air conditioning system, without increasing the hospital's electricity demand during peak summer–rate periods. The chiller provides cold for 48 000 ft^2 (14 630,4 m^2).
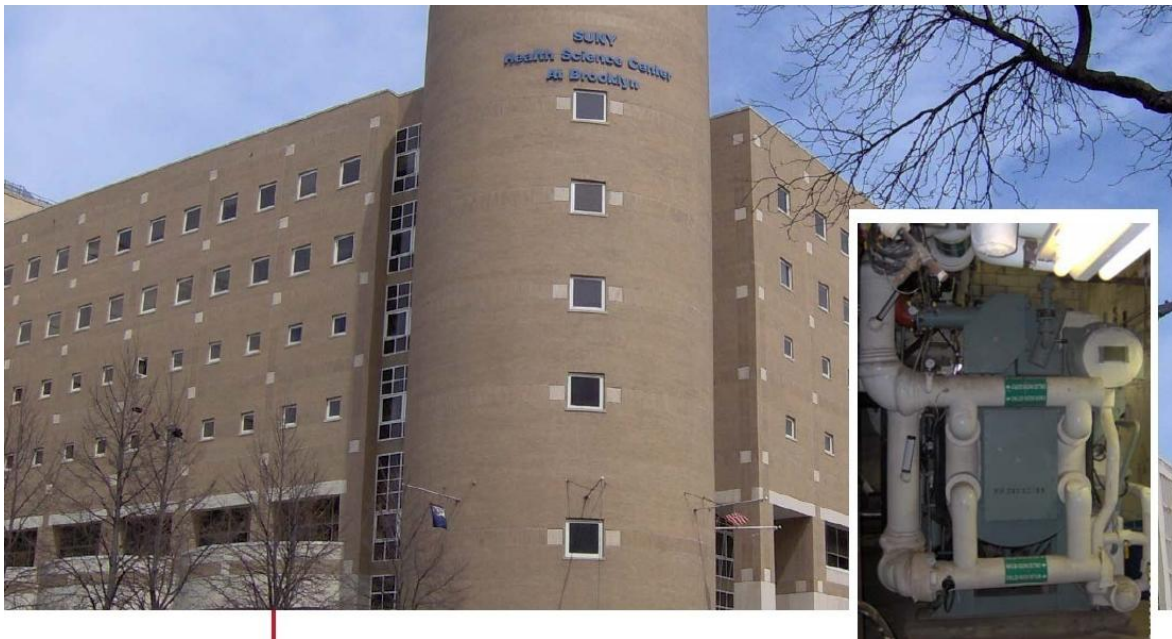
Figure 12 Double - effect steam - fired absorption chiller.

Appendix A contains a Summary of Published Past LiBr Development Efforts

3.4   Climate Considerations

In many applications, such as in solar energy technology, space cooling technology and heating and cooling applications an accurate climatic database is required. In the current project the accuracy of ambient air temperature (dry bulb temperature) and humidity are crucial. Preferable conditions for humans and machines to work are:

- Temperature between 18 °C – 25 °C

- Relative humidity below 50%.

If the environment within a building is too cold or too warm, with respect to this temperature band, then occupants soon will feel uncomfortable. Due to lower thermal comfort, machines could be less productive and occupants may even become ill.

Absorption technology seems to be a clever application for utilizing low grade geothermal heat to achieve thermal comfort at low energy cost. This technology has the potential to significantly reduce CO_2 and greenhouse gasses emission.

3.4.1 China – Shanghai

Shanghai is the largest city in China in terms of population and one of the largest metropolitan areas in the world, with over 20 million citizens. It is located at 31°12'0'' N and 121°30'0'' E. Shanghai sits on the Yangtze River Delta on China's east coast, roughly equidistant between Beijing and Hong Kong. The city has many rivers, canals, streams and lakes which are required for absorption technology, and is known for its rich water resources as part of the Taihu drainage area. The climatic data from the National Institute of Metrology, based on a one year concept for Shanghai, is characterized by a series of 8760 hourly outdoor data as dry bulb temperature.
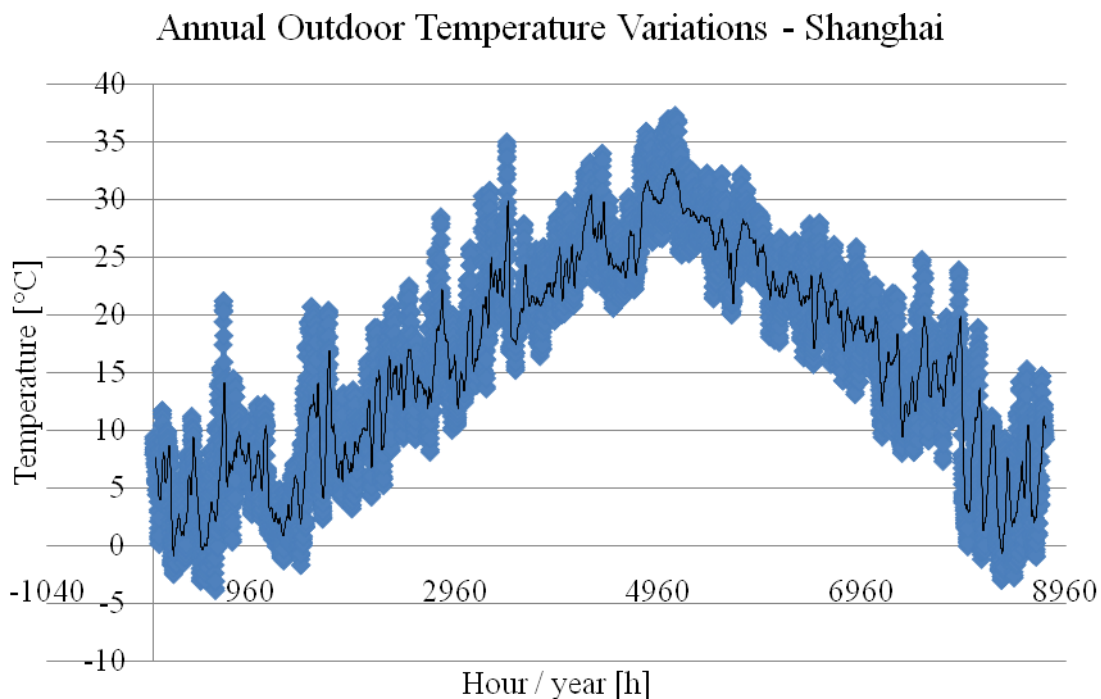
Figure 13 Annual Outdoor Temperature Variations - Shanghai, China

(http://www.eere.energy.gov/)

China has a humid subtropical climate and experiences four distinct seasons. Shanghai lies in the subtropical monsoon climate belt that is warm and moist with ample sunshine and a long plant growing season. In winter, cold northerly winds from Siberia can cause night time temperatures to drop below 0 °C, and although not usually associated with snow, the city can receive one or two days of snowfall per year. Summer in Shanghai is very warm and humid, with occasional downpours and thunderstorms. Spring and autumn is generally sunny and dry, Shanghai experiences, on average, 1778 hours of sunshine per year, with the hottest temperature ever recorded at 40 °C and the lowest at 12 °C. The average

number of rainy days is 112 per year. Since the coal consumption of Shanghai proper is more than 20 million tons per year, the urban temperature and rainfall are higher than on the outskirts of the city. In Shanghai the major demands for fresh water involve the direct use of precipitation in the growth of corps and pasture and the use of water from rivers for domestic, industrial, irrigation, thermal power, navigation, fishing, recreation and other purposes (Duan Shao-bo).
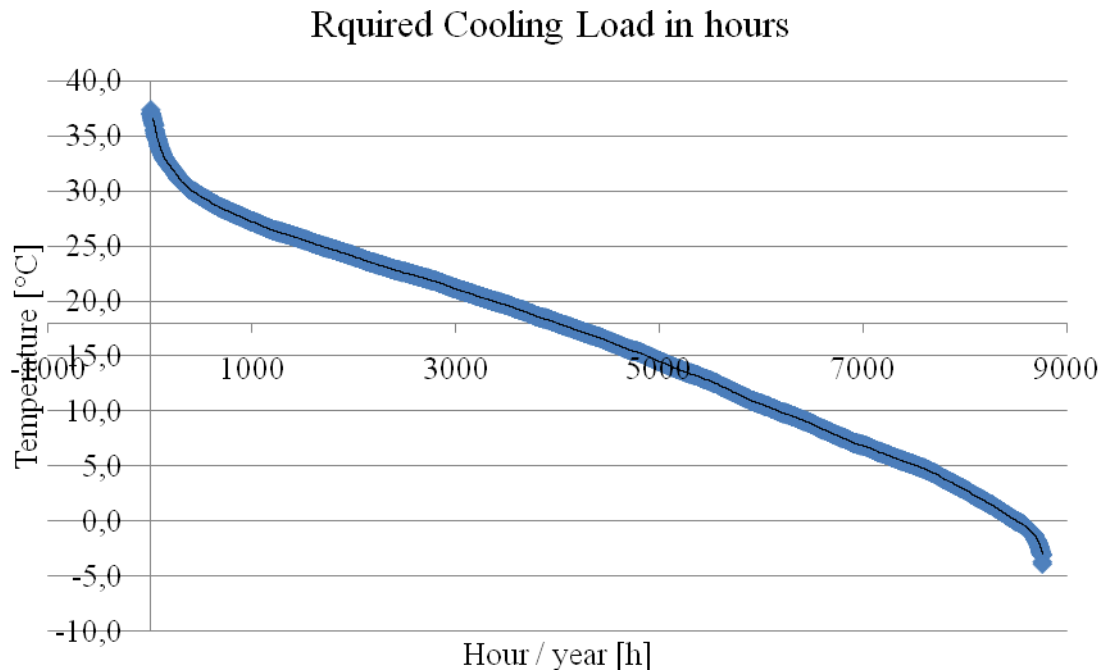
Figure 14 Required Cooling Load - Shanghai, China (http://www.eere.energy.gov/)

3.4.2 United Arab Emirates – Abu Dhabi

Abu Dhabi has a hot and humid climate. The months of April through September are generally hot and humid with maximum temperatures averaging above 40 °C. The weather is cooler from November to March. During this period dense fog occurs due to high air humidity. Abu Dhabi is located at 24°28' N and 54°22' E, in the oil–rich and strategic United Arab Emirates. It borders with the Kingdom of Saudi Arabia (south) and the Sultanate of Oman (east). Abu Dhabi city is on an island located less than 250 meters from the mainland, and is joined with the mainland by bridge.

The climatic data from the National Institute of Metrology, based on a one year concept for Abu Dhabi, characterized by a series of 8760 hourly outdoor data as dry bulb temperature. Absolute and relative humidity have been used in this work.

Annual Outdoor Temperature Variations - Abu Dhabi
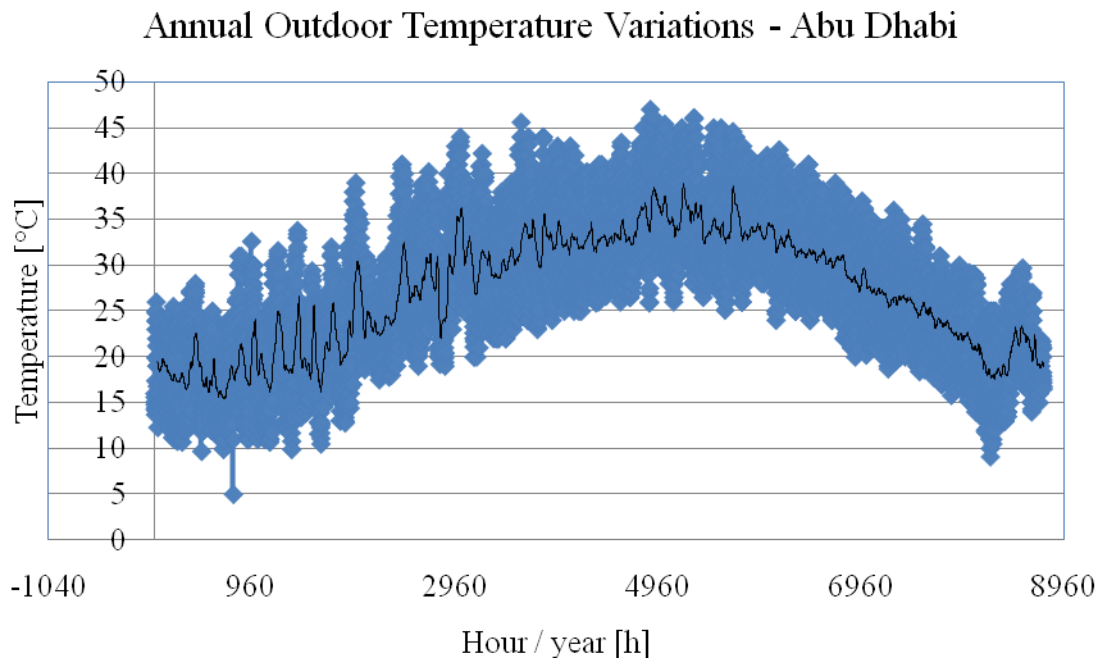
Figure 15 Annual Outdoor Temperature Variations - Abu Dhabi, UAE (http://www.eere.energy.gov/)

Required Cooling Load in hours
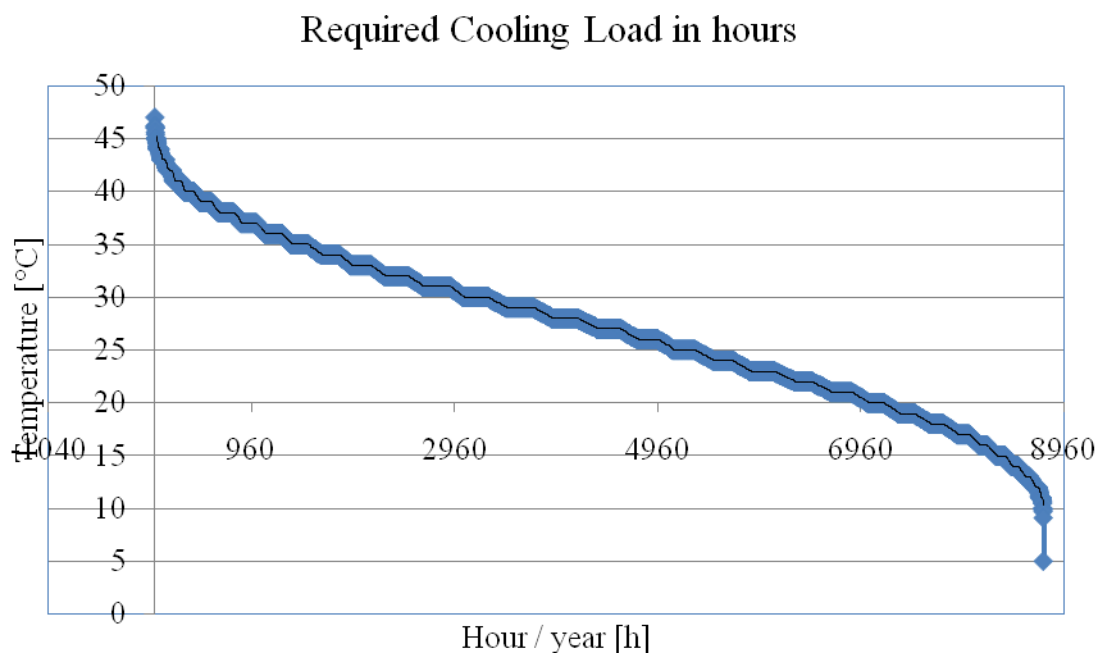
Figure 16 Required Coolind Load - Abu Dhabi, UAE (http://www.eere.energy.gov/)

Recent predictions say that by 2030 Abu Dhabi's gross population is projected to surge to 3,1 million from the current baseline figure of 930 000 in 2007, while annual tourist visits and residential units are set to rise to 7,9 million and 686 000 from 1,8 million and 180 000 respectively.

The population of Abu Dhabi is expected to increase from 930 000 in 2007 to 1,3 million in 2013, 2 million in 2020 and to 3,1 million in 2030 according to estimates made by the Plan Abu Dhabi 2030.

The Urban Structure Framework Plan, "Plan Abu Dhabi 2030", presents a coherent picture for the future of the City of Abu Dhabi as an environmentally, socially and economically sustainable community and as an increasingly important national capital. Abu Dhabi will see strong demand for residential units across all sectors of the market as the population grows. Therefore as fast as the number of residential units increase, the amount of absorption units will increase.

3.5 Summary

As it was discussed above, much of the water/lithium bromide absorption chiller development work has taken place in Japan, the US and China. The table (below) shows the temperature difference between Shanghai and Abu Dhabi. The values in the table do not include the effects of urban heat islanding (the human impact on the temperatures in urban areas), including the elevated temperatures. Many past developers have designed water/lithium bromide water absorption chillers for 35 °C ambient temperatures. However, it is important to note that, without demonstrating performance at higher temperatures, it could be not clear how absorption machines would behave at different ambient conditions.

Country	Region/City	Operation hours over 35°C		Operation hours over 30°C	
		Hours/Year	Percent of Year [%]	Hours/Year	Percent of Year [%]
USA	Central Texas/Dallas (Love Field)	531	6,06	<u>1691</u>	19,3
	Arizona/Phoenix (TMY2)	1042	11,89	<u>2298</u>	26,23
Japan	Osaka	18	0,21	<u>466</u>	5,32
	Sapporo	0	0,00	23	0,26
China	Shanghai	50	0,57	**408**	4,66
UAE	Abu Dhabi	1515	17,29	**3291**	37,57

Table 4 Comparison of ambient temperatures in different countries (http://www.eere.energy.gov/)

As discussed above, absorption technology is well developed in Asia and the US due to high HVAC demands and the competitiveness of this technology. Due to the favourable climate and geographical characteristics of China and UAE these regions are preferable in the foregoing project because of the high value of operation hours over 30 °C during the year and vicinity of big water reservoirs provided by these locations.

4 TOTAL EQUIVALENT WARMING IMPACT

4.1 Overview

Modern refrigeration technology, including mature absorption refrigeration technology, has coexisted with the more standard and widespread types of vapour compression systems more frequently than ever before. Only in the USA and Asia (Japan, China) is absorption technology widely available and respected, mostly due to its competitiveness with standard vapour compression systems, and concerns regarding the Global Warming Potential (GWP) and the more recently known Total Equivalent Warming Impact (TEWI) provide the opportunity to examine refrigeration and cooling applications (Robert Tozer, 2005). The total contribution of refrigeration/air–conditioning applications to the greenhouse effect is estimated according to the TEWI concept (DVK Conference, Hamburg, 1997).

TEWI considers both the direct refrigerant effect and the (indirect) primary energy impact on the equivalent carbon dioxide emissions. The Direct GWP effect is produced mainly by all halogenated refrigerants including HFCs such as HFC 134a, which is used further as a base case for centrifugal chillers. The Indirect GWP effect is attributed to indirect carbon dioxide emissions due to the energy usage for the duration of its useful life. The amount of CO_2 strongly depends on the type of electricity generation, power plant technology and the type of fuel which varies between countries.

In fact absorption units do not require one specified type of heat source – any type of heat source can be adjusted to absorption technology. The following consideration contains only solutions which are the most available on the current market. By using a water/lithium bromide solution, the direct refrigerant impact on TEWI is not considerable.
According to TEWI III (Oak Ridge National Laboratory, 1997), approximately 62 million refrigerators are manufactured annually. In addition, hundreds of millions of refrigerators are currently in operation. Furthermore, a significantly increasing demand is expected for developing countries in the near future, which will increase primary energy consumption, which in turn will increase the impact on TEWI. The use of air conditioners, seen from a global perspective, is the most widespread after refrigerators, and therefore makes a significant contribution in total to the greenhouse effect. The world installed charge of decentralised air conditioners is approximately four times greater than central water chillers (in 1991: estimated $2,11 \times 10^8$ kW worldwide for chillers). Around 80% of central water chillers are installed in the USA and Japan (Oak Ridge National Laboratory, 1997).

4.2 TEWI Allocation

To allocate TEWI values to electricity, heat and cold produced, it has been necessary to refer to thermo economic techniques (Lozano & Valero 1993). The thermodynamic component of such an analysis can be conveniently based on the second law of availability (Moran 1989) or the exergy concept.

Exergy

Exergy is a thermodynamic variable which quantifies the usefulness of energy. As a simple example of exergy, air at atmospheric conditions of temperature, pressure, and composition

contains energy but no exergy when it is chosen as the thermodynamic reference state known as ambient. Individual processes on Earth like combustion in a power plant often eventually result in products that are incorporated into a large atmosphere, so defining this reference state for exergy is useful even though the atmosphere itself is not at equilibrium and is full of long and short term variations (http://en.wikipedia.org).

An introduction to the theoretical concept of exergy is detailed in Appendix C.

TEWI Allocation

The TEWI allocation was based on the specific exergy costs of thermal and electric energy being identical. This implies that the TEWI allocations of thermal energy will be lower than electric energy.

$$\eta^b = P\!\!\Big/\!\!F$$

(4.2.1)

where:

η^b - exergetic efficiency

P - exergy of the product [kW]

F - exergy of the fuel [kW]

The inverse of the exergetic efficiency (η) is the specific exergy consumption, the ratio of the exergy of thermal and electric products with respect to the exergy of the natural gas fuel (driving force – hot water) (Lozano & Valero, 1993).

$$k = F\!\!\Big/\!\!P = 1\!\!\Big/\!\!\eta^b$$

(4.2.2)

k - specific exergy consumption [-]

The TEWI allocations are detailed below

The Carnot efficiency indicates the quality of energy, which is used in equation below (3.2.5) for exergy cost evaluation.

$$\eta_{Carnot} = \frac{T - T_0}{T}$$

(4.2.3)

$$CO_{2-wchp} = \frac{CO_{2-ng}}{\eta^b}$$

(4.2.4)

$$CO_{2-qchp} = \frac{CO_{2-ng}\,\eta_{Carnot}}{\eta^b}$$

(4.2.5)

Where applied subscripts describes:

w - electricity

q - thermal

chp - combined heat and power

ng - natural gas

4.3 TEWI Analysis

The objective is to evaluate the chiller unit with the lowest TEWI. A few types of chillers are considered, which are listed in the table below. To increase the ecological awareness of the reader, single stage absorption units are compared with double stage absorption units and centrifugal chillers. Double stage absorption units (direct-fired especially) were very popular in the U. S. and Japan in the 1970s due to the low cost of natural gas and oil (see chapter 1.2)

No.	Type	No. of stages	Heat source	Heat source temp.
I	Absorption	Single	Hot water	86 °C
II	Absorption	Single	Hot water	115 °C
III	Absorption	Double	Direct fired	1500 °C
IV	Absorption	Double	Steam	150 °C
V	Absorption	Double	Exhaust gas	190 °C
VI	Centrifugal chiller		Electric driven	-

Table 5 TEWI – types of considered chillers (R. Tozer, Z. Ure)

The Heat Dissipation Ratio (HDR), is the ratio of condenser and absorber heat with respect to the evaporator heat (Tozer 1991, 1992). HDR is an important factor, due to quantity of heat which is dissipative through cooling towers. In the case of chillers with a direct fired heat source, the efficiency of the burner has to be taken into account.

$$HDR = 1 + \frac{\eta_{burn}}{COP} = \frac{Q_a + Q_c}{Q_e}$$

(4.3.1)

η_{burn} - burner efficiency (in case of lack of combustion chamber it is assumed to be one)

The main energy input to the chiller unit depends on chiller type; in the case of absorption chillers it is heat load to the generator Q_{Input}, whereas for centrifugal chillers it is electric load Q_{el} (equation 1.4.2) to the compressor.

$$Q_{Input} = \frac{Q_e}{COP}$$

(4.3.2)

The exergy input B_{Input} is equal to the Carnot efficiency multiplied by Q_{Input}.

$$B_{Input} = \eta_{Carnot} Q_{Input}$$

(4.3.3)

The mass flow rate for the condenser and absorber is computed from equation

$$\dot{m}_{cw} = \frac{Q_e HDR}{c_{pcw} \Delta t_{cw}} = \frac{Q_a + Q_c}{c_{pcw} \Delta t_{cw}}$$

(4.3.4)

Where:

cw – cooling water,

The condenser water pump power is expressed as a function of condenser water mass flow, pressure drop, and pump motor efficiency.

$$W_{cw} = \frac{\dot{m}_{cw} \, \Delta p_{cw}}{\eta_p \eta_m}$$

(4.3.5)

The total electric power includes the power of the condenser and absorber water pump, cooling tower fan power, hot water pump power and power for the pump in the thermal compressor (solution pump in case of absorption chillers).

$$W = \sum W = W_{ch} + W_{hw} + W_{cw} + W_{ct}$$

(4.3.6)

In the case of energy cost calculation and TEWI calculation it is important to determine the load factor, the ratio of working hours being 8760 hours. Evaluation should be done with the same number of operational hours for each chiller unit (*eflh* and *poh*). For the compressor power of centrifugal chillers and the generator input to absorber chillers, Equivalent Full Load Hours (EFLH) were used. For the rest of the electrical equipment Plant On Hours (POH) were used. Thus it was required to evaluate the power corresponding to both these operational hours. As it was mentioned above, TEWI consists mostly of CO_2 emission (indirect effect). CO_2 emission strongly depends on electricity consumption, which strongly depends on cooling water mass flow, hot water flow, etc (table below).

To calculate TEWI it is required to obtain:

	Symbol/Unit	I	II	III	IV	V	VI
Type		Absorption					Centrifugal
Capacity	Q_e [kW]	2100					
No. of Stages	N	1		2			-
Heat source		Hot water		Direct fired	Steam	Heat recovery	Electric
Heat source temperature	[°C]	86	115	1500	150	190	-
Coefficient of Performance	COP	0,705	0,705	1,05	1,2	1,1	5,5
Burner efficiency	η_{burn}	-		0,88	-		
Heat dissipation ratio	HDR	2,42	2,42	1,84	1,83	1,91	1,18
Dead state temperature	T_{ref} [°C]	25					
Carnot efficiency	η_{Carnot}	0,17	0,23	0,83	0,30	0,36	1
Energy input	Q_{Input} [kW]	2838	2978	2000	1750	1909	381

Exergy input	B_{Input} [kW]	482	691	1664	517	680	382[*]
Auxiliary chiller power	W_{ch} [kW]	10	8	15	11	11	5
Hot water pump	W_{hw} [kW]	3	2	-	1	-	

	Symbol/ Unit	I	II	III	IV	V	VI
Condenser water temperature difference	Δt_{cw} [°C]	7,0	7,0	8,3	8,3	5,5	5,5
Condenser water flow	$\dot{m}_{cw}$ [kg/s]	174	173	111	110	174	107
Condenser water pressure drop	Δp_{cw} [kPa]	200	200				
Pump efficiency	$\eta_p \eta_m$	0,665	0,665				
Condenser water pump power	W_{cw} [kW]	52	52	33	33	52	32
Cooling tower fan	W_{ct} [kW]	-	60	40	38	60	38
Electric power	W [kW]	124	122	88	83	123	457
Electric power	W_{eflh} [kW]	0					382
Electric power	W_{poh} [kW]	124	122	88	83	123	75

Table 6 TEWI - Primairly TEWI analysis (R. Tozer, Z. Ure)

[*] - Centrifugal compressor power [kW]

The table above provides detailed TEWI results for each type of introduced chiller.

A preliminary analysis of numbers contained in the table no. 6 indicates the following:
- The exergy input to both single and double effect absorption chillers is approximately the same, the only exception being the direct fired (III) double effect absorption chiller. Exergy input can be almost equal, because single stage ABS units have lower COPs and require a lower grade heat source, when double effect ABS units require a high grade heat source.
- Heat Dissipation Ratio: In practical applications this is the predominant factor in energy usage, as double stage absorption chillers benefit from lower HDR more than single stage chillers (Tozer & James, 1995).

- Cooling tower fan and condenser water pump power constitute major portions of the electricity consumed by ABS, due to the fact that it might occur that decreasing the temperature of the cooling water could be beneficial. A decline in cooling water inlet temperature will increase the temperature gap between the outlet and inlet cooling water temperature and/or decrease the cooling water mass flow, thus a decline in electricity consumption will occur.

4.4 TEWI and Energy Calculations

It was assumed that the amount of plant on hours (POH) was 50% greater than the equivalent full load hours (EFLH). The service life is assumed to be 10 years.

The electric energy load is based on both plant on hours and equivalent full load hours.

$$E_w = eflh \cdot W_{eflh} + W_{poh} \cdot poh$$

$$(4.4.1)$$

The quantity of thermal energy is evaluated on the equivalent full load hours of heat, which is used to run the chiller unit.

$$E_q = eflh \cdot Q_g$$

$$(4.4.2)$$

Feature	Symbol	Unit	I	II	III	IV	V	VI
Equivalent full load hours	$eflh$	[h/yr]	1000					
Plant on hours	poh	[h/yr]	1500					
Lifetime	T_{long}	[yrs]	10					
Electric energy	E_w	[kWh]	186588	182930	132453	124823	184795	494786
Thermal Energy	E_q	[MWh]	3021	2978	2000	1750	1909	-

Table 7 TEWI – Energy breakdown.

The European average for CO_2 emissions per kWh electricity is equal to 0, 47 kg CO_2/kWh, whereas the carbon dioxide emissions for natural gas and grid electricity are 0, 18 and 0, 53 kg CO_2/kWh. The table below provides the results of TEWI calculations for each previously introduced case.

Direct Global Warming Potential is evaluated by the formula:

$$D_{GWP} = GWP \cdot L_{out} \cdot T_{long} + GWP \cdot m_{Rch} \cdot 1 - \alpha_{ref}$$

$$(4.4.3)$$

Symbols used in equation (4.4.3) are described in table 8 (below)

and Indirect Global Warming Potential is evaluated by the formula:

$$I_{GWP} = T_{long} \cdot E_w \cdot \beta_e \qquad (4.4.5)$$

$$TEWI = D_{GWP} + I_{GWP} \qquad (4.4.6)$$

Feature	Symbol	Unit	I[2,4]	II[2,4]	III[2,3]	IV[2,*]	V[2,*]	VI[2]
Global Warming Potential	GWP	-						1300
Refrigerant charge	m_{Rch}	[kg]						450
Annual leakage	Y_{leak}	%						5
Annual purge	Y_{purge}	%			-			0,5
Annual service release	Y_{sr}	%						0,25
Leakage CO_2	L_{out}	ton CO_2/life						366,6
Refrigerant recovery factor	α_{ref}	-						0,75
CO_2 emission – grid	E_{grid}	kg CO_2/kWh			0,53[1]			
CO2 emission – EU average	E_{EU}	kg CO_2/kWh			0,47[2]			
CO2 emission – gas	E_{gas}	kg CO_2/kWh			0,18[3]			
CO2 emission – geothermal	E_{geo}	kg CO_2/kWh			0,05[4]			
Electricity consumed	E_w	MWh/yr	186	183	132	124,5	184,5	494,5
Indirect GWP	$I_{GWP-gas}$	ton CO_2/life	2384,7	2349,1	4220,4	2685,2	3157,9	2324,2
TEWI – EU average	*TEWI*	ton CO_2/life	2384,7	2349,1	4220,4	2685,2	3157,9	2824,8

Table 8 Assessment of TEWI

(* - CO$_2$ emission for heat recovery (and exhaust gases) is equal to 0, 12 kg CO$_2$/kWh

Total Equivalent Warming Impact

Type of chiller unit	1	2	3	4	5	6
Direct effect	0	0	0	0	0	500,6
Indirect effect	2384,7	2349,1	4220,4	2685,2	3157,9	2324,2
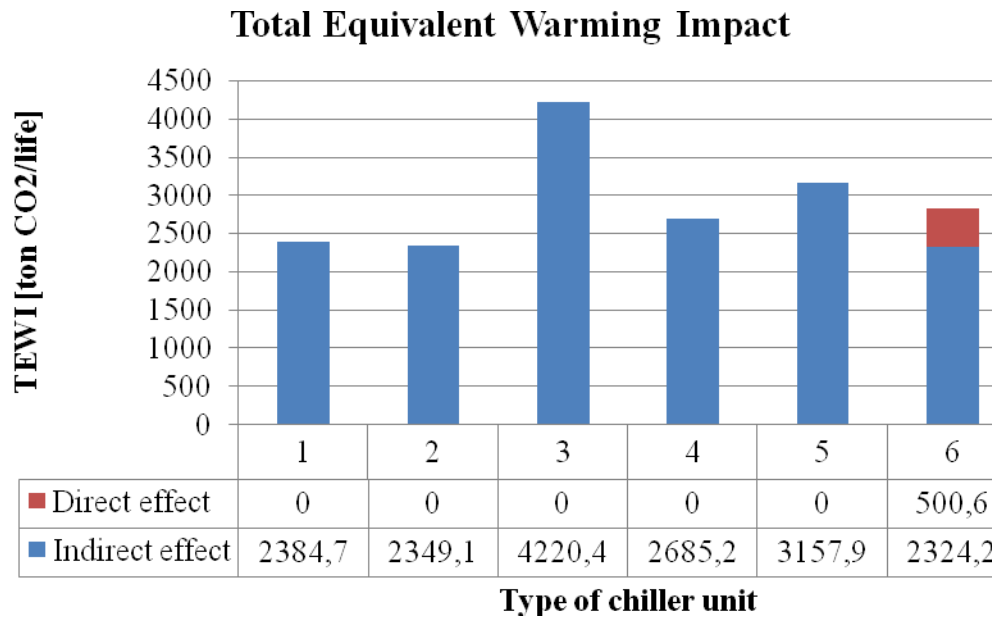
Figure 17 TEWI – Total equivalent warming impact, results for previously introduced six types of chillers

- Double stage direct fired absorption chillers have the highest TEWI, around 30% higher than conventional centrifugal chillers. Therefore double stage direct fired absorption chillers cannot be recommended. Despite the clearly visible TEWI disadvantage, in certain instances this option has proved to be economically viable (Tozer, 1994).

- Both single stage hot water driven absorption chillers resulted in the best TEWI values. It was possible to obtain such low TEWI value only with the assumption that the hot water (86 °C and 115 °C) comes from geothermal wells.

- Double stage heat recovery and steam powered absorption chillers obtained similar results, however the exhaust gas driven chiller resulted in a TEWI value 15% worse than steam driven chillers. This is mostly due to the fact that the exhaust gas driven chiller has a higher heat dissipation ratio, lower COP value and greater electricity consumption.

- Both the double stage steam powered absorption chiller and the centrifugal chiller obtained similar TEWI values, in the range of 500 – 2000 *eflh* (1000 – 2500 *poh*). However, the steam powered absorption chiller is environmentally viable only in a range up to 1500 *eflh*. Beyond the mentioned range (above 1500 *poh*), the thermal energy load to the generator is significantly high, which results in an increases in TEWI value (see crossed line IV and VI, figure no 18).
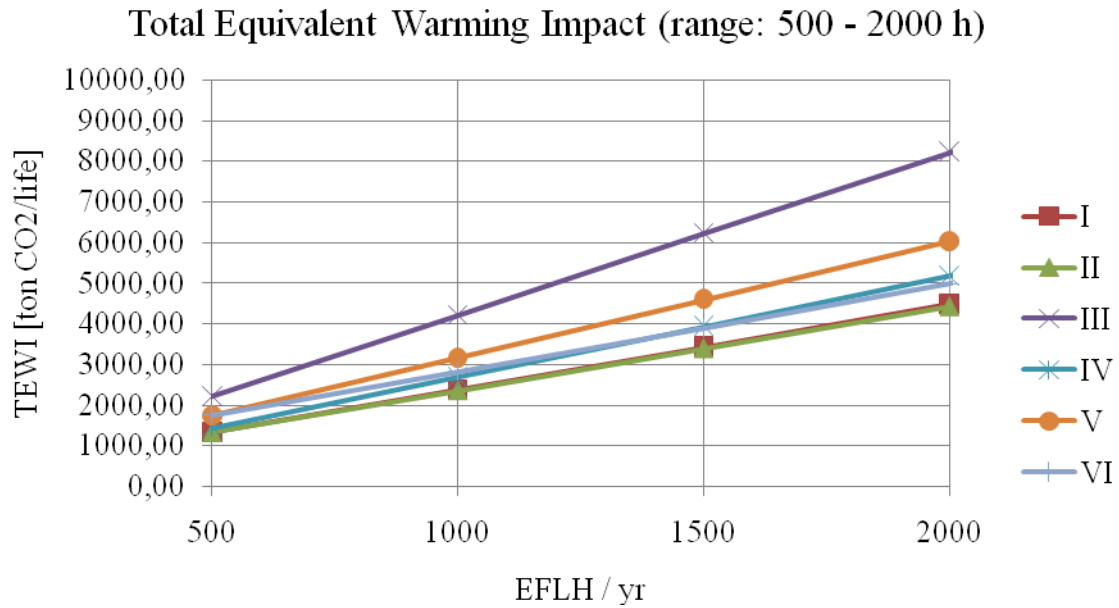
Figure 18 *TEWI - Total Equivalent Warming Impact as a function of eflh.*

Both figures above indicate that the most viable chillers in terms of TEWI are single stage absorption chillers powered by hot geothermal water. Climatic data for Shanghai confirms 3383 equivalent full load hours, and for Abu Dhabi 7189. Due to such high values of *eflh*, single stage absorption chillers powered by hot water appear to be the most viable option in the following aspect.
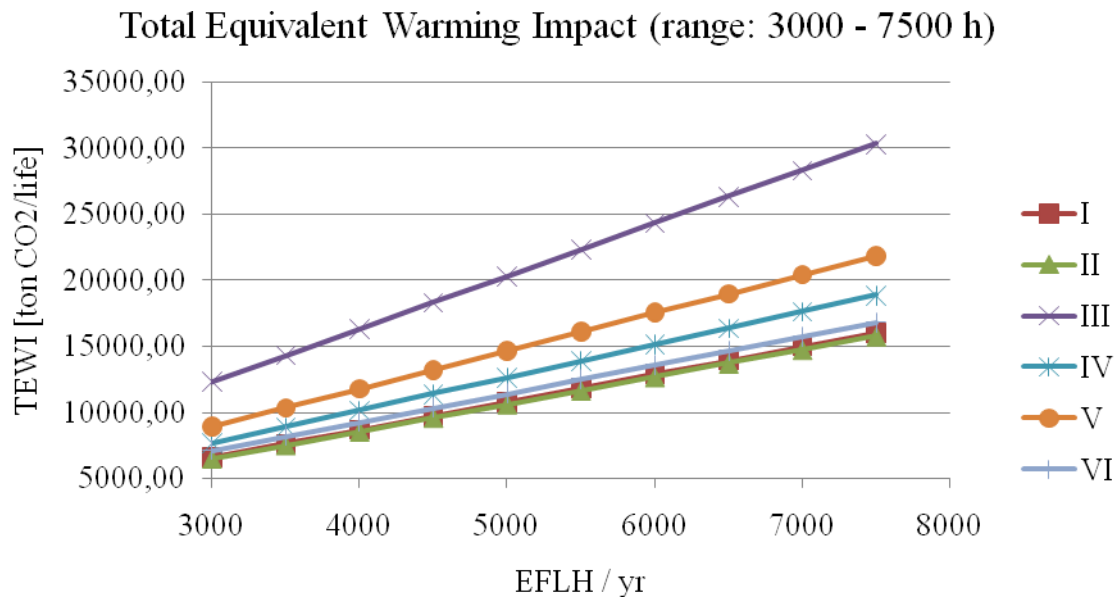
Figure 19 *TEWI - Total Equivalent Warming Impact as a function of eflh.*

35

4.5 Summary

The TEWI analysis is widely used for evaluating the contribution of refrigeration plant to the greenhouse effect over the entire lifetime of the application. However, TEWI is only one of the methods of holistic evaluation. In environmental management, the standardised eco–balance is used to assess a product over its predicted lifetime.

Over the past years emissions from halogenated hydrocarbons have been significantly reduced. The share of CFCs, HCFCs and HFCs in the greenhouse effect will drop by 64% before 2010 due to implementation the Montreal Protocol (CFC-11, CFC-12, CFC-113, CFC-114, CFC-115 and others are currently forbidden). However, measures for reducing emissions or improving efficiency were not taken into consideration in this scenario. Furthermore, the International Institute of Refrigeration (IIR) estimates that the radiative forcing share of HCFs can be reduced by approximately 91% by 2010, or it could even be reduced to zero.

The foregoing TEWI study has established that TEWI could be reduced through the use of water/lithium bromide single stage absorption chillers, powered by geothermal hot water. It appears to be a feasible scenario, especially due to the predicted growth of the HVAC / R market. However it could be profitable to remember that TEWI is not the only criteria for selecting a refrigeration or air–conditioning plant. Further criteria are operating and maintenance costs (O & M costs), energy costs, service costs, neutralization costs and safety costs. In addition, the multidimensional approach of an eco–balance leads to a clear picture of the applicability of the product. Except for the CO_2 emissions from the production of energy, other simultaneous effects due to energy consumption which have an impact on the environment are also considered.

5 SINGLE STAGE WATER/LITHIUM BROMIDE ABSORPTION MODEL

5.1 Overview

Figure 20 (below), shows the scheme of a single stage absorption machine. The main components of the absorption machine are the condenser, refrigeration expansion valve, evaporator, absorber, solution heat exchanger, solution pump, expansion valves and generator. The weak water/lithium bromide solution is pumped from the absorber to the solution heat exchanger. The solution heat exchanger is a counter – flow heat exchanger. The solution heat exchanger helps increase the temperature difference between the outlet from the generator (vapour) and the inlet to the absorber.

The absorption system has two pressure levels, high pressure and low pressure. The pressure in the high pressure circuit is around ten times greater than the pressure in the low pressure circuit. High pressure is around 7 kPa and low pressure is around 0,7 kPa. The low pressure allows the use of water as refrigerant. When using water as a refrigerant the lowest temperature in the whole loop might be 0 °C.

Heat exchanging occurs in heat exchangers and the solution heat exchanger is the only heat exchanger that does not exchange heat with the outside. The evaporator and generator gain heat from the outside. The condenser and absorber lose heat to the outside (surroundings). Heat from the condenser and absorber is removed by a water cooling circuit. A cooling circuit is required to condense water vapour, which leaves the generator and enters the condenser. The cooling circuit in the absorber is required to aid the removal of heat that is generated when vapour is absorbed into the strong mixture. The condenser and absorber cooling circuits are separated water loops. There are applications which have only one cooling circuit, in which cooling water is passed first through the condenser and then through the absorber.

Low pressure cold water enters the evaporator. Chilled water is the hot stream in this case. The hot stream loses its heat and low pressure cold water is changed into low pressure cold water vapour (evaporation process).
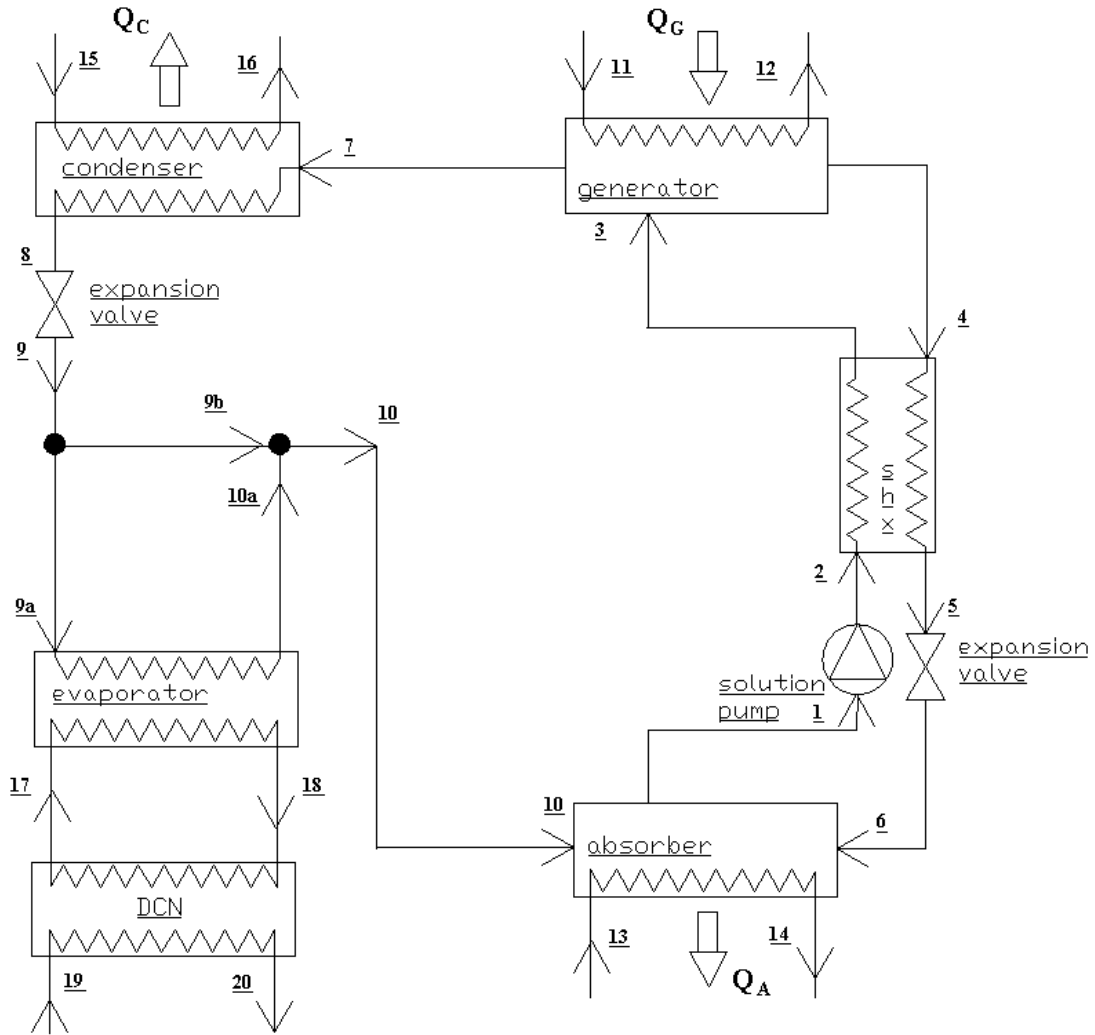
Figure 20 Scheme of single - stage absorption chiller

The most fragile area in the whole absorption machine circuit is the area where strong water/lithium bromide solution leaves the solution heat exchanger. It is due to the risk of crystallization, which may occur under specific conditions, related to temperature and water/lithium bromide concentration. Crystallization is the formation of solid salt crystals due to the (relatively) low temperature at the outlet from the solution heat exchanger and the (relatively) high concentration of lithium bromide salt in water. Crystallization is often the major issue in whole water/lithium bromide absorption technology. Crystallization has to be avoided, as it may cause the formation of a wet solid (crystals) in the absorption piping system, which finally may create a solid (crystal) jam in the piping system. Due to crystallization and jams in the network, the solution's mass flow may be blocked. Crystallization might be avoided by the addition of some heat to the mixture or thinning the water/lithium bromide solution down with the water from the evaporator.

5.2   Possible Constraints

The most probable obstacles which follow from absorption technology constraints are corrosion, crystallization, vacuum requirements and sophisticated working fluids.

5.2.1   Crystallization

It is obvious that the salt component precipitates when the mass fraction of salt exceeds the solubility limit in the solution. The solubility limit is a strong function of mass fraction and temperature (T_1, T_2, T_3, T_4, T_5, and T_6) and a weak function of pressure (P_{low}, P_{high}). Furthermore, crystal nucleation is a process sensitive to the presence of nucleation sites. If no suitable nucleation sites are present, super saturation may occur where the salt content of the liquid is greater than the solubility limit. Once crystals begin to form, the crystals can grow on themselves.
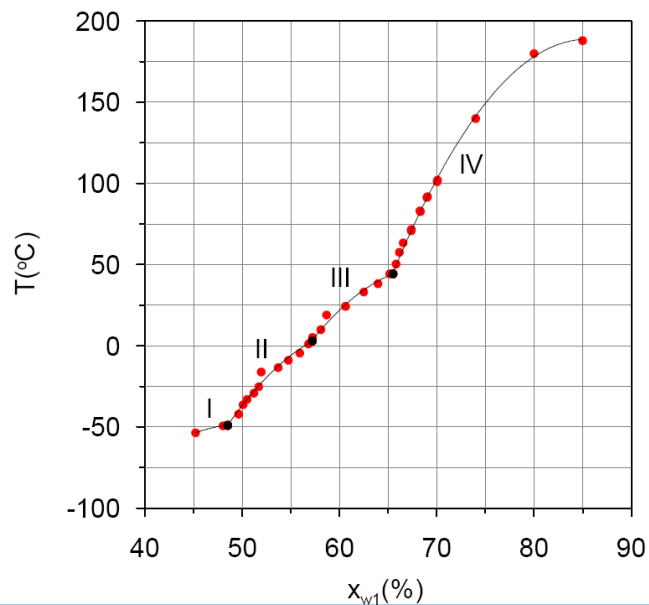
Figure 21 Solubility of LiBr in water (Dong-Seon KIM, 2007)

The formation of LiBr crystallization usually occurs when pressure drops after the valve in the strong solution. This occurs, not due to pressure drop itself, but due to:

- cooling occurring at the same time,

- when pressure drops after the valve, some of the solution boils off (due to the pressure drop) and the concentration of LiBr in liquid phase after the valve might be greater than LiBr concentration before the valve (in liquid phase)

- as the temperature decreases, the solubility properties of LiBr decreases, due to decrease of absorption properties of water,

The control strategy could contain a survey of the distance (temperature and/or solubility distance) from the temperature after the valve (stage 6) to the solubility limit itself. The figure above contains a solubility curve which was obtained from empirical equations (see Appendix) (Dong-Seon KIM, 2007).

Various methods can be used to minimize the possibility of crossing the phase boundary (saturated – supersaturated). The simplest method used is to ensure a sufficiently low temperature sink for cooling the absorber. Based on the properties of aqueous LiBr, low

temperatures in the absorber require lower absorber solution concentration and thus tend to avoid the phase boundary.
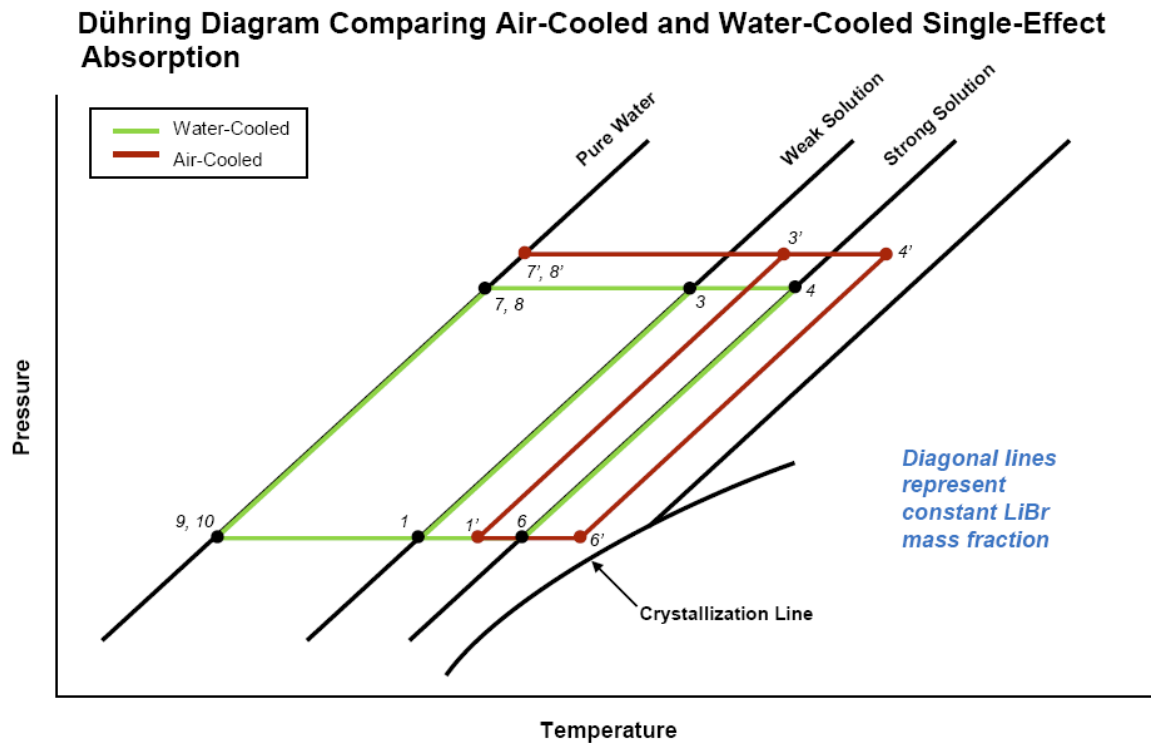
Figure 22 Duhring Diagram Comparing Air - Cooled and Water - Cooled Single - Effect Absorption Chillers (U. S. Department of Energy, 2005)

The crystallization characteristic of the fluid is the main obstacle to producing an air–cooled LiBr absorption machines. Air–cooled absorbers tend to run hotter than water–cooled units due to the relative heat transfer characteristics of the coolants. Particularly in hot climates, this presents a difficult design issue. Thus the project was carried out with an emphasis on water–cooled absorption machines.

The plot above contains the water–cooled and air–cooled thermodynamic cycles (P – T) of an absorption chiller machine. The only difference between air–cooled and water–cooled cycle is the medium used to condense water vapour in the condenser and the medium used to cool down the absorber. Water has a much higher thermal mass than air, thus heat exchangers can be used instead of expensive and big cooling towers in the case of air–cooled machines. Water–cooled cycle seems to be safer in the case of crystallization, due to lower temperatures in the cycle (7,8 and 7',8'). As pressure differences between points 1 and 3 (3') or 9 and 7 (7')show, COP can be higher and the electrical energy consumed by the (solution) pump is smaller (parasitic facilities like, fans or pumps).

Water–cooled absorption machines generally can operate year round without crystallization problems, due to small changes of cooling water temperature. Air–cooled absorption machines do not give such high stability in work as do water–cooled machines; this is due to air temperature and humidity variations. Crystallization – the slushy solid that forms – is wet and viscous. Thus the preferred recovery procedure is to raise the temperature of the portion that is crystallized to a point where the viscosity is reduced sufficiently so that the pumps can circulate the solution. Once the solution is circulating, it

can be easily diluted using water from the evaporator. It is highly recommended to avoid crystallization.

Due to the risk of corrosion and crystallization, some modifications to LiBr – H_2O chemistry are required. Hence modifications to LiBr –H_2O chemistry modifications are nothing new. Water–cooled absorption chillers have relied on 2 – ethyl hexanol for decades to promote heat and mass transfer (R. Sweetser, G. DeVault, 2000). Many crystallization inhibitors have been proposed in order to facilitate the use of air cooling (see figure 14). Unfortunately many crystallization inhibitors solve one problem just to create another, including:

- Temperature limitations

- Corrosion

- Toxicity/flammability

- Reduced heat and mass transfer (significantly in the absorber)

- Incompatibility with the conventional heat- and mass- transfer additive (2 – ethyl hexanol)

Appendix B contains Crystallization Inhibitors for Air – Cooled ABS

5.3 Absorption System

Absorption cooling solutions could be more favourable than compression machines. Air–conditioning systems use compression machines, which are energy wasteful and have impacts on stratospheric ozone depletion due to chlorofluorocarbons (CFC) and hydro fluorocarbon (HCFC) refrigerants (working fluids) (M. Balghouthuni, 2006). Geothermal (absorption) cooling is an attractive alternative since it has the advantage of removing the majority of harmful effects of traditional refrigeration machines. Absorption machines are the main cause of decline in electric peaks during the summer months. At present various types of solar, waste heat and geothermal–powered systems are available for cooling and refrigeration applications.

5.4 Household Subsystem

Energy demands and costs have increased rapidly in recent years all over the world, especially in China and United Arab Emirates. A huge part of consumed energy is used for the cooling and heating of residential, domestic and commercial buildings, which have gained an increasing interest due to higher comfort standards (requirements - demands) (Daniel H. Rosen, Trevor Houser, May 2007).

Thermal comfort is defined by ASHRAE (American Society of Heating, Refrigerating and Air – Conditioning Engineers) as a state of mind that expresses satisfaction with the surrounding environment (ASHRAE standard 55).

Thermal comfort is affected by heat conduction, convection, radiation and evaporative heat loss. Thermal comfort is maintained when the heat generated by human metabolism is allowed to dissipate, thus maintaining thermal equilibrium with the surroundings. Any heat gain or loss beyond this generates a sensation of discomfort. It has been long recognised that the sensation of feeling hot or cold is not just dependent on air temperature.

In arid – hot climates space cooling is needed for commercial buildings throughout the year, which might be responsible for about half of the total energy use in such buildings – such a situation is being observed in Hong Kong (Milorad Bojic, 2004). In cities like Hong Kong space cooling is required for 7 months a year in a residential buildings, and has become the dominant energy end – use in this type of building since the mid–1990s. A similar trend towards space cooling is being observed in Shanghai and in Abu Dhabi, United Arab Emirates. Therefore reducing energy use for space cooling in buildings is a key measure in energy conservation and environmental protection in these areas. Energy efficiency in this case is not only a matter of designing energy efficient buildings. It is also a matter of designing less energy–consuming and environmentally harmless space cooling solutions like absorption chillers.

Residential and commercial buildings are usually provided with powered ventilation systems. Exhaust air from inside is replaced continuously with fresh outside air, which must be treated to fulfil the thermal comfort standards (temperature and humidity). The exchange of inside air for outside air requires the expenditure of energy. Even in well–insulated buildings, ventilation losses of energy (heat or cold) are always larger than energy (heat or cold) gains and losses through building's external walls.

6 SINGLE STAGE WATER/LITHIUM BROMIDE ABSORPTION MODEL DEVELOPMENT

6.1 Model Input Parameters

Model input parameters are based on SANYO and YORK absorption machines. Input variables are reliable and widely available.

6.1.1 Global Variables

P_{loss} – pressure loss [kPa/m]

6.1.2 Condenser

T_{15} – cooling water inlet temperature [°C]

T_{16} – cooling water outlet temperature [°C]

P_{15} – cooling water inlet pressure [kPa]

P_{16} – cooling water outlet temperature [kPa]

m_{15} – cooling water (mass) stream [kg/s]

U_c – overall heat transfer coefficient [kW/Km^2] = [kW/°Cm^2]

6.1.3 Evaporator

T_{12} – chilled water inlet (return) temperature [°C]

m_{18} – chilled water (mass) stream [kg/s]

U_e – overall heat transfer coefficient [kW/Km^2] = [kW/°Cm^2]

6.1.4 Absorber

T_{13} – cooling water inlet temperature [°C]

m_{13} – cooling water (mass) stream [kg/s]

U_a – overall heat transfer coefficient [kW/Km^2] = [kW/°Cm^2]

6.1.5 Generator

T_{11} – hot water inlet temperature [°C]

m_{11} – hot water (mass) stream [kg/s]

U_g – overall heat transfer coefficient [kW/Km^2] = [kW/°Cm^2]

6.1.6 Solution Heat Exchanger

U_{shx} – overall heat transfer coefficient [kW/Km^2] = [kW/°Cm^2]

6.1.7 Household Heat Exchanger

T_{20} – air outlet (return) temperature [°C]

T_{19} – air inlet temperature [°C]

U_{hh} – overall heat transfer coefficient [kW/Km^2] = [kW/°Cm^2]

R_{air} – relative humidity [%]

6.2 Model Assumptions

6.2.1 Global Assumptions

- The constant solution level in the desorber. Steady – state model
- Pressure drop in the piping system is constant [kPa/m^2]
- No heat loss in the piping network – ideal insulation
- Refrigeration and solution expansion valve are isenthalpic (h = const)
- Constant (high) pressure level in the generator and the condenser
- Constant (low) pressure level in the evaporator and the absorber
- No leaks in the low and high pressure circuits
- The solution pump is adiabatic, and it is used to maintain the constant solution (concentration) level in the generator (desorber)

6.2.2 Condenser

- Cooling liquid is (sub cooled) water
- Inlet to the condenser is superheated water vapour
- Outlet from the condenser is (sub cooled) water, at pressure P_{high}

6.2.3 Evaporator

- Outlet from the evaporator is saturated water vapour, at pressure P_{low}
- Inlet to the evaporator is water (vapour – liquid water state), vapour flashes as liquid passes through refrigeration expansion valve, thus enthalpy $h_8 \cong h_9$

6.2.4 Absorber

- Outlet from the absorber is saturated water liquid (sub cooled) weak water/lithium bromide solution, thus vapour quality is equal to 0.
- Liquid in cooling circuit is sub cooled water

6.2.5 Generator

- Refrigerant liquid (water vapour) is free of LiBr salt
- Refrigerant which leaves generator is (superheated - saturated) water vapour
- Generator is powered by hot water

6.2.6 Solution Heat Exchanger

- Efficiency (effectiveness) of solution heat exchanger is constant

- Strong water/lithium bromide solution on the inlet is saturated liquid solution, thus vapour quality is equal to 0.
- Weak water/lithium bromide solution on the outlet is sub cooled liquid solution, state calculated from solution heat exchanger model

6.3 Model Development for Single Stage Absorption Machine

6.3.1 Overall

The absorption machine model is based on energy and mass balance equations. Each component (condenser, refrigeration expansion valve, evaporator, absorber, solution pump, solution expansion valve, solution heat exchanger, generator and household heat exchanger) is a separated module with mass and energy balance equations. The major task is to determine (estimate) the area of the heat exchangers. The overall heat transfer coefficient for each heat exchanger is estimated. Properties of enthalpy, pressure, specific volume, specific heat of water and steam, water/lithium bromide fraction, temperature, steam quality are determined from property relations and checked in REFPROP. The low and high pressure levels are computed from the saturated pressure of the water vapour (all values are determined from the EES and REFPROP libraries). Specific heat formulas turned out to be crucial in the model development stage. It seems to be the most accurate pathway for such calculation.

See **Appendix E** – EES code

7 MODEL BEHAVIOUR AND OPTIMIZATION CRITERIA

The Absorption chiller model consists of many various factors, equations, assumptions and mutual relations. Due to this, the model is sensitive to variation of any of these factors and/or assumptions. Thus such behaviour must be analyzed and optimized. The optimization in parameters is based on the idea that some combination of parameter values produces the ideal model behaviour over its life time. The maximization (or minimization) of some values is the goal of the optimization process. The next reasonable step is to expand the idea to several objectives. The function (equation) can be formulated as a combination of goals that produces the optimal model behaviour over its life time.

After each planning and simulation stage the model is revised. Once more the model is planned and simulated and revised to obtain the optimal (or close to optimal) behaviour. The main criterion in the optimization process in this case is the cost (area) of heat exchangers, electricity, water and others.

State point no.	T [°C]	h [kJ/kg]	m [kg/s]	P [kPa]	C_p [kJ/kg°C]	q [%]
1	34,21	77,65	11,0000	0,900		0
2	34,21	81,69	11,0000	6,671	2,066	0
3	58,72	132,37	11,0000	6,671	2,070	0
4	82,24	197,99	10,0522	6,671	1,925	0
5	53,43	142,53	10,0522	6,671	1,924	0
6	44,95	142,53	10,0522	0,900		58,6
7	71,26	2632,85	0,9478	6,671	1,895	100
8	38,12	159,62	0,9478	6,671	1,902	0
9	5,45	159,62	0,9478	0,900		5,5
10	5,45	2510,53	0,9478	0,900		100
11	86,00	360,23	110,0000	200,000		0
12	79,45	332,69	110,0000	149,338		0
13	28,00	117,48	60,0000	200,000	4,183	0
14	39,79	166,74	60,0000	149,338		0
15	28,00	117,48	62,0000	200,000	4,183	0
16	37,04	155,12	62,0000	149,338		0
17	12,00	50,55	100,0000	200,000	4,185	0
18	6,98	29,57	100,0000	250,663	4,193	0
19	35,00	85,30	119,7449	111,000	1,043	
20	18,00	34,41	119,7449	111	1,019	

Table 9 State points in 2100 [kW] ASB model

As it was mentioned, optimal (in the case of the ABS project) means the lowest possible cost. However, low initial cost could be obtained only with high annual operational cost. The cost of the ABS system is determined mainly by the cost of heat exchangers (evaporator, absorber, solution heat exchanger, generator and condenser).

7.1 LiBr-H₂O Influence

7.1.1 LiBr – H₂O Influence on COP
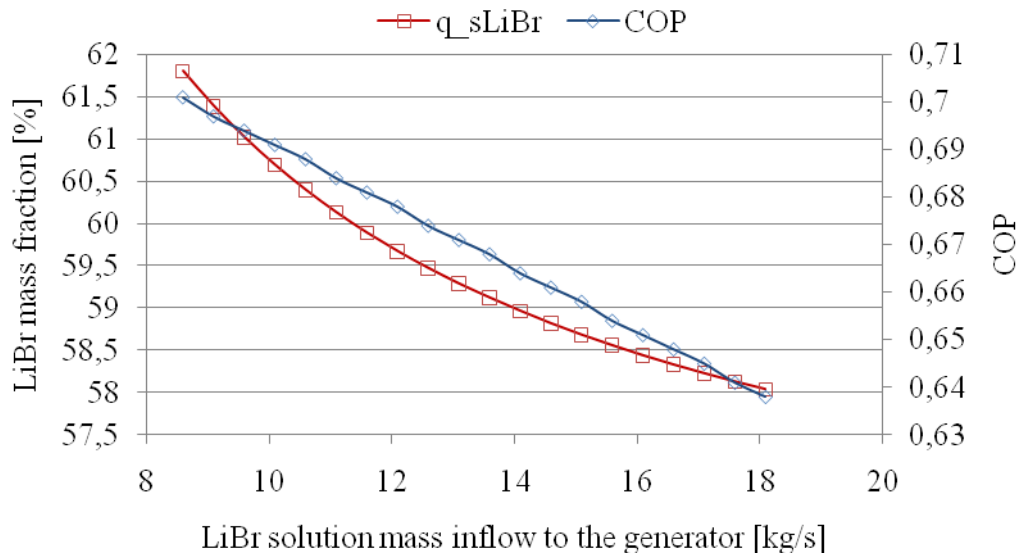
Figure 23 Variation of COP and LiBr mass fraction as a function of LiBr-H₂O mass flow.

The mass fraction can be defined in many different ways – including mole, mass fraction, concentration, etc. In this analysis mass fraction is defined as:

$$x_{LiBr} = \frac{mass \ \ of \ \ one \ \ component \ \ [kg]}{total \ \ mass \ \ of \ \ both \ \ components \ \ [kg]} \left(LiBr\% \ \ fraction \right)$$

(7.1.1)

It is important to note that COP is a strong function of the LiBr mass fraction (plot above). However due to crystallization possibility, it is impossible to obtain high COP. For given conditions (table 9):

-   As the LiBr mass fraction increases, the COP increases,

It occurs because of the extreme hygroscopic character of lithium bromide, which is used as a desiccant in absorption chiller machines. Hygroscopy is the ability of a substance to attract water molecules from the surrounding environment (generator and absorber). Therefore, the higher the LiBr concentration, the smaller the amount of water vapour that can be absorbed by the LiBr solution, thus a greater mass of water vapour can go to the condenser and evaporator to produce a cooling effect (as the water vapour stream increases, the cooling capacity and COP increase). The COP equation contains pump work and LiBr mass fraction is greater as the LiBr weak solution mass flow is smaller, thus COP increases as pump work decreases (eqn. 1.4.1), although the work of the pump is often very small compared to the heat input.

7.1.2 LiBr – H₂O Influence on Generator and Absorber Area
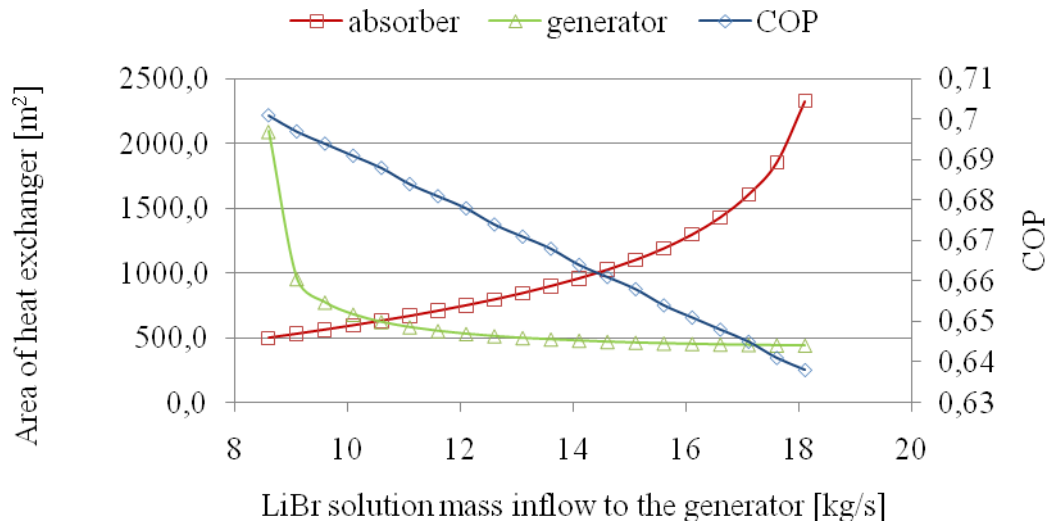
Figure 24 COP and heat exchanger area variations as a function of LiBr-H₂O mass flow.

Capital cost is the major criterion in the case of ABS systems, due to the high initial cost of their heat exchangers. An absorption machine consists of shell & tube and plate heat exchangers. Generator and absorber cost account for the greatest contribution to the initial costs of the absorption unit (table 11); it is mostly due to high unit cost [€/m²] of the shell & tube heat exchanger and the large area of both heat exchangers. Therefore the optimal (minimum) area of shell & tube heat exchangers must be found. The optimization process resulted in:

- Minimum area 1252 m² (shell and tube heat exchanger area),

- COP equal 0,684,

- LiBr mass fraction equal 60,134%,

- LiBr mass flow (weak solution) 11,1 kg/s,

- Circulation ratio equal to 11,71,

It is important to note that high (maximum) COP value does not assure an optimal (minimal) capital and operational cost of the venture.

7.2 Cooling Water Influence

7.2.1 Condenser Cooling Water Influence on COP and Chilled (district cooling) Water Temperature

The optimum design is considered to be the design which has a competitive capital cost and the lowest annual cost. The cost of brine and cold water has significant influence on annual cost.
Figure 24 indicates the condenser cooling water's influence on COP and chillerd (district cooling) water temperature. It is important to note that the figure presents the variation of

COP and chilled water temperature as a function of the solution heat exchanger efficiency for different cooling water mass flow. The figure can be described as follows:

- As the solution heat exchanger efficiency increases, the COP increases,
- As the cooling water mass flow increases, the chilled water outlet temperature decreases,
- As the cooling water mass flow increases, the COP increases,

For a given solution heat exchanger efficiency, several COP values can be obtained for one cooling water mass flow value.
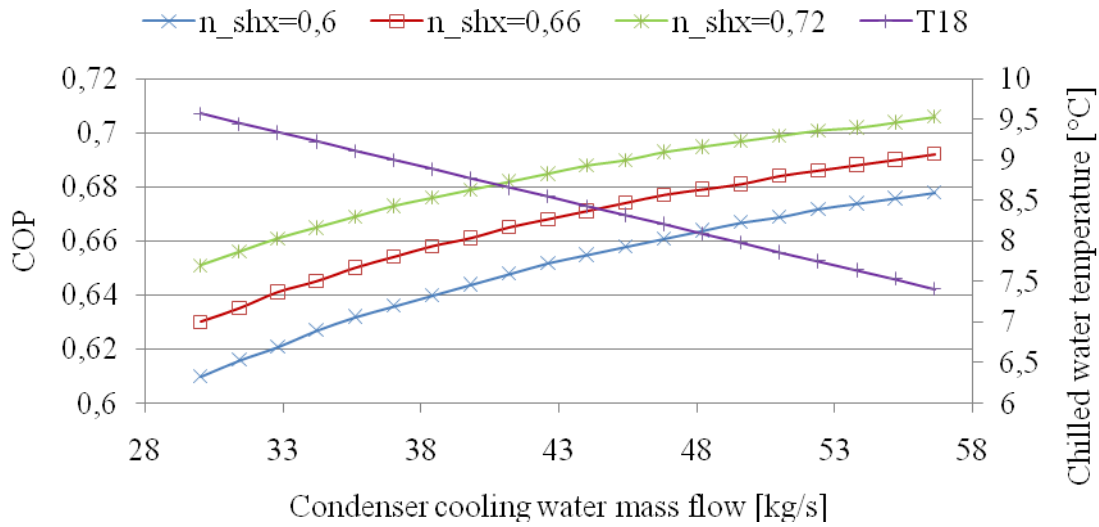
Figure 25 COP and chilled water temperature variations as a function of condenser cooling water mass flow and solution heat exchanger efficiency.

7.2.2 Condenser Cooling Water Influence on Solution Heat Exchanger Surface (and COP)

It was properly noted above that for a given solution heat exchanger efficiency, several COP's can be obtained. The COP of the absorption unit is the ratio of the change in energy flow at the output (evaporator), to the supplied energy flow (generator and pump). However in terms of annual cost, the condenser cooling water mass flow and solution heat exchanger area are much more important factors of total annual cost. Due to that, figure 25 can be considered as follows:
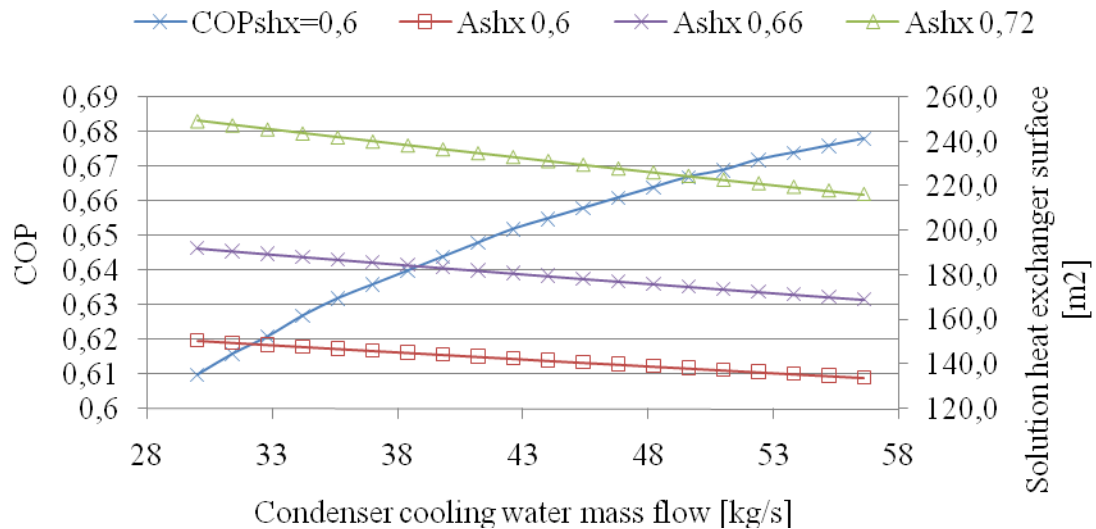
Figure 26 COP and solution heat exchanger surface variations as a function of condenser cooling water mass flow for different solution heat exchanger efficiency values.

- As the solution heat exchanger efficiency increases, the area of the solution heat exchanger increases,

- As the area of solution heat exchanger increases, the capital and annual cost of ABS chiller system increases,

- As the condenser cooling water mass flow increases, the solution heat exchanger area decreases,

- As the solution heat exchanger efficiency increases by 0,12, the area of solution heat exchanger increases around 70 [m^2], and COP increases around 0,05,

- As the area of the solution heat exchanger increases to 70 [m^2], the capital cost increases to 43 650 €, and the annual cost increases to 4 808 [€/year]. A COP increase of 0,05 is twice as beneficial (increase of total annual cost 2 415 [€/year]) compared to an increased solution heat exchanger area.

7.3 Chilled water influence

7.3.1 Chilled Water Influence on Evaporator Surface

The figure below represents a comparison between different chilled water (return) temperatures obtained in a simulation for COP = 0,684 with chilled water mass flow as a variable.

The figure can be described as follows:

- As the chilled water mass flow increases, the evaporator area decreases,

- As the chilled water (return) temperature increases, the minimum chilled water mass flow increases

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- As the chilled water (return) temperature increases, the evaporator area decreases,

- As the chilled water (return) temperature increases by 1 [°C] due to temperature loss in the piping system, the chilled water mass flow must be increased to 22 [kg/s] (evaporator area equal 300 [m^2], T_{17} = 15[°C]) to compensate for a 1 °C temperature loss.
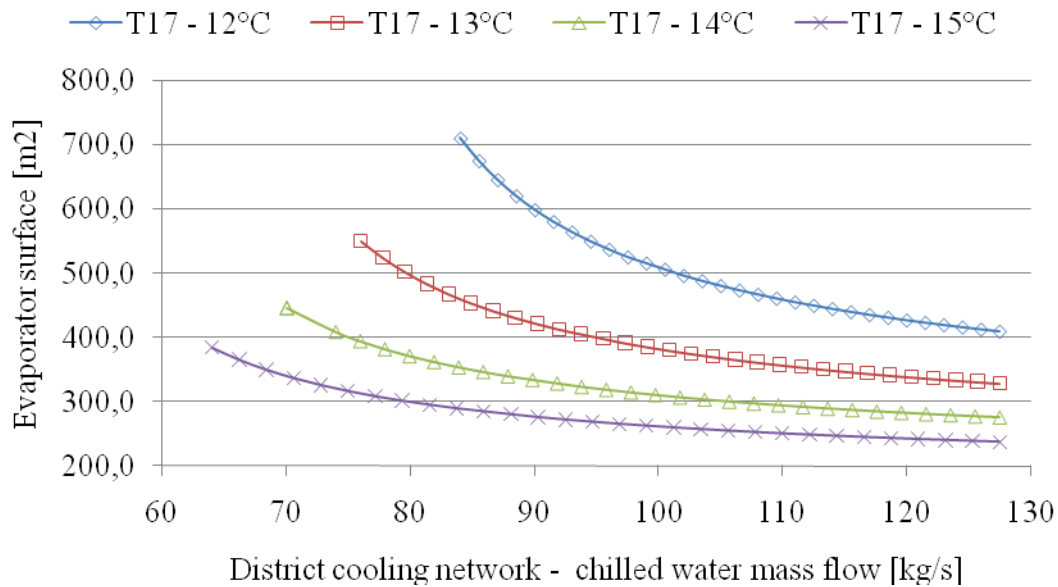
Figure 27 Evaporator area as a function of chilled water mass flow for different chilled water (return) temperatures.

As the chilled water (return) temperature increases, the evaporator area decreases. This is caused by an increase in the difference between the evaporator's outlet and inlet temperature ($\Delta T_{18,17} = T_{18} - T_{17}$). As the temperature difference increases, the heat flow through the evaporator surface increases and the total evaporator surface decreases.

7.3.2 Chilled Water Influence on Evaporator Outlet (DC inlet) Water Temperature

Figure 27 can be described as follows:

- As the evaporator outlet water temperature increases, the chilled water mass flow decreases (at evaporator outlet temperature),

- As chilled water mass flow decreases, the evaporator outlet temperature decreases,
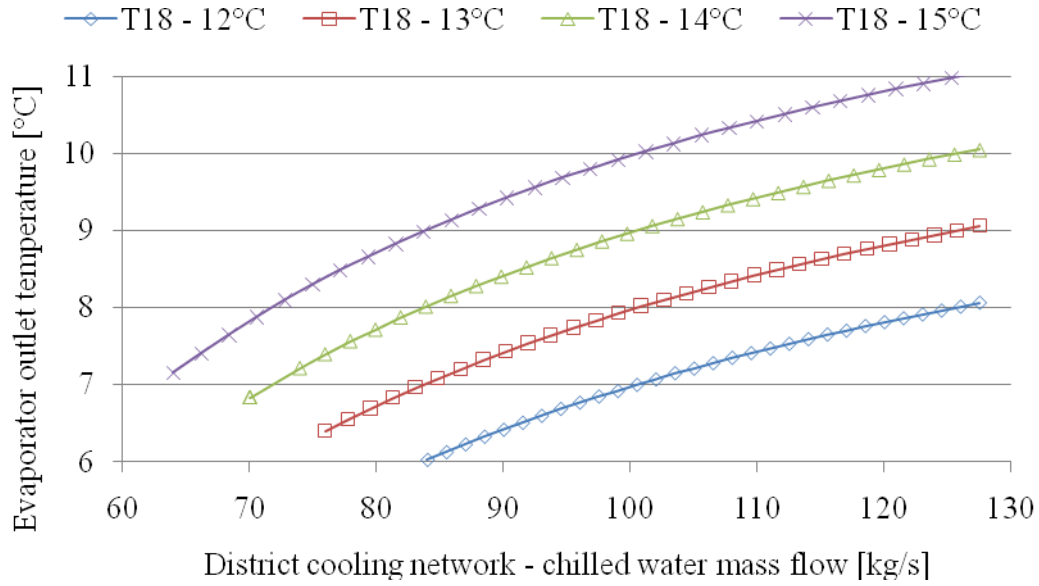
Figure 28 Evaporator's outlet temperature as a function of chilled water mass flow for different chilled water (return) temperatures.

Increasing the chilled water outlet temperature (inlet to the DC system) can significantly decrease the evaporator surface and chilled water mass flow. However, the chilled water (return) temperature can vary due to the cooling load to the building, ambient temperature, humidity, atmospheric pressure, etc. In general, DC systems are designed to work in supply/return temperature difference ΔT of 7 °C to 8 °C, so they can reach a DC inlet temperature of 4 °C to 7 °C and DC return temperature of 11 °C to 15 °C.

7.4   Hot Water (brine) Influence

7.4.1   Brine Influence on Generator Surface and Efficiency

Figure 28 presents generator surface and efficiency variations as a function of the brine mass flow. The figure can be described as follows:

- As the brine mass flow decreases, the generator efficiency decreases,

- As the brine mass flow decreases, the generator surface increases, due to constant value of total heat flow,

The optimum between generator area and brine mass flow occurs around efficiency equal to 0,43 – 0,47. Generator efficiency should not be higher, due to the area and cost of shell & tube heat exchangers. As above, the major optimization criterion for generator surface, efficiency and brine mass flow was annual cost.
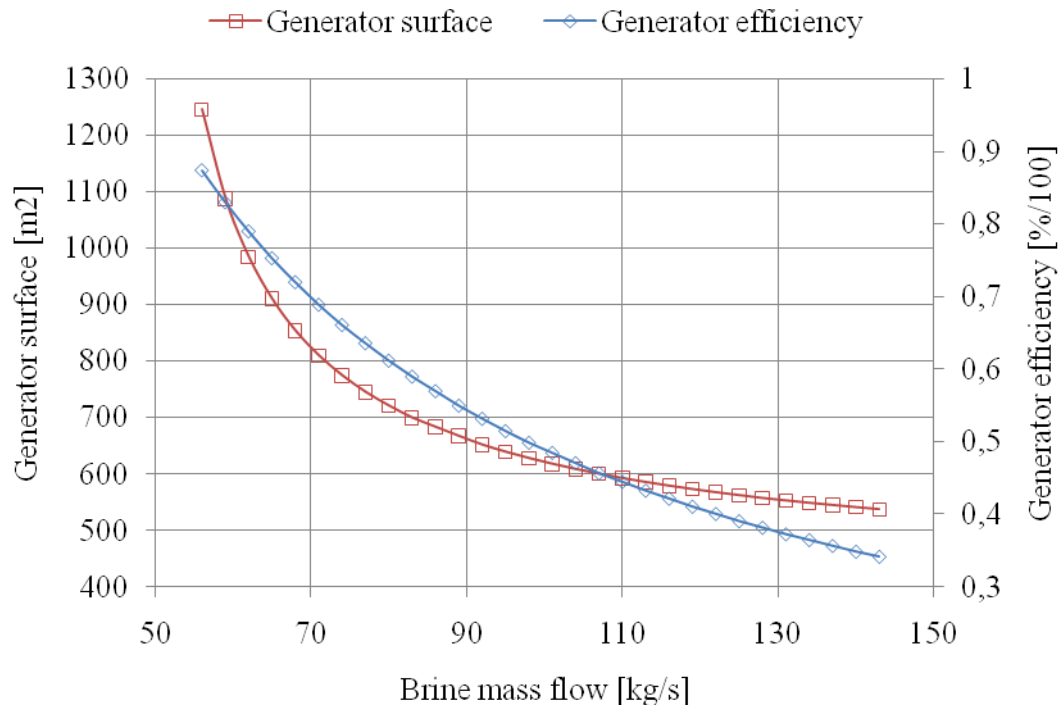
Figure 29 Generator efficiency and area as a function of brine mass flow.
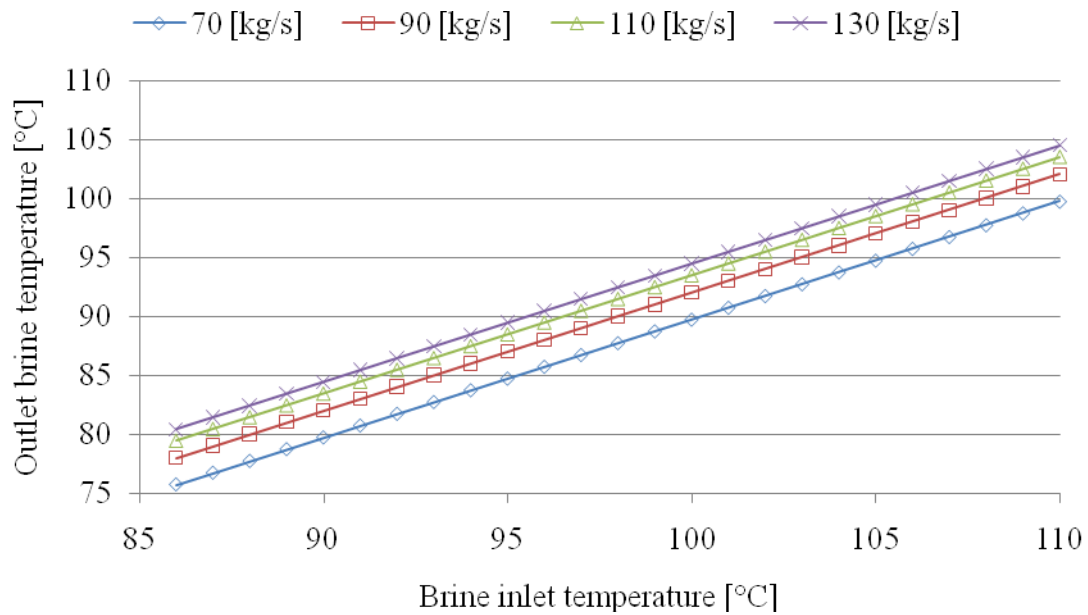
Figure 30 Outlet brine temperature as a function of brine outlet temperature for different brine mass flows.

The figure above presents the variation of the outlet brine temperature as a function of brine inlet temperature for different brine mass flows. This figure can be described as follows:

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- As the brine mass flow increase, the temperature drop (inlet – outlet) decrease,

A temperature drop in mass flow of 70 [kg/s] is equal to 10 [°C], for 130 [kg/s] it is equal to 5-6 [°C]. It is important to note that the cost of brine can be measured (indicated) in [€/m³°C], thus it is important to obtain an outlet brine temperature at a reasonable level. However, this issue is strongly dependent on brine cost ([€/m3] or [€/m³°C]).

7.4.2 Brine Influence on Generator Surface

The figure presents the variation of generator (shell and tube) heat exchanger area as a function of brine temperature for different brine mass flows. This figure can be described as follows:

- As the brine inlet temperature increases, the heat exchanger area decreases,

- As the brine mass flow increases, the heat exchanger area decreases,

It is important to note that generator surface is a strong function of brine inlet temperature. Therefore, to obtain low annual cost, generator surface, brine temperature and mass flow must be considered as mutual functions (relationship). In the cost analysis chapter it is shown how strong annual cost is dependent on brine mass flow and temperature.
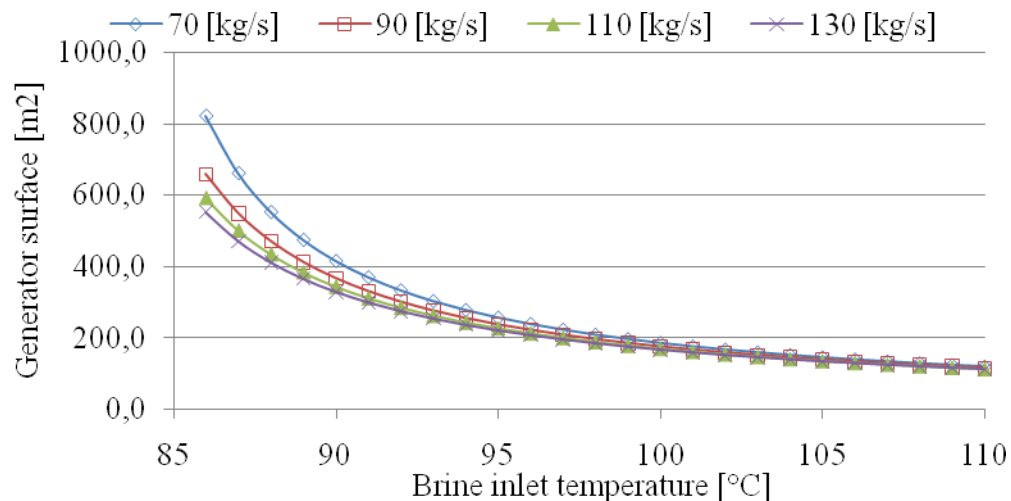
Figure 31   Generator surface as a function of brine inlet temperature for different brine mass flows.
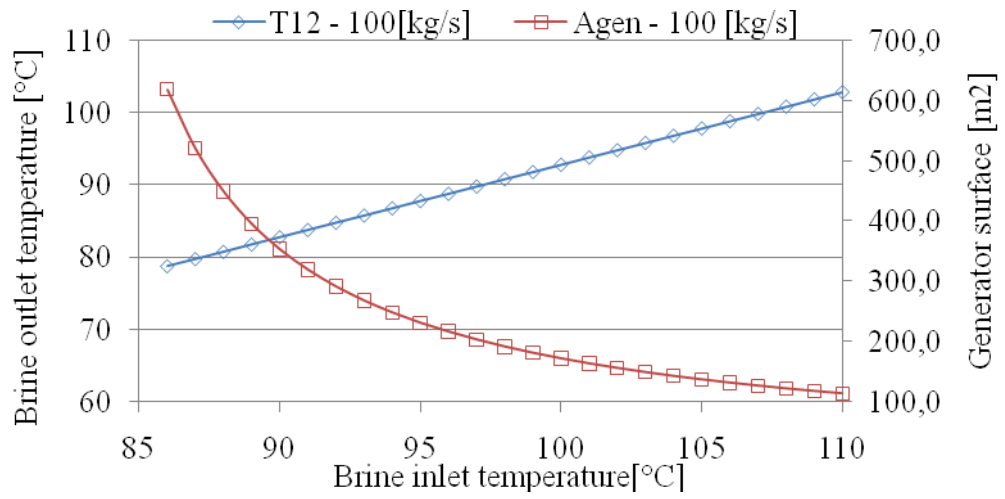
Figure 32 Brine outlet temperature and generator surface as a function of brine inlet temperature.

8 COST ANALYSIS

8.1 Investment costs of ABS

The costs of an absorption chiller plant (system) are investment costs and operational costs. The investment costs are mainly the cost of equipment (heat exchangers, pumps, valves) and piping system, mounting cost, and cost of the control system (sensors, PLC's, and other parts of control system). Operational costs are related to the operation of the system.
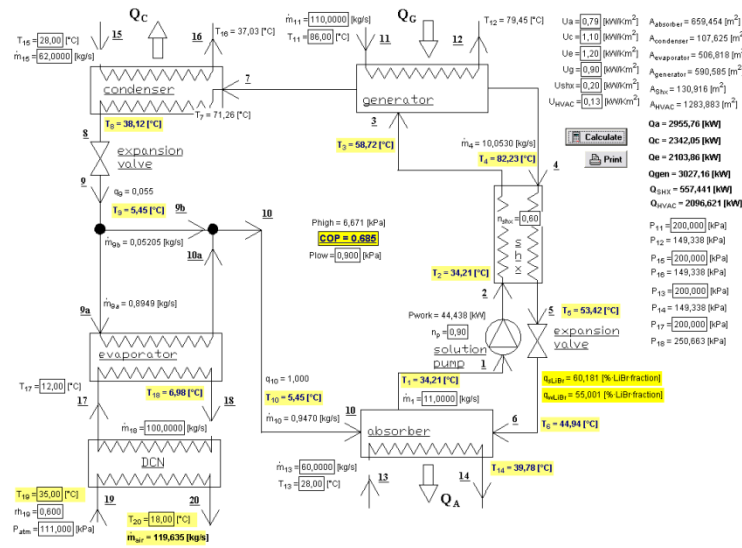
*Figure 33 Absorption chiller model (see **Appendix E**)*

The investment costs of heat exchangers are directly related to the required size of heat exchangers. The sizes of the modelled ABS systems (1000 [kW] and 21000 [kW]) are listed below, as well as the assumed approximation equations used to obtain these heat exchanger sizes.

Equipment	U value [kW/m²°C]	Type of heat exchanger	Size [m²] 1000 [kW]	Size [m²] 2100[kW]
Absorber	0,79	Shell and tube	258,6	638,1
Solution hex.	0,20	Plate	61,8	131,0
Generator	0,90	Shell and tube	274,3	588,0
Condenser	1,10	Plate	51,9	107,2
Evaporator	1,20	Plate	239,3	498,9
Solution pump	-		20,6 [kW]	44,4 [kW]
Household hex.	0,13	-	592,8	1147,1

Table 10 Heat exchanger sizes, pump size and overall heat transfer coefficient values.

The capital cost for equipment was calculated according to the equations below (8.1, 8.2, 8.3) (Olafsson, 2007):

$$I_{HeatExchanger\_0,1} = C_{0,1} \cdot A^{0,8}$$

(8.1)

$$I_{Pump} = C_2 \cdot (0,01 \cdot P_{work})^{0,6}$$

(8.2)

$$C_{PE,Y} = C_{PE,W} \cdot \left[\frac{X_Y}{X_W} \right]^{\alpha}$$

(8.3)

where:

I – investment cost of equipment,

C_0 – constant, indicating cost for each [m^2] of plate heat exchanging area,

C_1 – constant, indicating cost for each [m^2] of shell and tube heat exchanging area,

C_2 – constant, indicating cost for each 100 [kW],

A – area of heat exchanger [m^2],

P_{work} – Pumping power [kW],

$C_{PE,Y}$ – cost for purchased equipment Y,

$C_{PE,W}$ – cost for purchased equipment W,

X_Y – key variable for component involved, power in boilers, compressors, area of heat exchangers, volume of tanks

α – for each piece of equipment, $\alpha_{evaporator} = 0,54$; $\alpha_{compressor} = 0,95$,

The formula (8.3) is valid for pre–specified ranges, for:

- Evaporators: 10 – 1000 [m^2],

- Compressors: 0,05 – 8 [MW],

The estimated values C_0, C_1 and C_2 are listed below:

C0 – 623,58 €/m^2, for plate heat exchanger,

C1 – 902,35 €/m^2, for shell and tube heat exchanger,

C2 – 11 717,00 €/100 kW, not linear,

The capital investment costs were estimated and are presented in the table below (table 11). The table contains three different evaluation scenarios. Due to different cooling water inlet temperatures to the condenser and the absorber (cold water has significant influence on cooling capacity and COP). Abu Dhabi and Shanghai have hot – humid climate, which significantly increases the number of operation hours during the year.

Equipment	U value [kW/m²°C]	Type	Capital cost [€] 1000 kW	Capital cost [€] 2100 kW
Absorber	0,79	Shell and tube hex.	76 822,5	158 174,9
Solution hex.	0,20	Plate hex.	16 880,4	30 809,7
Generator	0,90	Shell and tube hex.	80 534,4	148 034,6
Condenser	1,10	Plate hex.	14 492,5	26 277,1
Evaporator	1,20	Plate hex.	49 897,4	89 727,5
Pump	-	-	2 474,2	5 206,8
Valve (both)	-	-	2 800,0	2 800
Total price for equipment	-	-	243 901,3	461 030,5

Table 11 Estimated price for heat exchanger and pumps.

I. Fixed capital investment (FCI)	1000 kW [€]	2100 kW [€]
A. Direct cost (DC)		
1. Onsite cost (ONSC) – Purchased equipment cost (PEC)		
1. Heat exchangers	238 627,1	453 024,0
2. Pump	2 474,2	5 206,8
3. Pipes in system (5% of 1+2)	12 195,1	23 051,5
4. Electrical control and monitoring system (30% of 1+2+3)	76 828,9	145 224,6
Total Onsite cost	332 925,3	629 306,7
2. Offsite cost (OFSC)		
1. Civil, structural and architectural work (20% of ONSC)	66 585,0	125 861,3
2. Service facilities (hot source and cold sink connection) (25% of ONSC)	82 231,3	157 326,7
3. Contingencies (15% of ONSC)	49 938,8	94 396,0
Total Direct cost (DC)	532 680,4	1 006 890,6
B. Indirect cost (IDC)		
1. Engineering and supervision (15% of DC)	79 902,0	151 033,6
2. Construction cost including contractor's profit (15% DC)	79 902,0	151 033,6
3. Contingencies (20% of DC)	106 536,0	201 378,1
Fixed capital investment, total (FCI)	799 020,5	1 510 355,9
II. Other outlays		
A. Start up cost (6% of FCI)	47 941,2	90 620,1
B. Working capital (5% of FCI)	39 951,0	75 516,8
C. Cost of licensing, research and development	20 000	25 000
Total capital investment (TCI)	906 912,7	1 701 472,8

Table 12 Breakdown of the Total Capital Investment cost for 1000 kW and 2100 kW ASB unit.

8.2 Annual Capital Cost – Annuity

The capital cost of investment can be divided into annual cost for each operating year. The term annuity is used to refer to any terminating stream of fixed payments over a specified period of time. The annuity or annual fractional payment of a project or component is calculated in the following manner:

$$An_j = C_{FC,j} \cdot \left(\frac{i_{eff}}{1 - \left(1 + i_{eff}\right)^{-n}} \right)$$

(8.2)

where:

An_j – Annuity (annual capital cost) for project j or component j

$C_{FC,j}$ – Fixed cost for project j or component j,

i_{eff} – annual effective rate of return,

n – number of years in which project j or component j is operated,

To establish annuity, the following will be assumed:

Type	n	i_{eff}
Absorption chiller	25	0,1
Centrifugal chiller	10	0,1

Table 13 Annual capital cost assumptions.

Power capacity	1000	2100	[kW]
Annual Onsite cost	36 667,7	69 329,5	
Annual Direct cost	58 684,4	110 927,2	
Annual FCI cost	88 026,5	16 641,5	[€/year]
Annual Other outlays cost	11 886,3	21 057,2	
Total annual cost (ABS unit)	99 912,8	187 448,0	
Total annual cost (Compression unit)	72592,7	130792,8	

Table 14 Estimated annual costs (Only initial costs are included – table 14).
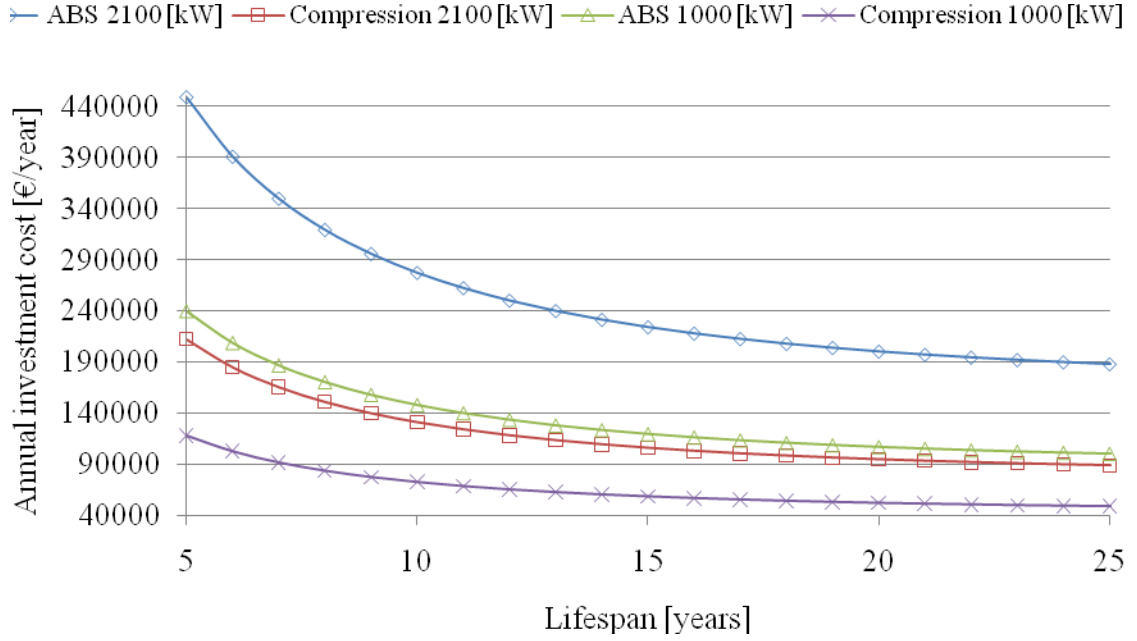
Figure 34 The annual investment cost variations as a function of lifetime for different cooling capacity values.

Figure 33 presents the variation of annual investment cost as a function of the lifetime, for a given evaporator cooling capacity (1000 kW and 2100 kW). The figure can be described as follows:

- As the cooling capacity increases, the annual investment cost increases. However, the annual investment cost per installed kW is constant or slightly decreases.

8.3   Operational Cost

The largest operational cost of an absorption plant is related to the cost of electricity, cost of cooling water (condenser and absorber) and cost of hot water (brine). Other significant costs are related to maintenance.

8.3.1   Cost of Electricity

To obtain the annual cost of electricity, the formula below (8.3.1) was used (Olafsson, 2007):

$$E_C = \left[\frac{10 \cdot \left(\dot{m}_{brine} + \dot{m}_{cooling} \right) \cdot 9{,}81}{0{,}6 \cdot 1000} + W_P \right] \cdot hr \cdot P_e$$

(8.3.1)

where:

E_C – electricity cost [€],

m_{brine} – stream of geothermal water (brine) [kg/s],

60

$m_{cooling}$ – stream of the cooling water (condenser and absorber) [kg/s],

W_P - power of the pump [kW],

hr – maximum utilization hours (dry bulb temperature above 20 °C/18 °C) [h],

P_e – price of electricity [€],

Total cost of electricity depends on the total cooling water, brine and solution mass flow.

Type	Unit	2100 [kW]	
$m_{absorber}$	[kg/s]	60,00	
$m_{condenser}$	[kg/s]	62,00	
m_{brine}	[kg/s]	110,00	
$m_{solution}$	[kg/s]	11,00	
$W_{abs+cond}$	[kW]	8,55	
W_{brine}	[kW]	7,71	
$W_{solution}$	[kW]	44,44	
hr	[h]	3383	7189
Annual average price (only pumps)	[€/MWh]	67,53	72,67
E_C	[€/year]	13 830,2	31 711,2

Table 15 Absorption chiller mass flow agreements.

The total annual cost consists of electricity cost, hot and cold water costs, operation and maintenance costs and annual capital cost. The annual cost of electricity depends on the amount of operating hours during the year and the local price of electricity. Due to significant electricity consumption, the price of electricity is an important factor.

It is assumed that both systems operate at the same COP at full and partial load. Such assumptions are almost true in both cases. The COP of a compression chiller is assumed to be COP = 5,5 for all loads (old technologies have COP values around COP = 3,0).

Location	Abu Dhabi	Shanghai	Abu Dhabi	Shangai
Type	Absorption chiller		Compression chiller	
Cooling capacity Q_e [kW]	2100			
Use of pumps/compressors [kW]	44,44		382,00	
Use of auxiliary pumps/fans [kW]	16,26		75,00	
Use of control system, etc. [kW]	31,51			
Max instantaneous load [kW]	93,21		489,51	
Annual electricity consumption [kWh]	670 069,0	315 321,4	3 519087,4	1 656012,3
Annual average price [€/MWh]	72,67	67,53	72,67	67,35

Annual cost [€/yr]	48 693,9	21 293,7	255 732,1	111 830,5
Annual CO_2 emission [ton CO_2/yr]	355,14	167,12	1865,12	877,67

Table 16 Annual cost and CO_2 emission.

Pressure losses in heat exchangers (absorber, condenser and generator) are estimated at 1 bar (10 [m]). Absorption chiller annual carbon dioxide emissions are around five times smaller than the amount of CO_2 emitted by compression chillers (with high COP = 5,5). Older compression machines with COP's in range of COP = 3, emit around ten times more CO_2 than absorption chillers, due to almost ten times higher electricity consumption.

8.3.2 Cost of Hot Water (brine)

To obtain the annual cost of hot water (brine), the formula below (8.3.2) was used:

$$E_{brine} = P_{brine} \cdot m_{brine} \cdot hr \qquad (8.3.2)$$

where:

P_{brine} – cost of how water (brine 86 °C) [€/m^3],

E_{brine} – hot water (brine cost) [€/yr],

The cost of hot water has to be estimated. It is assumed that hot source already exist and its temperature is above 86 °C (generator inlet temperature).

Due to the fact that ΔT_{abs}=6,56 °C, the cost of hot water is calculated as the cost for m^3 °C. Absorption systems can be coherent parts of district heating and cooling systems, however absorption units do not utilize the whole exergy of geothermal brine. Thus the cost of brine (hot water in a district heating network) is calculated as follows: cost of hot water (86 °C) times $\Delta T_{abs}/\Delta T_{DHS}$ (5,56 °C/45 °C).

The present sale price is 4,5 RMB/m3 in the market, where 8,3 RMB is equal 1 USD (Kyushu - Tohoku, 2000). Ratio USD/€ is equal 0,7817 (16 February 2009) thus cost of cold water in €/m^3 is equal:

$$P_{brine} = \frac{4,5}{8,3} \cdot 0,7817 \cdot k_n^{\,ny}$$

$$(8.3.3)$$

where:

k_n – annual cost growth rate, is assumed to be 10%/year,

ny – number of years (2000 to 2009),

thus:

$$P_{brine} = 0,9993 \ [€/m3]$$

Cooling capacity	2100	[kW]
Price of hot water (brine)	0,9993	[€/m^3]
Price of hot water (brine) for ΔT=6,56°C	0,1457	[€/m^3]

Usage	396	[m³/h]
Number of operating hours	7189	3383	[h]
Annual cost	414 716	195 189	[€/yr]
Temperature drop	6,56	°C

Table 17 Hot water – Annual average cost.

8.3.3 Cost of Cooling Water

To obtain the annual cost of cooling water (condenser and absorber), the formula below (8.3.3) was used:

$$E_{cooling} = P_{coldH_2O} \cdot m_{cooling} \cdot hr$$

(8.3.4)

where:

$E_{cooling}$ – cooling water cost [€],

$P_{coldH2O}$ – cooling water price [€/kg],

	Abu Dhabi	Shanghai	Unit
Cooling capacity	2100		[kW]
Price of cold water	0,1407		[€/m³]
Usage (absorber and condenser)	439,2		[m³/h]
Number of operating hours	7189	3383	[h/year]
Annual cost	444247	209054	[€/year]

Table 18 Cooling water – Annual cost

The cost of cold water is taken from http://www.csinvest.gov.cn/tzcs_jycb_gysy.asp, where the cost of cold water is equal to 0,18 USD/ton. The ratio of USD/€ is equal 0,7817 (16 February 2009), thus the cost of cold water in €/m³ is equal 0,1407. "The cost of industrial water is currently USD 0,18/ton, inclusive of a waste water discharge fee" (http://www.csinvest.gov.cn/tzcs_jycb_gysy.asp).

The cost of cold water is assumed to be the same in the case of Shanghai (data source) and Abu Dhabi, due to hurdles with the availability of such data.

It is assumed that a source of cold water exists and the source temperature is rather constant throughout the year. Cold water is pumped through the condenser and absorber (with 0,5 bar pressure loss).

8.3.4 Operational and Maintenance Cost (O&M)

To obtain the annual O&M (Operation and Maintenance) cost, the formula below (8.3.5) was used:

$$E_{O\&M} = TCI \cdot \xi + \left(\frac{E_{research} \cdot Q_e}{1000} \right)\left(\frac{hr}{8760} \right)$$

(8.3.5)

where:

> TCI – total capital cost [€],
>
> $E_{O\&M}$ – operation and maintenance cost [€],
>
> $E_{research}$ – cost of research, license and development,
>
> ξ - annual operational cost is considered to be equal to 0,5% in the case of ABS, and 3% in the case of compression chillers.

Absorption units contain only one moving device (solution pump), which is controlled by computer, thus operation and maintenance becomes less time consuming with compression systems. Compression chiller units require regular inspections due to refrigerant leakage from the circuit; also the compressor, which is the major part in the system, requires regular inspections and changes of its oil filter.

Type	Absorption chiller		Compression Chiller		Unit
Power capacity	2100				[kW]
Number of operating hours	7189	3383	7189	3383	[h/year]
Operational cost	43 072,9	20 269,3	43 203,3	20 330,6	[€/year]
Personnel cost	8 507,4	8 507,4	51 785,6	51 785,6	[€/year]
Total O&M cost	51 580,3	28 776,6	94 117,1	71 313,4	[€/year]

Table 19 O&M – Annual cost.

8.4 Total Annual Cost

The total annual cost is the sum of all of the annual costs which are listed in the table below. The obtained prices (results) can vary from real values due to a lack of information about current prices of electricity, hot water cold water and others used in calculations.

|	Absorption chiller		Compression chiller		Unit
Power Capacity	2100				[kW]
Number of operating hours	7189	3383	7189	3383	[h]
Annual capital cost	187 448		130 792,8		[€/year]
Electricity cost	48 693	21 293	255 732,1	111 830,5	
Hot water (brine) cost	414 716	195 189	-		
Cold water cost	444247	209054			
O&M cost	51 580	28 776	94 117,1	71 313,4	
Total cost	1 146 630	641 760	480642,0	313937,0	

Table 20 Breakdown of total annual costs.
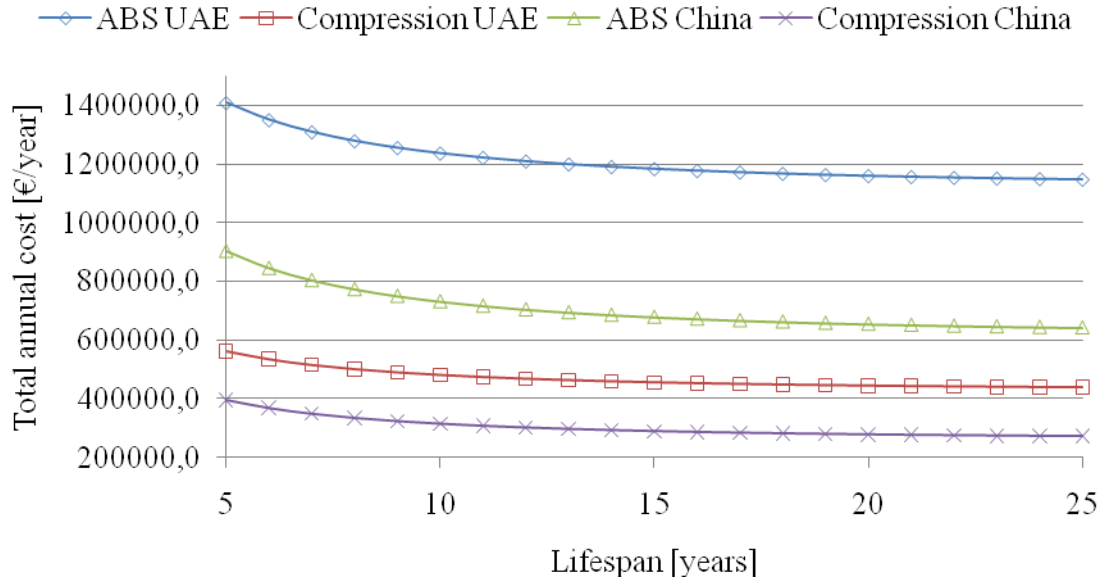
Figure 35 Total annual cost variations as a function of lifetime.

Figure 34 presents the variation of the total annual cost as a function of lifetime. The figure can be described as follows:

- The total annual cost of an ABS unit for Abu Dhabi and Shanghai is greater than the total annual cost of the compression unit for Abu Dhabi and Shanghai. It is mostly due to the fact that the ABS unit consumes great amounts of cold and hot water, which significantly increases the total annual cost,

- As the total amount of operating hours increases, the total annual cost increases (Abu Dhabi 7189 hours, Shanghai 3383 hours),

- As the lifetime increases, the total annual cost decreases. It is assumed that the lifetime of an ABS chiller is 15 years greater than that of a compression chiller (Table 20)

8.5 Possibility of Cost Optimization

It is important to note that absorption systems consume large quantities of water (both cold and hot), thus their costs largely contribute to the total annual cost.

	Absorption chiller		Compression chiller		Unit
Power Capacity	2100				[kW]
Number of operating hours	7189	3383	7189	3383	[h]
Annual capital cost	187 448,0		130 792,8		[€/year]
Electricity cost	48 693,9	21 293,7	255 732,1	111 830,5	

Hot water (brine) cost	41471,6	19518,9		
Cold water cost	44424,7	20905,4	-	
O&M cost	51 580,3	28 776,6	94 117,1	71 313,4
Total cost	373618,5[*]	277942,6[*]	480642,0	313937,0

Table 21 Total and detailed annual cost for 2100 [kW] chiller system.

[*] - Water price is assumed to be ten times smaller than the case in table 24 and figure 34.

	ABS UAE	CEN UAE	ABS CHINA	CEN UAE
Total annual cost =	373618,5	480642,0	277942,6	313936,7
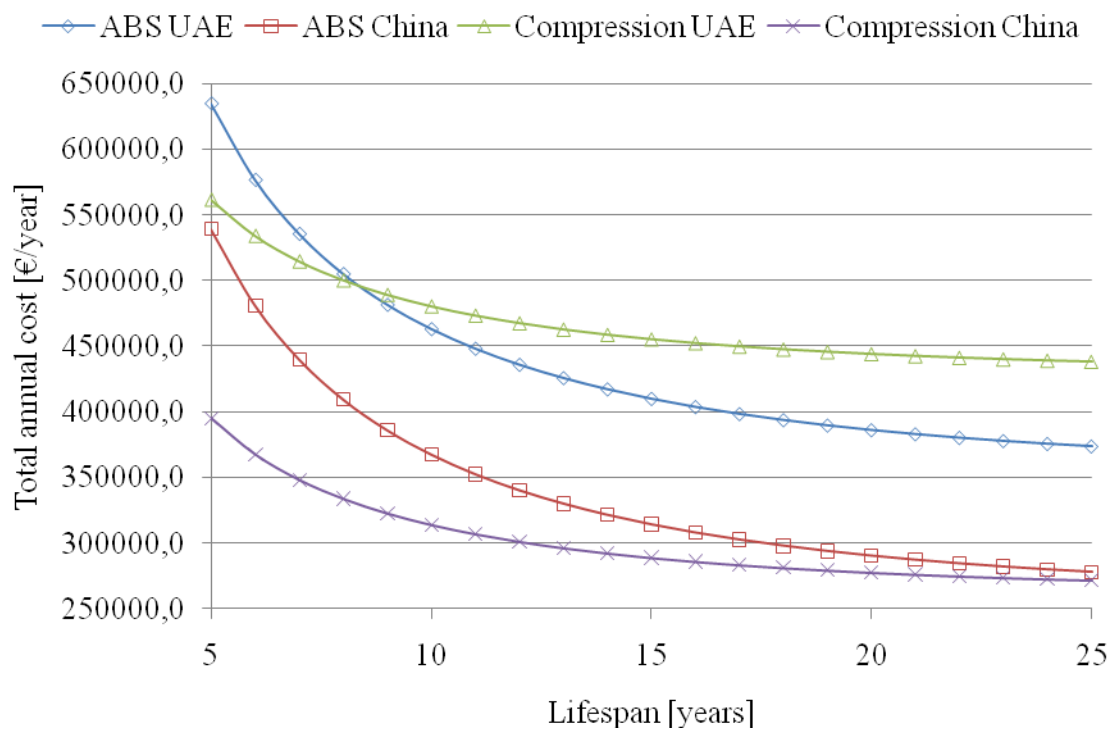
Figure 36 Total annual cost variation as a function of lifetime.

Figure 35 presents the variation of the total annual cost as a function of lifetime. In the case of normal (table 20) cost of hot and cold water an ABS system is not competitive with a compression system. The figure can be described as follows:

- As the hot and cold water price decreases, the ABS system competitiveness increases,

- As the investment lifetime increases, the total annual cost decreases,

- As the total amount of operating hours increases, the ABS system competitiveness increases,

The figure is clear–cut evidence that ABS technology can be competitive with compression technology, but only under some circumstances.

9   FURTHER STEPS – COST OPTIMIZATION (ESTIMATION)

During the last two chapters an absorption chiller has been extensively analyzed and discussed. However, not all of the optimization possibilities have been analyzed and discussed, therefore a few of them will be analyzed and briefly discussed below.

Optimum design is considered here as the design which provides the lowest capital and annual cost. In the case of this feasibility study the lowest annual cost is the most important criterion. However, annual cost is not the only optimization criterion – other factors may be in the range of interest. A system may be designed for maximum performance (COP in this case), lowest GWP, maximum lifetime and others.

9.1   Heat Sink

It was assumed that a heat sink exists and it consists of cold water at 28 °C. Thus, the system was optimized to work with cooling water at 28 °C. It is important to note that cooling water mass flow and temperature has great influence on the performance (water consumption, COP, area of heat exchangers, pump power, etc.) of absorption chillers. Therefore it is important to consider a case where heat sink is available at temperature 20 °C.

Equipment	Type of heat exchanger	Size [m^2] 28°C	Size [m^2] 20°C
Absorber	Shell and tube	638,1	320,7
Solution hex.	Plate	131,0	88,0
Generator	Shell and tube	588,0	295,5
Condenser	Plate	107,2	106,0
Evaporator	Plate	498,9	455,9
Solution pump		44,4 [kW]	21,9 [kW]

Table 22   Absorption chiller 2100 [kW]. Influence of cooling water on heat exchanger surfaces. .

Simulation and optimization shows a great influence of water temperature on heat exchanger sizes. It is important to note that as water temperature decreases from 28 °C to 20 °C, the shell and tube (absorber and generator) heat exchanger sizes decrease about 55%. It is a significant change, especially due to the high unit cost [€/m^2] of a shell and tube heat exchanger.

		Cooling water temperature	
Type	Unit	28°C	20°C
$m_{absorber}$	[kg/s]	60,00	30,80

m$_{condenser}$	[kg/s]	62,00	30,80
m$_{brine}$	[kg/s]	110,00	90,00
m$_{solution}$	[kg/s]	11,00	8,00
W$_{abs+cond}$	[kW]	8,55	4,32
W$_{brine}$	[kW]	7,71	6,31
W$_{solution}$	[kW]	44,44	21,9

Table 23 Absorption chiller 2100 [kW]. Influence of cooling water on mass flow agreements and auxiliary pumps.

The results indicate the possibility of decreasing the capital and annual cost (see total price for equipment). Lower cooling water temperature creates change by decreasing steams of cold and hot water. As cold and hot water mass flow decreases, the total annual operating cost decreases.

Equipment	U value [kW/m^2°C]	Type of heat exchanger	Capital cost [€] 28°C	Capital cost [€] 20°C
Absorber	0,79	Shell and tube	158 174,9	91 252,4
Solution hex.	0,20	Plate	30 809,7	22 420,0
Generator	0,90	Shell and tube	148 034,6	85 473,5
Condenser	1,10	Plate	26 277,1	26 015,8
Evaporator	1,20	Plate	89 727,5	83 526,0
Pump	-	-	5 206,8	2 563,5
Valve (both)	-	-	2 800	2 800
Total price for equipment	-	-	461 030,5	314 087,4

Table 24 Absorption chiller 2100 [kW]. Influence of cooling water on capital cost.

Lower cooling water temperature causes a decrease of total price for equipment from 461 030€ to 314 087€, which is more than 30% of the total price for equipment.

Power capacity 2100 [kW]	28°C	20°C	
TCI (Total Capital Investment)	1 701 472,8	1 167 134,4	
Total annual capital cost (ABS unit)	187 448,0	128 580,9	[€/year]
Total annual cost (Compression unit)	130792,8		

Table 25 Absorption chiller 2100 [kW]. Influence of cooling water temperature on total annual cost of <u>investment</u>.
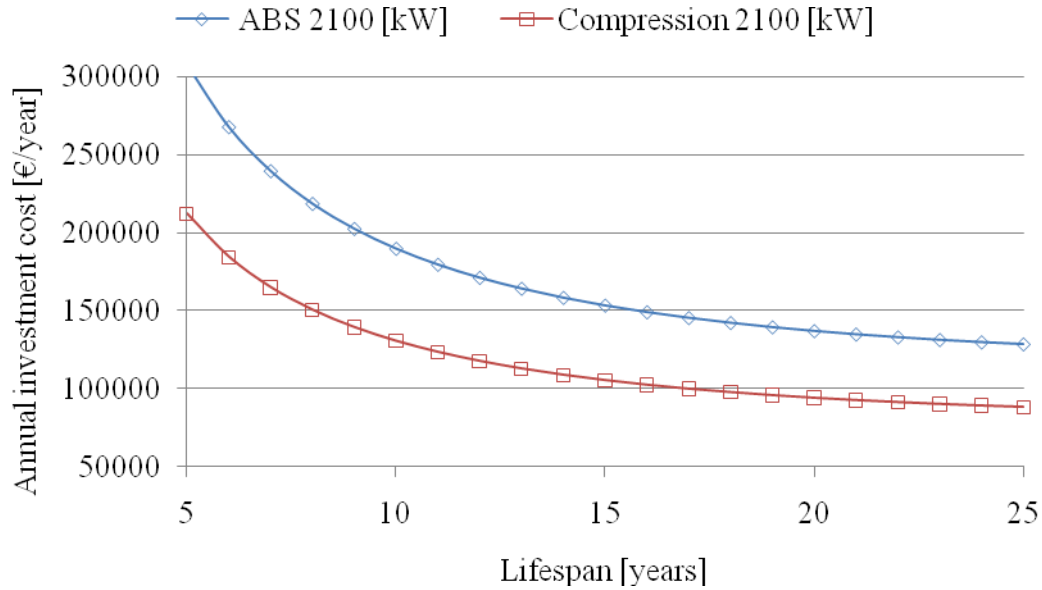
Figure 37 The annual investment cost variations as a function of lifetime (for 20 °C – cooling water temperature).

Figure 36 presents the variation of the total annual investment cost as a function of lifetime. The figure can be described as follows:

- As the lifetime increases, the total annual investment cost decreases,

- Annual investment cost of absorption chiller unit (25 years) is slightly smaller (table 25) than total annual cost of compression unit (10 years),

9.2   System Boundaries

An absorption cooling plant can be powered from a geothermal heat source as long as the absorption cooling plant is a coherent part of a district heating system. In the following discussion it has been assumed that a hot water source exists and has some specific price. The same rule applies to cold water. In its simplest terms, it has been assumed that our system boundary is shown by the **dark wide** line in the schematic figure shown below.
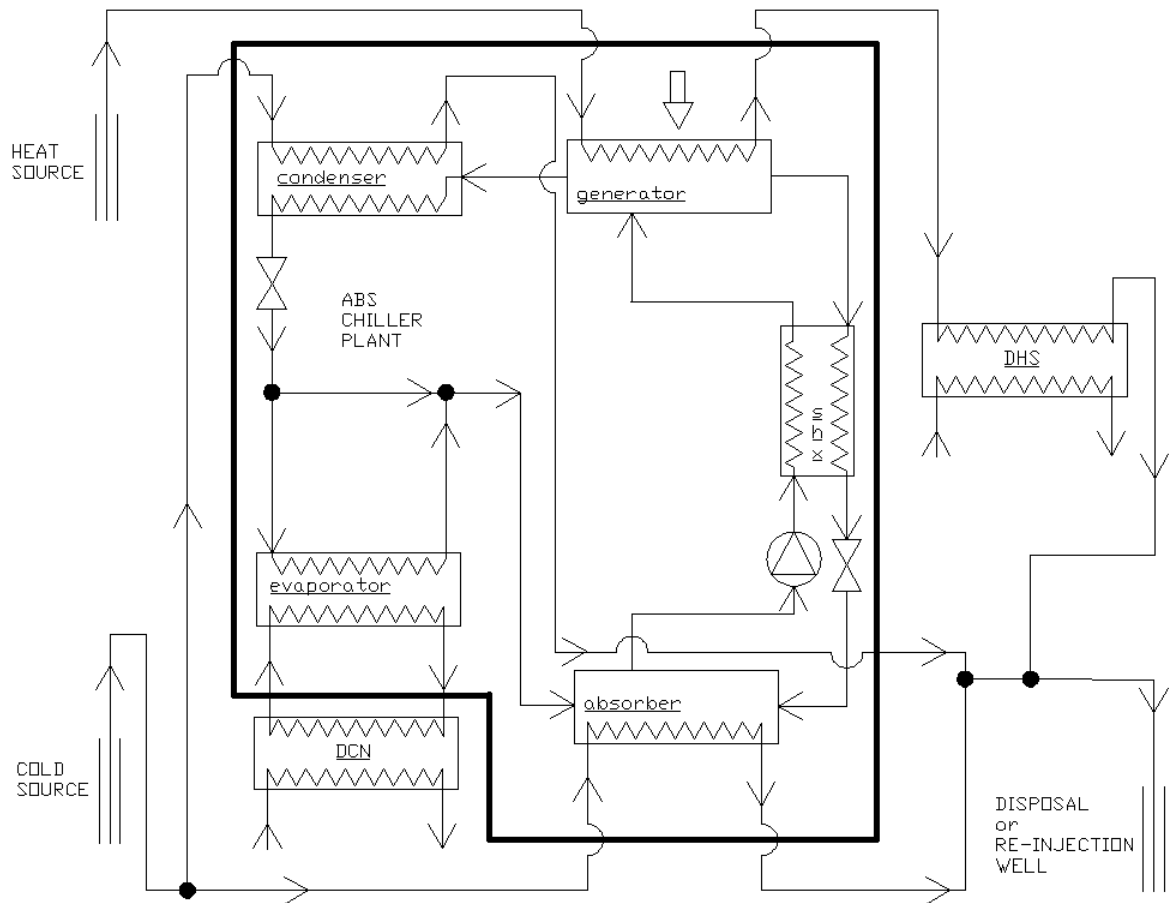
Figure 38 District heating & cooling network – simplified scheme.

"The whole of the Arabia peninsula has geothermal potential, as do other sedimentary basins in the world. That is if a success is tapping fluids at 120 °C or less. China is another example. The general conceptual reservoir model is then extremely simple. Temperature rises on the average by 30 °C/km in those basins. If annual mean surface temperature is 10-30 °C, you find your desired 90 $^+_-$ 5 °C at 2000-3000 m depth. In addition to temperature there has to be a horizontal geological formation that can easily conduit fluid between production and injection wells, say over a distance of more than 1 km. Name of the game is then to drill and develop such formations." (Grimur Bjornsson, 2009).

9.2.1 Drilling Cost Estimation

It is assumed that the geothermal well follows the values (Sverrir Thorhallsson, 2008):

- Fixed cost (casing, cementing, etc.) - 156 340 [€]
- Drilling cost   - 938 – 1407 [€/m]

"Sedimentary basin geothermal systems are well known. Sediments have low thermal conductivity (1 – 2 W/m°C). Crystalline basement rocks as well as limestone (carts) have higher thermal conductivity (2 – 4 W/m°C). Therefore, if we have a basin (2 – 3 km deep broad valley) in a fissured crystalline or limestone rocks filled with sediments, and normal regional heat flow then we will automatically have a much higher surface temperature gradient in the sediments than in the surrounding basement rocks (about twice as high). And consequently we have an accumulation of heat at the bottom of the sediments and

geothermal systems which can be utilized. The Paris basin is the best example, geothermal systems in and around Beijing are of same nature as well as some systems in central Europe. If the same geological conditions exist in south China (Shanghai) and in Abu Dhabi we most likely have geothermal systems at depth there (this has of course to be proven by drilling)" (Axel Björnsson, 2009).

Thus a geothermal reservoir is assumed to be as described above, with the temperature gradient 30 °C/km and depths up to 4 km. For simplification it is assumed that flow from one well equals 30 l/s, regardless of the chosen depth and temperature. This means that a well 1 km deep provides 30 l/s of 30 °C hot water (brine) at a cost from 1,094 10^6 € to 1,563 10^6 €. If the depth is 3 km, the associated flow is 30 l/s of 90 °C hot water (brine) at a cost from 2,970 10^6 € to 4,377 10^6 €. The deeper the well is, the smaller the total specific cost of the well (€/m).

Figure no. 39 presents the variation of Total Capital Cost (TCI) and drilling cost as a function of hot water (brine) temperature (depth). The figure can be described as follows:

- As the brine mass flow decreases from 90 l/s to 60 l/s, the Total Capital Cost increases (at 90 °C, line 60 L/s TCI_ABS is above line 90 L/s TCI_ABS),

- As the brine mass flow decreases, the drilling cost decreases due to the amount of wells which must be drilled (30 l/s/well)

- As the brine mass flow decreases, the Total Annual Cost (see Table no. 21) decreases due to lower electricity cost and hot water (brine) consumption,

Temperature	90	95	100	105	110	115	120	125	130	[°C]
Depth	3000	3167	3333	3500	3667	3833	4000	4167	4333	[m]

Table 26 Temperature change with depth.
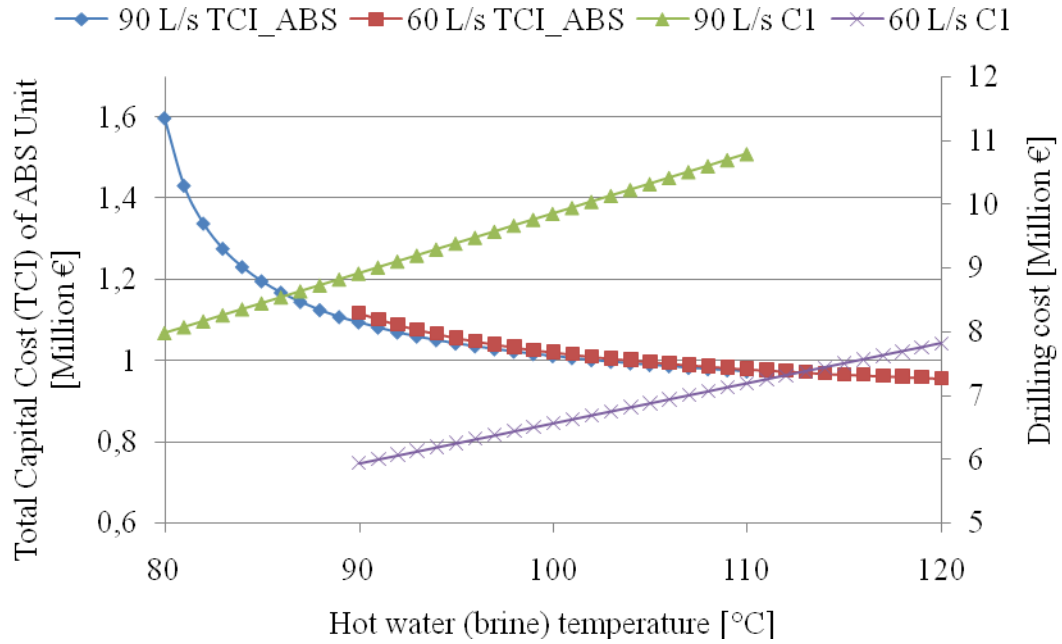
Figure 39 Total Capital Cost and Drilling Cost Variation as a function of hot water (brine) temperature (well depth).

Description of figure 39:

- 90 L/s TCI_ABS – Total Capital Cost of ABS unit for brine flow 90 l/s,
- 60 L/s TCI_ABS – Total Capital Cost of ABS unit for brine flow 60 l/s,
- 90 L/s C1 – Drilling Cost for brine mass flow 90 l/s,
- 60 L/s C1 – Drilling Cost for brine mass flow 60 l/s,

The calculations and figure above prove that well depth does not work (fit) as an optimization criterion (in case of drilling and TCI cost), due to limited deliverability from one well.

10 CONCLUSIONS

In this study a new computer model for calculation, simulation and optimization of an absorption chiller system has been developed. Using the (EES) computer simulation, after calculation the thermodynamic properties of each state point of the cycle, heat exchanger areas, COP, required mass flows and performance parameters of the system were calculated. The results of this calculation, simulation and optimization confirm that the system can be operated at a generator inlet (generating) a temperature of 86 °C and (condenser and absorber) cooling water temperatures of 20 °C to 28 °C. The COP was achieved at 0,731 (for 20 °C) and 0,684 (28 °C). The study shows the influence of working temperatures, mass flows and other factors on the heat exchanger surfaces and total annual cost. As the cooling water temperature decreases, the cooling water mass flow and heat exchanger areas decrease, while at the same time the COP increases.

This study indicates the agreements under which an optimal model can be obtained. Optimum design is considered here as the design which provides the lowest capital and annual cost.

The results show the great influence of cooling water temperature on the annual cost of the system and its performance.

This project can be considered as an indicator of the most environmentally friendly, sustainable and the most economically viable cooling application.

The following is a list of important conclusions from the study of a single stage absorption chiller system:

- From examining the published and available literature, the single stage absorption unit finds application in co–generation, waste heat and geothermal energy. The best method for designing a single stage absorption unit is to examine the heat exchanger area of all components and mass flows with temperatures,

- The cost of the single stage absorption chiller system may be lower by about 40%, due to lower cooling water temperature,

- The cost of heat (brine) and cold water for the absorption chiller is the major contributor to the annual operating cost (water, electricity, O&M),

- Absorption units powered by hot geothermal water have environmental advantages over the compression units and direct fired units,

- Current market situation,

Geothermal resources are already widely used in China for space heating and cooling. The Chinese geothermal market is large and is growing fast. However, most of energy consumed in China comes from coal and gas.

There is not much available knowledge about geothermal resources, utilization and its temperatures in the United Arab Emirates. Almost all energy consumed in United Arab Emirates is produced from fossil fuels (crude oil and gas). Thus it might not be easy for absorption chillers to compete against compression (electrical) chillers.

It is important to note that the depth of a geothermal well is not the best optimization criterion, due to its different deliverability in different conditions. Results show that a solution which consists of two wells (30 l/s each) with higher temperature geo–fluid is more beneficial to the entire venture than three wells with lower temperature.

Oil well data should be available to investigate temperature gradient, deliverability, porosity, permeability and others. However, such data are rather expensive and confidential.

A Total Equivalent Warming Impact (second part of the project) analysis on the wide range of single and double stage absorption chillers has been carried out to determine the most environmentally beneficial option. The analysis included the absorption chillers driven by geothermal hot water, the direct fired absorption chiller and electric driven chiller.

The best selections were found to be single stage geothermal hot water driven absorption chillers for the entire range of equivalent full load hours (*eflh*).

The next best options are the double-stage steam-driven absorption chiller and centrifugal electric-driven chiller. However, the steam powered absorption chiller is environmentally viable only in a range up to 1500 *eflh*. Beyond that range (above 1500 *poh*), the thermal energy load to the generator is significantly high, which results in an increase in TEWI value (see crossed line IV and VI, figure 14).

All calculations are based on average CO_2 emissions related to electricity generation. Further work is required to ascertain the annual impact of cooling technology on the Global Warming phenomena. A positive impact on the environment may be mainly achieved by avoiding the direct fired and electric chillers. Moreover, single stage (geothermal) hot water chillers are environmentally friendly and their contribution to the GWP is much smaller than in the case of electric chillers. Thus, single stage (geothermal) driven chillers should be used instead of electric or direct fired chillers.

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APPENDIX A – <u>SUMMARY OF PUBLISHED PAST LIBR DEVELOPMENT EFFORTS</u>

Developer	**Country**	**Year**	**Heat Source**	**No. of Stages**	**Intended Applications**	**Key Innovation**
Carrier Corporation	USA	1975 – 1984	Solar	Single	Residential / Light Commercial	Solution Chemistry
Hitachi	Japan	1988 – 1990	Direct Fired	Double	Residential / Light Commercial	Extended surface in absorber and emulsifier; spray absorber
Universitat Politecnica de Catalunya	Spain	Published 2002	Hot water	Single	Residential / Light Commercial	None Reported
Interotex/ Rotartica	England/ Spain	1989 – Present	Solar and Direct Fired	Single (Solar) and Double (Direct Fired)	Residential / Light Commercial	Rotating Heat Exchangers
TU Deft	Netherlands	2003 – Present	Solar	Half	Not Stated	Cycle

Developer	**Heat Source**	**No. of Stages**	**Nominal Capacity [Ton]**	**Nominal COP**	**Maximum Ambient Temperature Tested [°C]**
Carrier Corporation	Solar	Single	2, 3, 10, 20	0,71 – 0,75	Not Available
Hitachi	Direct Fired	Double	3	Not Available	Not Available
Universitat Politecnica de Catalunya	Hot water	Single	0,85	0,32 – 0,47	32,22°C
Interotex/ Rotartica	Solar and Direct Fired	Single (Solar) and Double (Direct Fired)	2,5	0,76 (Double) 0,67 (Single)	40,55°C (Double), 35°C (Single)
TU Deft	Solar	Half	3	Not Yet Tested	Not Yet Tested

APPENDIX B – CRYSTALLIZATION INHIBITORS FOR AIR – COOLED ABS

Developers	Description	Crystallization Temperature for p=0,127psia	Development Hurdle
Baseline	n/a	54 °C	n/a
Carrier Corporation	Carrol: LiBr – ethylene glycol/water. Additive: 1 – nonaylmine, later replaced with phenylmethylcarbinol	60 °C	Cannot survive high – generation temperatures in double stage ABS
Energy Concepts	(NaOH,KOH, CsOH)	Above 62,7 °C for 7,2 °C chilled water temperature	Corrosion
Yakazi	LiBr/LiCl/LiI/LiNO$_3$	Approximately 64 °C	Requires new corrosion inhibitor

Carrier's Carrol Solution

Carrier developed a solution that shows much promise for solar and air–cooled LiBr applications. The Carrol solution consists of LiBr, water, ethylene glycol and phenylmethylcarbinol. The ethylene glycol inhibits crystallization, and the phenylmethylcarbinol promotes heat and mass transfer. Temperature limitations prohibit using Carrol's solution in double–stage machines; however it is stable in the entire temperature range in single–stage machines. Carrol's solution was tested extensively in solar–fired absorption applications both in the laboratory and in the field.

Energy Concepts's Metal Hydroxide Sorbent

Energy Concepts developed a sorbent that is a metal hydroxide blend of NaOH, KOH and CsOH. This is not an additive as it completely replaces the LiBr sorbent. Interotex introduced this solution in their chiller, but some serious corrosion problems occurred. While they resolved one problem, they were stopped by another. Trace amounts of nitrogen (minor air leakage into the system) were reacting with trace amounts of hydrogen (from hydroxides) to form ammonia. The ammonia attacked the copper tubing used in their system. The copper, in turn, plugged orifices in the system. In the end Interotex was forbidden to work with this absorbent.

Yakazi's LiBr/LiCl/LiI Solution

Yakazi developed and patented a LiBr/LiCl/LiI solution for air–cooled applications that increases the allowable absorber and condenser operating temperatures by about 10 °C and 4 °C respectively. However they needed to use a new corrosion inhibitor, which permitted operation at greater temperatures of about 175 °C.
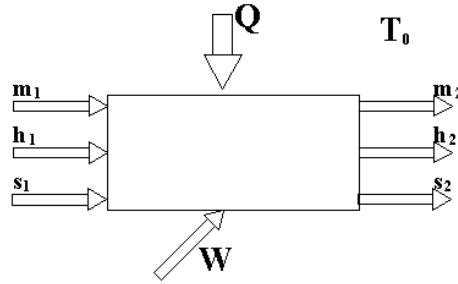
APPENDIX C – TEWI

Consider a system which works in a steady state. The equations which describe mass, energy and entropy balances are:

$$\dot{m}_1 = \dot{m}_2 = \dot{m}$$

C1

$$W = \dot{m}(h_2 - h_1)$$

C2

$$\phi_s = \dot{m}(s_2 - s_1) - \frac{\dot{Q}}{T} \geq 0$$

C3

where ϕ_s is the generated entropy due to internal irreversibility.

$\phi_s = 0$ indicates a reversible process (no entropy gain)

$\phi_s > 0$ indicates a irreversible process (entropy gain greater than 0)

T_0 is the ambient temperature, thus combines with equations C1, C3 gives

$$W = \dot{m}\left[\ h_2 - h_1\ -T_0\ s_2 - s_1\ \right] - Q\left(1 - \frac{T_0}{T}\right) + T_0\phi_s$$

C4

The equation above provides the exergy balance of the considered system, where

$T_0\phi_s$ exergy destruction

$Q\left(1 - \dfrac{T_0}{T}\right)$ exergy of heat

To produce a difference from state 1 to 2 on the mass flow in a system that only exchanges heat with the surroundings (T_0), a minimum quantity of will is required. It will be equal to the difference of exergy flow between states 2 and 1 which is equal to $\dot{m}\left[\ h_2 - h_1\ -T_0\ s_2 - s_1\ \right]$, when the process is internally reversible ($T_0\phi_s = 0$).

What is important, from the air–conditioning and cooling point of view, is to consider Q as the cooling capacity of a room at temperature T. The heat dissipated (gained) to the surroundings at T_0 is Q_0. The air–conditioning (refrigeration) plant works in a closed system. Thus equation C4 applied to this system, gives:

$$W = -Q\left(1 - \frac{T_0}{T}\right) + T_0 \phi_s$$

<div align="right">C5</div>

The equation above shows that the minimum quantity of work required for air–conditioning (refrigeration) is equal to $-Q\left(1 - \frac{T_0}{T}\right)$ which corresponds to the Carnot reversible cycle, where more work is required for colder rooms, in addition to extra work due to irreversibilities ($T_0 \phi_s > 0$).

The exergy change of a process is:

- Its ability to produce work
- Only conserved for reversible process and destroyed by irreversible process
- Equal to zero when at equilibrium with the environment
- Dependent on environment parameters
- Unlimited for reversible process due to the second law of thermodynamic
- A measure of quantity and efficiency of energy utilization

APPENDIX D – SOLUBILITY LIBR IN WATER

The crystallization data from Dong-Seon KIM (2007) were fitted with polynomial functions for the regions as shown in:
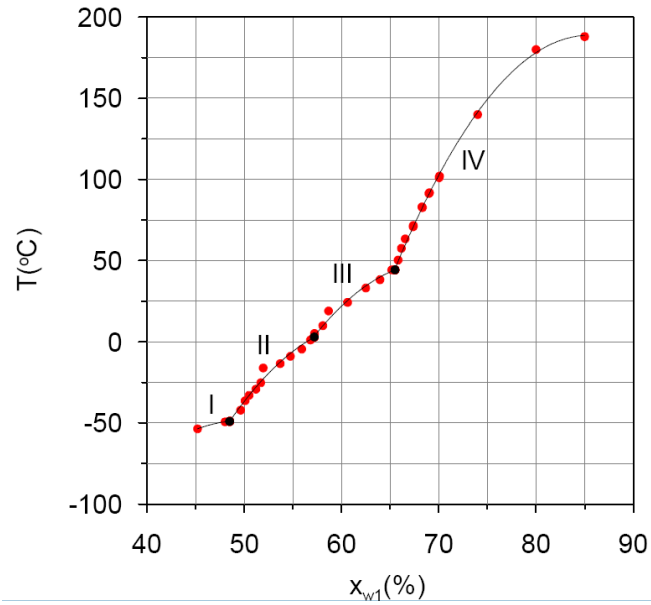
For the region I, where $x_w < 0.485$

$$T = -398.3 + 25.107 \times (100\ x_w) - 0.253 \times (100\ x_w)^2$$

For the region II, where $0.485 < x_w < 0.572$

$$T = -919.4 + 38.51957477 \times (100\ x_w) - 0.3080928653 \times (100\ x_w)^2$$
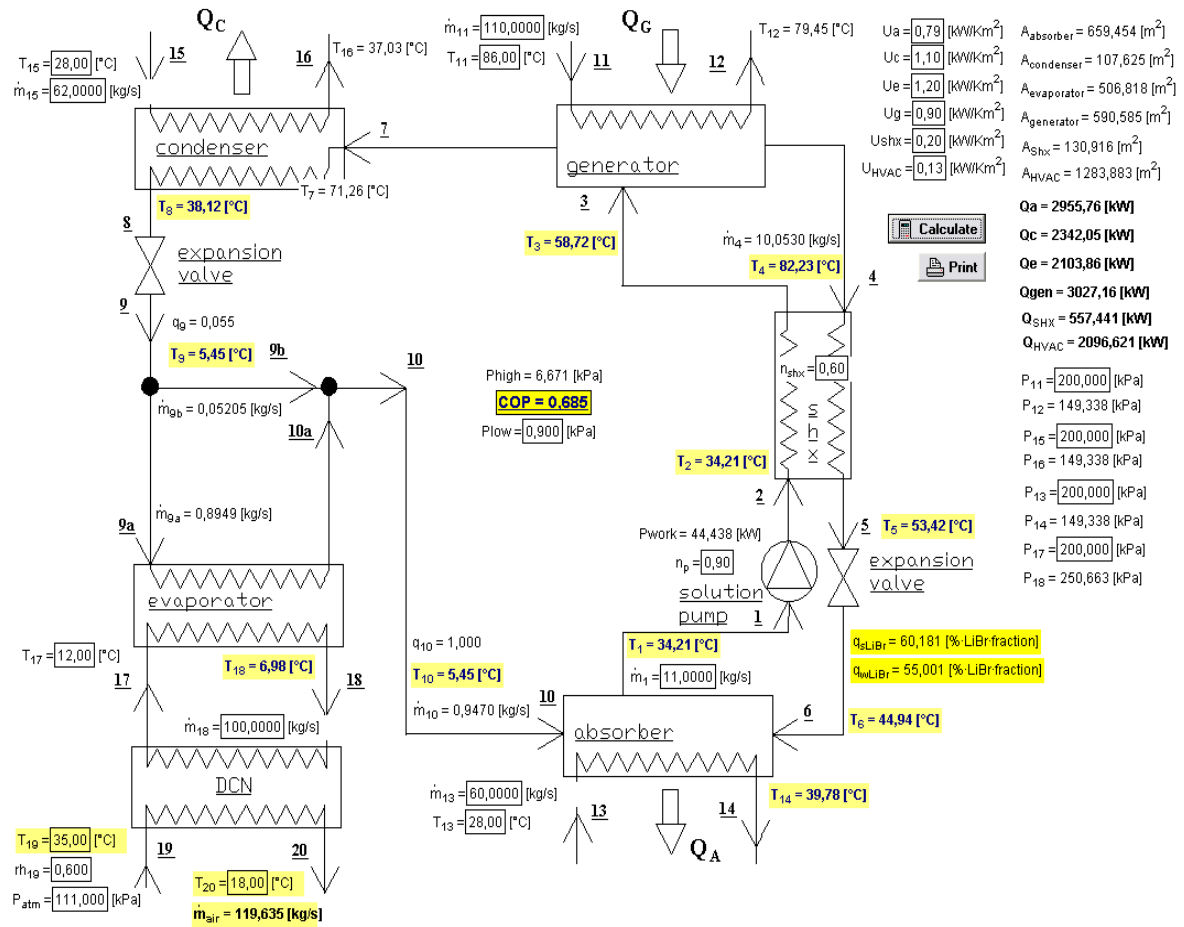
For the region III, where $0.572 < x_w < 0.655$

$$T = -1159.4 + 42.7386184 \times (100\ x_w) - 0.308288545 \times (100\ x_w)^2$$

For the region IV, where $x_w > 0.655$

$$T = -4486.9 + 130.3204838 \times (100\ x_w) - 0.8697209191 \times (100\ x_w)^2$$

where T in K and x_w in LiBr mass fraction.

APPENDIX E – EES CODE

"M. Sc. Thesis Model - Absorption Chiller"
"!Unit System - SI!"
SI=2
"!All input data!"
{Uc=1,8[kW/Km^2]   "Overall Heat Transfer Coefficient"
Ue=2[kW/Km^2]   "Overall Heat Transfer Coefficient"
Ua=2[kW/Km^2]   "Overall Heat Transfer Coefficient"
Ug=1,6[kW/Km^2]}   "Overall Heat Transfer Coefficient"
{Ushx=2[kW/Km^2]}   "Overall Heat Transfer Coefficient"
{T[11]=86[C]   "Generator inlet temperture - diagram"
m_dot[11]=1[kg/s]   "Brine mass flow - diagram"
P[11]=301,325[kPa]}   "Brine inlet pressure"
P[11]=P[12]+P_loss   "Brine outlet pressure"
m_dot[11]=m_dot[12]
{T[15]=28[C]   "Cooling water inlet temp - condenser - diagram"
m_dot[15]=0,3[kg/s]   "Cooling water mass flow - conenser - diagram"
P[15]=301,325[kPa]}   "Cooling water inlet pressure - condenser - diagram"
P_loss=0,5*101,325[kPa] "Pressure loss"
P[16]=-P_loss+P[15]   "Cooling wter outlet pressure - condenser"
{T[13]=28[C]   "Cooling water inlet temperature - absorber - ab"
m_dot[13]=0,3[kg/s]   "ab"
P[13]=301,325[kPa]}   "ab"
P[14]=-P_loss+P[13]   "ab"
{T[17]=16[C]   "ab"
m_dot[18]=0,6[kg/s]   "ab"
P[17]=301,325[kPa]}   "ab"

P[18]=P_loss+P[17]   "ab"
{n_shx=0,6}   "Solution Heat Exchanger Efficiency"
{n_p=0,9}   "Pump efficiecny"
{m_dot[1]=0,056}   "kg/s - Water/LiBr stream"
{Plow=0,9[kPa]} "Low Pressure Level"

"! Assumptions !"
q_wLiBr=X_LIBR('SI';T[1];Plow)
q_sLiBr=X_LIBR('SI';T[4];Phigh)
Pcondensating_steam_7=Pressure(water;T=T[7];x=1)   "This has to be above Phigh"
T_Pinch_condenser=5 [C]
Pcondenser_capability=Pressure(water;T=T[15]+T_Pinch_condenser;x=0)
Pressure_drop_condenser= 1,638{1,568} [kPa]   "This is preprogrammed at a pinch of 5C in the condenser. It may not be correct, but it should be between 3-7C"
Phigh=Pcondenser_capability+Pressure_drop_condenser
q[8]=0
q[10]=1

"°°°°°°°°°S°O°L°U°T°I°O°N°°°°°°°°°°"
"! Condenser !"
m_dot[7]=m_dot[8]
m_dot[15]=m_dot[16]

cp_[15]=Cp(Water;T=T[15];P=P[15])   "specific heat - inlet water"
cp_15m16=m_dot[15]*cp_[15]   "avg. spec. heat. - due to neglegible cp diff."

h[15]=Enthalpy(Water;T=T[15];P=P[15])
h[16]=Enthalpy(Water;T=T[16];x=0)
h[7]=Enthalpy(Water;T=T[7];P=Phigh)

T[16]=Temperature(Water;x=0;h=h[16])
T[7]=T_LIBR('SI';Phigh;q_wLiBr)
T[8]=temperature(water;h=h[8];x=0)   "T[8]=temperature(water;h=h[8];x=0)   doesn't   make   any difference"

Qc=cp_15m16*(T[16]-T[15])
{Qc=m_dot[15]*(h[16]-h[15])}
Qc=m_dot[7]*(h[7]-h[8])

T7saturationtemp=temperature(steam;x=0,1;P=Phigh)
h7saturationtemp=enthalpy(steam;T=T7saturationtemp+0,001;P=Phigh)
H_superheatdiff=(h[7]-h7saturationtemp)
h_sat_liquid_condenser=enthalpy(water;T=T7saturationtemp;x=0)

Tsubcooling=0,001
h[8]=h_sat_liquid_condenser

diff_cond=ln((T[8]-T[15])/(T[7]-T[16]))
LMTD_condenser=((T[8]-T[15])-(T[7]-T[16]))/diff_cond
Qc=Uc*A_condenser*LMTD_condenser

"! Expansion Valve - Throttle !"
h[8]=h[9]
m_dot[8]=m_dot[9]
T[9]=Temperature(Water;P=Plow;h=h[9])

"! Evaporator !"
m_dot[17]=m_dot[18]
m_dot[9]=m_dot_9a+m_dot_9b   "9b vapour stream"
m_dot[10]=m_dot_9b+m_dot_10a

m_dot_10a=m_dot_9a
q[9]*m_dot[9]=m_dot_9b

cp_[17]=Cp(Water;T=T[17];P=P[17])
cp_[18]=Cp(Water;T=T[18];P=P[18])
Cmin_evap=m_dot[18]*cp_[18]   "Due to neglegible change, cp is set as constant"

h[17]=Enthalpy(Water;T=T[17];P=P[17])
h[18]=Enthalpy(Water;T=T[18];P=P[18])

Qe=Cmin_evap*(T[17]-T[18])
Qe=m_dot_9a*(h[10]-h[9])

h[10]=Enthalpy(Steam;x=1;P=Plow)   "assumption: vapour quality is equal to 1"
T[10]=Temperature(Steam;x=q[10];h=h[10])

diff_evap=ln((T[17]-T[9])/(T[18]-T[10]))
deltaT_LMTD_EVAP=((T[17]-T[9])-(T[18]-T[10]))/diff_evap
Qe=Ue*A_evaporator*deltaT_LMTD_EVAP

q[9]=Quality(Water;T=T[9];h=h[9])

"! Absorber & Cooling Circuit in the Absorber!"
m_dot[13]=m_dot[14]
m_dot[10]+m_dot[6]-m_dot[1]=0 "Mass balance eqn"

cp_[13]=Cp(Water;T=T[13];P=P[13])
Cmin_13=cp_[13]*m_dot[13]
h[13]=Enthalpy(Water;T=T[13];P=P[13])
h[14]=Enthalpy(Water;T=T[14];P=P[14])

{h[0]=H_LIBR('SI';T[1];q_wLiBr)}   "every formula with footnote [0] is a test case"

Qa=m_dot[10]*h[10]+m_dot[6]*h[6]-m_dot[1]*h[1]
Qa=Cmin_13*(T[14]-T[13])

CALL Q_LIBR(h[6];Plow;q_sLIBR;SI:q6b;T6b;Xl6b;hl6b;hv6b)
T6b=T[6]
T[1]=T_LIBR('SI';Plow;q_wLiBr)
q6b=q[6]   "Strong Water/LiBr solution quality"
"! check quality, concentration, pressure, etc ALL LiBr Features!"

diff_abs=ln((T[6]-T[14])/(T[1]-T[13]))
deltaT_LMTD_Absorber=(T[6]-T[14]-T[1]+T[13])/diff_abs
Qa=Ua*A_absorber*deltaT_LMTD_Absorber

"! Solution Pump !"
h[2]=h[1]+Pwork/m_dot[1]   "REF: ScienceDirect - Solar Tunisia"
m_dot[2]=m_dot[1]   "Pump - TEMPERATURE CONSTANT - ISENTROPIC PUMP"
T[1]=T[2]   "REF: ScienceDirect - Solar Tunisia"

Pwork=m_dot[1]*v1*(Phigh-Plow)/n_p   "Power required"
v1=V_LIBR('SI';T[1];q_wLiBr)   "Specific volume"

"! Solution Expansion Valve !"
h[6]=h[5]   "h[5] computed from cp properties"
m_dot[6]=m_dot[5]   "h[6] put to call procedure and obtain T[6]"

"! Generator & Solution Heat Exchanger EQUATIONS"
m_dot[2]=m_dot[3]

m_dot[4]=m_dot[5]

Qgen=m_dot[11]*(h[11]-h[12])
Qgen+m_dot[3]*h[3]=m_dot[7]*h[7]+m_dot[4]*h[4]
{T[2]=T_LIBR('SI';Phigh;q_wLiBr) }   "Due to this equation ..."
m_dot[3]*q_wLiBr=m_dot[4]*q_sLiBr   "Stength balance eqn"
h[11]=Enthalpy(Water;T=T[11];P=P[11])
h[12]=Enthalpy(Water;T=T[12];P=P[12])

n_shx=(T[4]-T[5])/(T[4]-T[2])
{m_dot[4]*(h[4]-h[5])=Qshx1
m_dot[2]*(h[3]-h[2])=Qshx1}
cp_minSHX*(T[4]-T[5])=cp_maxSHX*(T[3]-T[2])   "Twin eqn"
{m_dot[3]*h[3]+m_dot[5]*h[5]=m_dot[4]*h[4]+m_dot[2]*h[2]}   "Twin eqn"
Q_SHX=m_dot[4]*(h[4]-h[5])

Tdel2=0,00002
h[2]=H_LIBR('SI';T[2];q_wLiBr)
h2del2=H_LIBR('SI';T[2]+Tdel2;q_wLiBr)
h2delta=h2del2-h[2]
cp_[2]=h2delta/Tdel2

Tdel3=0,00002
h[3]=H_LIBR('SI';T[3];q_wLiBr)
h3del3=H_LIBR('SI';T[3]+Tdel3;q_wLiBr)
h3delta=h3del3-h[3]
cp_[3]=h3delta/Tdel3

Tdel4=0,00002
h[4]=H_LIBR('SI';T[4];q_sLiBr)
h4del4=H_LIBR('SI';T[4]+Tdel4;q_sLiBr)
h4delta=h4del4-h[4]
cp_[4]=h4delta/Tdel4

Tdel5=0,00002
h[5]=H_LIBR('SI';T[5];q_sLiBr)
h5del5=H_LIBR('SI';T[5]+Tdel5;q_sLiBr)
h5delta=h5del5-h[5]
cp_[5]=h5delta/Tdel5

cp_min_stream=cp_[5]/2+cp_[4]/2
cp_min_stream=cp_45
cp_minSHX=m_dot[4]*cp_45
cp_max_stream=cp_[3]/2+cp_[2]/2
cp_max_stream=cp_23
cp_maxSHX=m_dot[2]*cp_23   "This min and max values are ok, I checked this in solution window"

"Area of generator - heat exchanger"
diff_gen=ln((T[11]-T[4])/(T[12]-T[7]))
deltaT_LMTD_GEN=(T[11]-T[4]-T[12]+T[7])/diff_GEN
Qgen=Ug*A_generator*deltaT_LMTD_GEN

Eff_d=(T[11]-T[12])/(T[11]-T[7])

"Area of Solution heat exchanger"
TypeHX$='counterflow'
Ntu=HX(TypeHX$;epsilon;c_dot_h;c_dot_c;'Ntu')
Ntu=(Ushx*A_Shx)/cp_minSHX
epsilon=n_shx
c_dot_h=cp_minSHX

c_dot_c=cp_maxSHX

"! COP - Coefficient of Performance !"
COP=Qe/(Qgen+Pwork)

"Arrays Table"
Phigh=P[7]
Phigh=P[8]
Phigh=P[3]
Phigh=P[4]
Phigh=P[2]
Phigh=P[5]
Plow=P[9]
Plow=P[10]
Plow=P[1]
Plow=P[6]

"Household heat exchanger"
m_dot_air=m_dot[20]
m_dot[20]=m_dot[19]
{P_atm=101,325}
{rh[19]=0,7}
P_atm=P[19]
P[19]=P[20]
rh[20]=rh[19]-0,05   "Due to some dehumidifiers, filters stuff and etc"
cp_[19]=Cp(AirH2O;T=T[19];r=rh[19];P=P_atm)
cp_[20]=Cp(AirH2O;T=T[20];r=rh[20];P=P_atm)
cp_19m20=(cp_[19]+cp_[20])/2
cp_dot_19m20=m_dot_air*cp_19m20
m_dot[17]*(h[17]-h[18])=m_dot_air*(cp_19m20*(T[19]-T[20]))

cp_dot_minHVAC=min(Cmin_evap;cp_dot_19m20)
Q_HVACmax=cp_dot_minHVAC*(T[19]-T[18])
Q_HVAC=m_dot[17]*(h[17]-h[18])

efficiency_HVAC=n_HVAC
n_HVAC=Q_HVAC/Q_HVACmax
Ntu=HX(TypeHX$;n_HVAC;cp_17m18;cp_19m20;'Ntu')
Ntu=(U_HVAC*A_HVAC)/cp_dot_minHVAC
h[19]=Enthalpy(AirH2O;T=T[19];r=rh[19];P=P_atm)
h[20]=Enthalpy(AirH2O;T=T[20];r=rh[20];P=P_atm)

"! Economic - Feasibility Analysis !"
C_1=623,58 [Euro/m2]   "Crossflow - fluid unmixed heat exchanger - Plate heat exchanger"
C_2=902,35 [Euro/m2]   "Shell and Tube n passes heat exchanger"
C_3=11717,00 [$/kW]   "Price for installed pump power""Price for 100kW "
In_Shx=C_1*A_Shx^0,8 "Initial price for solution heat exchanger - plate hex. "

In_pump=C_3*Pwork/100   "Initial price - pump"
In_SumHex=C_2*(A_absorber^0,8+A_generator^0,8)   "Initial   price - for shell and tube heat exchangers - absorber and generator"
In_Absorber=C_2*A_absorber^0,8   "Initial price - for absorber"
In_Generator=C_2*A_generator^0,8   "Initial price - for generator"
In_Condenser=C_1*A_condenser^0,8   "Initial price for condenser - plate hex."
In_Evaporator=C_1*A_evaporator^0,8   "Initial price for evaporator - plate hex."
In_valve1=1400 "Initial price for SolHex Valve"
In_valve2=1400 " Initial price for RefEx Valve"
In_plate=In_Condenser+In_Evaporator   "Initial price for plate hex"
In_ALL_HEX=In_SumHex+In_plate+In_Shx

In_SumTot=In_Shx+In_SumHex+In_pump+In_plate
In_Equipment=In_SumTot+In_valve1+In_valve2   "Intial price for all equipment"

In_PipingSys=0,05*In_Equipment "Initial cost of piping system"
In_Electricity=0,3*(In_Equipment+In_PipingSys)   "Initial cost of electricity"
In_OnSiteCost=In_Equipment+In_PipingSys+In_Electricity"Initial On-Site Cost"

In_OFFSC1=0,2*In_OnSiteCost   "Civil, structural and architectural work ..."
In_OFFSC2=0,25*In_OnSiteCost "Service facilities ..."
In_OFFSC3=0,15*In_OnSiteCost "Contingencies ..."
In_OFFSC_SUM=In_OFFSC1+In_OFFSC2+In_OFFSC3   "Total off-site cost ..."

In_DcTotal=In_OnSiteCost+In_OFFSC_SUM   "Total Direct Cost"
In_DC1=In_DcTotal*0,15   "Engineering and Supervision ..."
In_DC2=In_DcTotal*0,15   "Construction including ..."
In_DC3=In_DcTotal*0,2 "Contingencies ..."

In_FCI=In_DC1+In_DC2+In_DC3+In_DcTotal   "FCI total cost"

                    "Other outlays"
Oth_C1=0,06*In_FCI   "Start - up cost"
Oth_C2=0,05*In_FCI   "Working capital"
Oth_C3a=20000[$]   "Research and STH for 1000 kW ABS"
Oth_C3b=25000[$]   "Research and STH for 2100 kW ABS"

In_TCI=In_FCI+Oth_C1+Oth_C2+Oth_C3b   "Total INvestment CaPital - last field in the table"

"! Annual Cost !"
n_y=25 "Lifespan of ABS Unit"
i_n=0,1 "Annual effective rate of return"
P_df=0,12   "Price of hot water DF - Driving Force"
P_cw=0,06   "Price of cold water"
P_elCHI=67,53   "[Euro/MWh]""Current price from 12 Jan 2009"
P_elUAE=72,67   "[Euro/MWh]""Calculated"
M_uh_Ch=3383 "Maximum utilization hours - China - number of hours when temp is above 20°C"
M_uh_Uae=7189"Maximum utilization hours - UAE - number of hours when temp is above 20°C"
                    "Pumps work usually aournd 500h/yr more"
                    "Chkdsk for UNITS - ONE MORE TIME"

CRF=(i_n)/(1-((1+i_n)^(-n_y)))   "Capital Recovery Factor"
An_j=In_TCI*CRF

In_OpCh_Maint_PE=In_TCI*0,005   "Operational cost"
In_OpCh_Maint_OC=Oth_C3b*(Qe/1000)*(M_uh_Ch/8760)   "Personel cost"
In_OpUae_Maint_PE=In_TCI*0,005   "Operational cost"
In_OpUae_Maint_OC=Oth_C3b*(Qe/1000)*(M_uh_Uae/8760)   "Personel cost"
In_OpCh_Maint=In_TCI*0,005+Oth_C3b*(Qe/1000)*(M_uh_Ch/8760)   "0,5% for ABS, and 3% for
Compression Chiller"
In_OpUae_Maint=In_TCI*0,005+Oth_C3b*(Qe/1000)*(M_uh_Uae/8760)   "Annual   O&M=0,5%
Investment Costs / Th. p60"

P_controls=Qe/80   "Power of control devices"
P_pump=Pwork   "Solution pump"
P_cond_abs=(5*9,81*(m_dot[15]+m_dot[13]))/(0,7*1000)   "Cold water pump power"
P_brine=(5*m_dot[11]*9,81)/(0,7*1000)   "Brine pump power"
P_auxiailary=(10*9,81*(m_dot[15]+m_dot[11]+m_dot[13]))/(0,7*1000)   "Auxiailary power"
P_Load_Max=P_controls+P_pump+P_auxiailary   "Total power Load - Max Load"
Ann_el_consCHI=P_Load_Max*M_uh_Ch   "Annual electricity consumption - China"
Ann_el_consUAE=P_Load_Max*M_uh_Uae   "Annual electricity consumption - UAE"

In_elec$ch=(P_Load_Max)*M_uh_Ch*P_elCHI*0,001   "m*kg/s*m^3/kg = kW"
In_elec$Uae=(P_Load_Max)*M_uh_Uae*P_elUAE*0,001   "5m due to press losses"
                        " 0,001 to convert price from E/MWh to E/kWh"
                        "Qe/80 => power (in kW) consumed by controls and sth"
IannCh_cw=P_cw*M_uh_Ch*(m_dot[15]+m_dot[13])*3600/1000   "Like below"
IannCh_df=P_df*M_uh_Ch*m_dot[11]*3600/1000   "Like below"

IannUae_cw=P_cw*M_uh_Uae*(m_dot[15]+m_dot[13])*3600/1000 "Annuatl cost of cold water"
IannUae_df=P_df*M_uh_Uae*m_dot[11]*3600/1000   "Annual cost of hot water"

In_AnnualCost_Ch=An_j+In_OpCh_Maint+In_elec$ch+IannCh_cw+IannCh_df   "Sum   Annual   costs
China"
In_AnnualCost_Uae=An_j+In_OpUae_Maint+In_elec$Uae+IannUae_cw+IannUae_df " - // - UAE"

"!!!!!!!!!! 3rd Stage!!!!!!!!!!"
"Particular elemts - China"
Z_CI_Ch_Absorber=In_Absorber*CRF/(M_uh_Ch*3600)
Z_OM_Ch_Absorber=In_OpCh_Maint*(In_Absorber/(In_SumTot+In_valve1+In_valve2))/(M_uh_Ch*3600
)
Z_Ch_Absorber=Z_CI_Ch_Absorber+Z_OM_Ch_Absorber

Z_CI_Ch_Shx=In_Shx*CRF/(M_uh_Ch*3600)
Z_OM_Ch_Shx=In_OpCh_Maint*(In_Shx/(In_SumTot+In_valve1+In_valve2))/(M_uh_Ch*3600)
Z_Ch_Shx=Z_CI_Ch_Shx+Z_OM_Ch_Shx

Z_CI_Ch_Generator=In_Generator*CRF/(M_uh_Ch*3600)
Z_OM_Ch_Generator=In_OpCh_Maint*(In_Generator/(In_SumTot+In_valve1+In_valve2))/(M_uh_Ch*360
0)
Z_Ch_Generator=Z_CI_Ch_Generator+Z_OM_Ch_Generator

Z_CI_Ch_Condenser=In_Condenser*CRF/(M_uh_Ch*3600)
Z_OM_Ch_Condenser=In_OpCh_Maint*(In_Condenser/(In_SumTot+In_valve1+In_valve2))/(M_uh_Ch*36
00)
Z_Ch_Condenser=Z_CI_Ch_Condenser+Z_OM_Ch_Condenser

Z_CI_Ch_Evaporator=In_Evaporator*CRF/(M_uh_Ch*3600)
Z_OM_Ch_Evaporator=In_OpCh_Maint*(In_Evaporator/(In_SumTot+In_valve1+In_valve2))/(M_uh_Ch*3
600)
Z_Ch_Evaporator=Z_CI_Ch_Evaporator+Z_OM_Ch_Evaporator

Z_CI_Ch_valve1=In_valve1*CRF/(M_uh_Ch*3600)
Z_OM_Ch_valve1=In_OpCh_Maint*(In_valve1/(In_SumTot+In_valve1+In_valve2))/(M_uh_Ch*3600)
Z_Ch_valve1=Z_CI_Ch_valve1+Z_OM_Ch_valve1

Z_CI_Ch_valve2=In_valve2*CRF/(M_uh_Ch*3600)
Z_OM_Ch_valve2=In_OpCh_Maint*(In_valve2/(In_SumTot+In_valve1+In_valve2))/(M_uh_Ch*3600)
Z_Ch_valve2=Z_CI_Ch_valve2+Z_OM_Ch_valve2

Z_CI_Ch_Pump=In_pump*CRF/(M_uh_Ch*3600)
Z_OM_Ch_Pump=In_OpCh_Maint*(In_pump/(In_SumTot+In_valve1+In_valve2))/(M_uh_Ch*3600)
Z_Ch_Pump=Z_CI_Ch_Pump+Z_OM_Ch_Pump

"Particular elements - UAE"
Z_CI_Uae_Absorber=In_Absorber*CRF/(M_uh_Uae*3600)
Z_OM_Uae_Absorber=In_OpUae_Maint*(In_Absorber/(In_SumTot+In_valve1+In_valve2))/(M_uh_Uae*3
600)
Z_Uae_Absorber=Z_CI_Uae_Absorber+Z_OM_Uae_Absorber

Z_CI_Uae_Shx=In_Shx*CRF/(M_uh_Uae*3600)

Z_OM_Uae_Shx=In_OpUae_Maint*(In_Shx/(In_SumTot+In_valve1+In_valve2))/(M_uh_Uae*3600)
Z_Uae_Shx=Z_CI_Uae_Shx+Z_OM_Uae_Shx

Z_CI_Uae_Generator=In_Generator*CRF/(M_uh_Uae*3600)
Z_OM_Uae_Generator=In_OpUae_Maint*(In_Generator/(In_SumTot+In_valve1+In_valve2))/(M_uh_Uae*3600)
Z_Uae_Generator=Z_CI_Uae_Generator+Z_OM_Uae_Generator

Z_CI_Uae_Condenser=In_Condenser*CRF/(M_uh_Uae*3600)
Z_OM_Uae_Condenser=In_OpUae_Maint*(In_Condenser/(In_SumTot+In_valve1+In_valve2))/(M_uh_Uae*3600)
Z_Uae_Condenser=Z_CI_Uae_Condenser+Z_OM_Uae_Condenser

Z_CI_Uae_Evaporator=In_Evaporator*CRF/(M_uh_Uae*3600)
Z_OM_Uae_Evaporator=In_OpUae_Maint*(In_Evaporator/(In_SumTot+In_valve1+In_valve2))/(M_uh_Uae*3600)
Z_Uae_Evaporator=Z_CI_Uae_Evaporator+Z_OM_Uae_Evaporator

Z_CI_Uae_valve1=In_valve1*CRF/(M_uh_Uae*3600)
Z_OM_Uae_valve1=In_OpUae_Maint*(In_valve1/(In_SumTot+In_valve1+In_valve2))/(M_uh_Uae*3600)
Z_Uae_valve1=Z_CI_Uae_valve1+Z_OM_Uae_valve1

Z_CI_Uae_valve2=In_valve2*CRF/(M_uh_Uae*3600)
Z_OM_Uae_valve2=In_OpUae_Maint*(In_valve2/(In_SumTot+In_valve1+In_valve2))/(M_uh_Uae*3600)
Z_Uae_valve2=Z_CI_Uae_valve2+Z_OM_Uae_valve2

Z_CI_Uae_Pump=In_pump*CRF/(M_uh_Uae*3600)
Z_OM_Uae_Pump=In_OpUae_Maint*(In_pump/(In_SumTot+In_valve1+In_valve2))/(M_uh_Uae*3600)
Z_Uae_Pump=Z_CI_Uae_Pump+Z_OM_Uae_Pump

"Hole – Geothermal Well "
hg_1=938,04   "Euro/m - drilling cost"
hg_2=(hg_1+hg_3)/2   "Euro/m - drilling cost"
hg_3=1407,06   "Euro/m - drilling cost"
AIO=156340   "All In One - Initial costs - which are constant"

T_gradient=0,3/10   "0,3°C/10m"

depth_d=T[11]*(1/T_gradient)   "Well depth at tempetarure T[11]"

Well_hg1=(depth_d*hg_1+AIO)/1000000   "Cost of well at hg_1 scenario - Mln Euro"
Well_hg2=(depth_d*hg_2+AIO)/1000000   "Cost of well at hg_2 scenario - Mln Euro"
Well_hg3=(depth_d*hg_3+AIO)/1000000   "Cost of well at hg_3 scenario - Mln Euro"
                "References: Data source: ISOR"