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COST ESTIMATION OF USING AN ABSORPTION REFRIGERATION SYSTEM WITH GEOTHERMAL ENERGY FOR INDUSTRIAL APPLICATIONS IN EL SALVADOR

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ABSTRACT

El Salvador is interested in improving the efficient utilization of its available resources, especially those that are environmentally friendly. Natural geothermal resources are available and have been used for commercial electrical production since 1975. Implementation of a project of installing and operating a cooling and refrigeration facility, using geothermal fluid as the heating source in combination with an absorption cycle, is being considered. The Lithium Bromide absorption cycle is thought to be feasible for cooling with a COP (coefficient of performance) of about 0.6. A refrigeration absorption cycle with a mixture of ammonia and water cannot be easily adapted to the climatic conditions of El Salvador, as it needs low cooling temperatures for equipment, such as condenser and absorber. It is recommended that a combined application of a cooling and a refrigeration cycle is studied in more detail.

1. INTRODUCTION

El Salvador is located in Central America. Its capital city, San Salvador, is located 13°50′ north of the Equator and 88°55′ west of Greenwich, facing the Pacific Ocean, as shown in Figure 1. Due to its location, the climate is tropical, with ambient temperatures ranging from 16 to 36°C in the wet and dry seasons.

FIGURE 1: Map of El Salvador, Central America

Electrical generation is distributed as shown in Figure 2, with generation from thermal sources being the strongest contributor at 42 %. (UT, 2007). The electrical demand over the next five years is expected to be covered by expansion of the existing types of electricity generation in addition to coal usage, shown in Figure 3 (Rodríguez and Herrera, 2007).

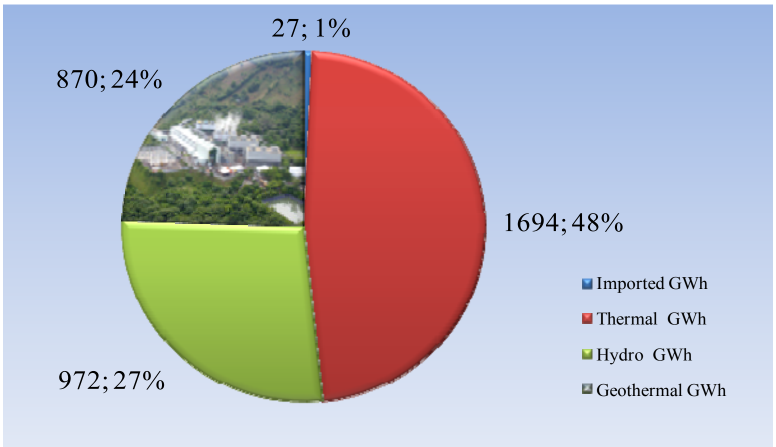


FIGURE 2: Electricity generation by resource in GWh for El Salvador, Jan-Aug 2007

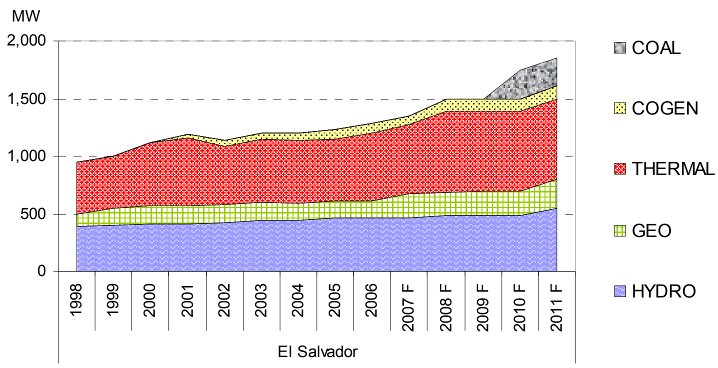


FIGURE 3: Forecast of installed electricity capacity by resource for El Salvador for 2007 to 2011

The National Authority has developed an agenda to make more efficient use of electricity and to generate energy savings. This commitment has resulted in the issuance of rules for the use of lighting systems, air conditioners, refrigeration and motors (Ministry of Economy, 2007). The local economy includes the production of food which also includes the cooling and refrigerating costs for storing the products meant not only for consumption within the country, but also for export to foreign markets such as USA with the TLC (TLC: Free Market Treaty between El Salvador and USA).

In El Salvador, the Ahuachapán and Berlín geothermal fields are under commercial production for a total electricity generation of 167 MW which is supplied to the grid. In this report, the same geothermal energy is proposed for use in combination with refrigeration and cooling technology using an absorption cycle to offer an environmentally friendly alternative for the storage and maintenance of food for the private sector that in turn will attract business. Some private investors have been running their own refrigerated rooms for more than ten years under international regulation certifications and have gained experience in the technology of food storage; they represent a potential sector that may be interested in participating in this particular project.

There are other possible applications using the same absorption principle and geothermal energy, such as district cooling, hydroponic growing of plants and refrigeration or cooling for other products than food that will be left out of the present study.

2. AGRICULTURAL AND INDUSTRIAL SECTOR IN EL SALVADOR

El Salvador, like any other tropical country in the Central American region, has a variety of food production, including meat, fruits and crops. Potential users for refrigeration facilities are therefore found in this sector. The need for refrigeration also includes the ability to preserve food when there are delays between delivery and consumption in the national and international markets. Figure 4 shows the level of production of different meat for the Central American region for 2005 (FAO, 2005a).

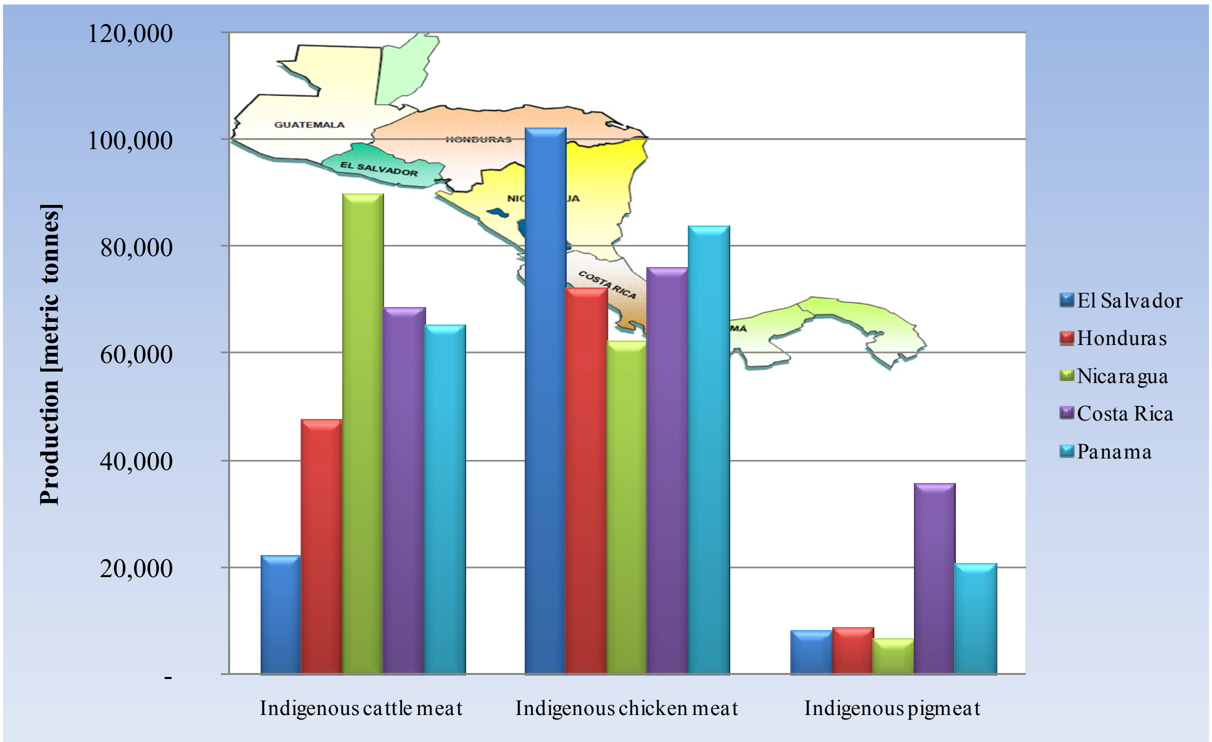


FIGURE 4: Central America, various types of meat production in 2005

Regarding fish products, more than 40% of the national production is exported: alive, dry, chilled or frozen as shown in Figure 5 (FAO, 2005b). The national fish meat production comes from several sources, such as industrial fishing, local fishing and aquaculture, as shown in Table 1.

The total import and export of fish meat for 2003 is shown in Table 2, in metric Tonnes. Imports cover 14% of the national total supply (FAO, 2005b).

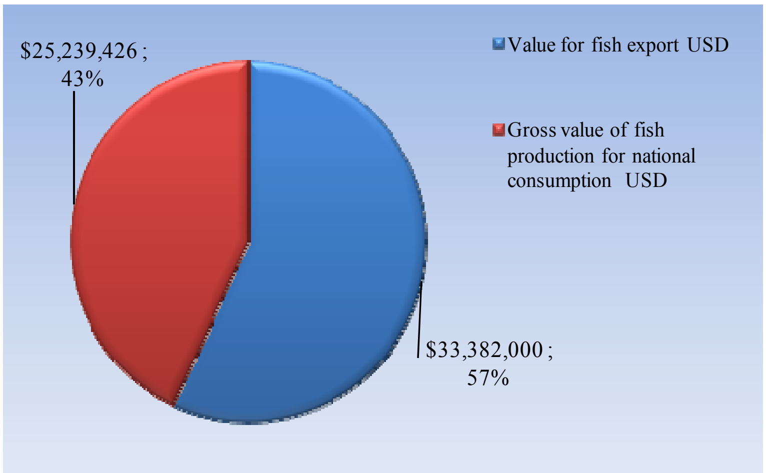


FIGURE 5: Gross value of fish production in El Salvador in 2003

TABLE 1: El Salvador fish production per source (metric Tonnes)

	2000	**2001**	**2002**	**2003**
Industrial fishing	2,099	2,407	2,008	14,813
Marine handline fishing	4,566	5,044	12,007	11,038
Continental handline fishing	2,830	2,774	2,664	2,673
Aquaculture	260	395	782	1,130
Total production	9,755	10,620	17,461	29,654

TABLE 2: El Salvador fish supply in 2003 (metric Tonnes)

	Production [Tonnes]	Export [Tonnes]	Import [Tonnes]	Total supply [Tonnes]	Supply/Capita [Tonnes]
Fish for alimentation	29,477	12,435	2,884	19,926	3.36
Fish for other purposes	118.7	-	4,697		
Live fish	580.0	5	156		

Along with fish, vegetables, aquatic invertebrates, bovine animals, that to some extent require refrigeration, are exported as shown in Figure 6 (ITC, 2005). Aquatic invertebrates are a main volume of product exportation; in 2004 exports exceeded 185% that of fish. Vegetables are also getting more important, increasing in both production and exportation. In 2004, exportation increased by 184% over that of 2003. This commodity may though need less refrigeration than the rest.

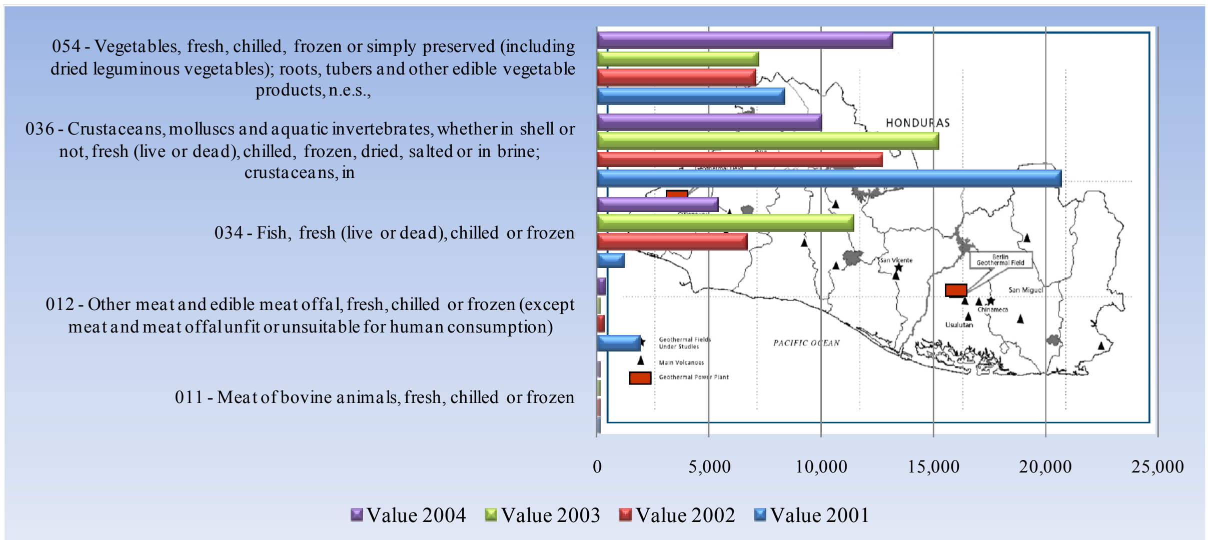


FIGURE 6: El Salvador export history for various frozen or preserved products (thousand USD)

Figure 7 shows some typical agricultural products for the tropical climate zone (FAO, 2005a). If cooling using an absorption system is found to be feasible as a part of a growing facility, the production could face changes by including untraditional products that until now are imported. But for this particular case further studies have to be carried out to estimate the necessary conditions for cooling a plant during growth.

3. COOLING AND REFRIGERATING FUNCTIONS

As was mentioned previously, the need for cooling and refrigeration is connected to the ability for slowing down product decomposition in order to allow for production, transportation, and storage. Decomposition is, by definition, the result of bacterial and fungal action on a biological media, which in turn feeds on dead organisms. There are methods for preventing food decomposition. Some of them date from ancient times. These include drying food, and using specific chemicals added to water, such as potassium nitrate or sodium nitrates to lower the temperature. Drying food involves reducing the amount of water contained in the fibres of the food, needed by the decomposing organisms, by evaporation. Refrigeration freezes the water contained in the food fibres, thus slowing the growth of bacteria. Table 3 shows storage times for refrigerated foods (ASHRAE, 2002).

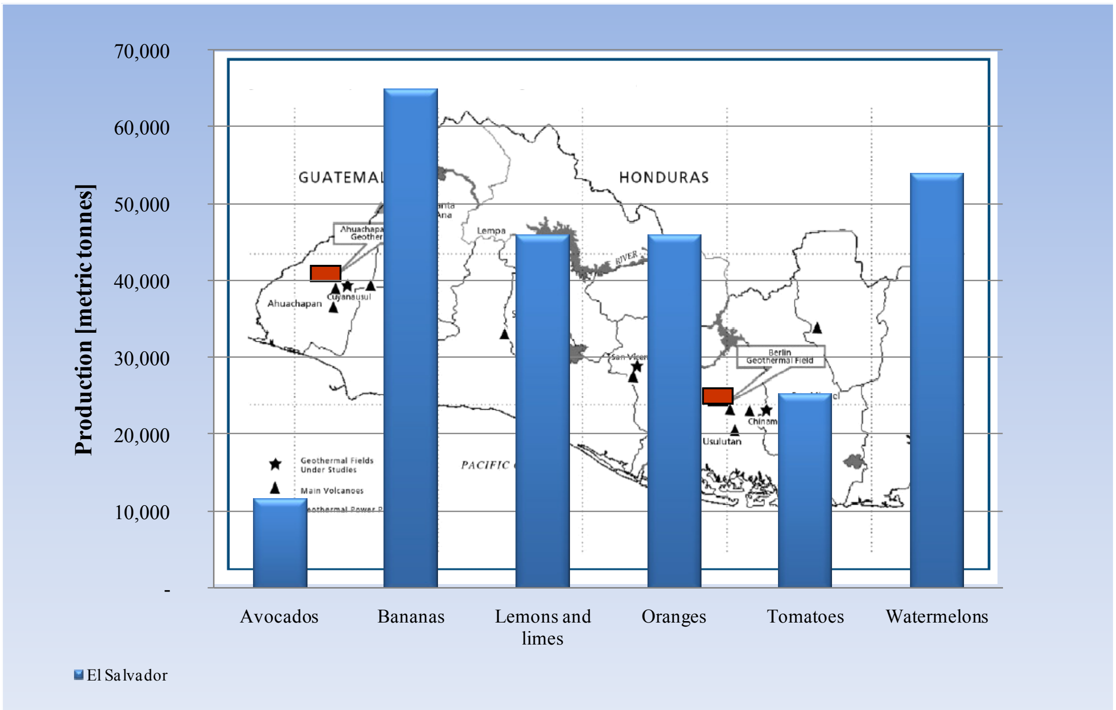


FIGURE 7: Production of various typical agricultural items for El Salvador in 2005 (metric Tonnes)

TABLE 3: Sample of storage time for refrigeration of food

Product	Storage temperature [°C]	Relative humidity [%]	Approx. storage life[a]
FISH			
Haddock, cod, perch	-0.5 - 1	95 - 100	12 days
Hake, whiting	0 - 1	95 - 100	10 days
Halibut	-0.5 - 1	95 - 100	18 days
Herring, kipper	0 - 2	80 - 90	10 days
Herring, smoked	0 - 2	80 - 90	10 days
Mackerel	0 - 1	95 - 100	6 - 8 days
Menhaden	1 - 5	95 - 100	4 - 5 days
Salmon	-0.5 - 1	95 - 100	18 days
Tuna	0 - 2	95 - 100	14 days
Frozen fish	–30 - –20	90 - 95	6 - 12 months
SHELLFISH[A]			
Scallop meat	0 - 1	95 - 100	12 days
Shrimp	-0.5 - 1	95 - 100	12 - 14 days
Lobster American	In seawater indefinitely		
Oysters, clams (meat and liquid)	5 - 10	100	5 - 8 days
Oyster in shell	0 - 2	95 - 100	5 days
Frozen shellfish	5 - 10	90 - 95	3 - 8 mo.
BEEF			
Beef, fresh, average	–2 - 1	88 - 95	1 week
Beef carcass			
Choice, 60% lean	0 - 4	85 - 90	1 - 3 weeks
Prime, 54% lean	0 - 1	85	1 - 3 weeks
Sirloin cut (choice)	0 - 1	85	1 - 3 weeks
Round cut (choice)	0 - 1	85	1 - 3 weeks
Dried, chipped	10 - 15	15	6 - 8 weeks
Liver	0	90	5 days
Veal, lean	–2 - 1	85 - 95	3 weeks
Beef, frozen	–20	90 - 95	6 - 12 months

Product	Storage temperature [°C]	Relative humidity [%]	Approx. storage life[a]
MEAT (miscell.)			
Rabbits, fresh	0 - 1	90 - 95	1 - 5 days
DAIRY PROD.			
Butter	0	75 - 85	2 - 4 weeks
Butter, frozen	–23	70 - 85	12 - 20 mo.
Cheese, cheddar			
Long storage	0 - 1	65	12 months
Short torrage	4	65	6 months
Processed	4	65	12 months
Grated	4	65	12 months
Ice cream, 10% fat	–30 to –25	90 - 95	3 - 23 mon.
Premium	–35 to –40	90 - 95	3 - 23 mon.
Milk			
Fluid, pasteurized	4 - 6		7 days
Grade A (3.7%)	0 - 1		2 - 4 months
Raw	0 - 4		2 days
Dried, whole	21	Low	6 - 9 months
Dried, non fat	7 - 21	Low	16 months
Evaporated	4		24 months
Evapor. unsweet.	21		12 months
Condens., sweet.	4		15 months
Whey, dried	21	Low	12 months
EGGS			
Shell	–1.5 - 0b	80 - 90	5 - 6 months
Shell, farm cooler	10 - 13	70 - 75	2 - 3 weeks
Frozen,			
Whole	–20		1 year plus
Yolk	–20		1 year plus
White	–20		1 year plus
Whole egg solids	1.5 - 4	Low	6 - 12 mon.

PORK			
Pork, fresh, average	0 - 1	85 - 90	3 - 7 days
Pork,carcass,47% lean	0 - 1	85 - 90	3 - 5 days
Pork, bellies,35% lean	0 - 1	85	3 - 5 days
Pork,fatback,100% fat	0 - 1	85	3 - 7 days
Pork, shoulder, 67% lean	0 - 1	85	3 - 5 days
Pork, frozen	–20	90 - 95	4 - 8 months
Ham, 74% lean	0 - 1	80 - 85	3 - 5 days
Ham, light cure	3 - 5	80 - 85	1 - 2 weeks
Ham, country	10 - 15	65 - 70	3 - 5 months
Ham, frozen	–20	90 - 95	6 - 8 months
Bacon, med. fat class	3 - 5	80 - 85	2 - 3 weeks
Bacon,cured,farm sty.	16 - 18	85	4 - 6 months
Bacon, cured, packer style	1 - 4	85	2 - 6 weeks
Bacon, frozen	–20	90 - 95	2 - 4 months
Sausage, links or bulk	0 - 1	85	1 - 7 days
Sausage,coun.smoked	0	85	1 - 3 weeks
Frankfurters,average	0	85	1 - 3 weeks
Polish style	0	85	1 - 3 weeks
LAMB			
Fresh, average	–2 - 1	85 - 90	3 - 4 weeks
Choice, lean	0	85	5 - 12 days
Leg, choice, 83% lean	0	85	5 - 12 days
Frozen	–20	90 - 95	8 - 12 months
POULTRY			
Poultry, fresh, average	–2 - 0	95 - 100	1 - 3 weeks
Chicken, all classes	–2 - 0	95 - 100	1 - 4 weeks
Turkey, all classes	–2 - 0	95 - 100	1 - 4 weeks
Turkey breast roll	–4 - –1		6 - 12 months
Turkey frankfurters	–20 - –10		6 - 16 months
Duck	–2 - 0	95 - 100	1 - 4 weeks
Poultry, frozen	–20	90 - 95	12 months

Yolk solids	1.5 - 4	Low	6 - 12 months
Flake album.solids	Room	Low	1 year plus
Dry spray albumen solids	Room	Low	1 year plus
CANDY			
Milk chocolate	–20 - 1	40	6 - 12 mon.
Peanut brittle	–20 - 1	40	1.5 - 6 mon.
Fudge	–20 - 1	65	5 - 12 mon.
Marshmallows	–20 - 1	65	3 - 9 months
MISCELLANEOUS			
Alfalfa meal	–20	70 - 75	1 year plus
Beer, keg	1.5 - 4		3 - 8 weeks
Beer, bottles& cans	1.5 - 4	65 or less	3 - 6 months
Bread	–20		3 - 13weeks
Canned goods	0 - 15	70 or lower	1 year
Cocoa	0 - 4	50 - 70	1 year plus
Coffee, green	1.5 - 3	80 - 85	2 - 4 months
Fur and fabrics	1 - 4	45 - 55	Sev. years
Honey	10	1	year plus
Hops	–2 - 0	50 - 60	Sev. months
Lard (without antioxidant)	7	90 - 95	4 - 8 months
Maple syrup	0	90 - 95	12 - 14 mo.
Nuts	0 - 10	65 - 75	8 - 12 mon.
Oil, vegetable,salad	21		1 year plus
Oleomargarine	1.5	60 - 70	1 year plus
Orange juice	–1 - 1.5		3 - 6 weeks
Popcorn, unpopped	0 - 4	85	4 - 6 weeks
Yeast, baker's compressed	-0.5 - 0		
Tobacco, hogshead	10 - 18	50 - 65	1 year
Bales	2 - 4	70 - 85	1 - 2 years
Cigarettes	2 - 8	50 - 55	6 months
Cigars	2 - 10	60 - 65	2 months

4. DESCRIPTION OF BASIC CONCEPTS OF ABSORPTION SYSTEMS

4.1 Ammonia and water absorption system

The whole thermodynamic process for an ammonia refrigeration absorption cycle is shown in Figure 8, based on Ólafsson (2007). In Figure 8, the x value refers to the ammonia content mixed with water, while T is the temperature of the mixture. For a given pressure, the bubble and dew curves are given; the bubble curve refers to the temperature at which the first drop of liquid will start to boil, and the dew curve shows the temperature at which the first condensate drop will start to be formed in the vapour phase.

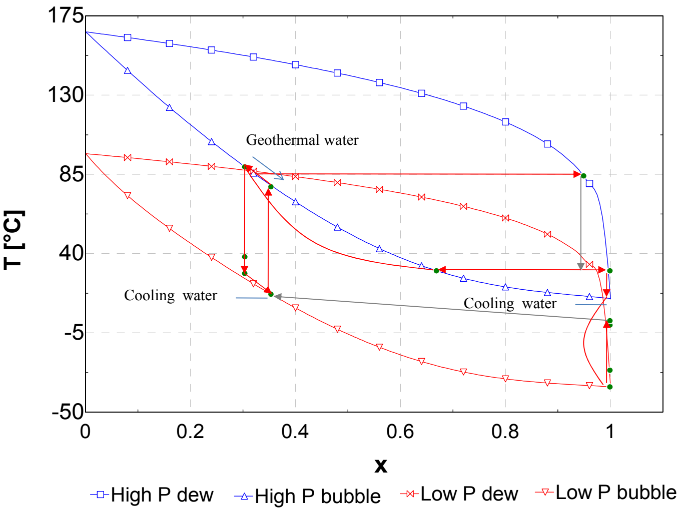


FIGURE 8: Ammonia refrigeration absorption cycle diagram

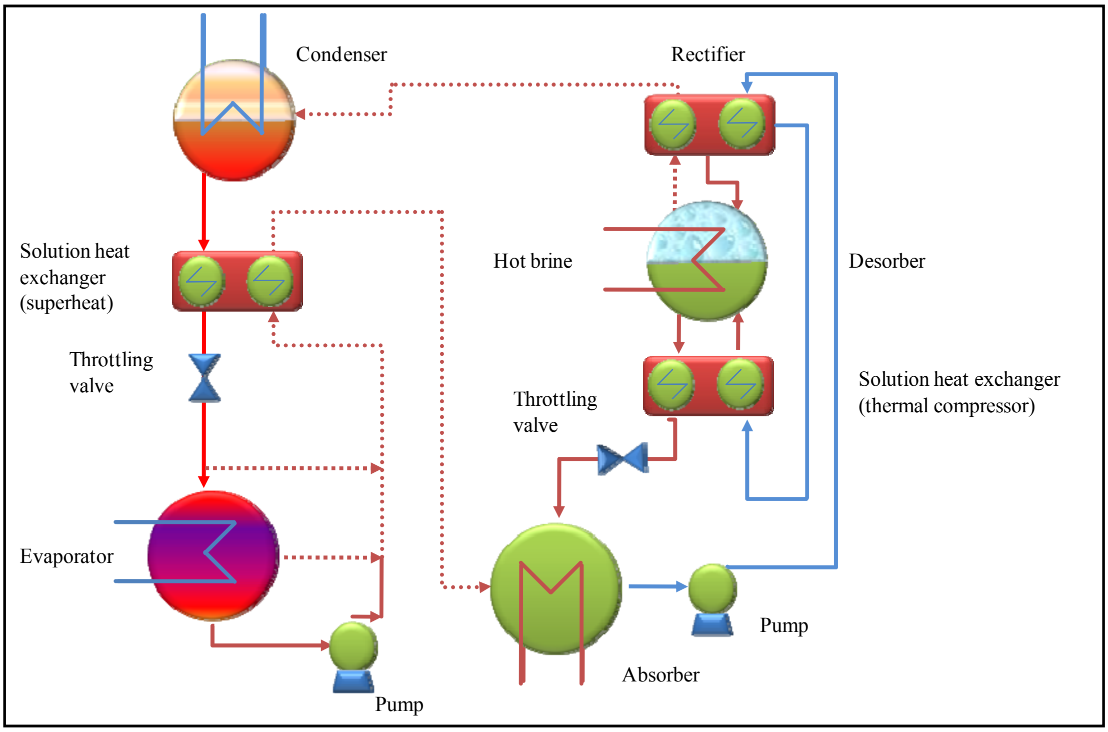


FIGURE 9: Typical absorption cycle components

Figure 9 shows a typical absorption cycle component diagram where the interconnections between the pieces of equipment can be observed. The principles of the various components are described below:

Heat exchangers: To calculate the heat exchanger area, Equation 1 is used:

$$Q = UA\Delta T_m = U \times A \times \frac{\left((T_{h2} - T_{c2}) - (T_{h1} - T_{c1})\right)}{\ln\left[\dfrac{T_{h2} - T_{c2}}{T_{h1} - T_{c2}}\right]} \tag{1}$$

where Q = Heat transfer in heat exchanger [W];
U = Overall heat transfer coefficient [W/(m^2 ºC)];
A = Area [m^2];
T_{h1} = Input temperature of hot fluid [ºC];
T_{h2} = Output temperature of hot fluid [ºC];
T_{c2} = Input temperature of cold fluid [ºC];
T_{c1} = Output temperature of cold fluid [ºC].

The estimated U values for the heat exchanger are shown in Table 4:

TABLE 4: Assumed U values for heat exchangers [kW/m^{2}K]

Heat exchangers	**U - value [kW/(m^{2}K)]**	**Type**
Condenser	1.1	Shell & tube
Solution hx. Thermal compr.)	1.8	Plate hx
Solution hx. (NH_3 superheat)	0.2	Plate hx
Absorber	0.785	Shell & tube
Desorber	1.2 boiling / 0.8 heating	Shell & tube
Rectifier	0.2	Shell & plate
Evaporator	1.2	Shell & tube

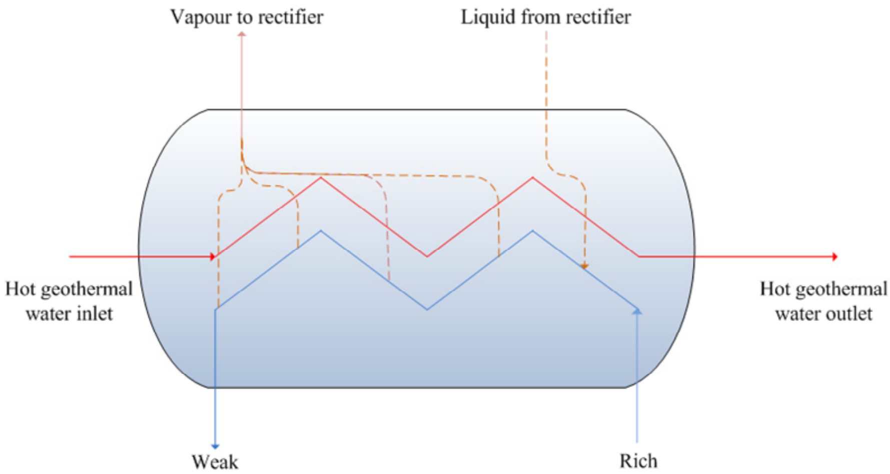


FIGURE 10: Generator or desorber, schematic diagram for the ammonia reference cycle

Desorber generator: In the desorber or generator, the heat source exchanges heat with the working fluid: ammonia - water or Lithium Bromide – water mixture, as shown in Figure 10. The heat gained by the working fluid is used to evaporate the refrigerant substance while most of the carrying fluid remains liquid due to the evaporation temperature difference.

Once the refrigerant changes in phase, the vapour and liquid are separated into two different pipes with the vapour connected to the rectifier and the liquid connected to the pre-heater and the absorber. When the mixture is heated, the vapour reaches an intermediate refrigerant concentration, whereas the liquid reduces its refrigerant concentration. The general equations are the following (Björnsdóttir, 2004):

The energy balance:

$$m_{rich} * h_{rich} + Q_{geothermal} = m_{weak} * h_{weak} + m_{vap-NH_3} * h_{vap-NH_3} \tag{2}$$

the ammonia balance:

$$m_{rich} * x_{rich} = m_{weak} * x_{weak} + m_{vap-NH_3} * x_{vap-NH_3} \tag{3}$$

and the mass balance:

$$m_{rich} = m_{weak} + m_{vap-NH_3} \tag{4}$$

where m = Mass [kg];
h = Enthalpy [kJ/kg];
Q = Energy [kJ]);
x = Vapour fraction.

Rectifier: The vapour coming from the absorber does not work well as a refrigerant because it still has some water content. Therefore, the fluid is passed into the rectifier, to cool and condense the remaining water, becoming a richer, almost pure, solution in the process. The condensed water is collected and sent back to the desorber. Equations 5 and 6 show the energy and mass balance, respectively, for this process:

$$m_{NH_3H_2O-hot_{in}} \times h_{NH_3H_2O-hot_{in}} = m_{vap-NH_3} \times h_{vap-NH_3} + m_{H_2O-liq} \times h_{H_2O-liq} + Q_r \tag{5}$$

$$m_{NH_3H_2O-hot_{in}} = m_{vap-NH_3} + m_{H_2O-liq} \tag{6}$$

Condenser: The condenser is a heat exchanger in which the refrigerant coming from the Desorber is cooled down to liquid form by rejecting heat to the cooling media (cooling water that may come from a river or a cooling tower) as shown in Figure 11. The vapour refrigerant is transformed into liquid in order to extract more heat from the environment to be cooled or refrigerated, and to achieve a lower ambient temperature. The condenser is then connected to a heat exchanger and a throttle valve.

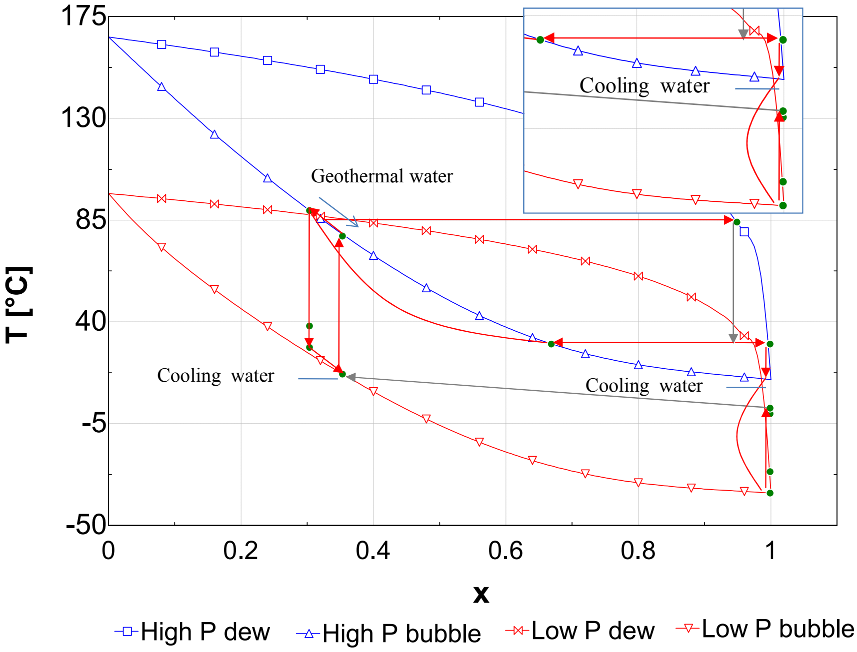


FIGURE 11: The thermodynamic cycle for ammonia (condenser)

The pinch point in the condenser heat exchanger can be located either on the cold end where refrigerant vapour enters the equipment or at the point where the refrigerant starts to condense. The energy balance equation for the condenser is given according to Equation 7:

$$m_{NH_3} \times (h_{NH_3-vap} - h_{NH_3-liq}) = m_{cool} \times (h_{cool-in} - h_{cool-out}) \tag{7}$$

Figure 12 shows the temperature profile of the refrigerant in the condenser.

The throttling valve is located between the condenser and the evaporator. Its main function is to drop the pressure in the liquid refrigerant and, thus, reduce its boiling temperature which allows for the heat exchange between fluid and refrigeration space.

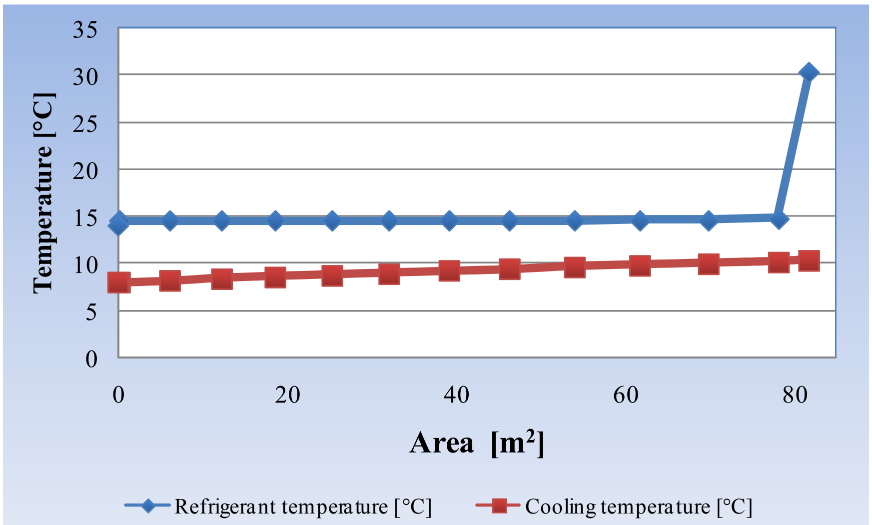


FIGURE 12: Condenser temperature profile

The evaporator is a unit where the refrigerant takes heat from a secondary fluid which is in direct contact with equipment in the room being refrigerated or cooled. At one point, it receives low-temperature and low-pressure liquid refrigerant which, when heated, changes into vapour and leaves the equipment. At another point, it receives the secondary fluid leaving the facility; when it passes the evaporator, it is cooled and returns to the same facility to maintain certain ambient conditions. The evaporation process could be at constant temperature and pressure, or present a slight temperature difference when there is a presence of another substance in the refrigerant.

There are two types of evaporators, the flooded and the direct type. Special care must be taken in the selection of evaporators due to problems related to leakages of refrigeration liquid into the compressor or leakages from the oil compressor into the refrigerant.

The energy balance equation in the evaporator is according to Equation 8:

$$m_{NH_3-liq} \times (h_{NH_3-liq-in} - h_{NH_3-vap-out}) = m_{air} \times (h_{air-out} - h_{air-in}) \tag{8}$$

In *the absorber* the refrigerant is absorbed back into the carrying fluid, in order to carry the refrigerant in a liquid form from a low-pressure state to a high-pressure state. In the heat exchanger, two different fluid flows take place: vapour refrigerant coming from the evaporator, and weak liquid carrying the solution coming from the generator. Cooling fluid is also passed through the exchanger. In this case, the outlet cooling water coming from the condenser, being some degrees hotter, will then be used to cool the fluid in the absorber. The cooling part of the absorber is needed in order to eliminate heat arising while condensing the vapour refrigerant, and to achieve a higher concentration of ammonia in the rich solution. If the condensing sink temperature supplied is not low enough, it will result in a lower concentration of the rich solution and the efficiency of the whole cycle will be reduced.

The energy balance in the absorber is:

$$m_{weak} \times h_{weak} + m_{ref-vap} \times h_{ref-vap} = m_{rich} \times h_{rich} + Q_{absorber} \tag{9}$$

the ammonia balance is:

$$m_{weak} \times x_{weak} + m_{ref-vap} \times x_{ref-vap} = m_{rich} \times x_{rich} \tag{10}$$

and the mass balance is:

$$m_{weak} + m_{ref-vap} = m_{rich} \tag{11}$$

The pump and the throttling valve: The pump increases the pressure of the rich solution coming from the absorber at low temperature, and sends it to the rectifier, solution heat exchanger and desorber. The throttling valve reduces the pressure of the weak solution from the high-pressure equipment to the pressure of the absorber.

The solution heat exchanger receives the weak solution coming from the desorber and going to the throttling valve on one side, and on the other the rich solution coming from the rectifier and going to the desorber. Its function is to increase the coefficient of performance (COP).

4.2 Lithium-Bromide water cooling absorption cycle

The cooling absorption cycle works using the same principle as the absorption refrigeration cycle. The main difference is that in this cycle, water is used as the refrigerant, not as the carrying fluid. Because of this, if no antifreeze is added to the water, the minimum cooling temperature that the cycle can achieve is the freezing temperature of water. Lithium Bromide solution works as the carrying fluid and has a higher boiling temperature than water. Neither the rectifier nor the solution heat exchanger between the condenser and evaporator are needed in the system as shown in Figure 13.

Similar equations apply for the components. The desorber or generator balances are shown in Equations 12, 13 and 14:

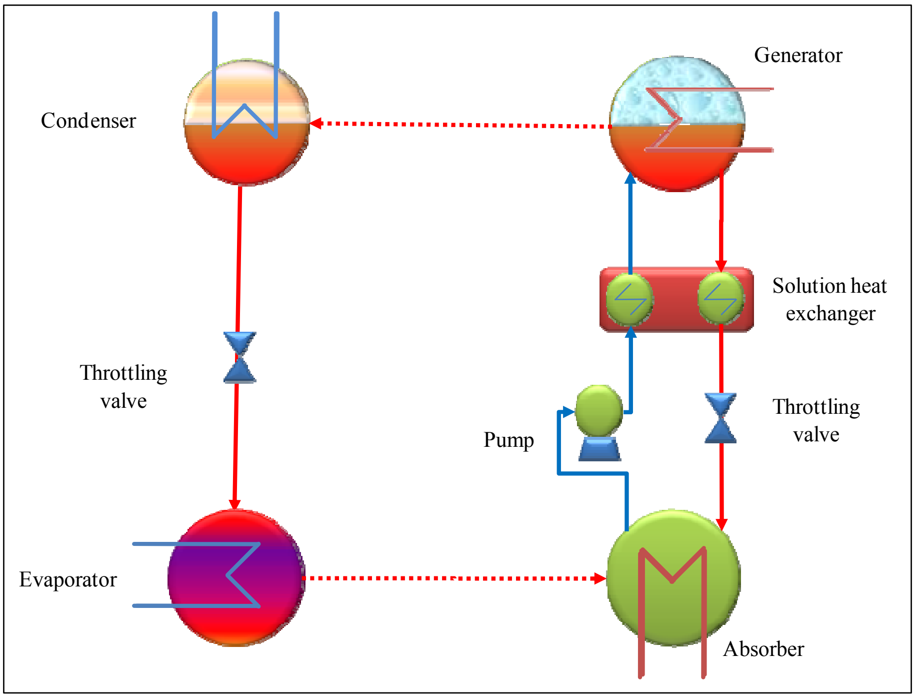


FIGURE 13: Absorption cooling cycle with Lithium Bromide system components

$$m_{rich} = m_{steam} + m_{weak} \tag{12}$$

$$m_{rich} \times h_{rich} + Q_{geothermal} = m_{steam} \times h_{steam} + m_{weak} \times h_{weak} \tag{13}$$

$$m_{rich} \times x_{rich} = m_{weak} \times x_{weak} + m_{steam} \times x_{steam} \tag{14}$$

Figure 14 shows the temperature profile for the generator in the cooling absorption cycle.

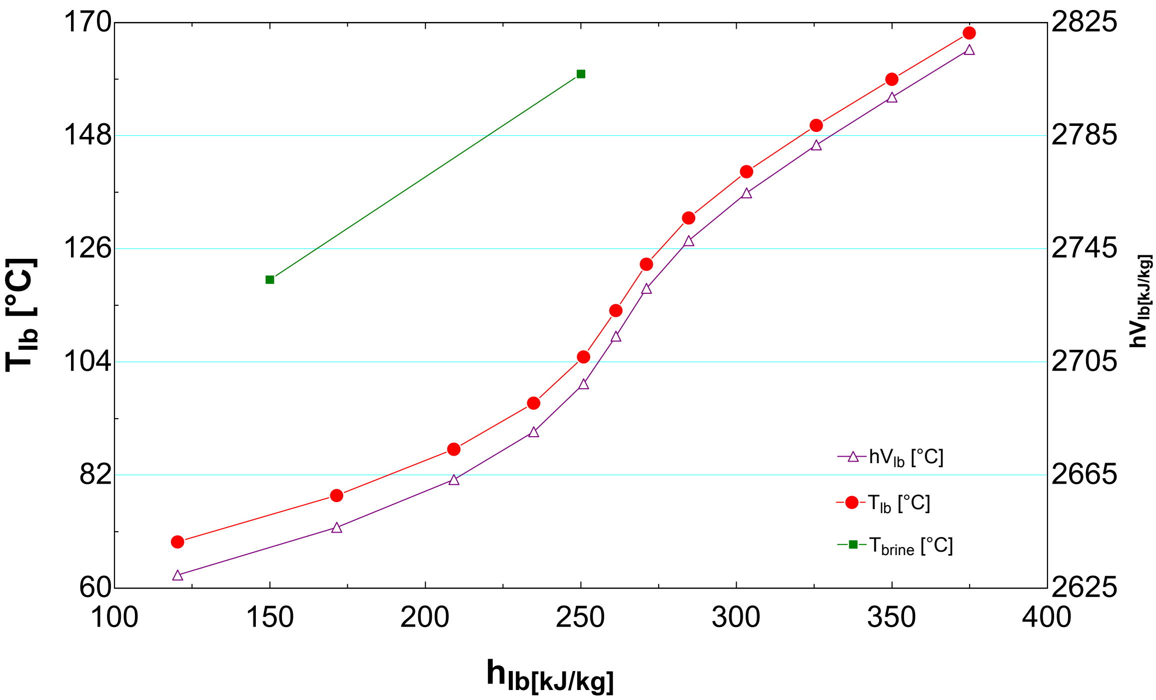


FIGURE 14: Desorber temperature profile for the Lithium Bromide cooling cycle

Condenser balance equations are given in Equations 15, 16 and 17:

$$m_{steam} = m_{liquid} \tag{15}$$

$$m_{steam} \times h_{steam} = m_{liquid} \times h_{liquid} + Q_{condenser} \tag{16}$$

$$m_{steam} = m_{liquid} \tag{17}$$

Figure 15 shows the temperature profile for the condenser in the absorption cooling cycle.

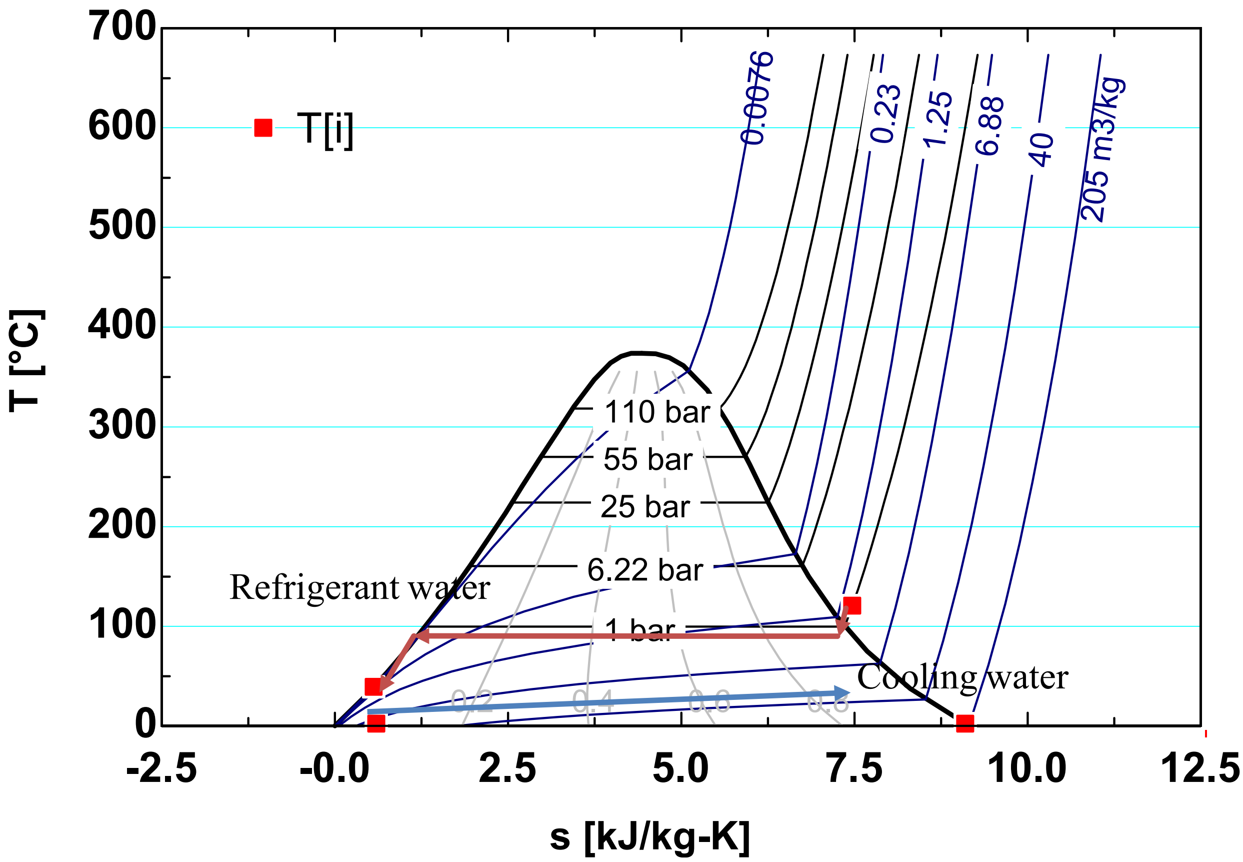


FIGURE 15: Condenser temperature profile

The evaporator balance equations are given in Equations 18 and 19:

$$m_{liq} \times h_{liq} + Q_{evaporator} = m_{vap} \times h_{vap} \tag{18}$$

$$m_{steam} = m_{liquid} \tag{19}$$

And the absorber balance equations are given in Equations 20, 21 and 22:

$$m_{steam} + m_{weak} = m_{rich} \tag{20}$$

$$m_{steam} \times h_{steam} + m_{weak} \times h_{weak} = m_{rich} \times h_{rich} + Q_{absorber} \tag{21}$$

$$m_{steam} \times x_{steam} + m_{weak} \times x_{weak} = m_{rich} \times x_{rich} \tag{22}$$

5. RESULTS

5.1 Project objective

The purpose of the project is to estimate the cost of installing and operating an absorption refrigeration system for the Ahuachapán and Berlín geothermal fields in El Salvador using geothermal brine as the heat source with ammonia and water as the carrying fluid, and using it to refrigerate beef (sample product) on the site of the power plant. A technical evaluation of the cooling cycle using geothermal brine as a heat source with a Lithium Bromide water mixture as the carrying fluid with no specific product will be included for both geothermal fields. The cooling load will just be assumed.

5.2 Berlín and Ahuachapán field fluid characteristics

Berlín geothermal field generates 94 MWe gross, but a new binary power station is under construction which will increase the gross generation by 9 MW. The field is producing with 13 wells and reinjection into 16 wells. The mass flow of the field is shown in Figure 16 (LaGeo S.A. de C.V., 2007). The production drop in Figure 16 is due to the fact that unit 3 was under maintenance, and the wells that feed this unit were closed.

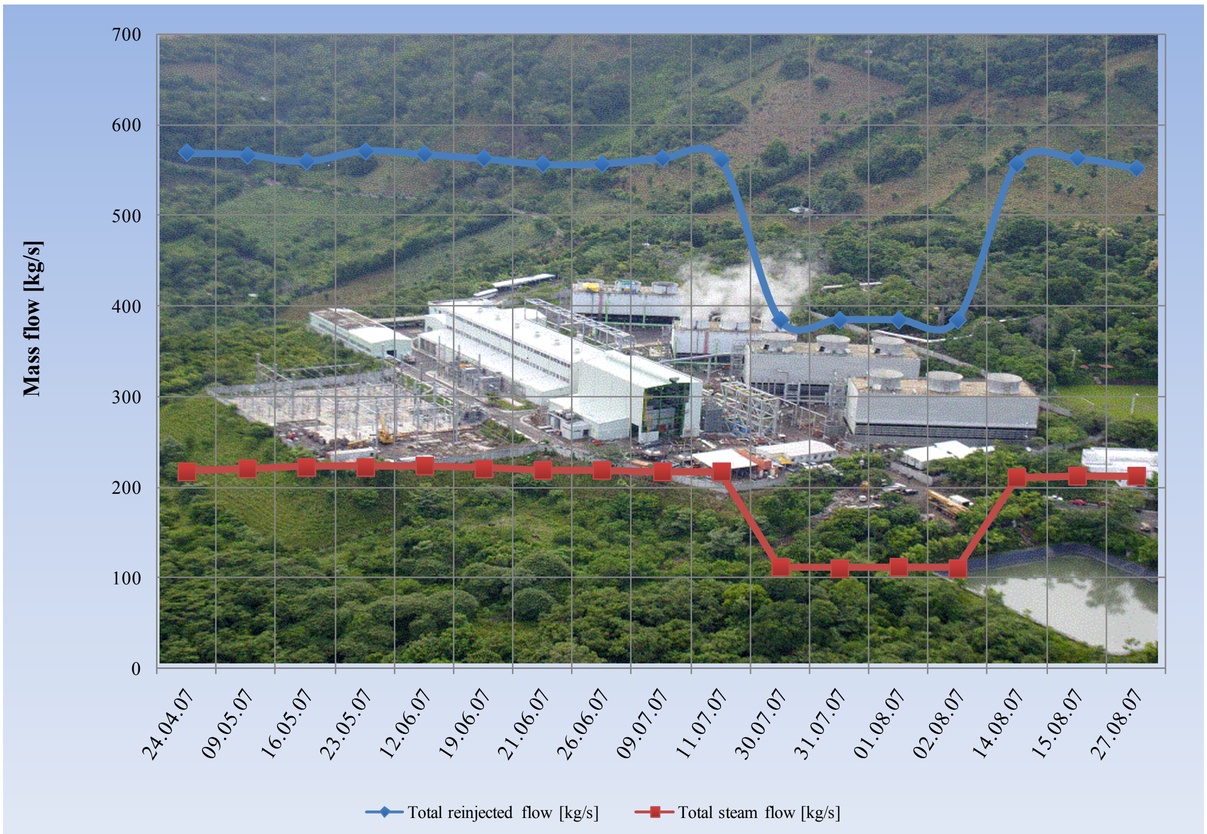


FIGURE 16: Berlín geothermal field mass flow production [kg/s]

So far all the water is being reinjected into the field at separation temperature, but when the new binary plant goes into operation, a brine mass flow of about 382 kg/s will be directed to the heat exchangers of the binary plant and its temperature will drop from 180 to 140°C which, due to silica deposition, is the minimum working temperature for the reinjection system. Hence, the available geothermal water that can be taken into account for the absorption refrigeration system is around 187 kg/s at a separation temperature of 180°C, shown in Figure 17. The experiments carried out on the Berlín geothermal brine for silica deposition are summarized in Table 5 (Molina et al., 2005). They

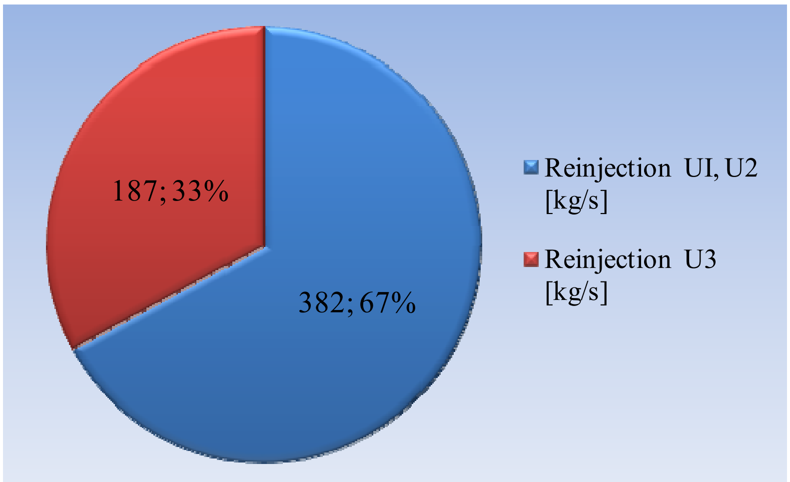


FIGURE 17: Berlín reinjection distribution [kg/s]

show that dosing with HCl to maintain a pH between 5.5 and 6, the temperature of 140°C gives a safe lower value for reinjection.

TABLE 5: Berlín experimental tests for silica deposition

Rates for tubing size [“)	Brine treatment option	Silica deposition range [g/Tonne]	
		From	To
1/4	HCl, pH 5.0	1 at 150ºC	1 at 90ºC
	HCl, pH 5.5	1 at 150ºC	3 at 90ºC
	Millisperse 5 ppm	3 at 120ºC	3 at 65ºC
	Drew 20 ppm	1 at 140ºC	5 at 100ºC
	Millisperse 10 ppm	1 at 140ºC	12 at 90ºC
	Drew 10 ppm	Insufficient data to assess	
3/4	HCl, pH 5.0	1 at 150ºC	1 at 90ºC
	HCl, pH 5.5	1.5 at 150ºC	1.5 at 65ºC
	Millisperse 5 ppm	1 at 120ºC	2 at 65ºC
	Drew 20 ppm	1 at 140ºC	3 at 100ºC
	Millisperse 10 ppm	1 at 140ºC	4 at 100ºC
	Drew 10 ppm	Insufficient data to assess	

The Ahuachapán geothermal field is located in the western part of El Salvador. There are 16 production wells and 4 reinjection wells for steam and water production, which is shown in Figure 18 (LaGeo S.A. de C.V., 2007).

Ahuachapán geothermal field is experiencing an increase in its mass flow production due to project “Ahuachapán Optimization”. This has meant drilling of more production wells in order to replace the present production wells with new ones, and shift the reservoir production zone. Ultimately, it will also result in some increase in the total mass flowrate due to larger diameter production casings in the new wells. Ahuachapán geothermal power station is also expected to have its own binary cycle. Some part of the separated geothermal water need to go directly to the binary plant heat exchangers without going to the second flash process, as shown in Figure 19. Subsequently, all the water in Ahuachapán geothermal field will be used for electricity production. The separation temperature of the water will then be reduced from about 165°C to about 95°C. If a cooling or refrigeration cycle were to be used in Ahuachapán field, it would have to be supplied from wells currently producing electricity, and care must then be taken to evaluate the reservoir response to extra mass extraction.

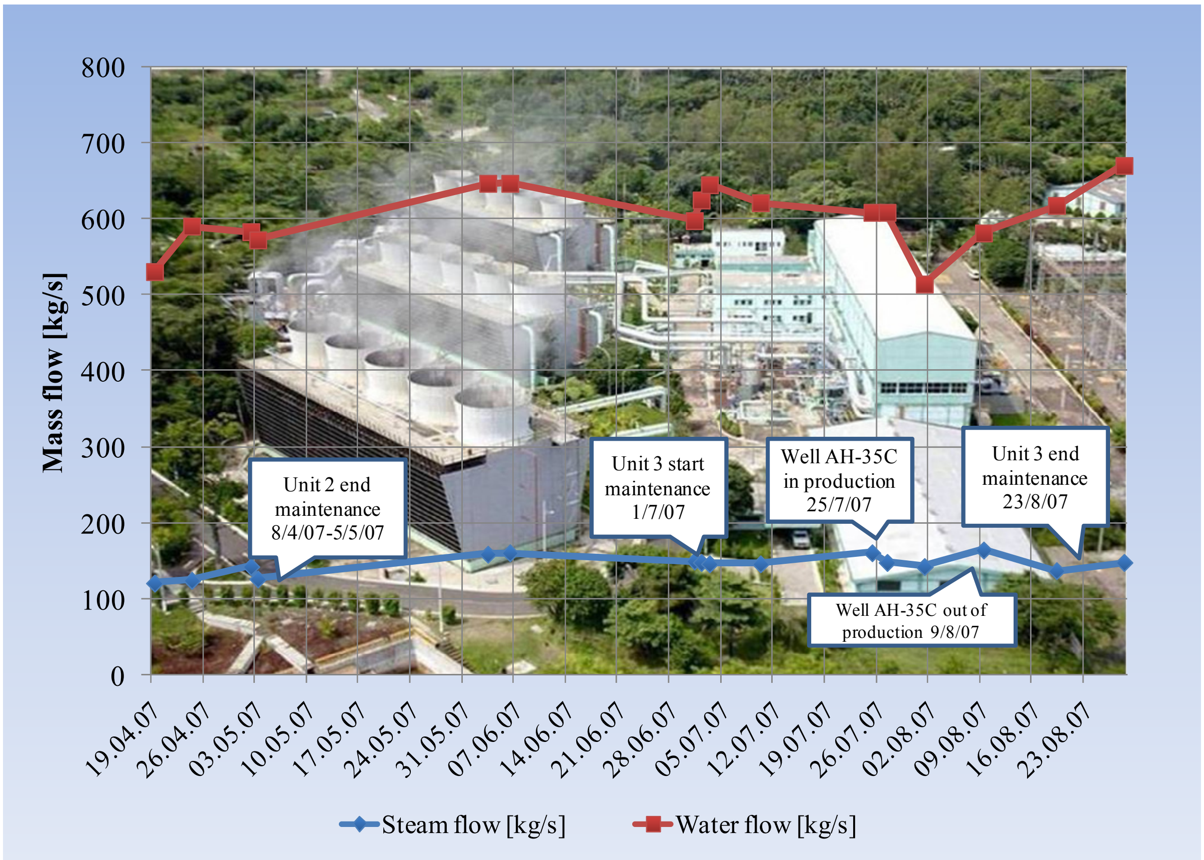


FIGURE 18: Ahuachapán mass flow production [kg/s]

Silica deposition tests shown in Table 6 (Guerra et al., 2005), revealed that for the Ahuachapán geothermal brine being cooled down to 90°C, it would require acid dosing. This temperature will be considered a limit for the use of any cooling or refrigerating cycle.

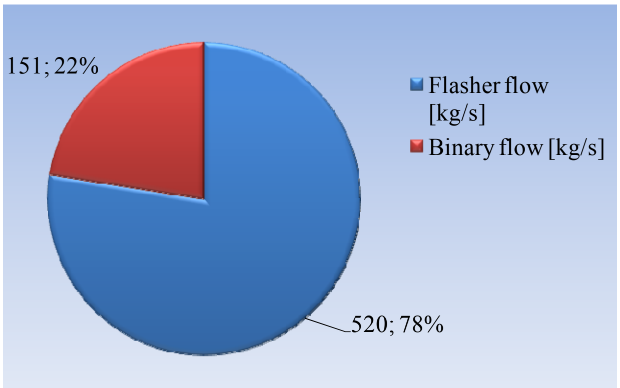


FIGURE 19: Ahuachapán brine use distribution [kg/s]

5.3 Cooling load example for refrigeration application

For the case of a refrigerating load for a typical chilling process above freezing, calculations were taken from the ASHRAE refrigeration handbook (ASHRAE, 2002). It included the inputs and assumptions for the product to be chilled, trucks with offal, chilled from 38 down to 1°C within 2 hours, as shown in Table 7.

The heat gain from the room surfaces is calculated with Equation 23.

$$q = UA\Delta T \tag{23}$$

where q = Heat gain through walls, floors and ceiling [kW];
A = Outside area of section [m^2];
ΔT = Temperature difference between outside air and air of the refrigerated space [°C];
U = Overall heat transfer coefficient [W/m^2 K].

TABLE 6: Ahuachapán silica deposition trials

Tubing size ["]	**Brine treatment option**	**Silica deposition rates range [g/Tonne]**	
		From	**To**
1/4	Untreated brine	0.21 to 105°C	1.34 to 60°C
	HCL to pH = 6.0	1.01 to 90°C	0.16 to 80°C
	HCL to pH = 5.5	0.55 to 90°C	0.2 to 80°C
	Millisperse 5 ppm	0.24 to 90°C	0.22 to 80°C
	Millisperse 10 ppm	0.26 to 90°C	1.09 to 80°C
	Drew 10 ppm	0.25 to 90°C	0.07 to 80°C
	Drew 20 ppm	0.23 to 90°C	0.15 to 80°C
3/8	Untreated brine	0.23 to 105°C	0.15 to 60°C
	HCL to pH = 6.0	1.11 to 90°C	0.21 to 80°C
	HCL to pH = 5.5	0.35 to 90°C	0.21 to 80°C
	Millisperse 5 ppm	0.02 to 90°C	0.32 to 80°C
	Millisperse 10 ppm	0.54 to 90°C	0.04 to 80°C
	Drew 10 ppm	0.15 to 90°C	0.17 to 80°C
	Drew 20 ppm	0.29 to 90°C	0.66 to 80°C

TABLE 7: Refrigeration load calculation assumptions

Refrigeration assumptions	
Product average initial temperature [ºC]	38
Product average final temperature [°C]	1
Room temperature [°C]	-18
Outdoor temperature [ºC]	26
Outdoor RH [%]	70
Room area [m^2]	2500
Volume of the room [m^3]	10000
Room air changes per 24 hours	12
Room gain [W/m^{2}K]	0.426
Time to refrigerate [hours]	2

The overall heat transfer coefficient U is calculated with Equation 24:

$$U = \frac{1}{\frac{1}{h_i}+\frac{x}{k}+\frac{1}{h_o}} \tag{24}$$

where x = Wall thickness [m];
k = Thermal conductivity of wall material [W/m K];
h = Inside surface conductance [W/m^2 K];
h_o = Outside surface conductance [W/m^2 K].

The surface conductance is assumed the same as for still air. The room gain the U value in W/m^{2}K is given, not calculated. For the amount of mass for the offal and truck, the results are shown in Table 8. The heat gain from air infiltration is calculated with Equation 25:

$$q_i = \frac{(h_i-h_r)\times V\times RAC}{(24\times 3600)} \tag{25}$$

where h_i = Enthalpy of infiltration air [kJ/kg];
h_r = Enthalpy of refrigerated air [kJ/kg];
V = Volume of the refrigerated space [m^3];
RAC = Room air changes in 24 hours.

TABLE 8: Mass conversions for the offal and truck sample

Mass conversions	
kg of meat/truck	330
kg of truck	180
Number of trucks	34
Time to refrigerate [hours]	2
kg/s of meat	1.558
kg/s of truck	0.85

The change in enthalpy can be calculated from the psychrometric chart, shown in Figure 20, with the temperatures of the refrigerated air and the outdoor air, considering that the relative humidity of the air inside the room is 100%. The results of the calculation are shown in Table 9.

TABLE 9: Result of volumetric energy calculation

From psychrometric chart	
Heat from infiltration air [kJ/m^3]	95.3

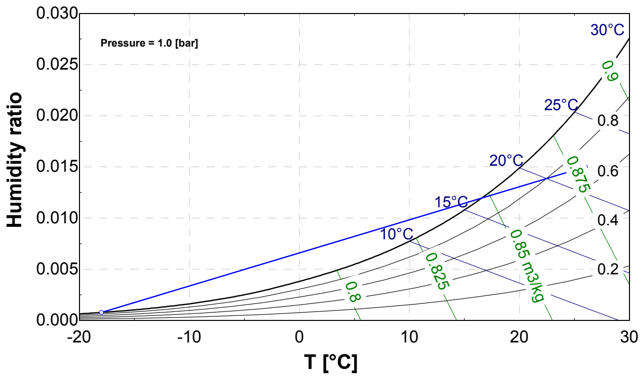


FIGURE 20: Psychrometric chart

Other assumptions are related to the heat gained for the electrical motors and lighting system. The values are shown in Table 10.

TABLE 10: Assumption for the electrical motors and lighting system

Assumptions	
Fan and motor [kW]	7.5
Light [kW]	0.2

The general formula for the refrigeration load is shown in Equation 26:

$$q_r = m_m \times c_m \times (t_1 - t_2) + m_t \times c_t \times (t_1 - t_2) + q_w + q_i + q_m \tag{26}$$

The definitions are given in Table 11, which shows the results of the calculation.

TABLE 11: Heat load calculation results

q_r	Refrigeration load [kW]	414.5
m_m	Mass flow of meat [kg/s]	1.6
m_t	Mass flow of trucks [kg/s]	0.9
c_m	Specific heat of meat [kJ/kg K]	3.14
c_t	Specific heat of truck, containers or platforms (0.5 for steel) [kJ/kg K]	0.5
t_1	Average initial temperature [ºC]	38
t_2	Average final temperature [ºC]	1
q_m	Heat gain from the meat [kW]	181.0
q_t	Heat gain from the trucks [kW]	15.725
q_w	Heat gain through room surfaces [kW]	46.86
q_i	Heat gain through infiltration [kW]	132.36
q_m	Heat gain from equipment and lighting [kW]	38.5
t_m	Time to refrigerate [hours]	2

It is always a recommended to have a safety factor of 20% over the estimated value, to allow for the possible discrepancies between the actual and design situations; the final results are shown in Table 12.

TABLE 12: Cooling calculation with safety factor

Factor used for computing refrigeration load [%]	20
Refrigeration load, q_r [kW]	497.4

5.4 Thermodynamic absorption cycles for the Berlín and Ahuachapán geothermal fields

Figure 21 shows the thermodynamic processes for the Berlín geothermal field for the ammonia-water case, while Figure 22 shows the cooling and refrigeration process for the Ahuachapán geothermal field.

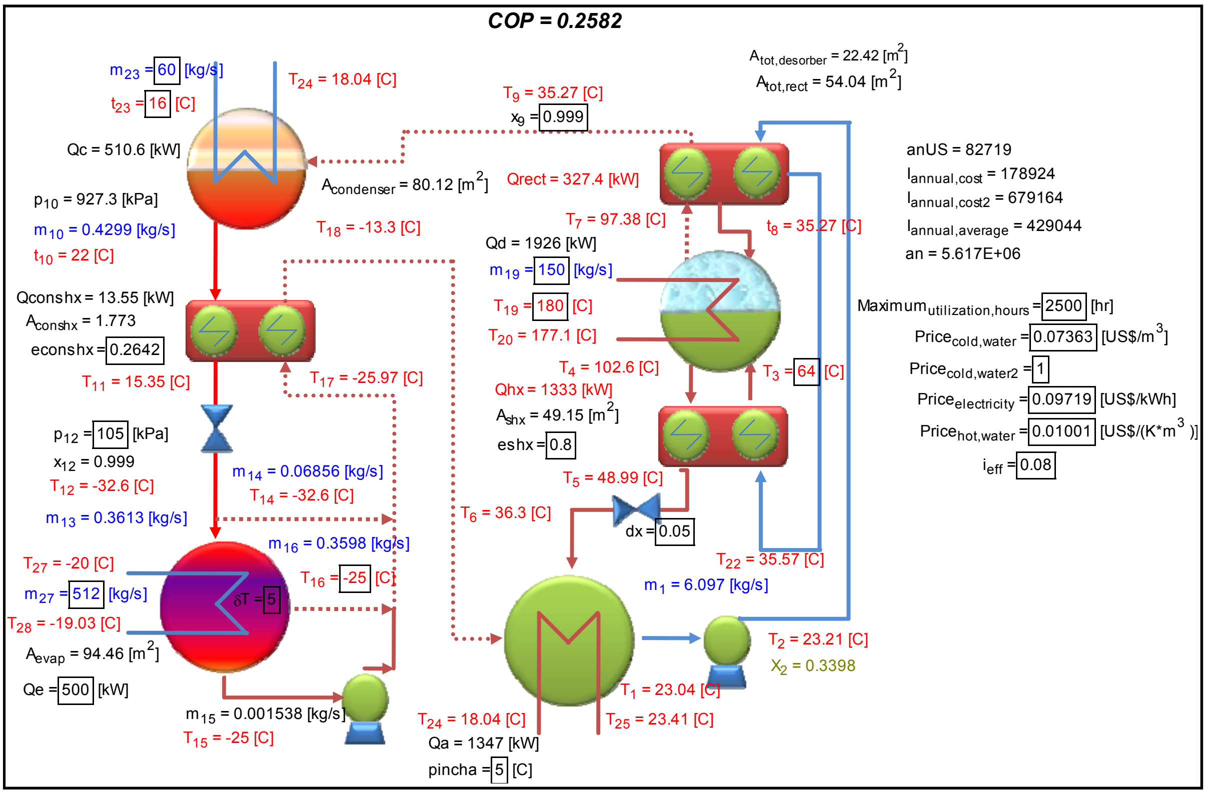


FIGURE 21: Berlín ammonia water absorption refrigeration process

Figure 23 shows that due to the tropical environmental conditions, the COP was reduced to very low values when trying to increase the cooling temperature for the condenser and absorber for either Berlín or Ahuachapán. For conditions in El Salvador, a cooling water temperature higher than 28-29°C will be required, assuming that a cooling tower is used. Therefore, a cooling absorption refrigeration cycle in combination with an ammonia refrigeration cycle is proposed in order to provide the low cooling water temperature required. A schematic configuration is shown in Figure 24.

It is important to take into consideration the following assumptions:

- The cost of the cooling absorption refrigeration cycle is not included in the cost of operation and installation; it will have to be added on in further calculations.

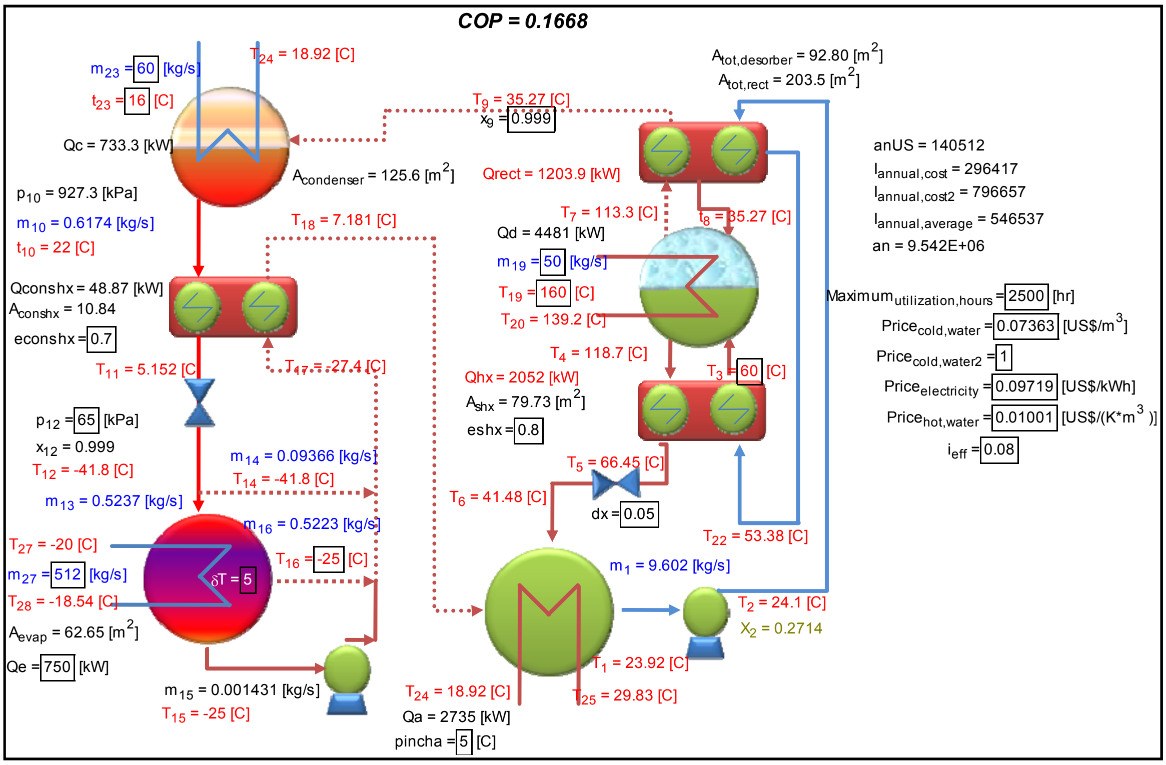


FIGURE 22: Ahuachapán ammonia refrigeration process

- The cost of installing and operating the absorption refrigeration cycle is estimated for temperatures of T = 8 and 16°C.

- It is important to note that the evaporator heat load and evaporator temperatures for the cooling cycle will not be in correspondence with those of the refrigeration cycle for the condenser and absorber. Further calculations will be needed to continue with this option. The Lithium Bromide cooling cycle will only address the basic concepts of the system for an assumed case.

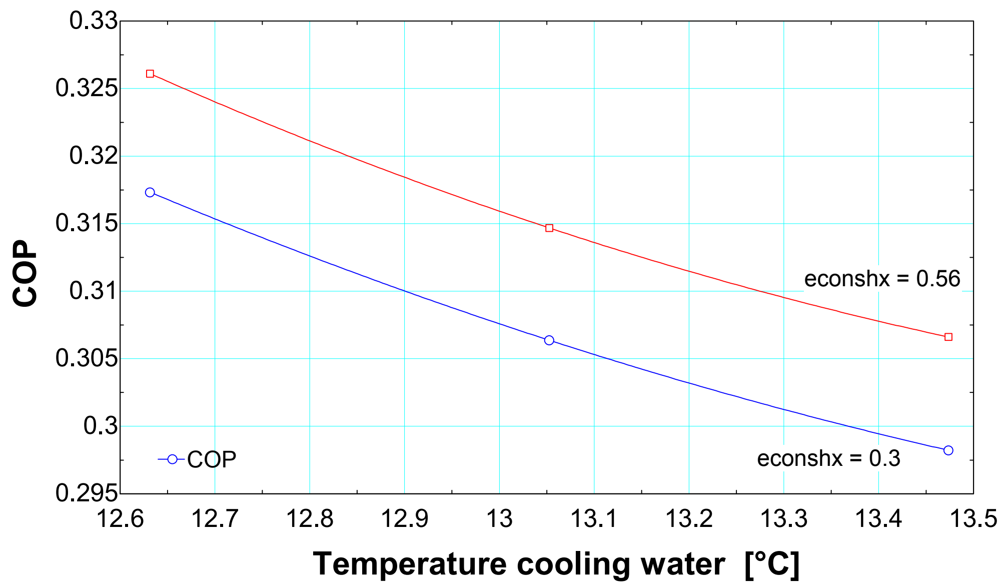
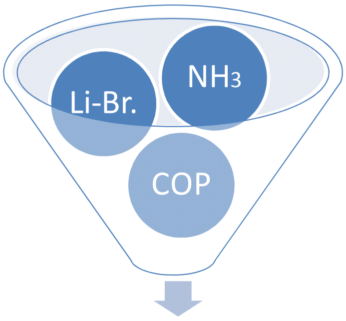


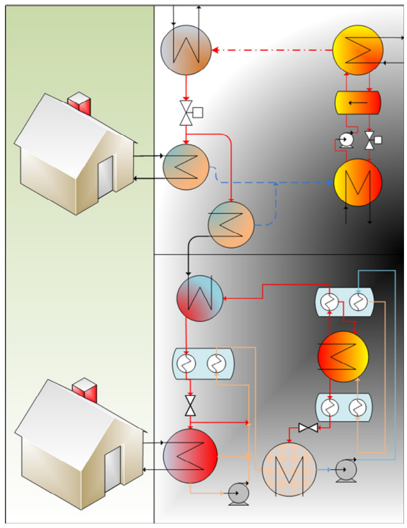
FIGURE 23: Refrigeration COP sensibility with respect to
the cooling temperature at condenser and absorber;
econshx = Effectiveness of the condenser solution for the heat exchanger



Propose of a combined solution with Li-Br and NH_3 systems.

Continue with refrigeration partial cost estimation plus a cost sensibility analysis respect to cooling temperature.

Cooling cost to be out of the estimation.



Combined cooling and refrigeration cycle

FIGURE 24: Proposed configuration for providing low water temperature in tropical climate

5.5 Alternatives for location of the cooling facility

The absorption cooling system should be located in the vicinity of the power station or some distance away from it in order to provide better access to mains roads for easier transportation of products. The case is only analyzed for the Lithium Bromide absorption system where the refrigerant substance is water instead of ammonia for the refrigeration cycle.

The first option is to have the condenser and the evaporator away from the power station, as shown in Figure 25, at a site close to main access roads. This location has the following implications:

- A second cooling tower must be installed in the cooling facility; the cooled water will be used for the condenser;
- A network of pipelines carrying steam at very low pressure will be part of the facility installation and operation;
- A pump could be used in the system to overcome pressure drop in the steam pipeline from the power station to the cooling storage facility.

The second option is to have all the equipment at the power station site, and a heat exchanger in the cooling storage site facility for cooled water, as shown in Figure 26.

This option has the following implications:

- The same cooling tower for the condenser and the absorber;
- A pump might be required to overcome the pressure drop in the cooled water pipeline from the power station to the storage cooling facility some kilometres away;
- A well insulated network of pipelines for carrying cooled water must be part of the installation and operation.

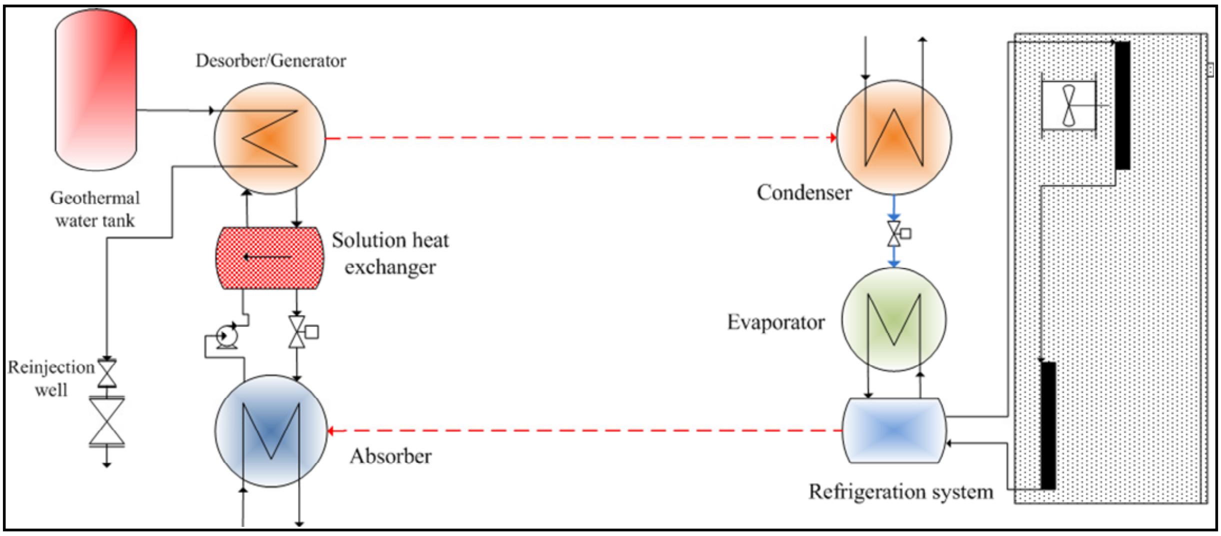


FIGURE 25: First case with condenser and evaporator located away from the power station

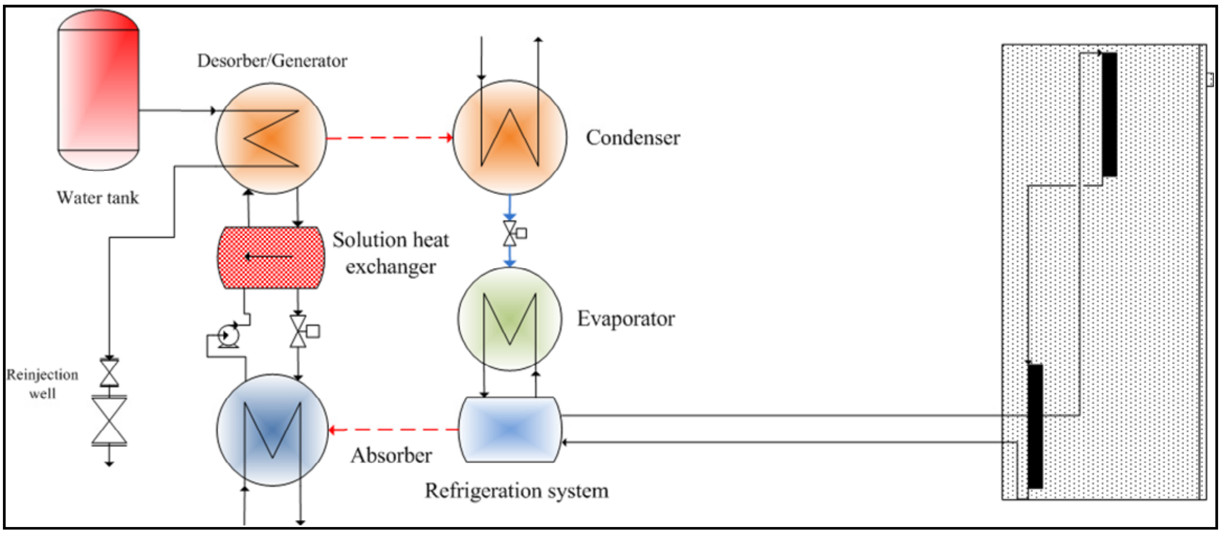


FIGURE 26: Second case with all the main cooling equipment at the power station site

To evaluate the conditions for both scenarios, the same program in EES - Engineering Equation Solver (F-Chart Software, 2004) was used for the evaluation of the technical aspects for the Lithium Bromide cooling cycle, and a pressure drop in the pipelines added to the calculation. Assumptions considered in the pressure drop calculation include:

- There is no change in the level of the terrain; i.e. all is flat to avoid including the piezometric pressure part of the Bernoulli equation;
- The pipes are well insulated and no temperature drop to the environment is taken into account;
- The Colebrook-White equation was used for the calculation of the friction factor of the pipes (F-Chart, Software, 2004).

Results from the EES calculations for the Berlín geothermal field program are shown in Figure 27 (Rogowska and Szaflik, 2005).

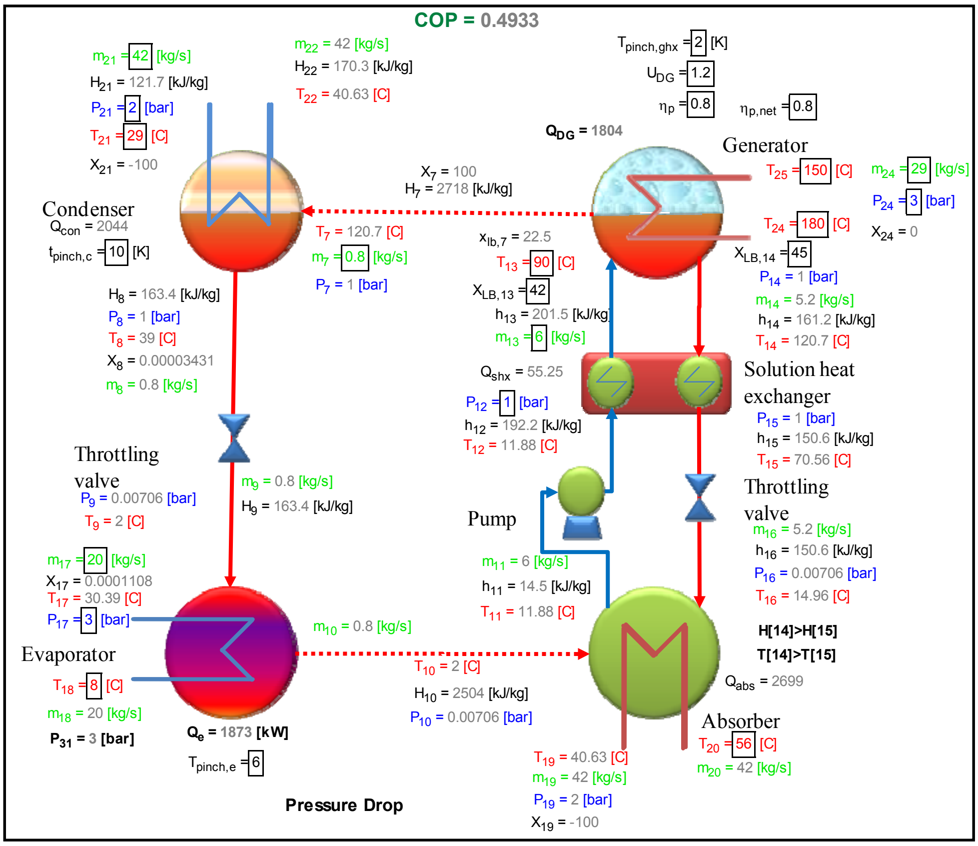


FIGURE 27: The Lithium Bromide cooling process for the Berlín field

Similarly, the results from the EES calculations for the Ahuachapán field are shown in Figure 28.

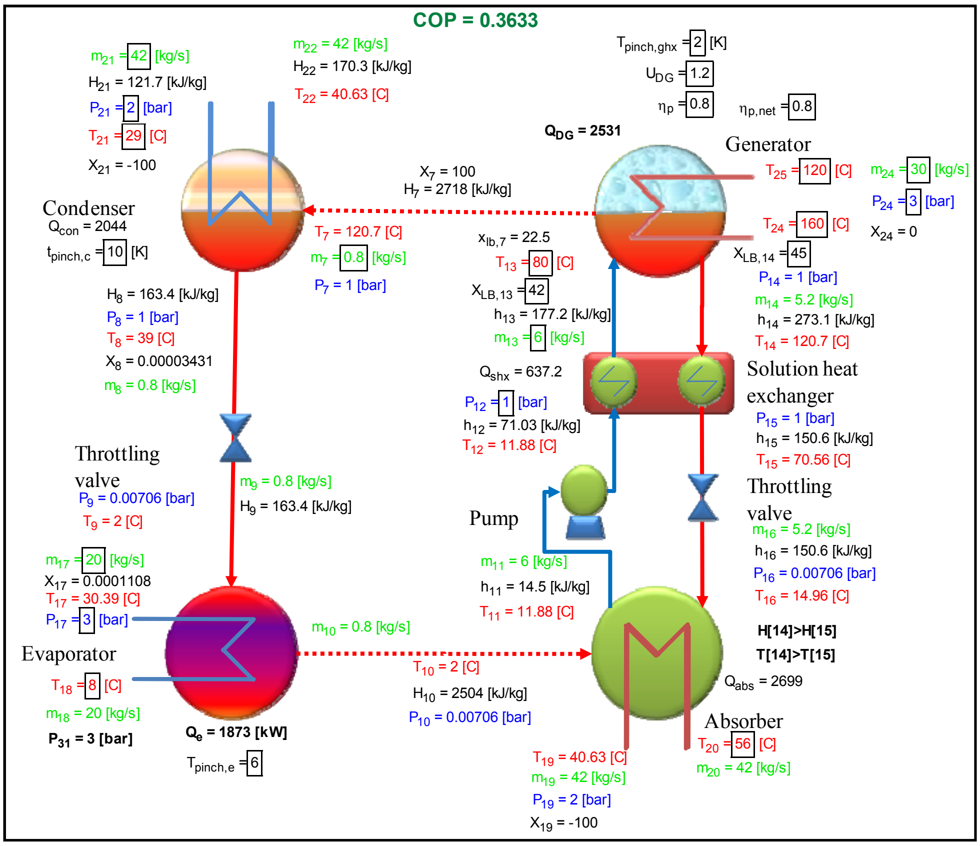


FIGURE 28: The Lithium Bromide cooling process for the Ahuachapán field

The results are addressed at the end of this main section. It can though be stated that the Lithium Bromide cycle for cooling uses low refrigerant pressures. Because of this, the specific fluid volume is large, resulting in large diameter pipelines which are more expensive. A cooled water gathering system results in smaller pipeline diameters and is therefore the better option.

5.6 Capital and operating cost estimations

5.6.1 The Berlín geothermal field

The capital cost for equipment was calculated according to Equations 27, 28 and 29 (Ólafsson, 2007):

$$I_{heatExchanger} = C_o * A^{0.8} \tag{27}$$

$$I_{pump} = C_1 * (0.01 * P)^{0.6} \tag{28}$$

$$C_{PE,Y} = C_{PE,W} * \left[\frac{X_Y}{X_w}\right]^{\alpha} \tag{29}$$

where I = Investment cost of equipment;
C_o = Constant, indicating cost for each m^2 of heat exchanging area;
C_1 = Constant, indicating cost for each 100 kW of pumping power;
A = Heat exchanging area [m^2];
P = Pumping power [kW];
$CP_{E,Y}$ = Cost for purchased equipment Y (as W);
X_Y = For component involved, power in boilers, compressors, etc.
α = For each piece of equipment, $\alpha_{evaporator} = 0.54$; $\alpha_{compressor} = 0.95$; etc.

The $CP_{E,Y}$ formula is valid for the following ranges:

Evaporators: 10 - 1000 m^2;
Compressors: 0.05 - 8 MW.

The estimated values are:

C_o = USD 736.2; for plate heat exchangers;
C_o = USD 1,104 .4; for shell and tube heat exchangers;
C_1 = USD 11,044.4; per 100 kW, not linear.

The capital costs were then estimated and are shown in Table 13. It is important to note that there are different scenarios evaluated and calculated due to the sensibility of the temperature at the condenser and absorber inlets on the total cost. El Salvador has warm climate conditions and this will limit the coefficient of performance operation of the refrigeration cycle.

The annual cost from investment is calculated with Equation 30:

$$An_j = C_{FC,j} * \left\{\frac{i_{\text{eff}}}{1-(1+i_{\text{eff}})^{-n}}\right\} \tag{30}$$

where An_j = Annuity (capital cost) for the project;
$C_{FC,j}$ = Fixed cost for project;
i_{eff} = Annual effective rate of return;
n = Number of years where the project is operated.

TABLE 13: Berlín field capital costs for the refrigeration cycle

	Investment cost (8°C) 0.5 MW [USD]	**Investment cost (16°C) 0.5 MW [USD]**
Condenser	37,459	37,312
Solution hx. (thermal compressor)	14,564	16,536
Solution hx. (NH_3 superheat)	5,121	1,783
Absorber	74,894	83,927
Desorber	7,895	12,028
Rectifier	14,345	27,160
Evaporator	47,053	42,009
Pumps	-	-
Thermal compressor pump	1,473	1,473
Evaporator bleed pump	1,473	1,473
Annual cost from investment – annual capital cost	**204,277**	**223,701**

The cost breakdown structure for the Berlín field is shown in Table 14.

TABLE 14: Berlín field, cost breakdown for the refrigeration cycle

I. Fixed Capital Investment (FCI)	**0.5 MW power T=8°C [USD]**	**0.5 MW power T=16°C [USD]**
A. Direct cost (DC)		
1. Onsite cost (ONSC) – Purchased equipment cost (PEC):		
1. Heat Exchangers	201,329	220,755
2. Pumps	2,403	2,992
3. Pipes in system (5% of 1 +2)	10,187	11,187
4. Electrical control & monitoring system (30% of 1+2+3)	64,176	70,480
Total onsite cost:	*278,095*	*305,414*
2. Offsite cost (OFSC)		
1. Civil structural and architectural work 20% of ONSC	55,619	61,083
2. Service facilities (hot source &codl. sink connection) (25% ONSC)	69,524	76,353
3. Contingencies (15% of ONSC)	41,714	45,812
Total direct cost (DC)	**444,952**	**488,662**
B. Indirect cost (IDC)		
1. Engineering and supervision (15% of DC)	66,743	73,299
2. Construction cost including contractor´s profit (15% of DC)	66,743	73,299
3. Contingencies (20% of DC)	88,990	97,732
Fixed capital investment, total	**667,428**	**732,992**
II. Other outlays		
A. Start up cost (6% of FCI)	40,046	43,980
B. Working capital (5% of FCI)	33,371	36,650
C. Cost of licensing, research and development, {2,3,4}	29,452	29,452
Total capital investment (TCI)	**770,297**	**843,074**

The estimated parameters for the calculation of the annual equivalent investment cost are shown in Table 15:

TABLE 15: Estimated parameters for the annual equivalent investment cost

Type	N	i_{eff}	Annual cost factor
Absorption refrigeration	25	0.1	11%

To calculate the cost of electricity, the formula in Equation 31 was used (Ólafsson, 2007):

$$E_c = \left[\frac{5*[m_{\text{geo}}+m_{\text{ctower}}]*9.81}{0.6*1000} + W_p + W_{p2} + \frac{Q_e}{80}\right] * hr * P_e \tag{31}$$

where E_c = Electricity cost [USD];
m_{geo} = Mass flow of hot geothermal water [kg/s];
m_{ctower}: = Mass flow of the cooling water for the condenser and absorber equipment [kg/s];
W_p = Work of the pump [kW];
hr = Maximum utilization hours [hour];
P_e = Price of electricity [USD].

For calculating the hot water cost, the formula in Equation 32 was used:

$$HW_c = P_{hw} \times m_{hw} * (T_{in-hot} - T_{out-hot}) \times 3.6 \times hr \tag{32}$$

where HW_c = Hot water cost [USD];
P_{hw} = Price of hot geothermal water [USD/(kg °C)];
m_{hw} = Mass flow of hot geothermal water [kg/s];
T_{in-hot} = Inlet temperature of hot geothermal water to the desorber [°C];
$T_{out-hot}$ = Outlet temperature of hot geothermal water coming from desorber [°C];
hr = Maximum utilization hours [h].

For calculating the cost of the cold water, the formula in Equation 33 was used:

$$CW_c = P_{cw} \times hr \times 3.6 \times m_{ctower} \tag{33}$$

where CW_C = Cold water cost [USD];
P_{cw} = Cold water price [USD/kg];
m_{ctower} = Mass flow rate of cooling water going to the condenser, absorber [kg/s].

For calculating the operation and maintenance cost, the formula in Equation 34 was used:

$$OM_c = TCI * 0.005 + \left[\frac{2E6*Q_e*hr}{(1000*8000)}\right] \tag{34}$$

where OM_C = Operation and maintenance cost [USD];
TCI = Total capital investment [USD];
Q_e = Evaporator heat [kW];

Hence, the total annual costs can be calculated for the Berlín refrigeration cycle (Table 16).

According to Figure 29, the change in the cooling inlet temperature in the condenser and absorber will generate a reduction in the COP (Coefficient of performance of the cycle) and will also have a significant impact on the cost of the equipment, mainly on the rectifier with about a 90% increase.

TABLE 16: Total annual cost results for the Berlín refrigeration cycle

Total cost	**0.5 MW power T=8°C [USD]**	**0.5 MW power T=16°C [USD]**
Annual capital cost	76,299.00	83,117.00
Electricity cost	6,966	7,956
Hot water cost	22,416	32,403
Cold water cost (1)	29,820	36,447
Cold water cost (2)	405,000	495,000
Operation and maintenance	8,674	9,038
Total with cw 1	144,175	168,961
Total with cw 2	519,355	627,514
Total annual cost	**331,765**	**398,238**

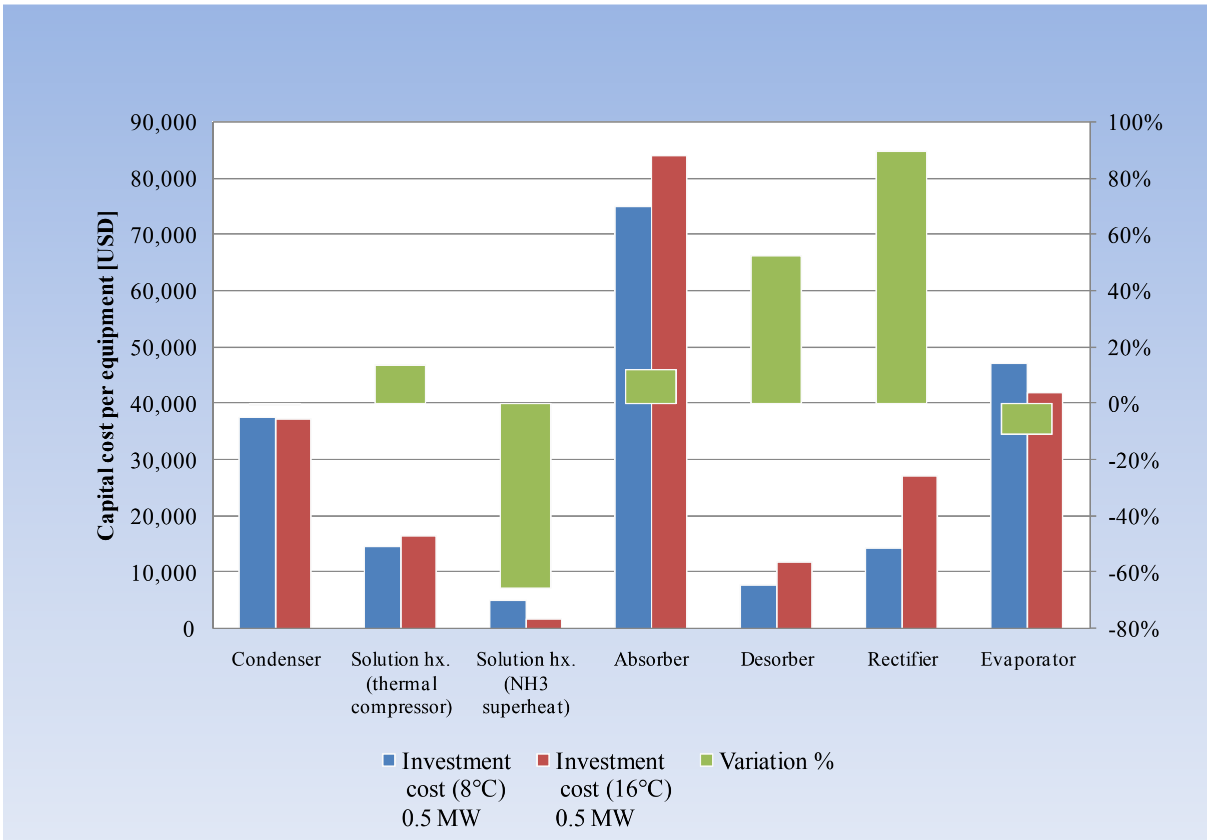


FIGURE 29: Berlín refrigeration capital cost with condenser inlet temperature and variation of it (right hand side)

5.6.2 Capital and operating cost estimation for the Ahuachapán geothermal field

The same calculation method was applied for the case of Ahuachapán geothermal field, and the results are shown in Tables 17, 18 and 19.

TABLE 17: Capital costs for the refrigeration cycle for the Ahuachapán field

	Investment cost (8°C) 0.75 MW [USD]	**Investment cost (16°C) 0.75 MW [USD]**
Condenser	58,122	52,756
Solution hx. (thermal compressor)	21,068	24,454
Solution hx. (NH_3 superheated)	7,705	7,435
Absorber	129,465	154,433
Desorber	15,435	41,417
Rectifier	25,701	77,622
Evaporator	44,539	30,246
Pumps	-	-
Thermal compressor pump	1,473	1,473
Evaporator bleed pump	1,473	1,473
Annual cost from investment - annual capital cost	**304,981**	**391,309**

TABLE 18: Ahuachapán, cost breakdown for the refrigeration cycle

I. Fixed capital investment (FCI)	**0.75 MW power T=8°C [USD]**	**0.75 MW power T=16°C [USD]**
A. Direct cost (DC)		
1. Onsite cost (ONSC) purchased equipment cost(PEC):		
1. Heat exchangers	302,036	388,364
2. Pumps	3,116	3,871
3. Pipes in system (5% of 1 +2)	15,258	19,612
4. Electrical control & monitor system (30% of 1+2+3)	96,123	123,554
Total onsite cost:	*416,533*	*535,401*
2. Offsite cost (OFSC)		
1. Civil structural and architectural work 20% of ONSC	83,306	107,080
2. Service facilities (hot source & cold sink connection) (25% ONSC)	104,133	133,850
3. Contingencies (15% of ONSC)	62,480	80,310
Total direct cost (DC)	**666,452**	**856,641**
B. Indirect cost (IDC)		
1. Engineer. and supervis. (15% of DC)	99,968	128,496
2. Construction cost including contractor´s profit (15% of DC)	99,968	128,496
3. Contingencies (20% of DC)	133,290	171,328
Fixed capital icome, total	**999,678**	**1,284,961**
II. Other outlays		
A. Start up cost (6% of FCI)	59,981	77,098
B. Working capital (5% of FCI)	49,984	64,248
C. Cost of licensing, research and development, {2,3,4}	29,452	29,452
Total capital investment (TCI)	**1,139,095**	**1,455,759**

TABLE 19: Total annual cost results for Ahuachapán refrigeration cycle

	0.75 MW power T=8°C [USD]	**0.75 MW power T=16°C [USD]**
Annual capital cost	110,847	140,512
Electricity cost	6,664	8,235
Hot water cost	38,028	93,507
Cold water cost (1)	29,820	39,760
Cold water cost (2)	405,000	540,000
Operation and maintenance	12,819	14,402
Total with cw 1	198,178	296,416
Total with cw 2	573,358	796,656
Total annual cost	**385,768**	**546,536**

Figure 30 shows the increase and decrease of equipment costs when changing the cooling temperature at the entrance of the condenser and absorber.

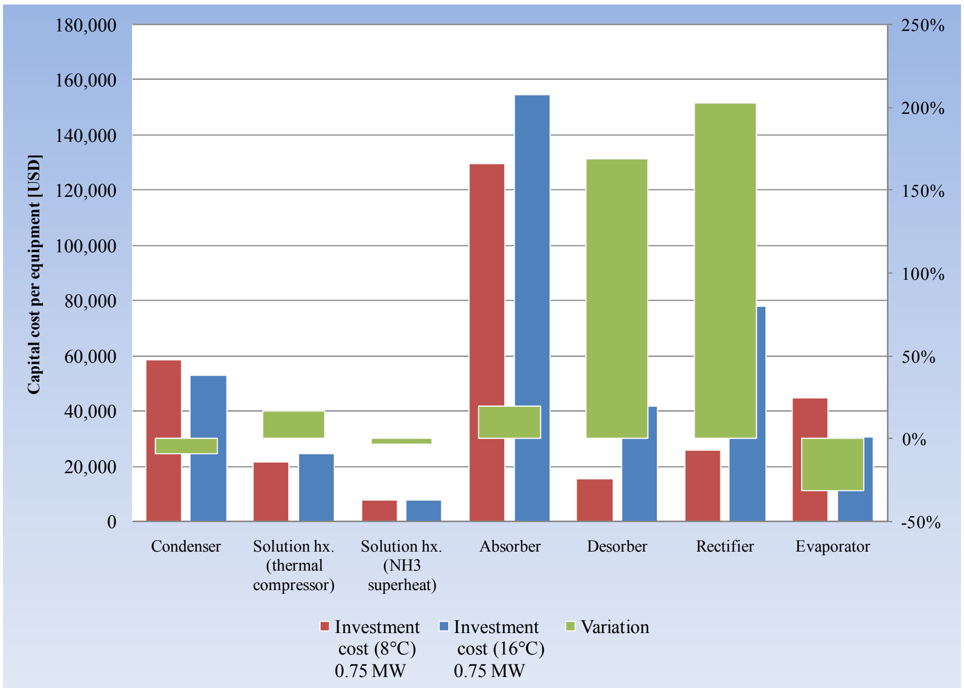


FIGURE 30: Ahuachapán refrigeration capital cost variation with condenser inlet temperature

Table 20 shows the comparison of the costs of installing and operating the ammonia and water absorption refrigeration cooling cycle for the two fields, while Figure 31 shows the same results graphically. It is important to note that the refrigeration load for the Ahuachapán refrigeration cycle is 0.75 MW, which is different from the Berlín geothermal cycle which has a refrigeration load of 0.5 MW. Therefore, a better comparison will result from temperature changes in the cooling water.

TABLE 20: Cost comparison for the refrigeration projects in the Berlín and Ahuachapán fields

	T=8°C [USD]	**T=16°C [USD]**
Berlín – total annual cost	331,765	398,238
Ahuachapán – total annual cost	385,768	546,536

5.7 Results

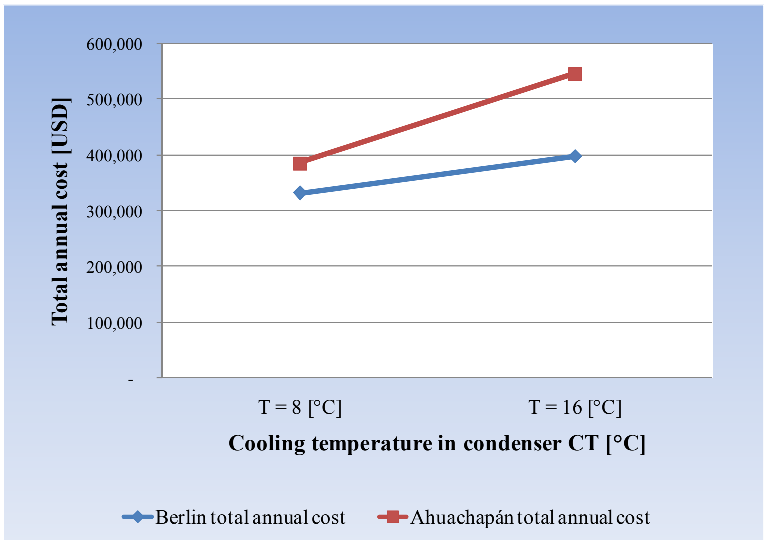


FIGURE 31: Comparison of the total annual cost vs. cooling temperature for both plants

- For the cooling cycle with the Lithium Bromide fluid, the temperature for the cooling fluid at the condenser and absorber could be increased to reasonable values for a cooling tower.

- When evaluating the location of the equipment in the cooling cycle, stationing the evaporator and condenser away from the power station resulted in greater pipe diameters for the gathering system due to the low pressure of the steam and high specific volume.

- Changing the cooling water temperature from 8 to 16°C, led to a reduction in the COP for the Berlín field, from 0.44 to 0.25; and the cost increased about 20% with respect to the total annual cost.

- Changing the cooling water temperature from 8 to 16°C, led to a reduction in the COP for the Ahuachapán field from 0.4 to 0.16; and the cost increased about 41%, with respect to the total annual cost.

6. CONCLUSIONS

The study shows that the refrigeration absorption cycle is not adaptable to tropical climates under the report's considerations in which a cooling tower is assumed to be the supplier of cold fluid to the condenser and absorber equipment, where temperatures need to be as low as possible. However, there is an alternative: a combined cascading scenario of the cooling and refrigerating cycle. Here, the cooling facility could be supplying, not only the refrigeration cycle, but also to other users who might need it for storage of daily products, cured meat, or as was proposed at the beginning of this report, to hydroponic plantations with no traditional products. The cooling cycle also has the flexibility of being placed at a distance from the power station, by transporting the cooled water through a well insulated gathering system. In general, a further study is needed to evaluate the feasibility of installing and operating the cooling load cycle with a Lithium Bromide mixture. From a simple technical point of view, there are acceptable thermodynamic conditions for the continued study.

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