AALBORG UNIVERSITY
STUDENT REPORT

Absorption Heat Pump for Air Conditioning Driven by Fuel Cell Waste Heat

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SYNOPSIS: Air conditioning in electric vehicles is nowadays a challenge. An alternative as substituting the traditional heat pump, which runs with electricity, by an absorption heat pump, which is heat driven, could be a solution in order to the range of the vehicle not being reduced by switching on the air conditioning. In this project, this alternative system will be modelled and validated, and a size optimisation will be carried out, with the purpose of studying its feasibility. As a first step, the cooling necessities of a vehicle dealing with the worst possible ambient conditions are studied. Moreover, various sets of equations that provide the thermodynamic properties of the fluids involved are given and validated.

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Appendix:	8
Supplements:	-

By signing this document, each member of the group confirms that all group members have participated in the project work, and thereby all members are collectively liable for the contents of the report. Furthermore, all group members confirm that the report does not include plagiarism.

Summary

Absorption heat pumps are able to provide cooling effect employing heat as input. This heat can be waste heat from sources as fuel cells, or solar heat. The advantage this kind of heat pump has over the conventional is that they do not need electricity to run. In the case of the application in electric vehicles, traditional heat pumps use electricity from their battery, decreasing the range of the vehicle, while absorption ones do not affect the range.

The source of heat for the absorption heat pump concerning this project is a fuel cell. If a hybrid vehicle is considered, a fuel cell is what provides electricity for the motor to work, producing at the same time a high amount of heat that can be used as input in the generator of the absorption heat pump.

The cooling necessities of a vehicle depend on the environmental conditions to which they are exposed. In order to know which these necessities in different situations are, a model will be developed in EES that allows to calculate them as a function of various inputs as the ambient temperature, the velocity of the outside air, or the velocity of the car while driving.

Once the cooling demand for the proposed system is known, a model of the system will be carried out and validated, what allows to compute the heat input required by the generator to meet the cooling demand in the evaporator. Available data in literature are used for the correspondent validation.

The model of the system has been written in this case using EES, which contains the properties of the fluids present in the cycle in an external library. With the purpose of facilitating the possible development of models of the same system in other kind of software that do not contain these properties, various ways of calculating them have been studied and validated.

Finally, the application of the absorption heat pump to a vehicle requires that the size of the system is small enough to fit in it. For this reason, a size optimisation as a function of one of the parameters of the system has been carried out. There it can be seen that the proposed system seems to be feasible, though further studies could be useful.

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Nomenclature

Roman Letter Symbols

A	Area, m^2
C	Mass flow rate multiplied by the correspondent specific heat, kW/K
COP	Coefficient of Performance
cp	Specific heat, kJ/kgK
d	Diameter, m
d_h	Hydraulic diameter, m
e	Error
EES	Engineering Equation Solver
EPA	United States Environmental Protection Agency
EV	Electric Vehicle
F	Friction factor
F	Correction factor
FCV	Fuel Cell Vehicles
h	Enthalpy, kJ/kg
h	Convection coefficient, kW/m^2K
$\bar{h}$	Mean convection coefficient, kW/m^2K
h’	Enthalpy of saturated pure liquid water, kJ/kg
h^e	Specific excess enthalpy, kJ/kg
H_m^e	Molar excess enthalpy of the mixture, kJ/kg
ICE	Internal Combustion Engines
k	Conduction coefficient, kW/mK
L	Length, m
LMTD	Logarithmic Mean Temperature Difference, K
M	Molar mass, kg/mol
$\dot{m}$	Mass flow rate, kg/s
NTU	Number of Transfer Units
Nu	Nusselt number
p	Pressure, kPa
Pr	Prandtl number
Q	Heat (cooling) capacity, kW
R	Radius, m
R	Thermal resistance, K/kW
Re	Reynolds number
t	Thickness, m
t	Time, s
T	Tempetarure, K
U	Overall heat transfer coefficient, kW/m^2K
v	Velocity, m/s
V	Volume, m^3
w	Mass fraction of LiBr in a water/LiBr solution, kg LiBr/kg solution
x	In a flat plate, distance to the edge, m

x	Molar fraction of LiBr in a water/LiBr solution, mol LiBr/mol solution

Greek Letter Symbols

α	Ratio of total transfer area on the air side of the exchanger to total volume of the exchanger
Δ	Denotes difference
ε	Efficiency
μ	Viscosity, kg/ms
ν	Kinematic viscosity, m^2/s
ρ	Density, kg/m^3
θ	Temperature at which the saturation pressure of water equals the pressure of the water/LiBr solution at temperature T and composition x, K

Subscripts

abs	Absorber
CF	Counter-flow heat exchanger
cond	Condenser
cond	Conduction
conv	Convection
evap	Evaporator
env	Environment
eq	Equilibrium
ext	External
gen	Generator
H	High
HEx	Heat Exchanger
in	Inside the cabin
int	Internal
L	Low
lam	Laminar
max	Maximum
min	Minimum
out	Out, outer
Rad	From radiation
rich	Refers to the water/LiBr rich solution
trans	Refers to the point where the flow changes from laminar to turbulent
turb	Turbulent
weak	Refers to the water/LiBr weak solution

1. Introduction

1.1. Background

The autonomy of electric vehicles (EV) is still a challenge and it is also significantly reduced when some accessories that are running with electricity, as the air conditioning, are operating. The idea to solve this problem is to implement a system where the air conditioning runs not with electricity, but with heat. This way the autonomy of the car will not be affected by the comfort necessities.

For the air conditioning to run with heat, the conventional heat pump consisting of a compressor, a condenser, a valve and an evaporator, should be substituted by an absorption heat pump. The main components of this pump are a generator, a condenser, an evaporator, an absorber and a heat exchanger, along with some valves and pumps to take the fluids to the different pressure levels. The conventional heat pump needs electricity to make the compressor work, while the absorption heat pump can be operated supplying heat to its generator.

The purpose is now to obtain heat that can be used by the generator. This can be achieved using the waste heat from a fuel cell. Furthermore, the fuel cell provides electricity as its main product, used as power supply for the electric motor of the vehicle.

To sum up, the proposed system takes advantage of the exhaust gases from the fuel cell to run an absorption heat pump that will provide the required air conditioning to the car without reducing its range.

In the next sections, a brief introduction about the electric cars and their principles will be given in order to explain which are the limitations they present, focusing in the range of the vehicle when connecting the air conditioning.

1.2. Electric cars

The first existing vehicles in the 19th century were in fact electrically powered, but during the last century, the internal combustion engine (ICE), mainly reliant on Otto and diesel cycles, has since dominated the world of transportation [1]. Unfortunately, their residual gases are very pollutant for the atmosphere, which has become a serious issue. The alternative to the internal combustion engine is to go back to the beginning and develop the electric vehicles [2].

A definition of electric vehicle could be formulated as a vehicle that draws its power from electricity stored in batteries [3]. This means that they use an electric motor [2].

The drive system, which is in charge of the transmission of the mechanical energy to the traction wheels for the vehicle to move, performs the same functions for an EV as for a conventional vehicle. Nevertheless, the components of this drive system are different in both cases. In fact, even if there can still be gears, a transmission that changes the gear input/output ratio to give a certain torque or power at certain speeds is not needed in EV. These vehicles utilize the electric motor to provide power to the wheels [2].

Electric vehicles and vehicles powered with internal combustion engines have many similar aspects. These similarities start with the chassis, since it can be reutilized from cars that had ICE to new electric cars. Also the interior of the vehicle remains as before, and the accessories both kind of vehicles have are almost the same [2].

The benefits of operating electric vehicles are numerous. The principal benefit is that they do not use gasoline or diesel to operate, but electricity, which reduces pollution and the dependence on combustible fuels. Another advantage of these vehicles is that they are more energy efficient than the internal combustion engine. At the same time, if regenerative braking is used, the lifetime of the brakes can be improved. Furthermore, including fuel cost, the maintenance cost is much lower than for the conventional cars [2].

Focusing in the main target of the EV, which is also their most important benefit, it could be said that they produce zero emissions. It is true that power plants, in charge of the electricity production, produce pollutants, but they are highly controlled by the government, and, in the best cases, this electricity can be produced by photo voltaic plants or wind power, which are not pollutant. Furthermore, since the electricity demand at night is lower than during the day, these power plants have an excess of production that could be used by EV owners to recharge their vehicles during that time. This would improve power plants efficiency [2].

The most important part of an electric car is its battery, which supplies power to run everything in the car, from the electric motor to the accessories like the radio or the air conditioner [2].

1.3. Batteries

1.3.1. Principles

A battery is a device that stores chemical energy and converts it into electricity. In consists of one or more cells connected electrically so that the necessary energy storage capacity can be met. Each electrochemical cell consists of two electrodes denominated the negative electrode and the positive electrode [4].

The negative electrode has overabundance of electrons, while the positive has a deficit of them. Connecting the two electrodes by an external circuit (a wire or an electrical cable), electrons will flow through it from the overcrowded electrode to the one with deficit, producing electricity [5].

A more detailed explanation about the batteries and how they work is given in the appendix.

1.3.2. Autonomy

The advantage of rechargeable batteries is precisely their disadvantage. Recharging the battery of an electric car can take several hours. The existing rechargers take between eight hours in the best cases up to ten hours in the worst ones to fully charge an electric car using the current supplied for general domestic use [6].

Once the car has been recharged, its autonomy depends on various factors, especially remarkable the type of battery that is being used [2].

Ranges of typical electric vehicles are shown in the table below. As it can be observed, ranges are always around 100 miles ($\approx 161\ km$), except in the case of Tesla models, whose ranges can go up to 300 miles ($\approx 483\ km$). The ranges in the table have been rated by the EPA (United States Environmental Protection Agency), except those where indicated [7].

Table 1. Overview of 2011-2012 Electric Vehicle models. Marked with an asterisk are manufacturer-estimated ranges, not rated by the EPA [7].

Brand	Model	Range (miles)
Tesla	Roadster	245
Nissan	LEAF	73 (100*)
Mitsubishi	i-MiEV	62
Ford	Focus	100*
Tesla	Model S	160/230/300*
Coda	Sedan	120*
Smart	ED	70-80*

Going into more details about the range of an EV, it is important to mention that it can be extended using a technology known as regenerative braking. This regenerative braking consists on recharging the battery on the fly by employing the kinetic energy of the brakes, and it can extend the range up to 15 per cent [8].

Even though, as seen in table 1, the most challenging obstacle of EV is still the storage capability of their batteries.

The EV range can significantly be reduced by switching on the air conditioning while driving. To operate the compressor of the air conditioning in a vehicle, a significant amount of power is required. Indeed, this amount of power could be even greater than that required to move a vehicle of a middle size at a constant speed of 56 km/h [9].

Table 3 shows how the air conditioning affects the range of five-passenger, lightweight-chassis EV with a Nickel metal hydride battery. In table 2 its characteristics are specified [9].

Table 2. EV Specifications [9].

Parameter	Value	Motor		Battery Pack	
Test Mass	1599 kg	**Max. Power**	75 kW (continuous) 135 kW (intermittent)	**Type**	NiMH
C_D*A	0.67 m^2	**Max. Torque**	271 kW (continuous) 488 kW (intermittent)	**Manufacturer**	Ovonic
Fixed Gear Ratio	6.7	**Max. Speed**	10000 rpm	**Pack Voltage**	327 V
Accessory Load	500 W			**Pack Energy**	30.4 kWh
				Pack Mass	412 kg

Table 3. Electric Vehicle Range [9].

	500 W	1500 W		2500 W		3500 W	
	Range (km)	Range (km)	Change from 500 W case	Range (km)	Change from 500 W case	Range (km)	Change from 500 W case
FUDS	175.9	147.7	-16%	125.5	-29%	108.9	-38%
HWFET	183.6	167.5	-9%	154.0	-16%	142.1	-23%
US06	116.0	107.6	-7%	102.5	-12%	95.3	-18%
SC03	174.3	146.9	-16%	126.8	-27%	111.2	-36%

In the Table 3 above, ranges and changes are estimated for the four different U.S. EPA certification procedures: FUDS (urban driving cycle), HWFET (highway driving cycle), US06 and SC03 [9].

Assuming a base electrical load of 500 W, if accessory loads as result of air conditioning in increments of 1000 W are considered, it can be observed that the range of the vehicle decreases considerably in all cases, compared to the reference range considered for the standard case [9].

Considering that conventional air conditioning loads can reduce the performance of the battery of an electric vehicle by nearly 40% in the worst cases [9], this project aims to study some aspects of an alternative system that can reduce the impact of the air conditioning in the autonomy of the EV.

2. The alternative system

2.1. Fuel Cell Vehicles

2.1.1. Principles

As electric vehicles, fuel cell vehicles (FCV) do not produce any pollutant gases while been driven. Water vapours are the only emission during the operation of the vehicle. What makes them different from EV is that their performance and refuelling time are much closer to conventional ICE vehicles [10]. Moreover, the volume of the energy storages that FCV need, compared to their relatives in battery-based cars, is much lower when the range of the vehicle is increased. This fact, which is related to the importance of the weight of the different components, is determinant to improve the autonomy of the vehicle [11].

Figure 1 illustrates the driving range in relation with the energy storage volume of different kind of vehicles [11].

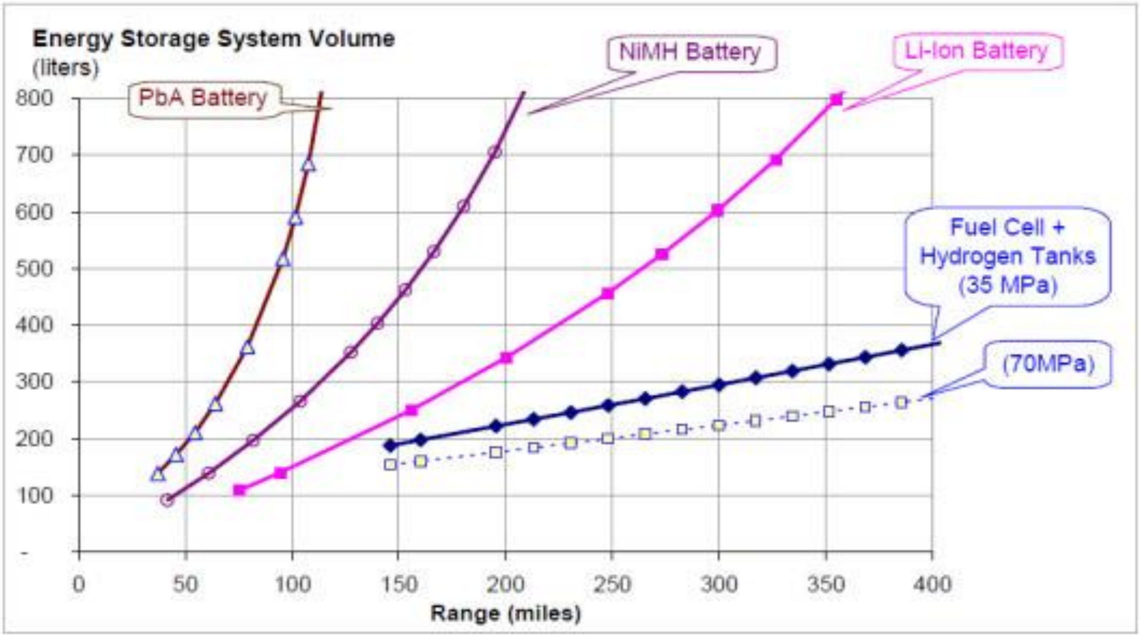


Figure 1. Driving range as a function of energy storage [11].

It can be observed that for the lead-acid and the NiMH batteries, the weight exponentially increases when larger distances want to be achieved. For Li-ion batteries, the curve is approximately linear until it reaches 250 miles, when it becomes exponential as well. In the other hand, fuel cell and ICE vehicles have a linear progression [11].

Conventional vehicles and electric vehicles have many similar aspects, and the same occurs with fuel cell vehicles. From the outside, they all look the same, but inside they present some new components. Between these components, the most remarkable is the fuel cell stack, which is in charge of producing electricity converting hydrogen gas and oxygen. This electricity is then employed to power the electric motor, which propels the car [12].

To feed the fuel cell stack, the vehicle also contains a hydrogen storage tank, where extremely high pressures are needed to keep as much hydrogen gas as possible and thus increase the driving range. Finally, FCV comprise a battery that provides the electric motor with supplementary electricity when needed [12].

In the case of the system proposed in this project, the heat produced by the fuel cell will be provided to the absorption heat pump to produce the cooling effect needed for air conditioning. If air conditioning requirements result on an excess of electricity production in the fuel cell, this would be directed to the battery to store it.

FCV require hydrogen to be run. Using hydrogen as fuel has the advantage of reducing the dependence of oil that world has [12]. A wide variety of sources are able to produce hydrogen [10], most commonly natural gas [11].

2.1.2. Fuel Cell

Fuel cells are electrochemical devices able to transform inputs as hydrogen and oxygen in outputs as electricity and heat. There are different types of fuel cells, among which Proton Exchange Membrane (PEMFC) fuel cells are the most appropriate for automotive applications [13].

The temperature of the exhaust gases needed in the generator of the absorption heat pump that composes the air conditioning system is around 150 °C and 200 °C, what requires a special kind of PEMFC, called HTPEM (High Temperature Proton Membrane Fuel Cell).

Further explanations about the fuel cells and their working principles can be found in the appendix.

2.1.3. Batteries

Even if fuel cells are the main power suppliers for running electric driven vehicles, batteries are needed as a supplementary power supply. PEM fuel cells have a fast start-up response, but still need between 5 and 30 seconds to reach a steady state of operation. For this reason, vehicles rely 100% on the battery during the start-up, which is also the warm-up period for the fuel cell [11].

Once the fuel cells have finished their warming-up, the power needed for cruising operation of the car is supplied by them, while batteries have some contribution when accelerating or climbing hills. As for EV, regenerative braking allows to recover kinetic energy and use it to charge the battery during braking [11].

3. Cooling Load of the Vehicle

3.1. Comfort

Comfort is defined in Oxford dictionary as “a state of physical ease and freedom from pain or constraint”.

More specifically, thermal comfort combines subjective and objective aspects related to how a person feel and heat and mass transfer rates. There are various kinds of magnitudes that affect thermal comfort. These are grouped in two: person-related magnitudes and environment-related magnitudes [14].

- Person-related: several aspects are included in it. Physical parameters as deep body temperature, metabolic dissipation rate or skin temperature are included, as well as subjective parameters as personal preferences, habits, previous accommodation or the mood [14];
- Environment-related: where the most important are air temperature and humidity and wind speed, but also the direct solar irradiance, sky temperature or wall temperature [14].

A scale of comfortableness or uncomfortableness depending on the amount of people in a significant group who feel comfortable or not under specific conditions is given below [15].

- Uncomfortable cold, which means that more than 95% of people feel cold;
- Cool, when around 75% of the people feel cold;
- Slightly cool, which is related to a 25% of people feeling cold;
- Comfortable, when less than 5% of people complain of being cold or warm;
- Slightly warm, which refers to a 25% of people feeling hot;
- Warm, when around 75% of the people complain of being hot;
- Uncomfortable hot, which means that more than 95% of the people feel hot.

In this case, the only parameter that is going to be studied in relation to the feeling of comfort of the passengers in a vehicle is the temperature inside the cabin.

3.2. Model

To assure comfort conditions inside the car, a model considering the sun radiation, the temperature outside the vehicle and a desired temperature inside its cabin has been written using EES (Engineering Equation Solver).

For the model, the worst possible outside conditions have been used. These conditions correspond to Phoenix, Arizona, where the outside ambient temperature is 49 °C and the solar load is 1 kW/m^2. The desired temperature taken for the cabin is 20 °C, and the dimensions of the vehicle have been taken from the Renault Kangoo Electric Vehicle in Aalborg University.

The equations used for the model are presented in the next section, as well as a discussion about the obtained results.

3.2.1. Assumptions

With the purpose of investigating extremes, two different cases are studied. In the first one, a black car is considered, in the second case, the car is white.

Radiation:

- For the initial conditions, as well as while the vehicle is parked, only heat form radiation is taken into consideration, considering that is much higher than the convective heat;
- 100% absorption is assumed in the case of the black car;
- 100% reflection is assumed in the case of the white car;
- Radiation from the sun does not reach all the surface of the vehicle at the same time and the angles of incidence also vary depending on its position, thus, to simplify the model, the areas affected by the radiation are assumed to be half of the sum of the areas of the panel and the windows in the first case, and half of the area of the windows in the second. In both cases, again for simplicity, the angle is considered to be 90° for the whole affected surface.

Convection:

- Natural convection inside the cabin of the vehicle;
- Forced external convection, for which the employed correlations correspond to those for mixed fluid (laminar and then turbulent when $Re > Re_{trans} = 5 \cdot 10^5$) over a flat plate;
- It is assumed that the flow is parallel to the sides and the ceiling of the vehicle, which, consequently, are the only surfaces affected by the forced external convection;
- The flow over the external surfaces taken as flat plates could be represented as:

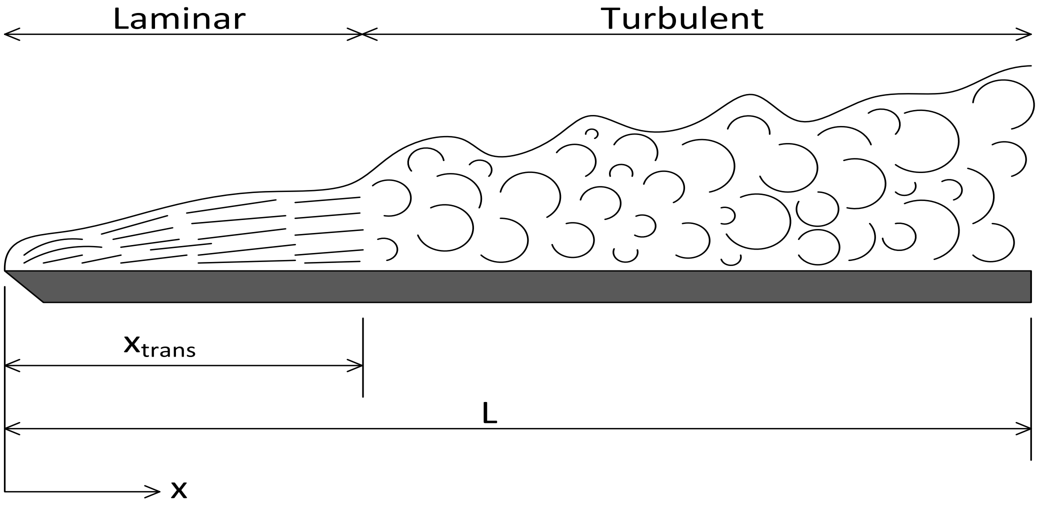


Figure 2.Mixed fluid over a flat plate.

Conduction:

- The panel is assumed to be composed by three layers, specifically steel, plastic and isolating air in the middle;
- The cross section of the panel could be represented as:

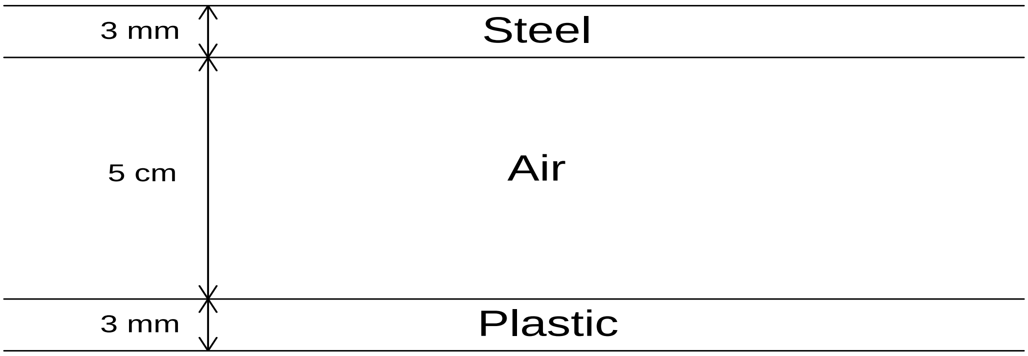


Figure 3. Cross section of the panel of the vehicle

3.2.2. Equations

Heat from radiation:

$$Q_{Rad}[kW] = Q_{Solar\ Load}[kW/m^2] \cdot A_{Rad}[m^2] \tag{1}$$

Where A_{Rad} is the area affected by the radiation as explained in the assumptions above, and $Q_{Solar\ Load}$ is the solar load considered for Arizona.

Heat equilibrium while parked:

When the cabin gets warmer than the ambient temperature, there is heat equilibrium and the same amount of heat that enters the car by radiation has to go out by conduction through the panel and windows, and by convection. Thus, $U_{out}A_{out}$ is calculated in the EES model as a combination of the two effects.

$$Q_{out}[kW] = Q_{Rad}[kW] \tag{2}$$

$$Q_{out}[kW] = U_{out}[kW/(m^2K)] \cdot A_{out}[m^2] \cdot (T_{eq}[K] - T_{env}[K]) \tag{3}$$

Heat gain while driving:

While the car is driving, it gains a high amount of heat by forced convection due to the velocity of the vehicle. The temperatures used for the calculation are the temperature of the environment and the desired for the cabin, what implies taking into consideration external forced convection, conduction through the panel and windows and internal natural convection to compute the thermal resistance. Equations showing this are presented below.

$$Q_{gain}[kW] = \frac{1}{R[K/kW]}(T_{env}[K] - T_{desired}[K]) \tag{4}$$

$$R[K/kW] = R_{conv,in}[K/kW] + R_{cond}[K/kW] + R_{conv,out}[K/kW] \tag{5}$$

$$\frac{1}{R_{cond}[K/kW]} = \frac{1}{R_{panel}[K/kW]} + \frac{1}{R_{window}[K/kW]} \tag{6}$$

$$\begin{aligned} R_{panel}[K/kW] = {} & \frac{t_{plastic}[m]}{k_{plastic}[kW/mK] \cdot A_{panel}[m^2]} + \frac{t_{air}[m]}{k_{air}[kW/mK] \cdot A_{panel}[m^2]} \\ & + \frac{t_{steel}[m]}{k_{steel}[kW/mK] \cdot A_{panel}[m^2]} \end{aligned} \tag{7}$$

$$R_{window}[K/kW] = \frac{t_{window}[m]}{k_{glass}[kW/mK] \cdot A_{panel}[m^2]} \tag{8}$$

$$R_{conv}[K/kW] = \frac{1}{h[kW/m^2K] \cdot (A_{panel}[m^2] + A_{window}[m^2])} \tag{9}$$

Where h is the convection coefficient and is taken as a fixed number for the internal convection (natural convection), and is calculated in the form of $\bar{h}$ as a result of the following formulas for the external convection (forced convection), for which mixed fluid has been considered:

$$\overline{Nu} = \frac{\bar{h}[kW/m^2K] \cdot L_{out}[m]}{k_{air}[kW/mK]} \tag{10}$$

$$\overline{Nu} = \left(0.037 \cdot Re^{4/5} - A\right) \cdot Pr_{air}^{1/3}, when\ 0.6 \leq Pr \leq 60\ and\ Re_{trans} \leq Re \leq 10^8 \tag{11}$$

$$A = 0.037 \cdot Re_{trans}^{4/5} - 0.664 \cdot Re_{trans}^{1/2} \tag{12}$$

$$Re = \frac{v_{air}[m/s] \cdot L_{out}[m]}{\nu_{air}[m^2/s]} \tag{13}$$

Equations (11) and (12) are taken from the book ‘Fundamentals of Heat and Mass Transfer’; Incropera, De Witt, Bergmann, Lavine; sixth edition, Wiley.

Considering that the average convection coefficient along the flat plate is [16]:

$$\bar{h} = \frac{1}{L}\int_0^L h dx \tag{14}$$

If there exists a mixed boundary layer, then the integration has to be done in two terms, one for the laminar region and another one for the turbulent region. The laminar region is situated between $0 \leq x \leq x_{trans}$, while the turbulent region is situated between $x_{trans} \leq x \leq L$. L represents the length of the plate, x the axis in which L grows, and x_{trans} is the point where flow changes from laminar to turbulent, and corresponds to the point where $Re = 5 \cdot 10^5$. This can be expressed as [16]:

$$\bar{h} = \frac{1}{L}\left(\int_0^{x_{trans}} h_{lam} dx + \int_{x_{trans}}^{L} h_{turb} dx\right) \tag{15}$$

Knowing that the local Nusselt numbers in the laminar and turbulent regions are respectively [16]:

$$Nu_x = 0.332 \cdot Re_x^{1/2} \cdot Pr^{1/3}, when\ Pr \geq 0.6 \tag{16}$$

$$Nu_x = 0.0296 \cdot Re_x^{4/5} \cdot Pr^{1/3}, when\ 0.6 \leq Pr \leq 60 \tag{17}$$

Substituting and integrating the equation for $\bar{h}$, equations (11) and (12) are obtained.

Initial cooling load to cool down the cabin from the inside temperature while parked to the desired temperature in a fixed time interval:

$$\dot{m}[kg/s] = \frac{\rho_{air}[kg/m^3] \cdot V_{in}[m^3]}{t[s]} \tag{18}$$

$$Q_{initial}[kW] = \dot{m}[kg/s] \cdot cp_{air}[kJ/(kgK)]\,(T_{in}[K] - T_{desired}[K]) \tag{19}$$

3.2.3. Results

After running the model written in EES, the values of the different cooling loads needed initially and while driving, as well as the initial inside temperature are known. Depending on the colour of the vehicle, the results are different because of the different absorption of the radiation. Results for a black and a white car will be explained.

3.2.3.1. Comparison

To make it simpler, a comparison between the results obtained for the two different vehicles used in the model is represented in the table below. The results shown are those for which an explanation will be given in the next section, this is, the equilibrium temperature reached by the cabin while the vehicle is parked and exposed to the solar radiation, the heat load from solar radiation, the heat gain due to the temperature difference between the ambient and the cabin, and the necessary initial cooling load to cool down the cabin from its initial temperature (the ambient temperature plus ΔT) to the one desired in a fixed time, in this case, of two minutes. In the same table, the inputs employed in the EES model and that can be modified when required are also shown.

Table 4. Inputs and results of the EES model for a black and a white vehicle.

<table>
<tr><th colspan="2">Inputs</th><th colspan="3">Results</th></tr>
<tr><th>Parameter</th><th>Value</th><th>Parameter</th><th>Black Vehicle</th><th>White Vehicle</th></tr>
<tr><td>Solar Load from radiation</td><td>1 kW/m^2</td><td>T_{eq}</td><td>381.4 K</td><td>332.9 K</td></tr>
<tr><td>T_{env}</td><td>49 °C</td><td rowspan="2">Q_{Rad}</td><td rowspan="2">8.541 kW</td><td rowspan="2">1.552 kW</td></tr>
<tr><td>$T_{desired}$</td><td>20 °C</td></tr>
<tr><td>v_{air}</td><td>8.33 m/s</td><td rowspan="2">Q_{gain}</td><td rowspan="2">5.993 kW</td><td rowspan="2">5.993 kW</td></tr>
<tr><td>v_{car}</td><td>100 km/h</td></tr>
<tr><td>ΔT</td><td>10 K</td><td rowspan="2">$Q_{initial}$</td><td rowspan="2">1.894 kW</td><td rowspan="2">1.894 kW</td></tr>
<tr><td>Initial time</td><td>120 s</td></tr>
</table>

3.2.3.2. Black Vehicle

As mentioned before, it is considered that the black surface of the vehicle behaves as a black body and consequently has 100% absorption.

Considering that radiation from the sun reaches half of the vehicle surface, this is, half of the surface of the panel and half of the windows, with 90° as the angle of incidence, and that the solar load in Arizona is 1 kW/m^2, then the necessary cooling load that the air conditioning has to provide to compensate radiation is 8.541 kW.

If a velocity of 8.33 m/s is considered as the air velocity outside the car when this is parked, and considering natural convection inside, the highest cabin temperature is 381.4 K. This temperature is a result of the heat equilibrium in the vehicle while parked, and is named in the model as T_{eq}. The velocity of the outside air has a great influence in this temperature, so the model allows to change that value in order to calculate the temperature depending on the actual outside conditions.

The initial cooling load necessary to cool down the cabin from the initial temperature to the one desired in two minutes results to be 1.894 kW. The initial temperature depends on the highest cabin temperature reached while parked and the time the door (or doors) of the vehicle are opened before going into it, lowering the cabin temperature down to the outside temperature in the best case. Once the cabin temperature reaches the one desired, this heat input is not necessary anymore.

For the calculation that gives the previous number, a temperature difference of 10 K between the initial conditions of the cabin and the environment has been considered. The two minutes time chosen could be changed to a shorter or longer time if required, changing as a consequence the initial cooling input that would cool down the cabin.

There is another cooling load that has to be provided by the air conditioning. This cooling load is in charge of compensating the heat gain due to the temperature difference between the cabin when is already cooled and the outside ambient. This cooling load has to be provided during the whole time the car is being driven. The same occurs to that that compensates radiation.

The heat gain is calculated for a car velocity of 100 km/h, though the EES model allows to change this value if calculations are desired for any other different speed. Considering the mentioned value of 100 km/h, the heat gain rate is 5.993 kW.

3.2.3.3. White Vehicle

In the case of the white car, 100% reflective effect is considered, which means that the solar load received by the car is only that due to the radiation through the windows. As said before, the area that has been considered to be affected by the solar radiation at the same time is half of the windows' area.

In such case, and for the same solar load of 1 kW/m^2 as before, the cooling load that the air conditioning has to provide to compensate the effect of the solar radiation is 1.552 kW.

While the car is parked the cabin increases its temperature until equilibrium between the heat from radiation and the heat loss due to the higher temperature inside than outside the car is reached. In this equilibrium, the cabin temperature is 332.9 K, which is much lower than the 381.4 K reached in the case of the black vehicle. The same as in that case, natural convection has been considered inside the car, and forced convection with a velocity of air of 8.33 m/s has been considered outside.

The initial cooling load and the one that compensates the heat gain rate that have to be provided by the air conditioning to cool down and maintain the cabin, respectively, at the desired temperature of 20 °C remain the same regardless of the colour of the vehicle. Thus, the initial load is 1.894 kW for a cooling time of two minutes and a temperature difference of 10 K between the environment and the initial conditions of the cabin, and the heat gain rate is 5.993 kW if 100 km/h is taken as the velocity of the vehicle.

3.2.4. Influence of ΔT

ΔT is the temperature difference between the environment and the cabin when the passengers go into the car and start driving. As explained before, this temperature difference depends on many factors like the cabin temperature reached while the car is parked or the time the doors of the vehicle are opened before the passengers actually go into it.

The results presented before are calculated for a ΔT of 10 K, but the initial cooling load that has to be provided to lower the temperature from the initial conditions to the desired in two minutes changes with this ΔT.

The following plot shows how the initial cooling load increases when the mentioned temperature difference increases. For the calculations, the initial cabin temperature has been considered to be in a range between 10 K higher than the ambient temperature and equal to it, in intervals of 2.5 K.

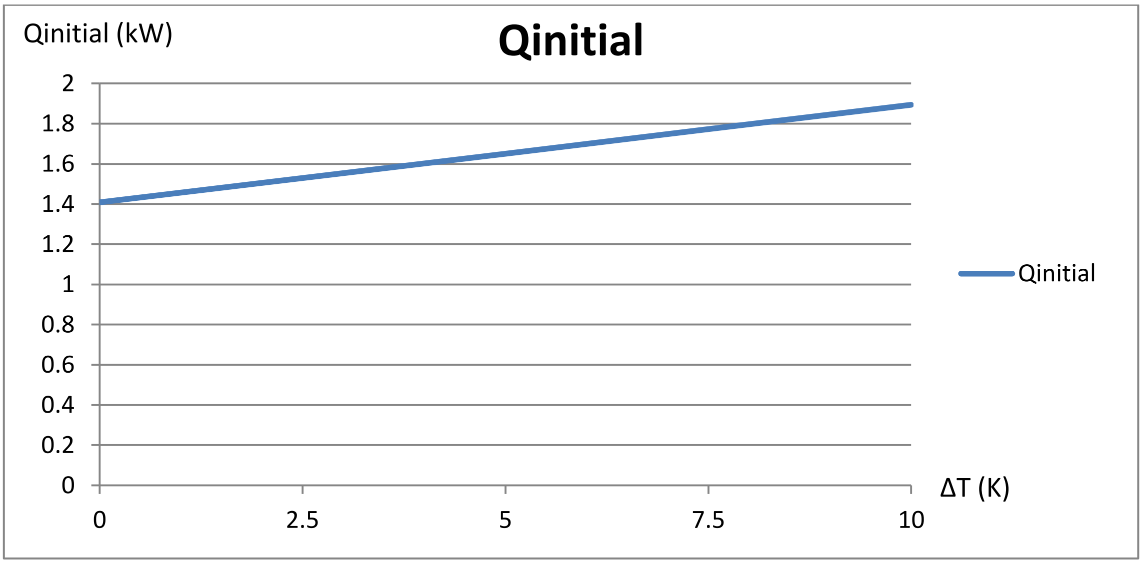


Figure 4. Variation of the initial cooling load as a function of ΔT.

4. Absorption technologies

4.1. Introduction

Global warming problem and the high necessity of energy have increased the interest in developing new thermally driven systems that are able to use waste heat to work. That way, not only waste heat can be recovered, but the power source that was used instead of it is not necessary anymore [17].

Industrial processes usually employ thermal energy produced by burning fossil fuel, which produces steam or heat that, after the processes, are thrown away to the environment as waste. This waste is, in most cases, a source of power that is not being exploited. In the specific case of refrigeration systems, this waste heat could be used to produce cooling, but what is usually used in its place is electricity, which emits CO_2 as a consequence of the combustion of fossil fuels, increasing the greenhouse effect [18].

The systems that are able to provide cooling employing waste heat are known as absorption refrigeration systems, while the conventional systems that run with electricity are vapour compression systems. Another disadvantage the second group has compared to the first one is that they commonly use chlorofluorocarbon (CFCs) refrigerants, which are also harmful emissions [18].

4.2. Principles

Binary solutions are the working fluids in absorption refrigeration. The two components of these binary solutions are refrigerant and absorbent. The principle of operation of the absorption process could be explained by the case of two vessels, one of them containing refrigerant vapours and the other containing a solution of refrigerant and absorbent. If the two vessels are connected to each other, then the vessel with the solution will absorb refrigerant vapours from the other vessel, reducing its pressure. The solution will consequently have a higher content of refrigerant, which means that it will become more dilute. This process is known as “absorption process”, and is characterized for being exothermic. Thus, it rejects heat to the surroundings [18].

For the separation process, heat should be provided to the solution, in which the refrigerant is dried so that its vapours separate from the solution and travel to the other vessel. The heat provided for this process is the waste heat mentioned in the introduction of the chapter and that allows the system to be environmentally friendly [18].

The separation process takes place at a higher pressure level than the absorption process. After the separation process, a condenser that converts the refrigerant vapours in liquid refrigerant is needed, and is followed by a valve that reduces the pressure to the absorption pressure again. Once in the low pressure level, an evaporator converts the liquid refrigerant in the refrigerant vapours present in the vessel for the absorption process [18].

A simple representation of the system described before is shown in the following figure:

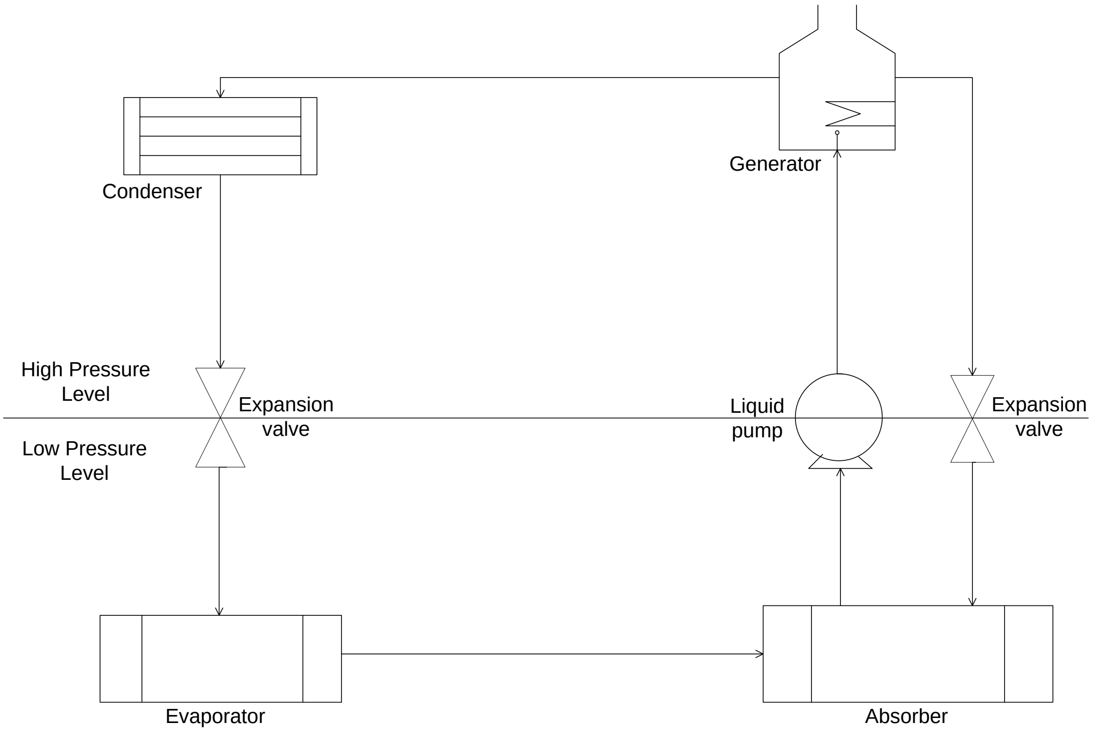


Figure 5. Absorption heat pump.

4.3. Working fluids

The working fluid of the absorption refrigeration system is one of the most important elements. Its chemical and thermodynamic properties determine the performance of the system. To choose the most appropriate working pair, there are some aspects that are necessary to consider. First of all, it is essential that the combination of the absorbent and refrigerant has a margin of miscibility in liquid phase for the operating temperature range. Other important requirements are that the mixture is non-toxic and non-explosive, and that is also chemically stable. There are some other desirable conditions as well, as high heat of vaporization of the refrigerant, environmentally friendly fluids or low-cost [18].

Although there is a wide variety of refrigerants and absorbents that could act as fluids, two combinations are the most common, ammonia/water and water/lithium bromide [18].

4.3.1. Ammonia/water

In the ammonia/water pair, ammonia acts as the refrigerant and water as the absorbent. Characteristics as high stability of the two components, high latent heat of vaporization of ammonia, low-cost or environmental friendliness make this pair one of the most important in refrigeration systems. Another advantage of this pair is that the freezing point of ammonia (the refrigerant) is -77 °C, what allows the applicability of the system when very low temperatures are required. Nevertheless, this pair also presents some disadvantages, as the necessity of employing a rectifier between the generator and the condenser due to the high volatility of both ammonia and water, or the corrosive action to copper [18].

4.3.2. Water/LiBr

In the water/LiBr pair, water acts as the refrigerant and LiBr as the absorbent. Due to the fact that water is the refrigerant, the applications of the system are limited to temperatures above 0 °C. An advantage of this pair compared to the previous is the non-volatility of LiBr, what eliminates the need of a rectifier between the generator and the condenser. Another important characteristic is the extremely high heat of vaporization of the refrigerant (water) [18].

Air conditioning application in a car requires temperatures above 0 °C, reason for which water/LiBr working fluid has been selected. In this section, different kind of configurations of absorption refrigeration technologies dealing with this working pair will be explained.

4.3.2.1. Single-Effect Absorption System

It is the simplest kind of absorption refrigeration system, and also the most commonly used. It consists basically in the same scheme as shown in figure 5, but including a heat exchanger between the absorber and the generator. This heat exchanger allows that the weak solution coming out of the generator preheats the rich solution from the absorber before entering the generator. Therefore, the heat input required in the generator is lower than that required without using the heat exchanger [18].

It is very suitable for those applications where small cooling capacities are required. The main disadvantage of this kind of absorption systems is its low COP [17].

4.3.2.2. Single-Effect Double Lift System

Usually, single-effect systems require heat sources above 90 °C, and if they work with lower temperatures in the heat source then the performance drops significantly. The single-effect double lift system allows the utilization of heat sources, mainly hot water, at temperatures down to 55 °C, without reducing the performance [17].

The configuration consists on a single-effect cycle to which is integrated a double lift cycle. The values of the COP varies between 0.35 and 0.7 depending on the proportion of high temperature heat used to power the single effect cycle and the low temperature heat used to power the double lift cycle. Anyway, the heat supplied to this system is more efficiently utilized than in the case of the single-effect system, what makes this system to provide a higher cooling effect for the same energy input [17].

The most important disadvantage of this system in comparison to the previous is the higher number of components that compose it, what increases its size and weight [17].

4.3.2.3. Double-Effect Absorption System

The double-effect system incorporates a high temperature generator in relation to the single-effect system. This additional generator operates at a higher temperature and pressure, and consequently the supplied heat is more efficiently utilized [17].

Normal COP values for this kind of refrigeration systems are in the range of 1.1 to 1.3, which are considerably higher than those obtained with the single-effect systems. This is due to the higher temperature heat input in the generator, while the rejection of heat and cooling at the condenser and evaporator respectively are at about the same temperatures as in the first case [17].

As in the case of the single-effect double lift systems, the major disadvantage of this configuration is the higher amount of devices in comparison to the single-effect system, which means bigger size of the system and higher weight.

4.3.2.4. Multi-Effect Absorption System

These systems are a generalization of the double-effect systems, trying to increase the COP as much as possible. The triple-effect cycle, for example, is able to achieve a COP of about 1.6, but it requires a heat input at a temperature significantly higher than 170 °C. Another case of multi-effect cycle is the seven-effect cycle, which using three absorption circuits is able to achieve COP values between 2.19 and 3.12. This seven-effect cycle employs water/sodium hydroxide solution pair for the high and medium temperature absorption circuits [17].

In the case of the cycle that is being studied in this report, the weight and size of the system are important factors, as they have a great influence in the performance of the electric vehicle. For this reason, the most adequate water/LiBr system of the different types explained is the single-effect absorption system.

5. Absorption heat pump

5.1. The cycle

A representation of the cycle of the absorption heat pump that has been explained in chapter 4 is shown below. Each stream has a number, which corresponds to the number used in the model of the heat pump implemented in EES. The properties (pressure level, temperature, enthalpy and mass flow rate) in each different point are also given. The inputs of the model for which the properties have been computed are the correspondent to those required by the white vehicle studied before, this is, Q_{evap}= 9.5 kW; T_{evap}= T_{10} = T_9 = 15 °C; T_{cond}= T_8 = 65 °C; T_{gen}= = T_7 = 80 °C; w_{Rich} = 0.58 kg LiBr/kg solution; w_{Weak} = 0.67 kg LiBr/kg solution.

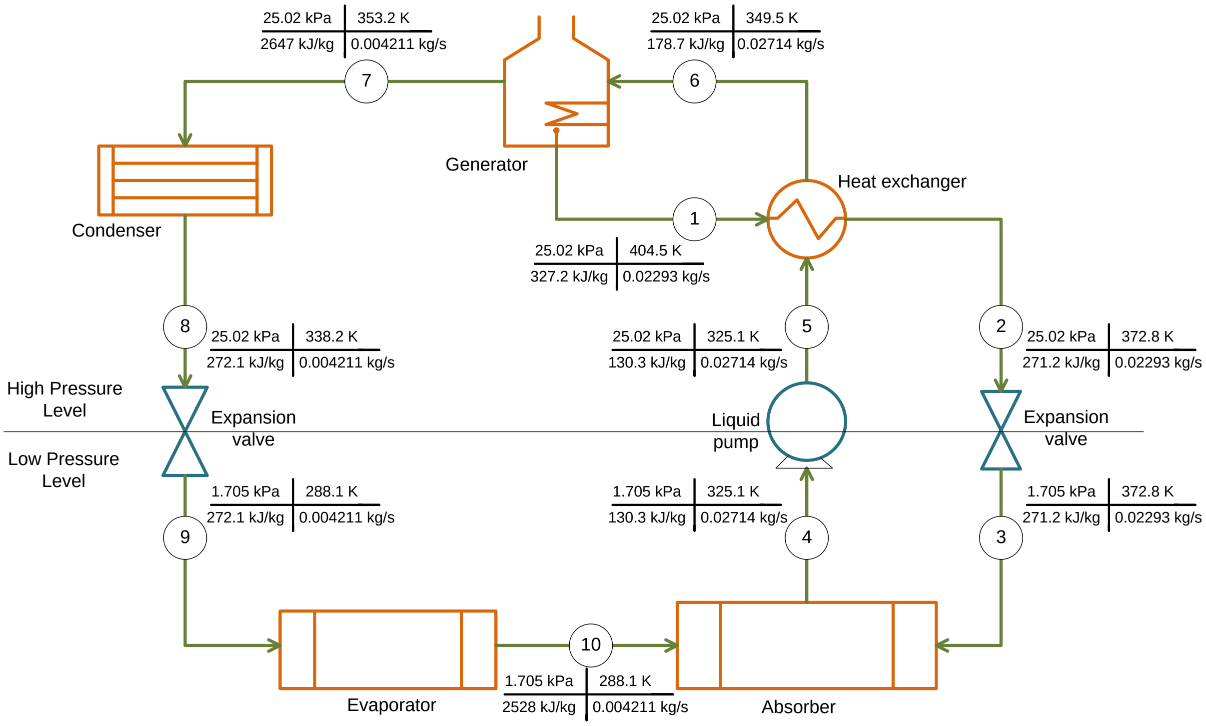


Figure 6. Cycle of the absorption heat pump.

5.2. Model

The model of the system has been implemented in EES. The model is based in the energy balances in each component of the system. Database in EES has been employed in order to calculate the properties of the fluids (water and water/LiBr mixture) involved in the cycle when required.

5.2.1. Assumptions

Before explaining the equations used in the model, the assumptions considered should be discussed. These assumptions are the following:

- Saturated liquid in points 1, 4 and 8;
- Saturated vapour in point 10;
- Superheated vapour at 80 °C in point 7;
- Mass fractions of the rich and weak solutions are considered to be fixed and equal to w_{Rich} and w_{Weak} respectively, and remain the same along the correspondent stream; this is:

$$w_1 = w_2 = w_3 = w_{Weak} \tag{20}$$
$$w_4 = w_5 = w_6 = w_{Rich} \tag{21}$$

- The temperature of the liquid water/LiBr mixture is independent of the pressure for a fixed composition, what in the cycle involves:

$$T_4 = T_5 \tag{22}$$
$$T_2 = T_3 \tag{23}$$

- The enthalpy of the liquid water/LiBr mixture is independent of the pressure for a fixed composition, what in the cycle involves:

$$h_4 = h_5 \tag{24}$$
$$h_2 = h_3 \tag{25}$$

- In the absorber, the absorption efficiency is considered to be equal to 1;
- There are no energy losses in the different devices.

5.2.2. Equations

Evaporator:

$$\dot{m}_{10}[kg/s] = \frac{Q_{evap}[kW]}{h_{10}[kJ/kg] - h_8[kJ/kg]} \tag{26}$$

Absorber:

$$Q_{abs}[kW] = \dot{m}_{10}[kg/s] \cdot h_{10}[kJ/kg] + \dot{m}_3[kg/s] \cdot h_3[kJ/kg] - \dot{m}_4[kg/s] \cdot h_4[kJ/kg] \tag{27}$$

$$\dot{m}_4[kg/s] = \dot{m}_{10}[kg/s] \frac{w_{Rich}[kg/kg]}{w_{Weak}[kg/kg] - w_{Rich}[kg/kg]} \tag{28}$$

$$\dot{m}_3[kg/s] = \dot{m}_4[kg/s] - \dot{m}_{10}[kg/s] \tag{29}$$

Heat Exchanger:

$$Q_{max}[kW] = C_{min}[kW/K] \cdot (T_1[K] - T_5[K]) \tag{30}$$

$$Q_{HEx}[kW] = \varepsilon \cdot Q_{max}[kW] \tag{31}$$

$$\begin{aligned} cp_1[kJ/(kgK)] \cdot \dot{m}_1[kg/s] \cdot (T_1[K] - T_2[K]) = \\ = cp_5[kJ/(kgK)] \cdot \dot{m}_5[kg/s] \cdot (T_6[K] - T_5[K]) \end{aligned} \tag{32}$$

$$NTU = \frac{U_{HEx}[kW/m^2K] \cdot A_{HEx}[m^2]}{C_{min}[kW/K]} \tag{33}$$

$$\varepsilon = \frac{1 - e^{-NTU\left(1 - \frac{C_{min}[kW/K]}{C_{max}[kW/K]}\right)}}{1 - c \cdot e^{-NTU\left(1 - \frac{C_{min}[kW/K]}{C_{max}[kW/K]}\right)}} \tag{34}$$

<u>Condenser:</u>

$$Q_{cond}[kW] = \dot{m}_7[kg/s](h_7[kJ/kg] - h_8[kJ/kg]) \tag{35}$$

<u>Generator:</u>

$$Q_{gen}[kW] = \dot{m}_7[kg/s] \cdot h_7[kJ/kg] + \dot{m}_1[kg/s] \cdot h_1[kJ/kg] - \dot{m}_6[kg/s] \cdot h_6[kJ/kg] \tag{36}$$

5.2.3. Energy Balance in the Heat Exchanger

In the heat exchanger, the energy balance in terms of $\dot{m}$, cp and ΔT has been used as an equation for the model. If, after obtaining the results, the energy balance in the heat exchanger is checked in terms of $\dot{m}$ and Δh, then it can be observed that the result obtained is not exactly as expected, as there is a small error.

In other words, the energy balance in the heat exchanger should satisfy:

$$\dot{m}_1[kg/s] \cdot (h_1[kJ/kg] - h_2[kJ/kg]) + \dot{m}_5[kg/s] \cdot (h_5[kJ/kg] - h_6[kJ/kg]) = 0 \tag{37}$$

Substituting the values of the mass flow rates and the enthalpies obtained with the EES model in the formula, the result in not zero.

A possible explanation for this could be the method used to calculate the enthalpies in the different points of the cycle. As it will be seen in following sections, the thermodynamic properties of the water/LiBr mixture can be calculated using different methods, but the results obtained are just an accurate approximation to the actual values.

As a consequence, the energy balance in the heat exchanger represented with the above equation gives a result more or less similar to zero depending on the method employed for the enthalpy calculations.

The errors have been computed for the peak cooling loads needed in the evaporator in the two cases studied, the black and the white vehicles.

5.2.3.1. Black Vehicle

The peak cooling load in this case is:

$$Q_{Rad} + Q_{gain} + Q_{initial} = 8.541 + 5.993 + 1.894 = 16.428\ kW \tag{38}$$

EES:

Running the EES model for a cooling load in the evaporator of 16.5 kW, the result obtained for the energy balance equation is:

$$\dot{m}_1(h_1 - h_2) + \dot{m}_5(h_5 - h_6) = -0.01709\ kW \tag{39}$$

Which, knowing that the heat capacity of the heat exchanger is 1.327 kW, represents an error of:

$$Error = \frac{0.01709}{1.327} = 1.29\% \tag{40}$$

Experimental method using 289 K as reference temperature:

The temperatures related to the heat exchanger, T_1, T_2, T_5 and T_6 when Q_{evap} = 16.5 kW are T_1 = = 404.5 K, T_2 = 385.9 K, T_5 = 325.1 K and T_6 = 339.4 K.

Substituting these temperatures and the mass fractions of lithium bromide in the rich and the weak solutions of the system, in the Matlab model written for the thermodynamic properties calculation of the mixture (see 6. Thermodynamic Properties), the enthalpies in the correspondent points are obtained.

If these enthalpies and the mass flow rates are introduced in the formula for the energy balance, the following is observed:

$$\dot{m}_1(h_1 - h_2) + \dot{m}_5(h_5 - h_6) = -0.04802\ kW \tag{41}$$

Which, for the same heat capacity of the heat exchanger as before, represents an error of:

$$Error = \frac{0.04802}{1.327} = 3.62\% \tag{42}$$

5.2.3.2. White Vehicle

In the case of the white vehicle, the peak cooling load of the evaporator is:

$$Q_{Rad} + Q_{gain} + Q_{initial} = 1.552 + 5.993 + 1.894 = 9.439\ kW \tag{43}$$

EES:

A cooling load of 9.5 kW is used to run the EES model, for which the energy balance results as:

$$\dot{m}_1(h_1 - h_2) + \dot{m}_5(h_5 - h_6) = -0.02975\ kW \tag{44}$$

The heat capacity of the heat exchanger is 1.303 kW, what implies an error of:

$$Error = \frac{0.02975}{1.303} = 2.28\% \tag{45}$$

Experimental method using 289 K as reference temperature:

Substituting in the Matlab model for enthalpy calculations the mass fractions of the rich and the weak solutions and the results obtained with EES, when Q_{evap} = 9.5 kW, of the temperatures of the points involved in the energy balance of the heat exchanger, this is, temperatures T_1 = 404.5 K, T_2 = 372.8 K, T_5 = 325.1 K and T_6 =349.5 K, the enthalpies in these four points are calculated.

Introducing the values of the correspondent enthalpies in the formula of the energy balance, as well as the mass flow rates in the two streams involved, the balance shows:

$$\dot{m}_1(h_1 - h_2) + \dot{m}_5(h_5 - h_6) = -0.04861\, kW \tag{46}$$

Compared to the heat capacity of the heat exchanger, which is still 1.303 kW, the error is:

$$Error = \frac{0.04861}{1.303} = 3.73\% \tag{47}$$

5.2.3.3. Conclusions

From the previous calculations, it can be concluded that the errors obtained when studying the energy balance of the heat exchanger between the rich and the weak solutions involved in the cycle could be a consequence of the method employed for the calculation of the thermodynamic properties, more specifically, the enthalpy. Moreover, this error could be also affected by round off adjustments.

Nevertheless, the errors are never higher than a 4%, so the results obtained can be considered as accurate. After these lines, a validation of the EES model developed is carried out, so that the accuracy of the results, compared to some real numbers, is checked.

5.3. Validation

The aim of this chapter is to validate the model of the proposed system comparing the results obtained when running it to those given in reference papers. For this purpose, the model has been run with the inputs (heat capacity of the evaporator and some temperatures) employed in reference [116], and the outputs obtained have been compared to the outputs given in the reference. The error between the two has also been calculated.

The table below shows the results of the validation when the inputs are:
w_{Rich} = 0.5672 kg LiBr/kg sol.; w_{Weak} = 0.6233 kg LiBr/kg sol.; Q_{evap} = 3.517 kW; T_{cond} = 40 °C; T_{evap} = 7 °C; T_{gen} = 90 °C.

Table 5. Model validation

	$\dot{m}_1$(kg/s)	$\dot{m}_4$ (kg/s)	T_2 (K)	T_6 (K)	Q_{gen} (kW)	Q_{abs} (kW)	Q_{cond} (kW)
Model	0.01366	0.01516	313.30	356.60	4.230	4.051	3.749
Reference [19]	0.01515	0.01664	323.15	346.67	4.532	4.301	3.748
Error	9.83%	8.89%	3.05%	2.86%	6.66%	5.81%	0.03%

From the table, it can be observed that the results obtained with the model for the inputs given in the reference are quite accurate, with errors around 10% in the case of the mass flow rates and considerably lower for the rest of the parameters. The validation of the model is, consequently, satisfactory.

To complete the validation, an energy balance of the system could be useful. This energy balance makes possible the comparison between the energy inputs and outputs in the system, evaluating their differences.

In the system proposed, the energy inputs are the heat capacities of the generator and the evaporator, since both are removing heat from the environment. The energy outputs are the heat capacities of the condenser and the absorber, which yield heat to the environment.

The equation of this energy balance can be expressed as:

$$Q_{gen}[kW] + Q_{evap}[kW] = Q_{cond}[kW] + Q_{abs}[kW] \tag{48}$$

The error between the inputs and outputs is defined as:

$$e = \left| \frac{Q_{input}[kW] - Q_{output}[kW]}{Q_{input}[kW]} \right| \tag{49}$$

The EES model has been run for two different blocks of inputs, those used in reference [19] and those mentioned in the cycle explanation, which will be called basic run from now on. The table below shows the results obtained for the different heat capacities when running the model for the two blocks of inputs.

Table 6. Energy balance

Input			Output		
Energy (kW)	**Basic Run**	**Model with inputs [19]**	**Energy (kW)**	**Basic Run**	**Model with inputs [19]**
Q_{gen}	13.80	4.230	Q_{cond}	10.00	3.749
Q_{evap}	9.50	3.517	Q_{abs}	13.33	4.051
Total	23.30	7.747	**Total**	23.33	7.800

From the numbers in the table, the errors for the two energy balances can be calculated using the equation mentioned before. Thus, the errors are:

Basic Run:

$$e = \left|\frac{23.30 - 23.33}{23.30}\right| = 0.18\% \tag{50}$$

Model with inputs in [19]:

$$e = \left|\frac{7.747 - 7.800}{7.747}\right| = 0.68\% \tag{51}$$

The errors below 1% obtained from the energy balance are an evidence of the accuracy of the model implemented in EES.

An explanation of why do those errors exist could be the assumptions made in the different points of the cycle, for example the saturated conditions of the fluid when going out of the absorber, or the absorber efficiency equal to 1. Furthermore, the different ways of calculating the thermodynamic properties in the model and in the reference could be also detonator of obtaining some different results, which leads in an error.

6. Thermodynamic properties

6.1. Introduction

Typical working pairs for absorption heat pump cycles are ammonia/water or water/lithium bromide mixtures. To develop good models of the heat pump and have accurate calculations of their performance, it is essential to study the behaviour of the thermodynamic properties of the working pair employed, under the conditions desired.

The absorption heat pump to which is referred the alternative system proposed is dealing with the water/lithium bromide working pair. As seen in the cycle description, there are two fluids involved along the different points of the cycle, water and water/lithium bromide mixture. Water appears as saturated vapour, saturated liquid and in liquid-vapour equilibrium, while water/LiBr mixture appears in saturated liquid state or as sub-cooled liquid.

Although there exist some database in computer software such as EES that can provide the thermodynamic properties mentioned above, there are some engineer software as MatLab that do not contain the water/LiBr mixture as a common fluid for which their thermodynamic properties can be given.

One of the main purposes of this project is to evaluate some property correlations used by various sources in order to allow the calculation of these properties without the necessity of disposing of the proper software.

As the values of the enthalpy depend on the reference taken for calculations, all the fluids participating in the cycle should be evaluated taking the same temperature reference. For this reason, not only the properties for the water/lithium bromide mixture are discussed, but also those for water. In the different cases or methods, the conditions under which the enthalpy of the fluid that is being studied is equal to zero are specified.

In the following sections, various correlations for the different properties of water in saturated conditions are given. Also three different ways of calculating the enthalpy of the water/LiBr mixture are explained, for which a MatLab code has been written to facilitate calculations. To conclude, a comparison between the different methods will be carried out, studying their range of validity. For this purpose, database in EES will be taken as reference.

6.2. Water

6.2.1. Pressure

The equation that correlates pressure and temperature is the same for saturated conditions, regardless if it is saturated vapour or saturated liquid. The reason for this is that temperature remains constant with the pressure when the phase change takes place.

The equation known as Antoine’s equation allows this calculation, and is written as [20]:

$$log_{10}[p_{sat.water}(T)] = A - \frac{B}{T + C - 273.15} \tag{52}$$

Where the units for T are Kelvin and the calculated pressure is in bar. The values of the constants A, B and C depend on the fluid. In the case of water, these values are [20]:

$$A = 5.11564$$

$$B = 1687.537$$

$$C = 230.17$$

6.2.2. Enthalpy calculation taking 298 K as temperature reference

The enthalpy equation for pure water varies depending on the phase. This means that there are two different equations, one referred to water in liquid phase and another referred to water in vapour phase.

The expression of the enthalpy of saturated liquid water corresponds to [19]:

$$h_{liq.water}(T) = T - 298.15 \tag{53}$$

Where T is given in K and the units of the resultant enthalpy are kcal/kg. The heat capacity of water is taken to be equal to 1kcal/kgK in the range of temperatures of the cycle. Multiplying the result before by 4.1868 is enough to obtain the enthalpy in SI units (kJ/kg).

For saturated water vapours, the expression for enthalpy calculations is [19]:

$$h_{vap.water}(T) = 572.8 + 0.417 \cdot (T - 273.15) \tag{54}$$

Where, as well as before, the temperature and enthalpy units are K and kcal/kg respectively.

These two equations are valid for temperatures below 100 °C. Since water in the absorption heat pump proposed always works under that temperature, the two expressions can be used in order to obtain the enthalpies of the different points of the cycle where water is the working fluid.

6.2.3. Enthalpy calculation taking 273 K as temperature reference

It is considered T = 273 K as reference for the calculations, which means that the enthalpy of liquid water in saturated conditions at that temperature should be equal to zero.

The equation for liquid water is given by [21]:

$$\begin{aligned} h_{liq.water}(T,p) &= -1295.609655091 + 560.4921679618 \cdot T - 43.7942784871 \cdot T^2 + \\ &+4.479367474097 \cdot T^3 + 1.268510765098 \cdot p - 2.34584584613 \cdot 10^{-4} \cdot p - \\ &-3.87145821 \cdot 10^{-2} \cdot p \cdot T^2 \end{aligned} \tag{55}$$

Where the units for the pressure are [p] = $\frac{bar}{10}$ and for the temperature are [T] = $\frac{K}{100}$, and the units of the calculated enthalpy are [h] = $\frac{kJ}{kg}$.

The equation for water vapours is given by [21]:

$$h_{vap.water}(T,p) = 1997.854583553 + 0.9857789261768 \cdot p + 185.4761288854 \cdot T - \\ -1.194203005662 \cdot T^2 + 0.3002594147055 \cdot T^3 - 5850.240980017 \cdot \frac{p}{T^3} - \\ -25665290.31128 \cdot \frac{p}{T^{11}} \tag{56}$$

Where the units of the enthalpy, pressure and temperature are the same as in the previous equation.

6.3. Water/Lithium bromide mixture

6.3.1. Pressure and temperature: Experimental method

This method is based in the implementation of equations that allow to calculate the enthalpy, temperature and pressure analysing the correlations found between them in a range of experimental data [22].

As EES database was mentioned before, it should be said that the equations below correspond to those that EES uses when asked for water/LiBr mixture properties.

It is of interest mention that the correlations in this experimental method are valid for temperatures between 273 K and 500 K, and mixture composition from 0% to 75% of LiBr in the mixture, given as mass fraction [22].

All the equations refer to saturated liquid conditions, considering for the sub-cooled liquid states that temperature and enthalpy remain constant, and equal to those for saturation conditions, with pressure changes.

Pressure:

The expression that allows to calculate the pressure as a function of temperature and composition of the mixture is the following [22]:

$$p(T,x) = p_{sat.water}(\theta) \tag{57}$$

$$\theta = T - \sum_{i=1}^{8} a_i x^{m_i} (0.4 - x)^{n_i} \left(\frac{T}{T_c}\right)^{t_i} \tag{58}$$

The units are Pa for the pressure and K for the temperatures named as T and θ.

In the previous equation and in the following in this experimental method, x is the molar fraction of LiBr in the mixture and its relation with the mass fraction (w) of LiBr in the mixture can be written as a function of the molar mass (M) of the two components. [22]:

$$x = \frac{w/M_{LiBr}}{w/M_{LiBr} + (1-w)/M_{water}} \tag{59}$$

The values of the constants involved are shown in table A1, which can be found in the appendix.

Temperature: substituting the values of t_i in the equation for the pressure and assuming that θ is a known parameter since the saturation temperature of pure water is known for a given pressure, it can be worked out the expression for the temperature as a function of θ and x.

The equation looks like:

$$T(\theta, x) = \frac{\theta + \sum_{i=1}^{4} a_i x^{m_i} (0.4 - x)^{n_i}}{1 - \sum_{i=5}^{8} a_i x^{m_i} (0.4 - x)^{n_i} \left(\frac{1}{T_c}\right)} \tag{60}$$

The values of the coefficients and exponents are the same as for the equation that characterizes the pressure.

6.3.2. Enthalpy calculation taking 298 K as reference

The reference temperature of 298 K is related to the temperature reference taken for the enthalpy calculation of water, which means that the enthalpy of pure water at 298 K is zero.

The equation that gives the enthalpy of the mixture at temperature T and composition w is given by [19]:

$$h_{H_2O/LiBr}(T, w) = 42.81 - 425.92w + 404.67w^2 + (1.01 - 1.23w + 0.48w^2)(T - 273.15) \tag{61}$$

Where the temperature and enthalpy units are K and kcal/kg respectively. As explained before, the units of the enthalpy will be kJ/kg after multiplying by 4.1868. The mass fraction expressed as w is given in kg LiBr/ kg mixture.

6.3.3. Enthalpy calculation taking 273 K as reference: Experimental method

The reference temperature of 273 K refers to the temperature at which the enthalpy of the water/LiBr mixture is zero for a composition of 50% of LiBr in the mixture, given in mass fraction. This method is the same as EES has implemented for the fluid properties calculation of the water/lithium bromide mixture.

The expression that correlates the enthalpy of the water/LiBr mixture with its temperature could be written as follows [22]:

$$h_{H_2O/LiBr}(T, x) = (1 - x)h'(T) + h_c \sum_{i=1}^{30} a_i x^{m_i} (0.4 - x)^{n_i} \left(\frac{T_c}{T - T_p}\right)^{t_i} \tag{62}$$

Where $h'(T)$ represents the enthalpy of saturated pure liquid water at temperature T (taking 273 K as reference for the calculation) and a_i, m_i, n_i and t_i are constant coefficients whose values correspond to the values shown in the table A2 in the appendix. T_c, T_p and h_c are also constants.

The units are Kelvin for temperatures and J/mol for enthalpies. To transform the units of the enthalpy to SI units (kJ/kg) it is necessary to operate as in the following formula:

$$h[kJ/kg] = h[J/mol] \cdot \frac{w[kg_{LiBr}/kg_{mix}]}{x[mol_{LiBr}/mol_{mix}] \cdot M_{LiBr}[kg_{LiBr}/mol_{LiBr}] \cdot 1000[J/kJ]} \quad (63)$$

6.3.4. Enthalpy calculation taking 273 K as reference: Theoretical method

This method consists of a description of the enthalpy of a mixture as a function of the enthalpy of the two fluids involved in the mixture and the value of the excess enthalpy due to the mixing process.

The reference temperature of 273 K is referred to the reference temperature of the pure components of the mixture. This means that at the mentioned temperature of 273 K, the enthalpies which are equal to zero are those of pure water and pure lithium bromide, but not that of the water/lithium bromide mixture.

Mathematically, the desired enthalpy is the sum of those of the pure components involved at the same temperature and pressure and the heat of mixing, which corresponds to the excess value [23].

The excess value mentioned before is explained as the difference between the enthalpies of a real mixture and an ideal mixture [23].

The expression for the enthalpy of the mixture as a function of the enthalpies of its components and the excess value is written as [23]:

$$h_{H_2O/LiBr}(T,p) = w \cdot h'_{LiBr}(T,p) + (1-w) \cdot h'_{water}(T,p) + h^e(T,p) \quad (64)$$

As in the experimental method, w corresponds to the mass fraction of lithium bromide in the mixture and $h'_{LiBr}(T,p)$ and $h'_{water}(T,p)$ are the enthalpies of the pure components, LiBr and water respectively, at temperature T and pressure p. The term $h^e(T,p)$ represents the specific excess enthalpy, which is calculated with the formula [23]:

$$h^e(T,p) = \frac{H_m^e}{M_m} \quad (65)$$

Where H_m^e is the molar excess enthalpy and M_m is the molar mass of the mixture.

The following equations could be used in order to calculate the correspondent values of both of them [23]:

$$H_m^e = (1-x)x\left[\left(a_h + \frac{b_h}{T}\right) + (1-x)\left(c_h + \frac{d_h}{T}\right)\right] \quad (66)$$

$$M_m = x\frac{M_{LiBr}}{k} + (1-x)M_{water} \quad (67)$$

Where k, a_h, b_h, c_h and d_h are constants for each specific mixture. The values of these constants for water/LiBr mixture are shown in table 7.

As for the previous method, x refers to the molar fraction of lithium bromide in the mixture.

Correlations for the calculation of $h'_{LiBr}(T,p)$ and $h'_{water}(T,p)$ are given by [23]:

$$h'_{water}(T,p) = a_{cp,water}(T - T_0) + \frac{1}{2}b_{cp,water}(T^2 - T_0^2) + \frac{1}{3}c_{cp,water}(T^3 - T_0^3) \qquad (68)$$

$$h'_{LiBr}(T,p) = a_{cp,LiBr}(T - T_0) + \frac{1}{2}b_{cp,LiBr}(T^2 - T_0^2) + \frac{1}{3}c_{cp,LiBr}(T^3 - T_0^3) \qquad (69)$$

Where $a_{cp,water}$, $b_{cp,water}$, $c_{cp,water}$, $a_{cp,LiBr}$, $b_{cp,LiBr}$ and $c_{cp,LiBr}$ are constant values, as shown in table 8.

The same as for water and for the previous method for enthalpy calculation in the water/lithium bromide mixture, temperature T = 273 K is taken as the reference temperature at which the enthalpy is equal to zero.

Table 7. Coefficients of the working pair for the excess enthalpy calculation [23].

Working pair	Correlation factor k	a_h (kJ/kmol)	b_h (kJK/kmol)	c_h (kJ/kmol)	d_h (kJK/kmol)
Water/LiBr	6.0	$-0.34362*10^5$	$0.33557*10^6$	$0.17733*10^7$	$0.23853*10^7$

Table 8. Coefficients of the absorbate and absorbant for enthalpy calculation [23].

Fluid	a_{cp} (kJ/kgK)	b_{cp} (kJ/kgK^2)	c_{cp} (kJ/kgK^3)
Water	$0.5331*10^1$	$-0.7297*10^{-2}$	$0.1157*10^{-4}$
Lithium bromide	$0.800*10^0$	$0.295*10^{-3}$	$-0.500*10^{-7}$

T_0 = 273 K

6.4. Validation

The three different methods of enthalpy calculation given have been validated using EES. For the validation, the results obtained with the different methods have been compared to those obtained with EES for various pressure levels and compositions of the mixture.

The three methods has been validated for a pressure range between 1 kPa and 45 kPa. In relation to the composition, water/LiBr absorption heat pumps usually work with mixtures that have mass fractions from 35% to 65% of lithium bromide. Therefore, the validation is also related to those compositions.

In the tables made for the comparison, the values of the enthalpies obtained in all the different cases employing the three methods are shown, as well as the results obtained with EES for the same inputs and the differences between them.

The difference between the correspondent method and EES is what allows to determine the accuracy of the first one. Varying the reference when calculating the enthalpy has as consequence a variation in the value for the specific enthalpy under the same conditions. If this difference remains almost constant for the entire range of pressures and temperatures of a mixture with a fixed composition, it could be said that the method is valid for the enthalpy calculation of that mixture with the given composition and in the pressure range studied.

The three tables used for the analysis are related to three different molar fractions (x=0.1; x=0.2; x=0.3), each of them with their correspondent mass fraction shown in brackets. In each case five different pressure levels are considered (1 kPa, 2 kPa, 15 kPa, 30 kPa and 45 kPa), showing their correspondent saturation temperature of the considered mixture in brackets.

The four methods represented in the tables are:

- Calculations employing EES for the composition and saturation temperature (shown in brackets in the table) of the mixture corresponding to the pressure given. In the table is referred as "EES".
- Experimental calculation method for enthalpies with 273 K as temperature reference. It corresponds to reference [22]. In the table is named as "Ref. 273 K (Experimental)".
- Enthalpy calculation method with 298 K as temperature reference. The name given in the table is "Reference 298 K", and is related to reference [19].
- Theoretical calculation method for enthalpies with 273 K as temperature reference. It is related to reference [23]. The correspondent name in the table is "Ref. 273 K (Theoretical)".

The first table, where x = 0.1, is shown below, while the other two tables are placed in the appendix.

Table 9. Values of the enthalpies for the different methods when x=0.1 and differences with the results obtained using EES.

x=0.1 (w=0.3488)						
Pressure		1 kPa (T=287 K)	2 kPa (T=298 K)	15 kPa (T=336 K)	30 kPa (T=352 K)	45 kPa (T=362 K)
Method	EES	23.170	51.990	154.400	196.900	224.000
	[22] Ref. 273 K (Experimental)	23.537	52.261	154.368	196.868	223.967
	[19] Reference 298 K	-200.171	-170.614	-67.798	-25.068	2.114
	[23] Ref. 273 K (Theoretical)	-7.824	24.040	135.687	182.525	212.481
Diff with EES	Ref. 273 K (Experimental)	-0.367	-0.271	0.032	0.032	0.033
	Reference 298 K	223	223	222	222	222
	Ref. 273 K (Theoretical)	31	28	19	14	12

The differences between the results from the various methods and those from EES are represented in the graph below for three pressures: 1 kPa, 15 kPa and 45 kPa.

On top of each bar, the correspondent number of the difference that is representing is shown, so that the evaluation of the graph is easier.

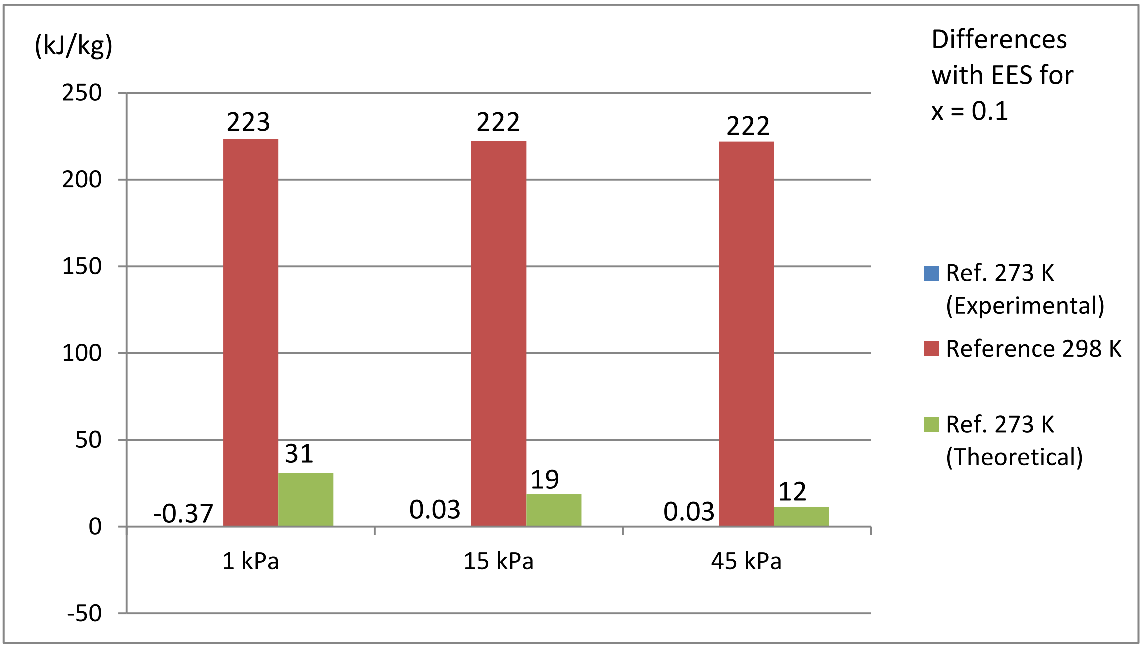


Figure 6. Differences between EES and the different methods when x = 0.1

From the table and the graph above it can be concluded that, for a composition of almost 35% of LiBr in the mixture, the method with a temperature reference of 298 K and the experimental method with 273 K as temperature reference are quite accurate for the enthalpy calculation of the mixture. In the other hand, the differences between the results of the third method (theoretical with 273 K as reference) and the results obtained with EES change considerably from the initial pressure of 1 kPa to the final pressure of 45 kPa. These changes indicate that the last method is not accurate for the enthalpy calculation of the mixture for the given composition.

In other words, an analysis of the results is:

- As expected, the differences between the EES results and those obtained with the experimental method with 273 K as reference are insignificant in the entire range of pressures considered. Thus, this method is accurate enough for enthalpy calculation of the mixture under the given conditions.
- The differences between the results obtained with the method with the temperature reference of 298 K and EES remain almost constant and equal to 222 kJ/kg in the whole range of pressures studied. This means that the results are the same, just needing and adjustment of a constant value of 222 kJ/kg. As before, it could be concluded that this method is accurate enough for enthalpy calculations of the mixture under the given conditions.
- In the third case, the difference between the enthalpy obtained with EES and with the theoretical method using 273 K as reference goes from 31 kJ/kg at a pressure of 1 kPa to 11.5 kJ/kg at a pressure of 45 kPa. The variation of almost 20 kJ/kg in the range of pressures studied is considered to be high enough not to consider this third method as accurate as the two before for enthalpy calculations under the given conditions.

In the following graph it can be observed that the pattern followed by the different methods is the same for all of them except for the theoretical. The lines for the EES (blue) results and the experimental (red) ones are almost coincident and the green line (method referred to the 298 K reference) is parallel to them, while the purple (theoretical method) one starts much more separated from the red one than it ends.

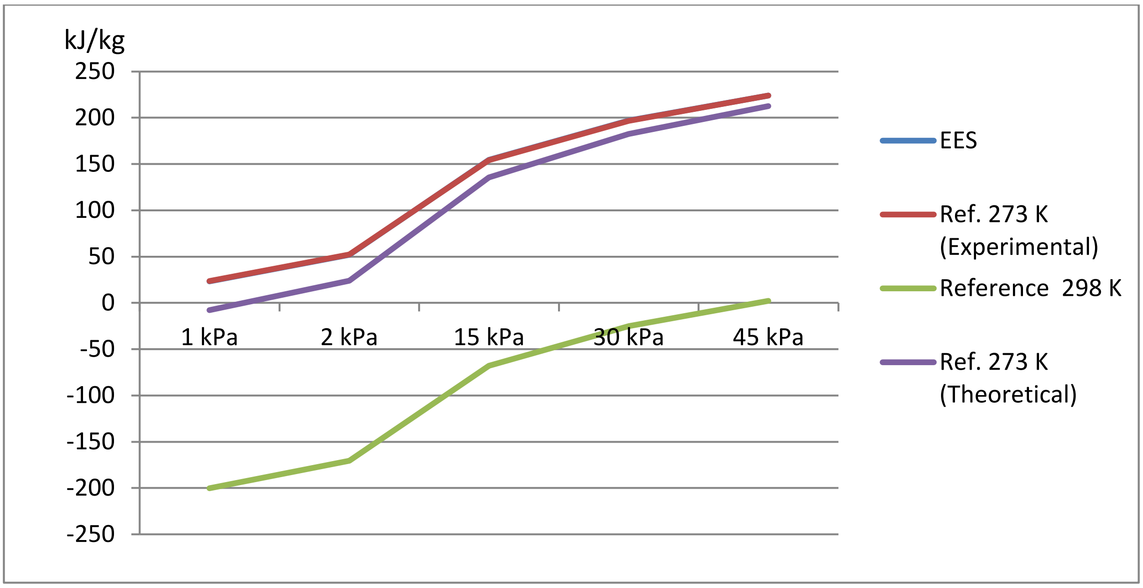


Figure 7. Pattern followed by the enthalpy when increasing pressure and x = 0.1

The values of the enthalpies when x = 0.2 for each method and their differences with EES are given in table A3 in the appendix. The following graph represents these differences.

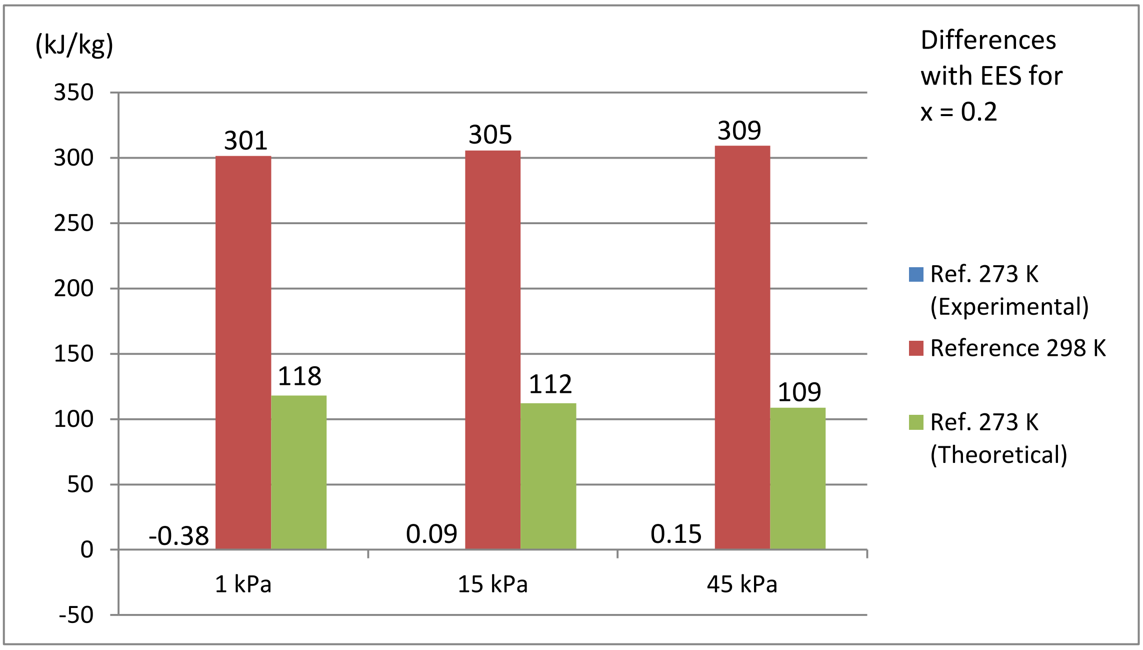


Figure 8. Differences between EES and the different methods when x = 0.2

In this case, the differences between the different methods and EES are almost constant for all of them. Consequently, it could be said that all of them are appropriate for the calculation of the enthalpies of the water/LiBr mixture in the range of pressures 1 kPa - 45 kPa when the mixture contains an approximately 55% of LiBr in mass fraction.

The graph below shows how the different methods follow all of them the same pattern, represented with parallel lines.

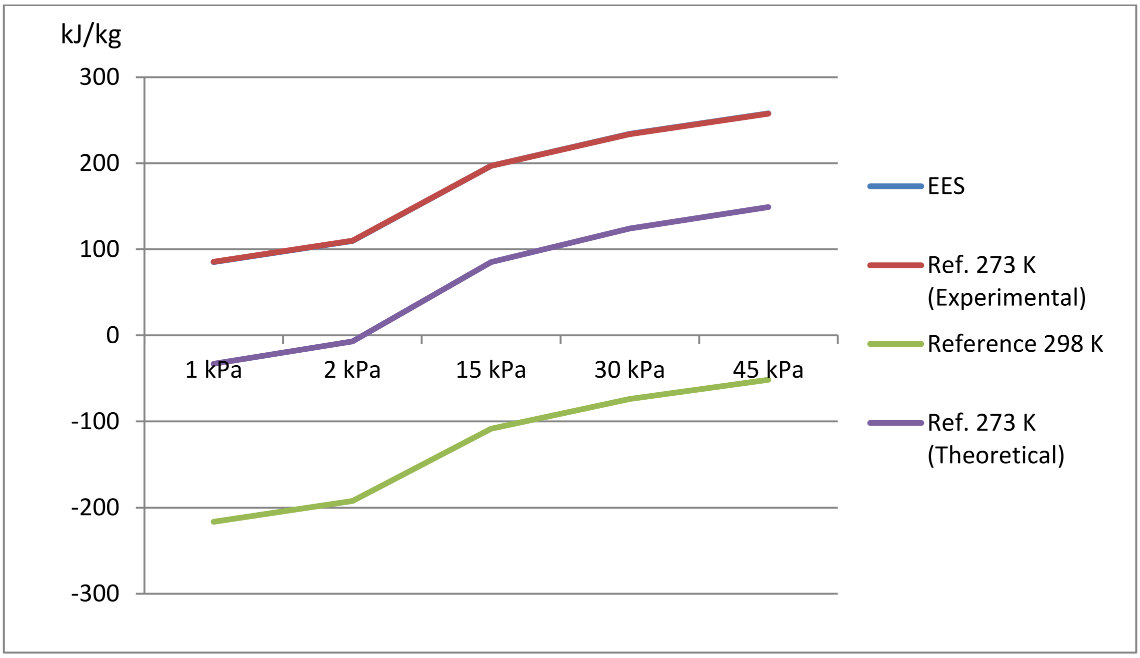


Figure 9. Pattern followed by the enthalpy when increasing pressure and x = 0.2

As in the previous case, the values of the enthalpies when x = 0.3 for each method and their differences with EES are given in table A4 in the appendix. The following graph represents these differences.

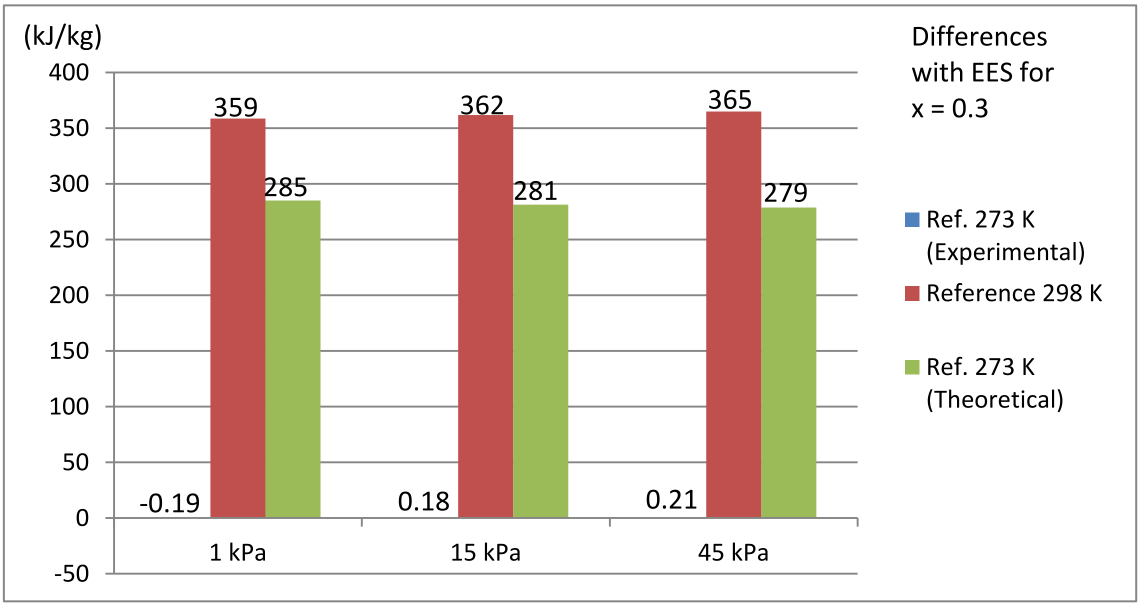


Figure 10. Differences between EES and the different methods when x = 0.3

The same as in the previous case, the differences for all the different methods with the EES results remain quite constant with the pressure increase, what makes possible to affirm that the three methods are able to give accurate values of the enthalpy of the mixture when it contains a 67% of LiBr in mass terms.

The graph showing the parallel growth of the enthalpy with the pressure for the different methods is given below.

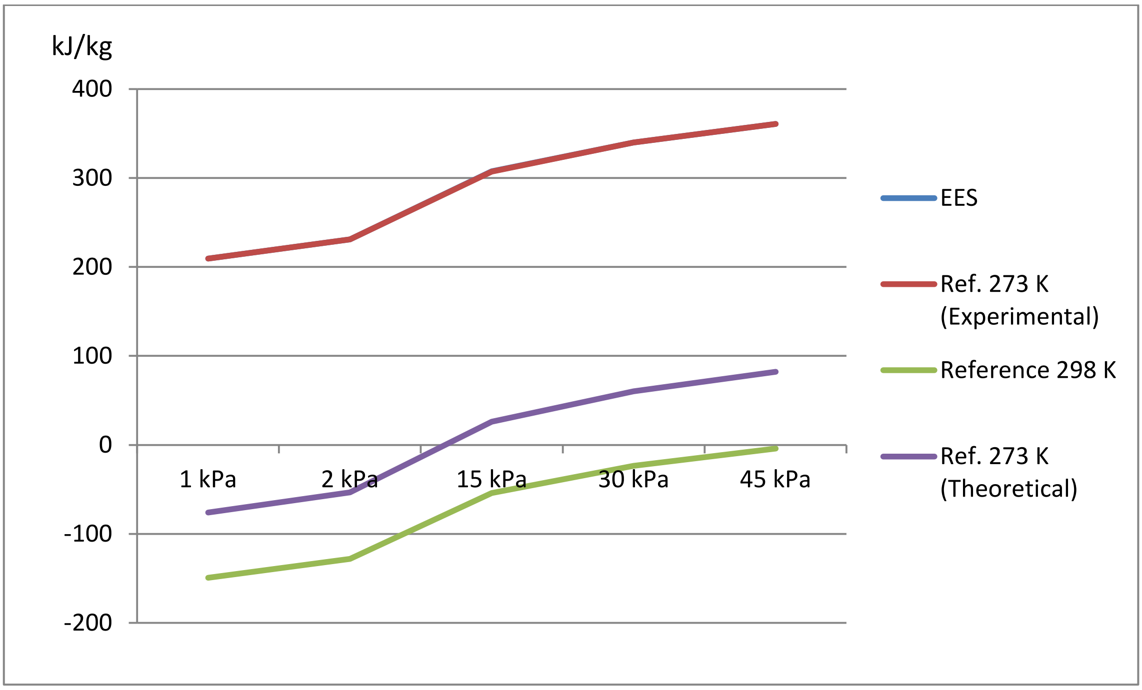


Figure 11. Pattern followed by the enthalpy when increasing pressure and x = 0.3

To complete the validation, it has to be taken in account that the references taken for the different methods are not only varying in the temperature. For instance, EES and thus the experimental method are taking as reference the water/LiBr mixture with a mass fraction of 0.5 of LiBr at 273 K, while the theoretical method is taking as reference the pure components (water and lithium bromide) at 273 K, and the method with 298 K as reference is also referring to the pure components at that temperature.

In the last two methods, the enthalpy of mixing has to be taken into consideration, and varies not only with the temperature chosen but also with the mass fraction of LiBr in the mixture.

To validate the different methods when working with various mass fractions, the heat capacities of the absorber and the generator of the cycle of the system proposed have been computed.

The absorber and the generator are working with both the rich and the weak solutions, what makes possible to check the accuracy of the three methods comparing the heat capacities obtained when these are calculated employing the resultant enthalpies from each of them.

The comparison has been made for two different pairs of rich and weak solutions. In one of the cases, the mass fractions are 0.5 and 0.6 of LiBr in the mixture respectively; and in the other case the mass fractions are 0.45 and 0.55 respectively. The tables below show this comparison. The experimental method is assumed to be the same as EES.

Four tables can be found, two of them related to the calculations in the absorber and another two related to the calculations in the generator. Below each table the values of the mass flow rate and temperature of the points involved in the heat capacity calculation can be found.

After them, an explanation of the conclusions obtained is given, analysing the validity of the three methods employed for the enthalpy calculations of the mixture, under the conditions taken for the absorption heat pump the system proposed is dealing with.

The equations for the heat capacities are the following:

$$Q_{abs}[kW] = \dot{m}_{10}[kg/s] \cdot h_{10}[kJ/kg] + \dot{m}_3[kg/s] \cdot h_3[kJ/kg] - \dot{m}_4[kg/s] \cdot h_4[kJ/kg] \quad (70)$$

$$Q_{gen}[kW] = \dot{m}_7[kg/s] \cdot h_7[kJ/kg] + \dot{m}_1[kg/s] \cdot h_1[kJ/kg] - \dot{m}_6[kg/s] \cdot h_6[kJ/kg] \quad (71)$$

Table 10. Comparison of the Q_{abs} calculated from the enthalpies from the different methods, when w_{Rich} = 0.5 and w_{weak} = 0.6.

ABSORBER		**w_{Rich}=0.5; w_{Weak}=0.6**			
Point		h_{10} (kJ/kg)	h_3 (kJ/kg)	h_4 (kJ/kg)	Q_{abs} (kW)
Method	EES=Ref. 273 K (Exp.) [22]	2528	148.1	76.49	6.242378
	[23] Ref. 273 K (Theoretical)	2528	-26.8889	-5.855	5.585
	[19] Reference 298 K	2424.5	-176.575	-209.144	6.302
Differences with EES	Ref. 273 K (Theoretical)				0.657
	Reference 298 K				-0.059

T_3 = 329.1 K; T_4 = 310.1 K; $\dot{m}_{10}$ = 0.00228 kg/s; $\dot{m}_3$ = 0.009119 kg/s; $\dot{m}_4$ = 0.0114

Table 11. Comparison of the Q_{abs} calculated from the enthalpies from the different methods, when w_{Rich} = 0.45 and w_{weak} = 0.55.

ABSORBER		**w_{Rich}=0.45; w_{Weak}=0.55**			
Point		h_{10} (kJ/kg)	h_3 (kJ/kg)	h_4 (kJ/kg)	Q_{abs} (kW)
Method	EES=Ref. 273 K (Exp.) [22]	2528	110.8	58.53	6.048
	[23] Ref. 273 K (Theoretical)	2528	-9.325	0.986	5.679
	[19] Reference 298 K	2424.5	-192.73	-210.469	6.149
Differences with EES	Ref. 273 K (Theoretical)				0.3682
	Reference 298 K				-0.102

T_3 = 321.2 K; T_4 = 303.2 K; $\dot{m}_{10}$ = 0.00228 kg/s; $\dot{m}_3$ = 0.00798 kg/s; $\dot{m}_4$ = 0.01026

Table 12. Comparison of the Q_{gen} calculated from the enthalpies from the different methods, when w_{Rich} = 0.5 and w_{weak} = 0.6.

GENERATOR		**w_{Rich}=0.5; w_{Weak}=0.6**			
Point		h_7 (kJ/kg)	h_1 (kJ/kg)	h_6 (kJ/kg)	Q_{gen} (kW)
Method	EES=Ref. 273 K (Exp.) [22]	2643	297.2	201.3	6.441
	[23] Ref. 273 K (Theoretical)	2643	129.051	127.407	5.750
	[19] Reference 298 K	2538	-35.414	-88.181	6.469
Differences with EES	Ref. 273 K (Theoretical)				0.691
	Reference 298 K				-0.028

T_1 = 404.9 K; T_6 = 366.2 K; $\dot{m}_7$ = 0.00228 kg/s; $\dot{m}_1$ = 0.009119 kg/s; $\dot{m}_6$ = 0.0114

Table 13. Comparison of the Q_{gen} calculated from the enthalpies from the different methods, when w_{Rich} = 0.45 and w_{weak} = 0.55.

GENERATOR		**w_{Rich}=0.45; w_{Weak}=0.55**			
Point		h_7 (kJ/kg)	h_1 (kJ/kg)	h_6 (kJ/kg)	Q_{gen} (kW)
Method	EES=Ref. 273 K (Exp.) [22]	2643	262.9	180.7	6.27
	[23] Ref. 273 K (Theoretical)	2643	151.094	133.425	5.863
	[19] Reference 298 K	2424.5	-48.021	-89.921	6.326
Differences with EES	Ref. 273 K (Theoretical)				0.407
	Reference 298 K				-0.056

T_1 = 393.4 K; T_6 = 355.2 K; $\dot{m}_7$ = 0.00228 kg/s; $\dot{m}_1$ = 0.00798 kg/s; $\dot{m}_6$ = 0.01026

From the tables, it can be observed that the enthalpy calculations with the experimental method and the method with 298 K as reference are giving similar results of the heat capacities. The conclusion from this is that those two methods are quite accurate and, consequently, any of the two could be used in order to know the enthalpies of the water/LiBr mixture in the points of the cycle where necessary.

The theoretical method with 273 K as reference gives values of heat capacities that are considerably different from those obtained when the enthalpies are calculated with EES. For this reason, this method is considered to be not enough accurate for the enthalpy calculations needed in the range of pressures and mass fractions of the cycle studied.

7. Components and Size Optimisation

7.1. Heat exchangers

Heat exchangers are devices in which heat from a fluid is transferred to another fluid without having direct contact between them. Temperature difference between the two fluids is what produces this heat transfer. It can occur involving a phase change or not. In the case that a phase change takes place, the heat exchangers involved are called evaporator, if liquid changes into vapour, and condenser, if vapour changes into liquid [24].

In the system that concerns this project, there are present various heat exchangers. These are the liquid-liquid heat exchanger in which the weak water/LiBr solution transfers heat to the rich water/LiBr solution; the condenser, in which hot air from the environment is used to condensate the refrigerant (steam into water); the evaporator, in which water evaporates before entering the absorber, and allows to cool down the temperature of the air coming in; the absorber, where the weak solution absorbs the steam from the evaporator reducing its temperature, heating the air from the environment; and the generator, in which the exhaust gases from the fuel cell heat the rich solution until the refrigerant evaporates from the mixture.

The heat exchanger between the rich and the weak solutions in the system is assumed to be a concentric heat exchanger, due to the small area required (seen in the results from the EES model), which could be considered a special case of shell-and-tube heat exchanger. For the generator, condenser, evaporator and absorber, the most appropriate heat exchanger in automotive applications is the compact heat exchanger. For this reason, and to have a better understanding of the types of heat exchangers that would be used in the proposed system, a brief explanation about their main characteristics will be given in this section.

7.1.1. Shell-and-Tube Heat Exchanger

Shell-and-tube heat exchangers are one of the most commonly used heat exchanger types. Its construction is based on one or more tubes inside a shell that usually has the same shape as a pipe [24]. There are different configurations depending on the number of tube and shell passes, being specially employed those with one shell and one tube pass, one shell pass and two tube passes, and two shell passes and four tube passes [16].

In these shell-and-tube heat exchangers is usual to find baffles, which are not only employed to support physically the tubes and reduce vibration, but to induce turbulent flow in the shell-side fluid and to increase the path length of the fluid, increasing its convection coefficient [16].

Shell-and-tube heat exchangers are characterized by one of the fluids flowing inside the tubes, while the other flows through the shell, what makes heat transfer possible between the two [25].

7.1.2. Concentric Heat Exchanger

Concentric heat exchangers are a particular case of shell-and-tube type, with an only tube inside a tubular shell. If the fluid flowing inside the tube and that flowing through the shell have the same direction, then the heat exchanger is co-current; in the other case, in which the two fluids flow in opposite directions, the heat exchanger is called counter-flow [25].

7.1.3. Compact Heat Exchangers

Compact heat exchangers are specially thought to be employed in automotive and aircraft applications, where size and weight are important factors. For this reason, they are specifically designed so that the heat transfer surface area is as high as possible per unit volume [25].

Gases have low heat transfer coefficients, what makes compact heat exchangers to be mainly used when gas-to-gas, liquid-to-gas or gas-to-liquid heat exchangers are required. In this case, the higher surface area counteracts the low heat transfer coefficient, allowing great heat transfer without the necessity of a heat exchanger of impressive dimensions [25].

The evaporator, condenser and absorber of the absorption heat pump for the air conditioning of the vehicle are all dealing with air, reason for which compact heat exchangers are the best option to achieve the required heat transfer minimizing the volume that they are occupying in the vehicle.

Cross-flow is the most common configuration for compact heat exchangers. This means that the two fluids have perpendicular flow to each other. In the case of a configuration of finned-tubes, one of the fluids flows inside the tubes, while the other flows perpendicularly through the space between the fins [25].

The parameter β (area density), defines the compactness of the heat exchanger. As minimum values, β has to be at least equal to 400 m^2/m^3 for liquids and at least equal to 700 m^2/m^3 for gases for the heat exchanger to be considered as compact [16].

7.2. Minimum Overall Heat Transfer Coefficients

As explained, the different heat exchangers involved in the system are concentric type for the heat exchanger between the rich and the weak solutions, and compact heat exchangers for the evaporator, the condenser, the generator and the absorber.

The overall heat transfer coefficients of each of them are different depending not only on the type but also on the fluids acting in them and the areas of heat transfer available in each case.

For the same heat capacity and same inlet and outlet temperatures of the fluids in a heat exchanger, the values of the heat transfer area (A) and the overall heat transfer coefficient (U) are inversely proportional. This is, following the formula below, when A decreases, U increases and viceversa.

$$Q = U \cdot A \cdot \Delta T_{LMTD} \tag{72}$$

Where ΔT_{LMTD} is the logarithmic mean temperature difference.

Since the size of the air conditioning in a car has to fit in a limited space, a minimum value of U has been calculated for each compact heat exchanger, employing space restrictions. In order to do so, an EES model has been developed, in which the different constraints have been included and could be modified if required.

7.2.1. Model

The procedure, assumptions and equations employed in the model for the overall heat transfer coefficient calculations are shown in this section.

7.2.1.1. Procedure

For the implementation of the model, the first step is the calculation of the volume of the solution heat exchanger, which is the only non-compact heat exchanger present in the system. For the calculation of its area (and consequently volume) and its overall heat transfer coefficient, an iteration loop has to be implemented.

Initially, a possible value of the overall heat transfer coefficient of the solution heat exchanger is assumed, as well as its length, from which the correspondent area can be easily computed. Employing this data and imposing equal velocities of the fluids in the two tubes of the heat exchanger, the diameters of the tubes are calculated.

The Reynolds numbers of the two fluids indicate that the heat exchanger is dealing with laminar flow, and so, assuming fully developed flow, the Nusselt numbers are fixed only depending on the relation between the diameters of the inner and outer tubes in the case of the annular flow. With the conduction coefficients given by EES, Nusselt numbers from reference [25], and diameters, convection heat transfer coefficient of the fluids are calculated, and so the correspondent U value of the heat exchanger. If this result in coincident with the first value assumed, the iteration is over, if not, another value should be assumed and the same process repeated.

There are different combinations of length and U that make the loop converge, from which the combination that allows to have reasonable numbers for the pressure drop in the tubes and for the overall heat transfer coefficients in the rest of the components has been chosen.

Once the dimensions of the solution heat exchanger are known, and taking as maximum desired volume of the system the same volume as that of the fuel cell system (more details are given in the assumptions section), a possible combination of the maximum volumes of the rest of the components could be given. From that estimation, the correspondent values of overall heat transfer coefficient of each of them can be computed and evaluated, and from the evaluation, the combination of volumes can be modified until all the results look reasonable.

7.2.1.2. Assumptions

The first assumption, which is a constraint for all the other calculations, is the total volume of the system. It is considered that the volume of the whole system, considering it as the sum of the volumes of the generator, the condenser, the evaporator, the absorber and the heat exchanger between rich and weak solutions, is not bigger than the volume of the fuel cell in charge of providing the electricity to the motor and the heat to the absorption heat pump.

Some data taken from the website of the company Serenergy show that the volume of a fuel cell system is approximately of 77 litres. Consequently, a volume of 0.077 m^3 has been considered as maximum volume for the absorption heat pump system.

To have an idea of the dimensions of the system, 0.077 m^3 correspond to the volume occupied by a cube whose sides have a length of 0.4254 m.

Another general assumption is that average overall heat transfer coefficients are considered, assuming they are not changing along the process that takes place in each heat exchanger due to temperature changes.

Rich-weak solution heat exchanger:

- A concentric heat exchanger has been considered between the rich and the weak solution, for which the two tubes involved have been considered to have no thickness, this is, the inner and outer diameter of the tubes are the same;
- The contribution of heat conduction in the walls of the tubes is negligible compared to convection for the calculation of the overall heat transfer coefficient of the heat exchanger, and consequently has not been considered;
- The length of the concentric heat exchanger is assumed to be 6.168 m and its overall heat transfer coefficient equal to 100 W/m^2K for the iteration loop to converge.

Condenser, evaporator, generator and absorber:

- The four of them are compact heat exchangers with flat tubes and continuous fins, where both fluids are unmixed;
- In all cases, the relation between heat transfer area of the air side and total volume of the heat exchanger is 886 m^2/m^3 [26];
- A pinch temperature of 5 K is considered in the evaporator;
- A pinch temperature of 10 K is considered in the condenser and in the generator;
- The temperature of the outlet air in the absorber has been considered to be 65 °C;
- Serenergy HTPEM fuel cell systems operate between 150 °C and 180 °C, and thus a temperature of 170 °C has been assumed for the exhaust from the fuel cell entering the generator;

- It is assumed that condenser, evaporator and absorber have the same total volume, while the volume of the generator is considered to be half of the volume of the other three.

7.2.1.3. Equations

Compact heat exchangers are cross-flow, reason why a correction factor has to be taken into consideration when computing the heat capacities as a function of the logarithmic mean temperature difference [25].

The correction factor, named F, depends on two other factors, P and R, defined as [25]:

$$P = \frac{T_{tube,2} - T_{tube,1}}{T_{shell,1} - T_{tube,1}} \tag{73}$$

$$R = \frac{T_{shell,1} - T_{shell,2}}{T_{tube,2} - T_{tube,1}} \tag{74}$$

Where 1 and 2 refer respectively to the inlet and outlet temperatures of the shell or the tube as indicated in the formulas.

The values for F are given by an existing function in EES for cross-flow heat exchangers. The correspondent logarithmic mean temperature difference employed in each case in which a cross-flow heat exchanger is involved is that as if it was a counter-flow type.

The expression for the heat capacity in this case includes the correction factor, having the following aspect:

$$Q[kW] = U[kW/(m^2K)] \cdot A[m^2] \cdot F \cdot \Delta T_{LMTD_{CF}}[K] \tag{75}$$

Heat exchanger:

$$NTU = \frac{U_{HEx}[kW/(m^2K)] \cdot A_{HEx}[m^2]}{C_{min}[kW/K]} \tag{76}$$

As seen in the equation for the model of the system, from which the area of the heat exchanger is calculated. Once that area is known, the volume of the heat exchanger can also be computed following the steps below. First of all, the internal diameter is calculated:

$$A_{HEx}[m^2] = \pi \cdot d_{int}[m] \cdot L_{HEx}[m] \tag{77}$$

It is imposed that both the rich and the weak solution have the same velocity:

$$v_{rich}[m/s] = v_{weak}[m/s] \tag{78}$$

The formula $\dot{m} = \rho \cdot v \cdot A$ allows to work out the value of the velocity in each case, and substituting the area by $A = \pi \cdot d^2/4$, the next two formulas are obtained, where it is taken in account that the rich solution flows through the external tube and the weak solution does it through the internal tube. Introducing equations (79) and (80) in equation (78), the relation between the diameters of the inner and the outer tubes can be found.

$$v_{rich}[m/s] = \frac{4 \cdot \dot{m}_5[kg/s]}{\rho_{rich}[kg/m^3] \cdot \pi \cdot (d_{ext}^2[m] - d_{int}^2[m])} \quad (79)$$

$$v_{weak}[m/s] = \frac{4 \cdot \dot{m}_1[kg/s]}{\rho_{weak}[kg/m^3] \cdot \pi \cdot d_{int}^2[m]} \quad (80)$$

The volume of the heat exchanger corresponds to the following formula, expressed as a function of its radius:

$$R_{ext}[m] = \frac{d_{ext}[m]}{2} \quad (81)$$

$$V_{HEx}[m^3] = \pi \cdot R_{ext}^2[m] \cdot L_{HEx}[m] \quad (82)$$

The known Nusselt numbers for fully developed laminar flow can be substituted in the next formula in order to compute the h value of the rich and the weak solutions.

$$Nu = \frac{h[kW/(m^2K)] \cdot d[m]}{k[kW/(mK)]} \quad (83)$$

The overall heat transfer coefficient of the heat exchanger can be worked out and calculated from:

$$\frac{1}{U_{HEx}[kW/(m^2K)]} = \frac{1}{h_{rich}[kW/(m^2K)]} + \frac{1}{h_{weak}[kW/(m^2K)]} \quad (84)$$

To make sure that the employed dimensions can actually be used, the pressure drop in the heat exchanger must be checked. Its formula corresponds to:

$$\Delta P[Pa] = \frac{1}{2} \cdot \rho[kg/m^3] \cdot v^2[m/s] \cdot \frac{L[m]}{d[m]} \cdot f \quad (85)$$

Where f is the friction factor and can be expressed as a linear function of the Reynolds number when dealing with laminar flow:

$$f = \frac{64}{Re} \quad (86)$$

$$Re = \frac{v[m/s] \cdot d_h[m] \cdot \rho[kg/m^3]}{\mu[kg/(ms)]} \quad (87)$$

Values of densities, viscosities and conduction coefficients of the weak and the rich solutions are taken from the EES library.

Reynolds number, convection heat transfer coefficient, pressure drop and friction factor are calculated for the weak and the rich solutions, modifying in each case the parameters involved in their calculation.

As the mass flow rates of both solutions are quite small, the velocities and so the Reynolds numbers are also low. This means that the flow is laminar. Assuming fully developed flow, the

Nusselt numbers of the rich solution (in an annulus) and the weak solution (in a circular tube), are [25]:

$$Nu_{rich} = 4.585$$

$$Nu_{weak} = 4.36$$

Common to compact heat exchangers:

$$V_{compact}[m^3] = V_{total}[m^3] - V_{HEx}[m^3] \tag{88}$$

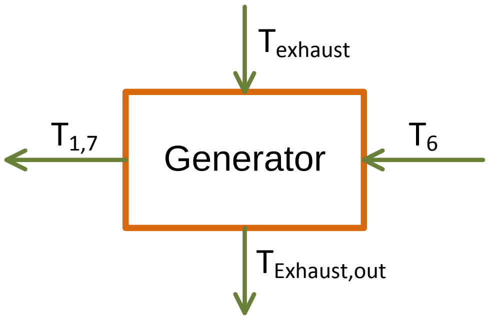
As explained in the assumptions, the relation between the volumes of the four compact heat exchangers is assumed, where the condenser, the evaporator and the absorber have the same total volume, while the volume of the generator is considered to be half of the volume of the other three. In such case, and taking the volume of the generator as reference, it could be said that the total volume occupied by the four compact heat exchangers is seven times the volume of the generator, what justifies the following calculation:

$$V_\alpha[m^3] = \frac{V_{compact}[m^3]}{7} \tag{89}$$

$$\alpha[m^2/m^3] = \frac{A_i[m^2]}{V_i[m^3]} \tag{90}$$

Where the volume of the generator is equal to V_α.

Generator:



$$Q_{gen}[kW] = U_{gen}[kW/(m^2K)] \cdot A_{gen}[m^2] \cdot F_{gen} \cdot \Delta T_{LMTD_{gen}}[K] \tag{91}$$

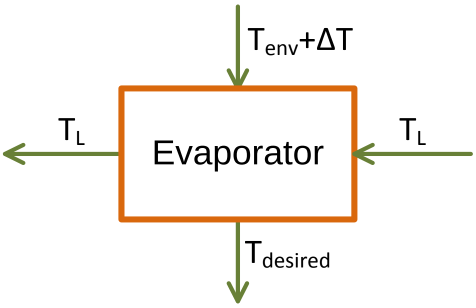
$$\Delta T_{LMTD_{gen}}[K] = \frac{\left(T_{exhaust}[K] - T_{1,7}[K]\right) - \left(T_{exhaust,out}[K] - T_6[K]\right)}{\ln\dfrac{\left(T_{exhaust}[K] - T_{1,7}[K]\right)}{\left(T_{exhaust,out}[K] - T_6[K]\right)}} \tag{92}$$

$$T_{1,7}[K] = \frac{\dot{m}_1[kg/s] \cdot cp_1[kJ/(kgK)] \cdot T_1[K] + \dot{m}_7[kg/s] \cdot cp_{water,gen}[kJ/(kgK)] \cdot T_7[K]}{\dot{m}_1[kg/s] \cdot cp_1[kJ/(kgK)] + \dot{m}_7[kg/s] \cdot cp_{water,gen}[kJ/(kgK)]} \tag{93}$$

$$V_{gen}[m^3] = V_\alpha[m^3] \tag{94}$$

$T_{1,7}$ makes reference to the mixing temperature of the two fluids going out of the generator, this is, the weak solution and the steam separated from the mixture.

Evaporator:



$$Q_{evap}[kW] = U_{evap}[kW/(m^2K)] \cdot A_{evap}[m^2] \cdot \Delta T_{LMTD_{evap}}[K] \tag{95}$$

$$\Delta T_{LMTD_{evap}}[K] = \frac{((T_{env}[K] + \Delta T[K]) - T_L[K]) - (T_{desired}[K] - T_L[K])}{\ln \frac{((T_{env}[K] + \Delta T[K]) - T_L[K])}{(T_{desired}[K] - T_L[K])}} \tag{96}$$

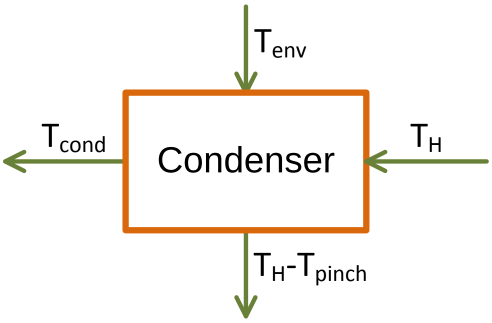
The correction factor in the evaporator is equal to one because only a phase change occurs.

As mentioned before, the volume of the generator is equal to V_α, while the evaporator, the condenser and the absorber are considered to have twice the volume of the generator:

$$V_{evap}[m^3] = 2 \cdot V_\alpha[m^3] \tag{97}$$

In a previous chapter, it was explained that the cabin temperature of the vehicle is considered to be 10 K higher than the ambient temperature. This is represented in model as T_{env}+ΔT.

Condenser:



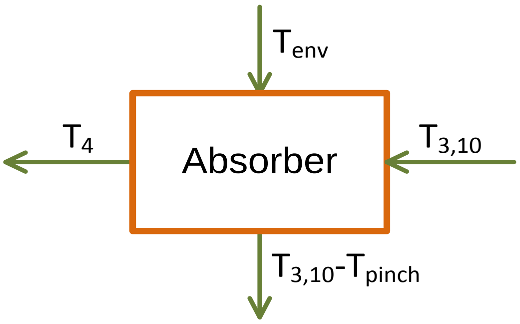
$$Q_{cond}[kW] = U_{cond}[kW/(m^2K)] \cdot A_{cond}[m^2] \cdot F_{cond} \cdot \Delta T_{LMTD_{cond}}[K] \tag{98}$$

$$\Delta T_{LMTD_{cond}}[K] = \frac{\left(T_H[K] - (T_H[K] - T_{pinch}[K])\right) - (T_{cond}[K] - T_{env}[K])}{\ln \frac{\left(T_H[K] - (T_H[K] - T_{pinch}[K])\right)}{(T_{cond}[K] - T_{env}[K])}} \tag{99}$$

$$V_{cond}[m^3] = 2 \cdot V_\alpha[m^3] \tag{100}$$

Where the pinch temperature corresponds to that between the cooling air when it goes out of the condenser and the superheated water entering it.

Absorber:



$$Q_{abs}[kW] = U_{abs}[kW/(m^2K)] \cdot A_{abs}[m^2] \cdot F_{abs} \cdot \Delta T_{LMTD_{abs}}[K] \tag{101}$$

$$\Delta T_{LMTD_{abs}}[K] = \frac{\left(T_{3,10}[K] - (T_{3,10}[K] - T_{pinch}[K])\right) - (T_4[K] - T_{env}[K])}{\ln \dfrac{\left(T_{3,10}[K] - (T_{3,10}[K] - T_{pinch}[K])\right)}{(T_4[K] - T_{env}[K])}} \tag{102}$$

$$T_{3,10}[K] = \\ = \frac{\dot{m}_3[kg/s] \cdot cp_3[kJ/(kgK)] \cdot T_3[K] + \dot{m}_{10}[kg/s] \cdot cp_{water,abs}[kJ/(kgK)] \cdot T_L[K]}{\dot{m}_3[kg/s] \cdot cp_3[kJ/(kgK)] + \dot{m}_{10}[kg/s] \cdot cp_{water,abs}[kJ/(kgK)]} \tag{103}$$

$$V_{abs}[m^3] = 2 \cdot V_\alpha[m^3] \tag{104}$$

As explained for the generator, $T_{3,\ 10}$ is the mixing temperature of the two fluids entering the absorber, this is, the weak solution and the steam coming from the evaporator.

About the pinch temperature, in this case corresponds to the temperature difference between the cooling air when it goes out of the absorber and the mixing temperature of the two fluids entering it.

7.2.2. Results

After running the EES model for the calculation of the minimum values of the overall heat transfer coefficients in the specified conditions of 49 °C ambient temperature, 20 °C desired temperature inside the cabin, and compact heat exchangers with a heat transfer area-total volume relation of 886 m^2/m^3, the results obtained show:

- The minimum overall heat transfer coefficient of the evaporator should be 28 W/m^2K, referred to the air-side surface area, which is indeed lower than those comprised between the range of typical values for water to air heat exchangers in finned tubes, which corresponds to 30 W/m^2K - 60 W/m^2K [25];
- The minimum overall heat transfer coefficient in the case of the condenser should be equal to 42 W/m^2K, referred to the air-side surface area as well, and which is also

comprised between the range of typical values for steam to air heat exchangers in finned tubes (30 W/m^{2}K - 300 W/m^{2}K) [25];

- The minimum overall heat transfer coefficient of the generator results to be equal to 110 W/m^{2}K, for which typical values are not known, but could be considered as reasonable taking as reference the typical values for steam to air;
- The minimum overall heat transfer coefficient that the absorber should have is equal to 111 W/m^{2}K and, as before, there are no typical values to take as a reference, but the heat exchange is between air and a mixture of steam and water/LiBr solution, so again it could be considered as reasonable taking the same reference data.

Even if the value obtained for the absorber could be considered, as said before, as reasonable, it seems slightly high. An explanation for this high value is the small LMTD the absorber is dealing with, consequence of the high temperature of the environment (49 °C).

If the ambient temperature is taken down to 35 °C, a more normal value, then the minimum overall heat transfer coefficient required by the absorber would be of 47 W/m^{2}K. In such case, the results obtained for the evaporator and the condenser are still comprised in their correspondent ranges of typical values.

In the case of the generator, the minimum overall heat transfer coefficient required could be brought down to a lower value by increasing the temperature of the exhaust gases coming from the fuel cell.

7.3. Size optimisation

After concluding, from the previous results, that the minimum overall heat transfer coefficients needed for the system to have a maximum volume of 77 L, are reasonable, a size optimisation could be carried out.

In order to analyse different cases, three procedures will be followed. First of all, maintaining as constants the minimum required overall heat transfer coefficients calculated above, the volume that the system would occupy in those conditions will be calculated when the pinch temperature in the evaporator is modified.

To continue, and again employing the same minimum required overall heat transfer coefficients, the volume occupied by the system will be studied depending on the maximum environmental temperature for which the system is designed.

To end the study, different minimum values for the overall heat transfer coefficients will be calculated, imposing lower volumes than 77 L and changing the extreme conditions for which the system is thought. It is important to consider that maybe the design of an air conditioning system for a Danish vehicle should be different to that for a vehicle that is going to operate in Arizona, which is the considered case for the worst ambient conditions. In all cases, a maximum heat capacity of 9.5 kW in the evaporator has been considered.

7.3.1. Studied cases

The different cases considered for which their correspondent minimum overall heat transfer coefficients have been calculated are shown in the table below. The nine different cases correspond to the combination of three ambient temperatures and three volumes established as the volume the system can occupy in each case.

Table 14. Combination of volumes and ambient temperatures considered for size optimisation.

		Volume		
		77 L	70 L	60 L
Ambient Temperature	49 °C	✓	✓	✓
	40 °C	✓	✓	✓
	35 °C	✓	✓	✓

In each case, the volume that the system would occupy maintaining U values constant and changing the environmental conditions to those not used in the calculation (for example in the first case this would be 40 °C and 35 °C) will be also studied.

7.3.2. Minimum required overall heat transfer coefficients obtained

The following table shows the results of the minimum overall heat transfer coefficients that are necessary in each case to make the system fit in the selected volume when the pinch temperature in the evaporator is 5 K.

Table 15. Minimum required overall heat transfer coefficients when w_{rich} = 0.58 kg LiBr/kg solution and T_{picnh} = 5 K.

		Volume		
		77 L	70 L	60 L
Ambient Temperature	49 °C	U_{abs} = 0.1912 kW/m^{2}K U_{cond} = 0.04304 kW/m^{2}K U_{evap} = 0.02825 kW/m^{2}K U_{gen} = 0.09459 kW/m^{2}K U_{HEx} = 0.1 kW/m^{2}K	U_{abs} = 0.2111 kW/m^{2}K U_{cond} = 0.04753 kW/m^{2}K U_{evap} = 0.0312 kW/m^{2}K U_{gen} = 0.1045 kW/m^{2}K U_{HEx} = 0.1 kW/m^{2}K	U_{abs} = 0.2481 kW/m^{2}K U_{cond} = 0.05586 kW/m^{2}K U_{evap} = 0.03667 kW/m^{2}K U_{gen} = 0.1228 kW/m^{2}K U_{HEx} = 0.1 kW/m^{2}K
	40 °C	U_{abs} = 0.07916 kW/m^{2}K U_{cond} = 0.03541 kW/m^{2}K U_{evap} = 0.03286 kW/m^{2}K U_{gen} = 0.09459 kW/m^{2}K U_{HEx} = 0.1 kW/m^{2}K	U_{abs} = 0.08742 kW/m^{2}K U_{cond} = 0.03911 kW/m^{2}K U_{evap} = 0.03629 kW/m^{2}K U_{gen} = 0.1045 kW/m^{2}K U_{HEx} = 0.1 kW/m^{2}K	U_{abs} = 0.1027 kW/m^{2}K U_{cond} = 0.04596 kW/m^{2}K U_{evap} = 0.04265 kW/m^{2}K U_{gen} = 0.1228 kW/m^{2}K U_{HEx} = 0.1 kW/m^{2}K
	35 °C	U_{abs} = 0.06311 kW/m^{2}K U_{cond} = 0.03135 kW/m^{2}K U_{evap} = 0.03631 kW/m^{2}K U_{gen} = 0.09459 kW/m^{2}K U_{HEx} = 0.1 kW/m^{2}K	U_{abs} = 0.06963 kW/m^{2}K U_{cond} = 0.03462 kW/m^{2}K U_{evap} = 0.0401 kW/m^{2}K U_{gen} = 0.1045 kW/m^{2}K U_{HEx} = 0.1 kW/m^{2}K	U_{abs} = 0.08191 kW/m^{2}K U_{cond} = 0.04069 kW/m^{2}K U_{evap} = 0.04713 kW/m^{2}K U_{gen} = 0.1228 kW/m^{2}K U_{HEx} = 0.1 kW/m^{2}K

- The U values obtained for the first combination of data, 49 °C and 77 L are in the range of typical values for the evaporator and the condenser, and a bit high for the absorber and the generator, but could be considered as possible to reach.
- About the minimum U values needed in the second combination, 40 °C and 77 L, it can be observed that, compared to the first situation, the values for the condenser and the evaporator are still in the range of typical values, for the generator it remains the same, and for the absorber is considerably reduced.

  This means that if the overall heat transfer coefficients needed before could not be reached, the system could alternatively be designed taking as reference the results obtained in this case if the vehicle is going to operate at 40 °C (the given temperature) or lower.
- In the case in which the system is designed thinking in using it with temperatures of around 35 °C or lower and a volume of 77 L, the minimum overall heat transfer coefficients are different from the ones before, what can be observed mainly in that correspondent to the absorber, which is again reduced. The other values remain almost constant and always in the range of possible values.
- Studying the case for 49 °C and a volume of 70 L, the U values obtained for the condenser and the evaporator are still in the range of typical values as happened in all the cases seen before, the value obtained for the generator is a bit higher than before, but it could again be considered as reasonable, and the value obtained for the absorber is, as in the first case, the highest one and that that will determine the feasibility or not of the system.
- For 40 °C and 70 L, the values of U obtained in this situation are within the range of typical values for the evaporator and the condenser, and remains constant in the case of the generator, compared to the previous calculation. The U value of the absorber is considerably lower than before, and could be consider like an easy value to obtain.
- Compared to the previous two systems for which different values of U have been calculated, if 35 °C and 70 L are used as the calculation parameters, this situation presents the same value for the generator, and values for the condenser and the evaporator that should be easy to obtain since are normal values for heat exchangers dealing with the fluids involved. The main difference is related to the absorber, where in this case the minimum U value required is lower than in the two previous cases, which can be considered as an advantage because it makes it easier to achieve.
- For the case with 49 °C and 60 L it can be seen that, as always before, there are no problems with the minimum values required for the condenser and the evaporator. The minimum U value required for the generator is a bit more exigent than before but it could still be considered as reasonable. The main challenge in this design is to achieve the U values needed in the absorber, since 248.1 W/m^2K seems too high.
- Referred to 40 °C and 60 L, in these conditions, and considering the range of typical values given at the beginning, all the U values obtained seem to be easily reachable.
- In the last case, where calculations are done for 35 °C and 60 L, all the U values obtained seem to be small enough to be reachable. The values for the condenser and the absorber are in the range of typical values and in the case of the absorber and the generator are reasonably small.

7.3.3. Influence of the pinch temperature in the evaporator in the total volume of the system

7.3.3.1. Case in which minimum U values are calculated for 49 °C and maximum volume of the system of 77 L

In the left column of table A5 in the appendix, the values of the obtained overall heat transfer coefficients in the mentioned conditions and when T_{pinch} in the evaporator is 5 K, are shown. Using those results and varying the pinch temperature, the COP values and the necessary volumes of the system in each case are evaluated, obtaining the following:

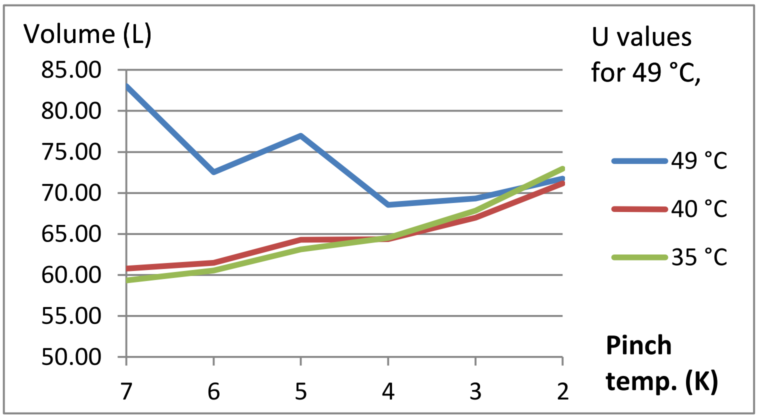


Figure 12. Volume of the system as a function of T_{pinch} in the evaporator. The minimum U values employed are those correspondent to 49 °C ambient temperature and 77 L of the system when T_{pinch} = 5 K.

- In the figure it can be observed that, for the actual case in which the ambient temperature is 49 °C, the total volume occupied by the system decreases when decreasing the pinch temperature from 7 K to 6 K, then increases when changing the same parameter from 6 K to 5 K, and after it decreases again when the pinch temperature is modified from 5 K to 4 K (the results shown in the figure can be found in table A5 in the appendix). This behaviour of the total volume as a function of the pinch temperature of the evaporator is not the expected and, consequently, it is necessary to look for an explanation that justifies it.

Procedure for the explanation:

First of all, the properties of the fluids in various points of the cycle must be checked employing the different pinch temperatures in the evaporator. When this is done, and after analysing the results, it can be concluded that the properties obtained in some points of the cycle when the pinch temperature is 5 K have big discrepancies compared to those obtained when the pinch temperature is 6 K or 4 K. At the same time, the results obtained when the pinch temperature is 6 K are quite similar to those obtained for 4 K, and also to those obtained for 3 K and 7 K.

From this analysis, it seems that there could be a problem with the EES calculations when the pinch temperature in the evaporator is exactly 5 K. To make sure about this, the same properties are calculated for 4,999 K and 5,001 K as pinch temperature. The results are, as expected, almost the same between them, and very similar to the results from the calculations with 4 K and 6 K.

The following figures show some of these discrepancies in the properties of the system:

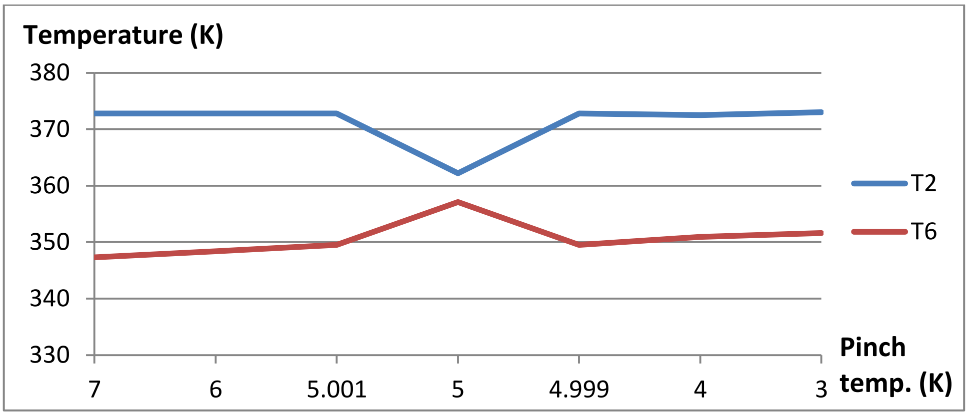


Figure 13. Variation of temperatures T_2 and T_6 as a function of the pinch temperature in the evaporator.

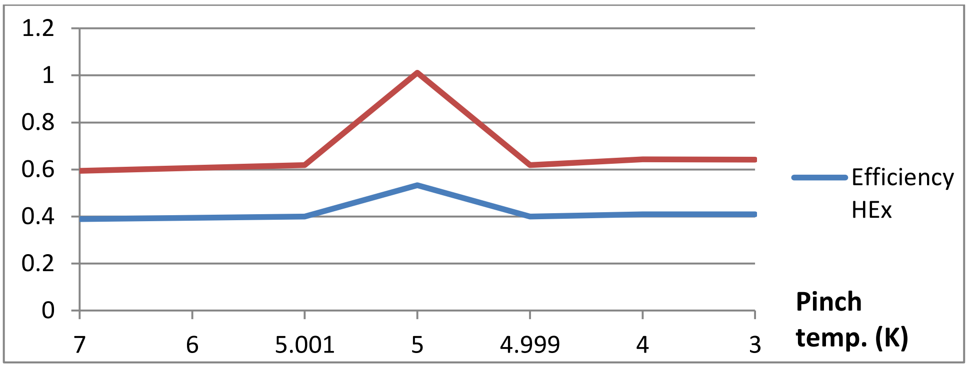


Figure 14. Variation of the efficiency and the NTU in the heat exchanger as a function of the pinch temperature in the evaporator.

Finally, and to confirm that the unexpected variances in the total volume are due to these discrepancies in some of the properties, the volume has been computed for a pinch temperature of 5,001 K, obtaining the following:

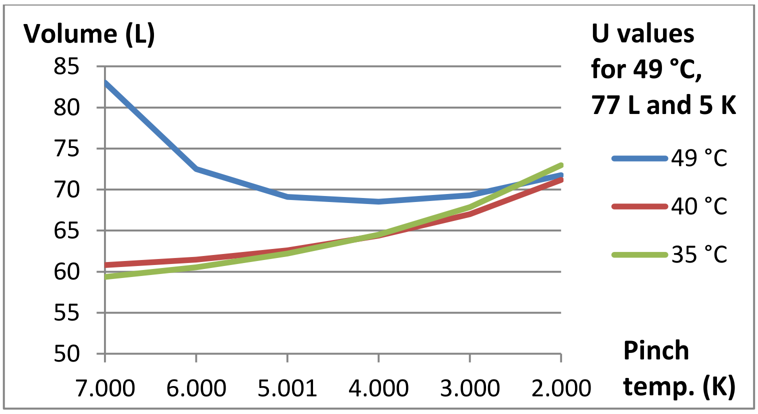


Figure 15. Volume of the system as a function of T_{pinch} in the evaporator. The minimum U values employed are those correspondent to 49 °C ambient temperature and 77 L of the system when T_{pinch} = 5 K.

- In this case, the total volume decreases in each 1 K pinch temperature change from 7 K to 4 K, and after increases in each temperature change from 4 K to 2 K. This means that when the calculations are done for 5,001 K instead of 5 K, the unexpected behaviour of the total volume as a function of the pinch temperature disappears. Table A6 in the appendix shows the actual numbers obtained. It can be finally concluded that the results that EES shows for 5 K pinch temperature in the evaporator are not correct when w_{rich} = 0.58 kg LiBr/kg solution.

New calculation of the overall heat transfer coefficients:

The minimum required overall heat transfer coefficients were computed for 5 K pinch temperature. As those minimum overall heat transfer coefficients are not correct, new overall heat transfer coefficients will be computed employing as pinch temperature 5,001 K.

Table 16. Minimum required overall heat transfer coefficients when w_{rich} = 0.58 kg LiBr/kg solution and T_{picnh} = 5.001 K.

		Volume		
		77 L	70 L	60 L
Ambient Temperature	49 °C	U_{abs} = 0.111 kW/m^{2}K U_{cond} = 0.04239 kW/m^{2}K U_{evap} = 0.02782 kW/m^{2}K U_{gen} = 0.1102 kW/m^{2}K U_{HEx} = 0.1 kW/m^{2}K	U_{abs} = 0.1224 kW/m^{2}K U_{cond} = 0.04674 kW/m^{2}K U_{evap} = 0.03068 kW/m^{2}K U_{gen} = 0.1215 kW/m^{2}K U_{HEx} = 0.1 kW/m^{2}K	U_{abs} = 0.1434 kW/m^{2}K U_{cond} = 0.05477 kW/m^{2}K U_{evap} = 0.03595 kW/m^{2}K U_{gen} = 0.1424 kW/m^{2}K U_{HEx} = 0.1 kW/m^{2}K
	40 °C	U_{abs} = 0.05587 kW/m^{2}K U_{cond} = 0.03488 kW/m^{2}K U_{evap} = 0.03237 kW/m^{2}K U_{gen} = 0.1102 kW/m^{2}K U_{HEx} = 0.1 kW/m^{2}K	U_{abs} = 0.06158 kW/m^{2}K U_{cond} = 0.03846 kW/m^{2}K U_{evap} = 0.03568 kW/m^{2}K U_{gen} = 0.1215 kW/m^{2}K U_{HEx} = 0.1 kW/m^{2}K	U_{abs} = 0.07216 kW/m^{2}K U_{cond} = 0.04506 kW/m^{2}K U_{evap} = 0.04181 kW/m^{2}K U_{gen} = 0.1424 kW/m^{2}K U_{HEx} = 0.1 kW/m^{2}K
	35 °C	U_{abs} = 0.04691 kW/m^{2}K U_{cond} = 0.03087 kW/m^{2}K U_{evap} = 0.03576 kW/m^{2}K U_{gen} = 0.1102 kW/m^{2}K U_{HEx} = 0.1 kW/m^{2}K	U_{abs} = 0.05173 kW/m^{2}K U_{cond} = 0.03404 kW/m^{2}K U_{evap} = 0.03943 kW/m^{2}K U_{gen} = 0.1215 kW/m^{2}K U_{HEx} = 0.1 kW/m^{2}K	U_{abs} = 0.06061 kW/m^{2}K U_{cond} = 0.03989 kW/m^{2}K U_{evap} = 0.0462 kW/m^{2}K U_{gen} = 0.1424 kW/m^{2}K U_{HEx} = 0.1 kW/m^{2}K

- Related to the U values obtained for the first combination of data, 49 °C and 77 L, it should be said that the minimum value needed in the evaporator is even below the range of typical values, which means that it should not be a problem to reach it; the result obtained for the condenser is in the range of normal values; and those required by the absorber and the generator are very similar and look reasonable. The minimum value required by the absorber could be reduced increasing the total volume of the system or if the ambient temperature is lower, and the value required by the generator could be reduced increasing the temperature of the exhaust gases coming into the generator.
- About the other 8 cases, the analysis of the results derives in the same conclusions as in the case where the values were computed using 5 K as pinch temperature, except in the case of the combination of 49 °C and 60 L, where in this occasion the necessary U

value in the absorber is not as high as in the previous calculation, and could still be reachable.

New calculations for the case in which minimum U values are calculated for 49 °C, volume of the system of 77 L and pinch temperature of 5.001 K in the evaporator:

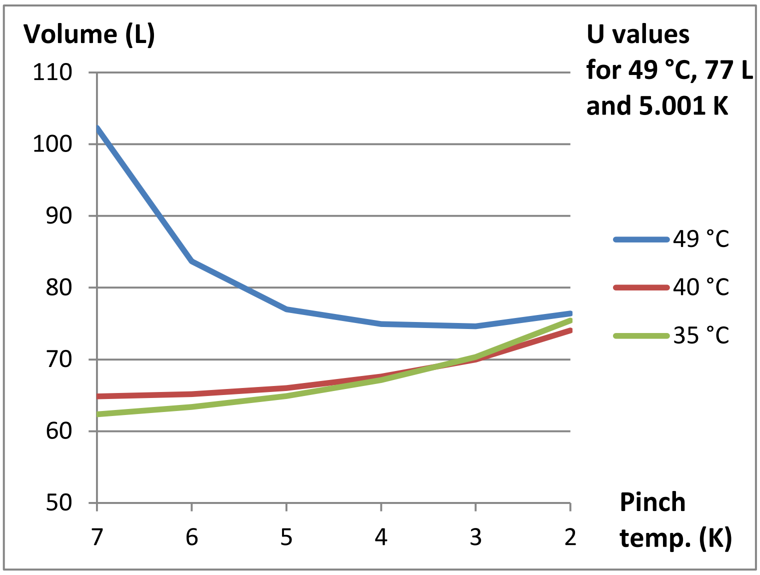


Figure 16. Volume of the system as a function of T_{pinch} in the evaporator. The minimum U values employed are those correspondent to 49 °C ambient temperature and 77 L of the system when T_{pinch} = 5.001 K.

Table 17. COP and volume of the system as a function of T_{pinch} in the evaporator. The minimum U values employed are those correspondent to 49 °C ambient temperature and 77 L of the system when T_{pinch} = 5.001 K.

T_{env} = 49 °C	T_{pinch} [K]	COP	V_{total} (49 °C) [L]	V_{total} (40 °C) [L]	V_{total} (35 °C) [L]
Q_{evap} = 9.5 kW	7	0.6820	102.3	64.86	62.37
U_{abs} = 0.0111 W/m^{2}K	6	0.6849	83.7	65.17	63.38
U_{cond} = 0.04239 kW/m^{2}K	5	0.6889	77	66.02	64.89
U_{evap} = 0.02782 kW/m^{2}K	4	0.6929	74.93	67.61	67.14
U_{gen} = 0.1102 kW/m^{2}K	3	0.6955	74.64	70	70.33
U_{HEx} = 0.1 kW/m^{2}K	2	0.6990	76.42	74.04	75.42

- From the table could be concluded that higher COPs are reached when lower pinch temperatures are employed, while the total volume occupied by the system would be reduced in more than 2 litres if a pinch temperature of 4 k or 3 K is used.
- Case in which the air conditioning system is designed for the conditions of the first case, but that is going to be used where temperatures are not higher than 40 °C: It can be observed that for a system with the same U values as before, if it is going to be used in less extreme conditions, its necessary total volume is considerably reduced, maintaining constant its COP when the pinch temperature is the same as in the previous case. In this specific situation, the minimum necessary volume is reached when the pinch temperature is increased until 7 K.

- Same system as in the two previous cases that is going to operate in an environment in which the temperatures are usually lower than 35 °C: In this occasion, the maximum total volumes obtained remain almost constant compared to when the ambient temperature is 40 °C, observing differences of less than 2.5 L for all the different pinch temperatures. As before, these volumes are considerably lower than in the first case and the lowest is obtained for a pinch temperature of 7 K.

7.3.3.2. Case in which minimum U values are calculated for 40 °C and maximum volume of the system of 77 L

In this case the desired volume for which the U values are calculated is again 77 L, but the environment is considered to be at 40 °C. In the left column of the table A7 presented in the appendix, the values of the obtained overall heat transfer coefficients in the mentioned conditions and when T_{pinch} in the evaporator is 5.001 K, are shown. Using those results and varying the pinch temperature, the COP values and the necessary volumes of the system in each case are evaluated.

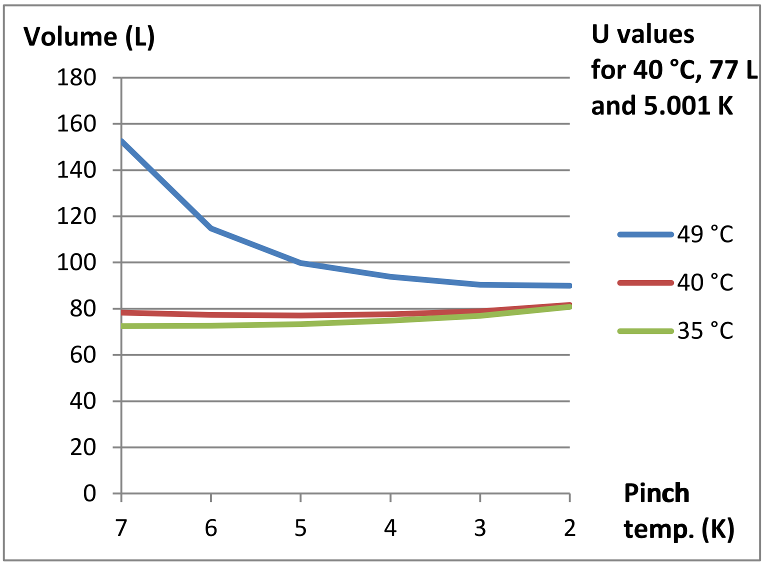


Figure 17. Volume of the system as a function of T_{pinch} in the evaporator. The minimum U values employed are those correspondent to 40 °C ambient temperature and 77 L of the system when T_{pinch} = 5.001 K.

- The performance of the system (COP) remains constant (values in table A7 in the appendix) compared to previous calculations, and, for the case for which the overall heat transfer coefficients are calculated, the pinch temperature for which the system occupies the least is 5 K.
- If a system with the same characteristics wants to be employed in countries in which 49 °C can be easily reached, then from the above table can be concluded that the overall heat transfer coefficients employed are too small for an air conditioning system that fits in a car. Nevertheless, as they are minimum values, if some higher overall heat transfer coefficients are reached, then the volume of the system will be reduced. In any case, it is not recommended to use a system designed for certain conditions in more extreme situations. In such case the volume needed would be higher.

- Same minimum overall heat transfer coefficients for a system that is going to be mainly used with temperatures below 35 °C: Opposite than before, now the conditions under which the system is going to be used are less extreme than those employed for its design, reason why the needed volumes obtained for the minimum U values considered are lower than in the case with 40 °C. The minimum value, as shown in the table, is reached for a pinch temperature of 7 K in the evaporator, and it could be even lower if it is increased to 8 K.

7.3.3.3. Case in which minimum U values are calculated for 35 °C and maximum volume of the system of 77 L

New minimum required values for the overall heat transfer coefficients are calculated maintaining the same 77 L as the volume that the system can occupy, but changing the operating conditions to 35 °C environmental temperature. These minimum U values can be seen in table 16 (new calculation of the overall heat transfer coefficients) or in table A8 (in the appendix).

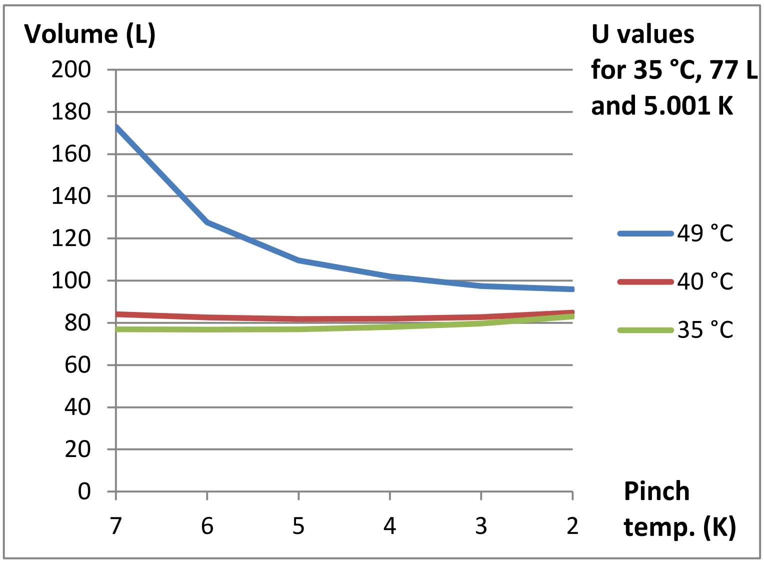


Figure 18. Volume of the system as a function of T_{pinch} in the evaporator. The minimum U values employed are those correspondent to 35 °C ambient temperature and 77 L of the system when T_{pinch} = 5.001 K.

- In the case of the design conditions the total volume of the system is minimized when the pinch temperature employed in the evaporator is 6 K. It should be also mentioned that for a pinch temperature of 5 K or 7 K, the total volume occupied is only 0.26 L higher.
- The same system as before, this is, with the same minimum overall heat transfer coefficients, but sized to be used with 49 °C ambient temperature: From the results it can be deducted that it is not recommendable to size this system to work under the specified conditions of 49 °C ambient temperature because the volumes obtained are all greater than desired. In case the system is going to deal with such temperatures, is better to design a different system in which the requirements of minimum overall heat transfer coefficients are more exigent.

- The same system as in the two previous cases, sized to be used with ambient temperatures around 40 °C: In this case it can be seen that the minimum volume occupied by the system if exactly the minimum overall heat transfer coefficients are used, is 81.84 L when the pinch temperature is 5 K. If that volume wants to be reduced, the overall heat transfer coefficients have to be increased.

7.3.3.4. Case in which minimum U values are calculated for 49 °C and maximum volume of the system of 70 L

A new set of minimum overall heat transfer coefficients needed are calculated for a system that is going to deal with 49 °C ambient temperature and whose volume is restricted to 70 L.

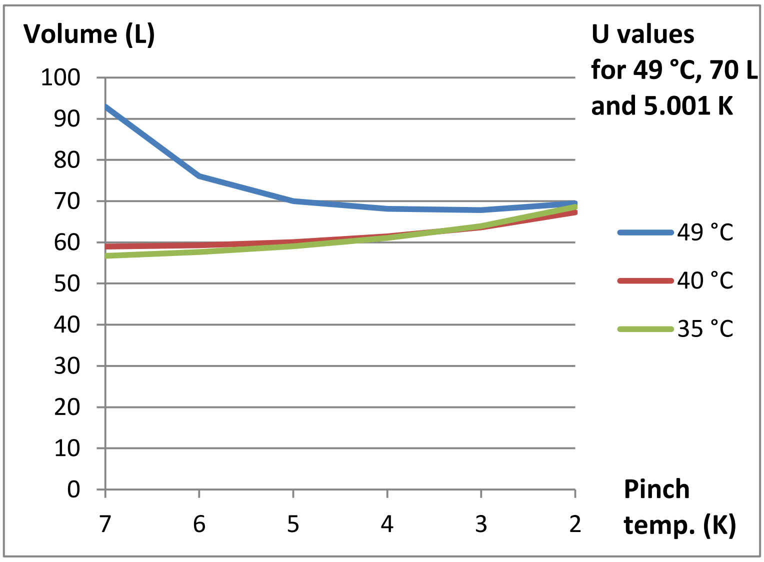


Figure 19. Volume of the system as a function of T_{pinch} in the evaporator. The minimum U values employed are those correspondent to 49 °C ambient temperature and 70 L of the system when T_{pinch} = 5.001 K.

- Studying the case taken for the calculation of the U values and assuming all of them could be reached, the pinch temperature in the evaporator that allows the system to occupy the minimum volume is T_{pinch} = 3 K, for which a system of 67.86 L would be enough.
- The same system as before that wants to be sized for ambient temperatures of about 40 °C: It can be seen that if the system wants to be sized to be used in less extreme conditions, the total volume needed by the system is lower, reaching its minimum, 58.98 L (or lower), for a pinch temperature of 7 K (or higher) in the evaporator.
- A system with the same U values as the previous two but sized to operate with 35 °C ambient temperature: Only 56.72 L are needed if a system with U values equal or higher than the specified is designed and sized to operate at 35 °C. The necessary pinch temperature in the evaporator to minimize the volume is 7 K. If higher pinch temperatures are tried, lower total volumes might be obtained.

7.3.3.5. Case in which minimum U values are calculated for 40 °C and maximum volume of the system of 70 L

New minimum required overall heat transfer coefficient values are calculated for a system designed to operate with an ambient temperature of 40 °C and that occupies 70L.

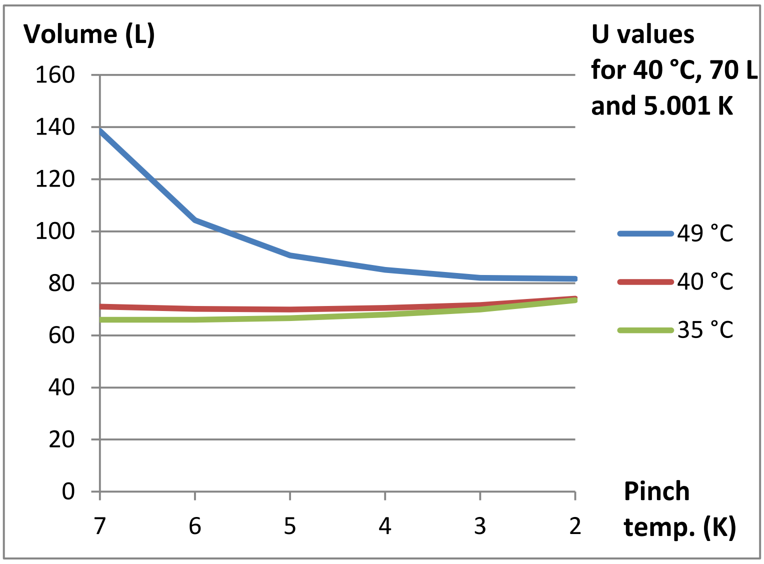


Figure 20. Volume of the system as a function of T_{pinch} in the evaporator. The minimum U values employed are those correspondent to 40 °C ambient temperature and 70 L of the system when T_{pinch} = 5.001 K.

- ○ In the design case, the total volume is imposed to be 70 L for 5 K pinch temperature in the evaporator, and is indeed the lowest volume that could be obtained under those conditions.
- ○ Same system sized to be used with 49 °C ambient temperature: Almost 82 L of volume are necessary in the best case, when T_{pinch} = 2 K, to make the system with the minimum overall heat transfer coefficients determined before meet the cooling demand if the environment is at 49 °C. Nevertheless, the volume could be reduced decreasing the pinch temperature; and the U values specified are minimum values, what implies that if higher values are achieved, the total volume needed by the system would be lower.
- ○ Same system sized to be used with 35 °C ambient temperature: When the pinch temperature in the evaporator is 7 K, a total volume of 65.97 L is needed for the system designed to work at 40 °C and with volume of 70 L to work at 35 °C.

7.3.3.6. Case in which minimum U values are calculated for 35 °C and maximum volume of the system of 70 L

New values of U are calculated for a system with a maximum volume of 70 L and initially thought to be used in places with 35 °C ambient temperature.

These values can be found in table 16 (new calculation of minimum overall heat transfer coefficients), in the column correspondent to 70 L and the row for 35 °C; or in table A11 in the appendix, where there are also shown the actual results obtained for the different volumes as a function of the pinch temperature in the evaporator.

The following graph shows the behaviour of this function when the system is sized for 49 °C, 40 °C and 35 °C ambient temperature.

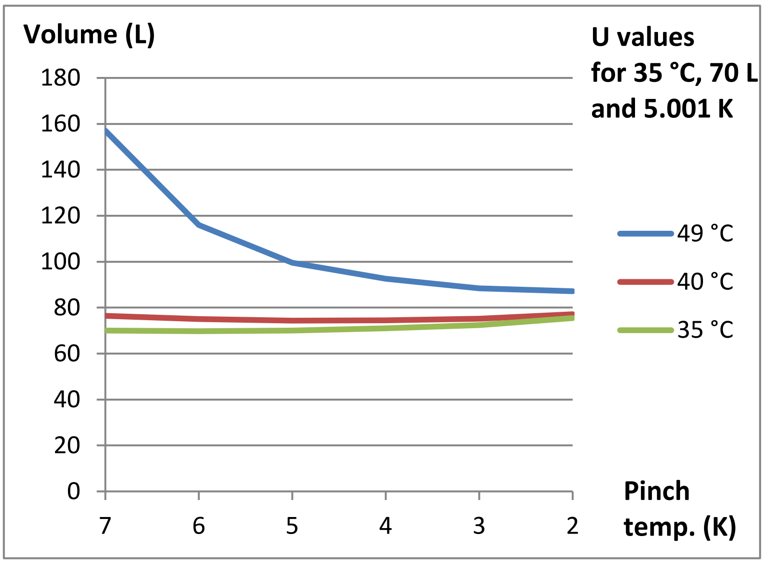


Figure 21. Volume of the system as a function of T_{pinch} in the evaporator. The minimum U values employed are those correspondent to 35 °C ambient temperature and 70 L of the system when T_{pinch} = 5.001 K.

- Under the design conditions, and related to the volume occupied by the system, if the pinch temperature in the evaporator is increased from 5 K to 6 K, then the volume needed is reduced from the initial 70 L to 69.76 L.
- Same system sized to work with 49 °C outside temperature: As seen in similar cases before, when a system is designed to be used in certain conditions and after is sized to be used in more extreme conditions, the necessary dimensions are higher than desired. In this case, the minimum sized required when the system has exactly the minimum U values, is 87.12 L if the pinch temperature in the evaporator is 2 K, and it could be lower reducing the pinch temperature to 1 K.

  In case that the U values achieved are higher than the minimum required, the volume needed will be lower. To know if the system would be feasible or not, those U values have to be known in order to calculate the actual volume.
- Same system sized to work with 40 °C outside temperature: In this case, the minimum volume, 74.39 L, is reached for a pinch temperature of 5 K, but is still higher than the desired volume that the system should occupy, 70 L.

  As explained before, if the actual values of the overall heat transfer coefficients are higher than the minimum, the volume occupied will be smaller.

7.3.3.7. Case in which minimum U values are calculated for 49 °C and maximum volume of the system of 60 L

Following the same criteria as in the two previous blocks of cases (same volume, different environmental temperature), new minimum values of U will be calculated now, using in this case 60 L as volume desired. In the first case, the design for a system which is going to deal with 49 °C outer temperature is considered.

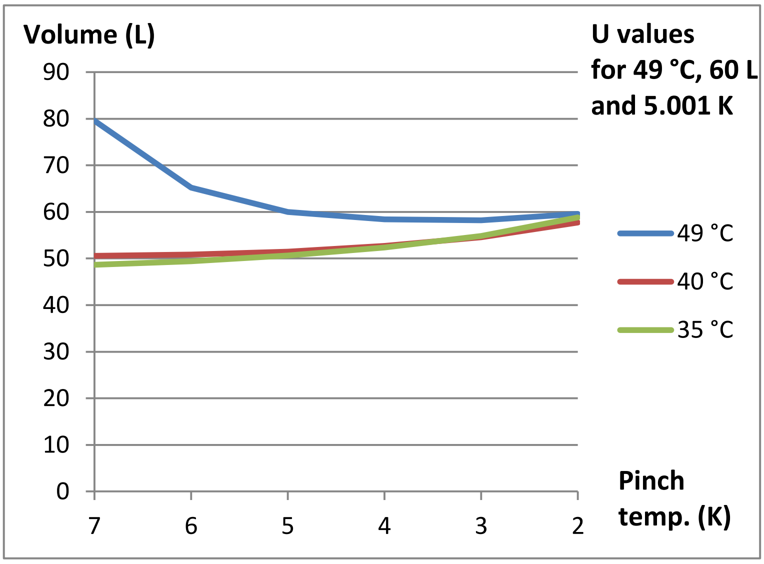


Figure 22. Volume of the system as a function of T_{pinch} in the evaporator. The minimum U values employed are those correspondent to 49 °C ambient temperature and 60 L of the system when T_{pinch} = 5.001 K.

- ○ In case all the values can be achieved, and the system employed, a pinch temperature in the evaporator of 3 K would be recommended in order to minimize its volume.
- ○ If the same system wants to be sized for an ambient temperature of 40 °C, then, if a system with the minimum U values required is feasible, the total volume occupied by it, in case it is sized to be used somewhere with 40 °C as maximum temperature, is quite small. For a pinch temperature of 7 K, the maximum volume of the system is 50.59 L. Moreover, the minimum volume has not been reached with this analysis, and could be even lower if the pinch temperature is increased over 7 K.
- ○ Sizing of the same system when used for a maximum temperature of 35 °C: The same conclusions as before could be taken, with the only difference that in this case the maximum volume needed by the system when the pinch temperature is 7 K is 48.66 L.

7.3.3.8. Case in which minimum U values are calculated for 40 °C and maximum volume of the system of 60 L

Taking 60 L as maximum volume of the system when it is thought to work with temperatures below 40 °C, new values of U are calculated, obtaining the results shown in table A13 placed in the appendix.

The figure below is a graphical representation of those results, in which it can be seen how the pinch temperature employed in the evaporator affects the total volume of the system when it is sized to work under different outside conditions.

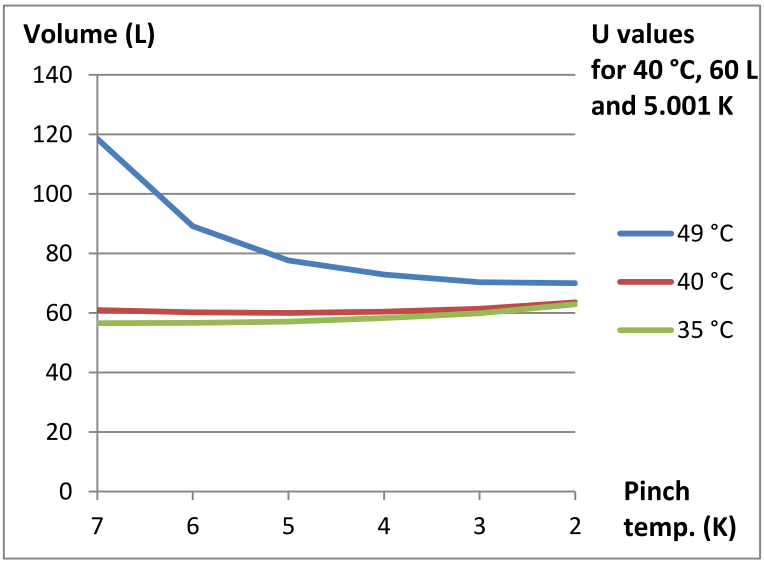


Figure 23. Volume of the system as a function of T_{pinch} *in the evaporator. The minimum U values employed are those correspondent to 40 °C ambient temperature and 60 L of the system when* $T_{pinch} = 5.001$ *K.*

- Referred to the design conditions and about the relation between pinch temperature and volume, the minimum volume, correspondent to 60 L, is obtained when the pinch temperature in the evaporator is actually 5 K.
- If the same system wants to be sized to be used with maximum environmental temperature of 49 °C: In this case the conditions are more extreme and consequently the needed size is greater than before, reaching its minimum, 70.02 L, when the pinch temperature is 2 K. As commented before, these values correspond to a system with U values equal to the minimum required, but the size would be smaller if higher values of U are reached.
- Same system sized for temperatures below 35 °C: If the system designed with constraints 40 °C maximum temperature and 60 L as volume is sized to be used with 35 °C as maximum environmental temperature, the volume needed with a pinch temperature of 7 K in the evaporator is 56.55 L, and there is the possibility that it could be reduced more if the pinch temperature is increased.

7.3.3.9. Case in which minimum U values are calculated for 35 °C and maximum volume of the system of 60 L

The last case for which minimum overall heat transfer coefficients will be calculated corresponds to that with 60 L as the volume occupied and 35 °C maximum ambient temperature.

As in all the previous cases, the table with the results is placed in the appendix (table A14), and the graph representing the variation of the volume respect to the pinch temperature of the evaporator is shown after these lines.

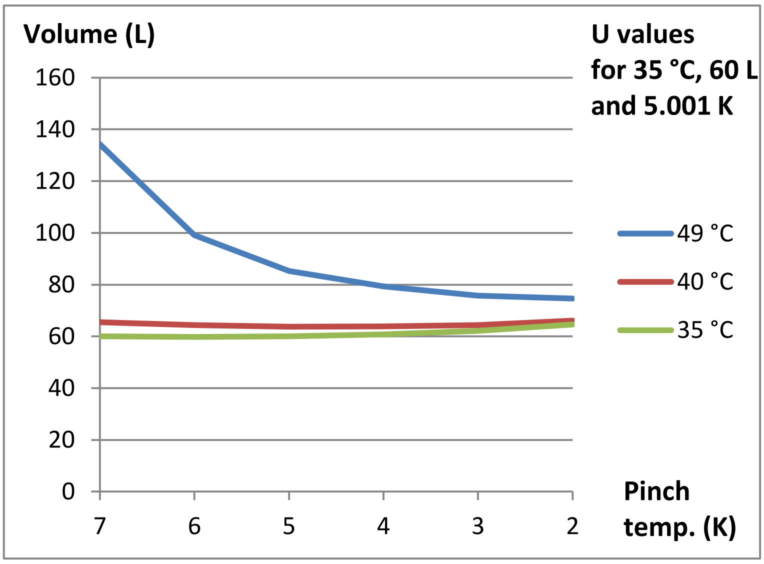


Figure 24. Volume of the system as a function of T_{pinch} *in the evaporator. The minimum U values employed are those correspondent to 35 °C ambient temperature and 60 L of the system when* T_{pinch} *= 5.001 K.*

- In the case employed to calculate the minimum required overall heat transfer coefficients, when the pinch temperature in the evaporator is 6 K, the volume occupied by the system is 59.79 L.
- Same system sized for 49 °C ambient temperature: Some higher U values should be reached in order to make the system feasible to be used with 49 °C ambient temperature. With the minimum overall heat transfer coefficients, the needed volume is at least 74.62 L, which corresponds to 2 K pinch temperature.
- Sizing of the same system to be employed when the environment can reach not more than 40 °C: The system occupies a minimum of 63.75 L when the pinch temperature is 5 K and the overall heat transfer coefficients are exactly the minimum required. In order to reduce the volume of the system, the U values of the different components should be increased.

7.3.4. Summary of results and conclusions

After evaluating the nine cases with different minimum required overall heat transfer coefficients, it is observed that the graphs representing the results of all of them have the same pattern.

In the three cases in which the minimum required U values are computed for 49 °C ambient temperature, the pinch temperature that allows to have the smallest system is 3 K when the system is actually sized to be used in an environment with 49 °C, and 7 K or higher when it is sized to work in less extreme conditions.

If the system is going to be used and thus sized to work with temperatures around 49 °C, then is recommended to use a system design for those conditions, with minimum U values high enough for the system to have a reasonable size.

In the three cases in which these U values were computed for these extreme conditions, lowering the pinch temperature in the evaporator from 5 K to 3 K reduces the total volume of the system between 1.5 L and 2.5 L depending on the case. In the other cases in which the minimum U values were calculated for less extreme conditions, the volume of the system needed if it is wanted to work with 49 °C is always considerably higher than that desired.

If the ambient temperature is 35 °C or 40 °C, the pattern followed by the volume as a function of the pinch temperature in the evaporator is very similar in both cases. The volume occupied by the total system is also quite similar for the two ambient temperatures regardless of the minimum U values employed.

Nevertheless, if the minimum required overall heat transfer coefficients are calculated for a system thought to work at 35 °C or lower, and after is sized to work at temperatures of around 40 °C, the volume required by the system is always a bit higher than desired.

When the system is designed to work with 40 °C ambient temperature and thus the minimum required U values and the size are calculated for those conditions, the pinch temperature that allows to have the smallest system is 5 K. It is not recommended to use that same system to work with 49 °C but, in such case, the most appropriate pinch temperature is 2 K or lower. If the outside temperature is not higher than 35 °C, then the size needed by this same system would be lower, reaching the lowest volume when the pinch temperature in the evaporator is 7 K or even higher.

A really similar analysis could be done for the cases in which the system is designed to work with 35 °C ambient temperature, reaching the smallest values of the total volume for 2 K in case of working at 49 °C, 5 K when working at 40 °C and 6 K when working at 35 °C.

To sum up, it is worth to say that the best pinch temperature for the system to be as small as possible depends on the conditions under which the system is designed to be used, not existing an absolute value common to all cases.

To know which the system that needs the lowest volume to provide the required cooling capacity of 9.5 kW is, it is necessary to have information about the overall heat transfer coefficients that can be achieved in each component and the environmental conditions. It is difficult to determine the most appropriate system for a general case, but it is always positive to have high values of U so the total volume of the system can be lowered.

7.3.5. Singularities in EES

As explained in the analysis of the variation of the total volume as a function of the pinch temperature in the evaporator, there is a singularity in EES when the properties of the system are computed for a pinch temperature of 5 K if the mass fraction of the rich solution is equal to 0.58 kg LiBr/kg solution.

The minimum overall heat transfer coefficients used for the volume calculations, and these volumes have thus been calculated employing 5.001 K as pinch temperature instead of 5 K.

What could be interesting to study now is if the singularity occurs for 5 K regardless of the mass fraction of the rich solution, or if it is the combination of the two what makes the singularity to appear.

In order to check this, the minimum required overall heat transfer coefficients needed by the system under the same conditions as before have been calculated employing 5 K as pinch temperature in the evaporator, and changing the mass fraction of the rich solution from w_{rich} = = 0.58 kg LiBr/kg solution to w_{rich} = 0.57999 kg LiBr/kg solution.

The obtained results are shown in the following table, and will be compared to those in the previous case.

Table 18. Minimum required overall heat transfer coefficients when w_{rich} = 0.57999 kg LiBr/kg solution and T_{picnh} = 5 K.

		Volume		
		77 L	70 L	60 L
Ambient Temperature	49 °C	U_{abs} = 0.1114 kW/m^{2}K U_{cond} = 0.04239 kW/m^{2}K U_{evap} = 0.02783 kW/m^{2}K U_{gen} = 0.11 kW/m^{2}K U_{HEx} = 0.1 kW/m^{2}K	U_{abs} = 0.1229 kW/m^{2}K U_{cond} = 0.04674 kW/m^{2}K U_{evap} = 0.03068 kW/m^{2}K U_{gen} = 0.1213 kW/m^{2}K U_{HEx} = 0.1 kW/m^{2}K	U_{abs} = 0.144 kW/m^{2}K U_{cond} = 0.05477 kW/m^{2}K U_{evap} = 0.03596 kW/m^{2}K U_{gen} = 0.1422 kW/m^{2}K U_{HEx} = 0.1 kW/m^{2}K
	40 °C	U_{abs} = 0.056 kW/m^{2}K U_{cond} = 0.03488 kW/m^{2}K U_{evap} = 0.03577 kW/m^{2}K U_{gen} = 0.11 kW/m^{2}K U_{HEx} = 0.1 kW/m^{2}K	U_{abs} = 0.06174 kW/m^{2}K U_{cond} = 0.03846 kW/m^{2}K U_{evap} = 0.03569 kW/m^{2}K U_{gen} = 0.1213 kW/m^{2}K U_{HEx} = 0.1 kW/m^{2}K	U_{abs} = 0.07235 kW/m^{2}K U_{cond} = 0.04507 kW/m^{2}K U_{evap} = 0.04182 kW/m^{2}K U_{gen} = 0.1422 kW/m^{2}K U_{HEx} = 0.1 kW/m^{2}K
	35 °C	U_{abs} = 0.04702 kW/m^{2}K U_{cond} = 0.03088 kW/m^{2}K U_{evap} = 0.03577 kW/m^{2}K U_{gen} = 0.11 kW/m^{2}K U_{HEx} = 0.1 kW/m^{2}K	U_{abs} = 0.05185 kW/m^{2}K U_{cond} = 0.03405 kW/m^{2}K U_{evap} = 0.03944 kW/m^{2}K U_{gen} = 0.1213 kW/m^{2}K U_{HEx} = 0.1 kW/m^{2}K	U_{abs} = 0.06076 kW/m^{2}K U_{cond} = 0.0399 kW/m^{2}K U_{evap} = 0.04621 kW/m^{2}K U_{gen} = 0.1422 kW/m^{2}K U_{HEx} = 0.1 kW/m^{2}K

The comparison between these minimum required U values and those obtained in the previous case, in which w_{rich} = 0.58 kg LiBr/kg solution and T_{pinch} = 5.001 K, shows that the differences between two sets of values computed for the same volume and ambient temperature conditions are negligible.

The conclusion that can be taken is that there must be a singularity in the EES routines for the thermodynamic properties of the water/LiBr solution that in the model of the system corresponds to the combination of having a rich solution with a composition of LiBr of 58% in mass terms, and a pinch temperature in the evaporator of exactly 5 K.

If any of the two parameters involved is modified and given a very close value, the singularity disappears and the results obtained can be considered as the actual results for the combination in which the singularity occurs.

7.4. Size optimisation changing assumptions

A new size optimisation can be carried out if the relation between the volumes of the different compact heat exchangers present in the system is modified.

In this case, and taking into consideration the limitations of the overall heat transfer coefficients in the different components, this is, in the evaporator the minimum required should be below 60 W/m^{2}K and in the condenser below 300 W/m^{2}K, a relation between the U values of the different components has been imposed, computing in each case the best relation between the areas, and thus volumes, that they should have.

It is known that:

$$Q_i[kW] = U_i[kW/(m^2K)] \cdot A_i[m^2] \cdot F_i \cdot \Delta T_{LMTD_i}[K] \tag{105}$$

Imposing $U_{cond} = U_{gen}$, then the expression that allows to compute the relation between the areas of the two components, condenser and generator, is as follows:

$$\frac{Q_{gen}[kW]}{A_{gen}[m^2] \cdot F_{gen} \cdot \Delta T_{LMTD_{gen}}[K]} = \frac{Q_{cond}[kW]}{A_{cond}[m^2] \cdot F_{cond} \cdot \Delta T_{LMTD_{cond}}[K]} \tag{106}$$

Following the same steps for the absorber, $U_{abs} = U_{gen}$ and:

$$\frac{Q_{gen}[kW]}{A_{gen}[m^2] \cdot F_{gen} \cdot \Delta T_{LMTD_{gen}}[K]} = \frac{Q_{abs}[kW]}{A_{abs}[m^2] \cdot F_{abs} \cdot \Delta T_{LMTD_{abs}}[K]} \tag{107}$$

In the case of the evaporator, the relation between its U value and that of the generator depends on the case that is being studied. This is because of the limitations in the overall heat transfer coefficient of the evaporator explained before, and the variation of the values of the correction factor and the logarithmic mean temperature difference when changing the ambient temperature.

In the cases in which the overall heat transfer coefficients want to be calculated to deal with environmental temperatures of 35 °C or 40 °C, the relation between the U value of the evaporator and that of the generator is established to be $U_{evap} = U_{gen}/2$, thus resulting in:

$$\frac{Q_{gen}[kW]}{A_{gen}[m^2] \cdot F_{gen} \cdot \Delta T_{LMTD_{gen}}[K]} = \frac{2 \cdot Q_{evap}[kW]}{A_{evap}[m^2] \cdot \Delta T_{LMTD_{evap}}[K]} \tag{108}$$

If the ambient temperature for which the U values are going to be calculated is 49 °C, two cases have been considered for the relation between the U of the evaporator and the generator. In the first one, it is imposed to be $U_{evap} = U_{gen}/2$ and consequently the expression that relates the areas of the two components is the same as before. In the second one, the relation considered between the U values is $U_{evap} = U_{gen}/3$, and so the relation between the areas is given by:

$$\frac{Q_{gen}[kW]}{A_{gen}[m^2] \cdot F_{gen} \cdot \Delta T_{LMTD_{gen}}[K]} = \frac{3 \cdot Q_{evap}[kW]}{A_{evap}[m^2] \cdot \Delta T_{LMTD_{evap}}[K]} \tag{109}$$

The reason why these two relations have been considered is that in case the U value of the evaporator required in the first case cannot be achieved, in the second one this value is reduced, so it could be easily reachable.

7.4.1. Studied cases and minimum U values obtained

Twelve different cases are studied, each of them with their correspondent calculation of U values and the influence of the pinch temperature of the evaporator in the required volume.

These twelve cases, with their correspondent calculated minimum U values are:

Table 19. Minimum required overall heat transfer coefficients when $w_{rich} = 0.58$ *kg LiBr/kg solution and* $T_{picnh} = 5.001$ *K, for different volume relations.*

		Volume		
		77 L	70 L	60 L
Ambient Temperature	49 °C (eq. 108)	$U_{abs} = 0.07561$ kW/m^{2}K	$U_{abs} = 0.08337$ kW/m^{2}K	$U_{abs} = 0.09769$ kW/m^{2}K
		$U_{cond} = 0.07472$ kW/m^{2}K	$U_{cond} = 0.08239$ kW/m^{2}K	$U_{cond} = 0.09654$ kW/m^{2}K
		$U_{evap} = 0.0378$ kW/m^{2}K	$U_{evap} = 0.04168$ kW/m^{2}K	$U_{evap} = 0.04884$ kW/m^{2}K
		$U_{gen} = 0.07562$ kW/m^{2}K	$U_{gen} = 0.08338$ kW/m^{2}K	$U_{gen} = 0.0977$ kW/m^{2}K
		$U_{HEx} = 0.1$ kW/m^{2}K	$U_{HEx} = 0.1$ kW/m^{2}K	$U_{HEx} = 0.1$ kW/m^{2}K
	49°C (eq. 109)	$U_{abs} = 0.08356$ kW/m^{2}K	$U_{abs} = 0.09213$ kW/m^{2}K	$U_{abs} = 0.108$ kW/m^{2}K
		$U_{cond} = 0.08258$ kW/m^{2}K	$U_{cond} = 0.09105$ kW/m^{2}K	$U_{cond} = 0.1067$ kW/m^{2}K
		$U_{evap} = 0.02785$ kW/m^{2}K	$U_{evap} = 0.03071$ kW/m^{2}K	$U_{evap} = 0.03599$ kW/m^{2}K
		$U_{gen} = 0.08357$ kW/m^{2}K	$U_{gen} = 0.09214$ kW/m^{2}K	$U_{gen} = 0.108$ kW/m^{2}K
		$U_{HEx} = 0.1$ kW/m^{2}K	$U_{HEx} = 0.1$ kW/m^{2}K	$U_{HEx} = 0.1$ kW/m^{2}K
	40 °C	$U_{abs} = 0.06028$ kW/m^{2}K	$U_{abs} = 0.06646$ kW/m^{2}K	$U_{abs} = 0.07788$ kW/m^{2}K
		$U_{cond} = 0.05954$ kW/m^{2}K	$U_{cond} = 0.06565$ kW/m^{2}K	$U_{cond} = 0.07693$ kW/m^{2}K
		$U_{evap} = 0.03015$ kW/m^{2}K	$U_{evap} = 0.03324$ kW/m^{2}K	$U_{evap} = 0.03895$ kW/m^{2}K
		$U_{gen} = 0.06028$ kW/m^{2}K	$U_{gen} = 0.06646$ kW/m^{2}K	$U_{gen} = 0.07788$ kW/m^{2}K
		$U_{HEx} = 0.1$ kW/m^{2}K	$U_{HEx} = 0.1$ kW/m^{2}K	$U_{HEx} = 0.1$ kW/m^{2}K
	35 °C	$U_{abs} = 0.05852$ kW/m^{2}K	$U_{abs} = 0.06452$ kW/m^{2}K	$U_{abs} = 0.07561$ kW/m^{2}K
		$U_{cond} = 0.05778$ kW/m^{2}K	$U_{cond} = 0.06371$ kW/m^{2}K	$U_{cond} = 0.07466$ kW/m^{2}K
		$U_{evap} = 0.02926$ kW/m^{2}K	$U_{evap} = 0.03226$ kW/m^{2}K	$U_{evap} = 0.0378$ kW/m^{2}K
		$U_{gen} = 0.05852$ kW/m^{2}K	$U_{gen} = 0.06452$ kW/m^{2}K	$U_{gen} = 0.07561$ kW/m^{2}K
		$U_{HEx} = 0.1$ kW/m^{2}K	$U_{HEx} = 0.1$ kW/m^{2}K	$U_{HEx} = 0.1$ kW/m^{2}K

It can be observed that, as expected, the minimum U values required by the different components of the system increase when the volume of the entire system, and thus the volume of each component, decreases.

Referred to the environmental temperature for which the system is designed, it is clear that it has an important effect on the minimum overall heat transfer coefficients required, making them decrease when the system works in less extreme conditions.

It is difficult to predict which of the two effects, temperature or volume, has a greater influence on the minimum U values required. Nevertheless, there are two combinations that give the most and the least exigent requirements for the minimum U values, and these correspond to the combination of 60 L and 40 °C, and 77 L and 35 °C respectively.

In the most exigent case for the evaporator, 48.84 W/m^{2}K have to be reached, which is a value in the range of typical values of this component and consequently could be considered as reachable. In the case of the condenser, absorber and generator, the minimum values of U needed in the most exigent case are around 108 W/m^{2}K, which can be treated as reasonable, knowing that the range of typical values in the condenser goes from 30 W/m^{2}K to 300 W/m^{2}K.

In the least exigent case, the minimum U values required in all the components seem small enough for the designer not to have any problem to reach them.

7.4.2. Influence of the pinch temperature in the evaporator in the total volume of the system

In the table below it is shown how the necessary volume of the system changes when changing the pinch temperature used in the evaporator.

Table 20. Volume occupied by the system as a function of T_{picnh} in the evaporator when the U values are those specified in table 19.

		Volume					
		77 L		70 L		60 L	
		T_{pinch} (K)	Volume (L)	T_{pinch} (K)	Volume (L)	T_{pinch} (K)	Volume (L)
Ambient Temperature	49 °C (eq. 108)	7	116.1	7	105.4	7	90.21
		6	87.99	6	79.96	6	68.5
		5	77	5	70	5	60
		4	72.51	4	65.93	4	56.54
		3	70.18	3	63.82	3	54.74
		2	70.05	2	63.7	2	54.64
	49 °C (eq. 109)	7	111.5	7	101.2	7	86.63
		6	86.41	6	78.54	6	67.26
		5	77	5	70	5	60
		4	73.6	4	66.92	4	57.37
		3	72.37	3	65.8	3	56.42
		2	73.5	2	66.84	2	57.3
	40 °C	7	78.18	7	71.07	7	60.9
		6	77.25	6	70.23	6	60.19
		5	77	5	70	5	60
		4	77.58	4	70.54	4	60.47
		3	78.96	3	71.79	3	61.54
		2	81.97	2	74.53	2	63.87
	35 °C	7	76.2	7	69.27	7	59.37
		6	76.31	6	69.38	6	59.46
		5	77	5	70	5	60
		4	78.45	4	71.33	4	61.14
		3	80.86	3	73.51	3	63
		2	85.11	2	77.37	2	66.3

In each combination of volume and ambient temperature a small table is shown, which represents the variation of the total volume as a function of the pinch temperature in the evaporator, when the minimum U values required are those correspondent to the combination of volume and temperature in which the table is placed.

The distribution of the table is the same as in the table showing the U values required in each case, so that it is easy to compare both tables and extract some conclusions.

In this size optimisation where the relation between the volumes is different from the first case studied, the variation of total volume of the system when changing the pinch temperature is only studied for the case for which the minimum U values are calculated, not showing the behaviour of the volume variation if the system wants to be employed and thus sized to work under different conditions.

It is observed that when the ambient temperature is high, a small pinch temperature reduces the necessary volume of the system, while when the ambient temperature is low, it is better to increase the pinch temperature in order to obtain the lowest volumes.

Comparing this table with the one presented before, in which the minimum values of the overall heat transfer coefficients of the different components could be seen, it can be concluded that when the compact heat exchangers have higher U values, the total volume occupied by the system is reduced, obtaining the minimum volumes for the most exigent combination before, 49 °C and 60 L, in any of its two variations.

In that case, if the U values for which the system has been designed are maintained as constant and the pinch temperature in the evaporator is brought down to 2 K in one case and to 3 K in the other, the volume occupied by the system would be 54.64 L and 56.42 L respectively, which are a quite lower than the 60 L occupied if 5 K are employed as pinch temperature.

Employing less exigent requirements for the U values, like those obtained with the combinations 49 °C and 70 L, and 49 °C and 77 L, the necessary volumes would be higher.

In case of having exactly the minimum U values required for the design combination 49 °C and 70 L, and for the relation between the areas of the evaporator and the generator shown in equation (108), the volume of the system would be 63.7 L for a pinch temperature of 2 K. In the same situation but for the relation between areas shown in equation (109), the volume of the system corresponds to 65.8 L, in this case when the pinch temperature is 3 K.

The analogue combinations in which the minimum U values were computed for the system to occupy 77 L working with 49 °C ambient temperature, result in 70.05 L for a pinch temperature of 2 K in the first case, and 72.37 L when T_{pinch} = 3 K in the second one.

In case the U values are higher than the minimum required, these necessary volumes would decrease.

These results are useful in order to decide the design of the system, because it allows to know which is the most appropriate pinch temperature that must be used depending on the ambient conditions for which the system is thought and the overall heat transfer coefficients the different components are able to reach.

It is difficult to say which system is the best without knowing first the conditions in which it is going to be used and the limitations of the overall heat transfer coefficients in each case.

7.5. Analysis of the size optimisation

In this section, the results from the size optimisation are studied, explaining the reason why the total volume of the system behaves in the way it does when the pinch temperature in the evaporator is modified.

To do so, the variation of the volume of the different components of the system as a consequence of reducing the pinch temperature in the evaporator is analysed.

Evaporator:

First of all, it will be studied how the different parameters in the expression for the cooling capacity of the evaporator are affected. This expression is:

$$Q_{evap} = \dot{m}_{10} \cdot (h_{10} - h_8)$$

If the pinch temperature in the evaporator is reduced, this means that the temperature at which the evaporation takes place is increased. If that temperature increases, the enthalpy of the saturated vapours also increases, and thus the enthalpy difference between points 10 and 8 is higher than before.

The cooling capacity in the evaporator is an input of the system, what implies that is considered as constant and equal to 9.5 kW.

Finally, and to keep Q_{evap} as constant, if the enthalpy difference increases, the mass flow rate in point 10 has to decrease, making also decrease the mass flow rate in all the different points of the system.

After concluding what is happening with the mass flow rates, is the turn to know how the size of the evaporator will be affected by the pinch temperature change. In this case, the equation that allows to evaluate that is the following:

$$Q_{evap} = U_{evap} \cdot A_{evap} \cdot \Delta T_{LMTD_{evap}}$$

Reducing the pinch temperature leads to a lower logarithmic mean temperature difference. As mentioned before, Q_{evap} stays constant, and the same occurs to U_{evap}, which is maintained as constant for each case of the size optimisation. In such case, the area, and thus the volume, of the evaporator has to increase in order to counteract the effect of the logarithmic mean temperature difference decrease.

To conclude, a reduction in the pinch temperature of the evaporator results in the necessity of a higher volume of the evaporator.

Condenser:

The analysis in the condenser is started from the same step as before, studying how the expression of its heat capacity is affected.

$$Q_{cond} = \dot{m}_7 \cdot (h_7 - h_8)$$

In this case, the condensation temperature and the temperature of the water vapours leaving the generator are fixed, and as a consequence, enthalpies in points 7 and 8 remain constant regardless of the pinch temperature.

Nevertheless, the mass flow rate, as seen before, is reduced if the pinch temperature in the evaporator is reduced, and this results in a reduction of the heat capacity of the condenser.

To evaluate what happens with the size of the condenser, the equation below must be analysed.

$$Q_{cond} = U_{cond} \cdot A_{cond} \cdot F_{cond} \cdot \Delta T_{LMTD_{cond}}$$

As said for the first energy balance, the temperatures before and after the condenser remain the same, and so the logarithmic mean temperature difference does not change its value. The same occurs to the correction factor F, which is a function only of the temperatures.

If U is also fixed for each case in which the pinch temperature is modified, and the heat capacity in the condenser is reduced when the pinch temperature in the evaporator is reduced, the area of the condenser has to decrease for the equation to be fulfilled.

Finally, the conclusion is that when the studied pinch temperature is reduced, the volume needed in the condenser is lower.

Generator:

The energy balance of the generator states:

$$Q_{gen} = \dot{m}_7 \cdot h_7 + \dot{m}_1 \cdot h_1 - \dot{m}_6 \cdot h_6$$

The mass flow rates reduce their values when the pinch temperature in the evaporator is reduced, as mentioned in the analysis of this component.

About the temperatures, temperature in point 7 is fixed, and temperature in point 1 is not affected by changes in the evaporating temperature. This way, the first two terms in the right side of the equation are reduced when the pinch temperature does so.

In that same situation, the temperature of the rich solution entering the generator, T_6, increases, also increasing its enthalpy. In comparison, the enthalpy increase is higher than the mass flow rate decrease, and consequently the term $\dot{m}_6 \cdot h_6$ increases.

Taking into consideration that this last term has a minus sign before, all the three terms in the right side of the equation make the capacity of the generator decrease.

To know which the necessary area of the generator is in this case, the variation of the different parameters in the expression below is studied.

$$Q_{gen} = U_{gen} \cdot A_{gen} \cdot F_{gen} \cdot \Delta T_{LMTD_{gen}}$$

The logarithmic mean temperature difference is a function of the temperatures in points 7 and 1 and the inlet temperature of the exhaust gases from the fuel cell, which remain all constant, of temperature in point 6 and of the outlet temperature of the exhaust gases.

This last temperature is calculated as a function of T_6, being $T_{exhaust,out} = T_6 + T_{pinch,gen}$, where the pinch temperature in the generator is fixed and remains constant. In the equation that determines the LMTD of the generator, the variation of T_6 is compensated by the variation of the outlet temperature of the exhaust gases, maintaining the LMTD as constant regardless of the evaporation temperature.

About the correction factor and the overall heat transfer coefficient, the same that occurs for the logarithmic mean temperature difference occurs for the correction factor, which also remains constant; and, as in all cases, the U value is fixed.

Finally, if as said before, the capacity in the generator is reduced, the area of this component is also reduced, consequently reducing its volume.

To conclude the analysis of the generator, its volume decreases when the pinch temperature in the evaporator decreases.

Absorber:

The energy balance in the absorber is expressed by:

$$Q_{abs} = \dot{m}_{10} \cdot h_{10} + \dot{m}_3 \cdot h_3 - \dot{m}_4 \cdot h_4$$

Analysing what happens with each parameter, the three mass flow rates decrease when the pinch temperature in the evaporator decreases, as seen in the analysis of the evaporator.

Also in that analysis it was seen that the enthalpy in point 10 increases when that situation takes place.

The temperature of the weak solution entering the absorber, T_3, remains almost constant, without been affected by changes in the evaporating temperature. In contrast, and as a result of the increase in T_{10} (temperature in the evaporator), the temperature of the rich solution leaving the absorber increases in such case.

The effect that the behaviour of these parameters has in the heat capacity of the generator is difficult to predict. Nevertheless, it can be said that in the first term in the right hand side of the energy balance, the decrease of $\dot{m}_{10}$ is compensated by the increase of h_{10}, remaining that term as constant. In the second term, the changes are produced by changes in the mass flow rate, and consequently that term decreases. Finally, in the third term, the increase of h_4 is more important than the reduction of mass flow rate, what makes that term to increase. The decrease of the second term, together with the increase of the third one, which is preceded by a negative sign, make the heat capacity of the absorber to be reduced when the pinch temperature in the evaporator does so.

To know how the volume of the absorber reacts, the following formula will be studied:

$$Q_{abs} = U_{abs} \cdot A_{abs} \cdot F_{abs} \cdot \Delta T_{LMTD_{abs}}$$

As a consequence of the increase in the evaporating temperature and that the temperature of the weak solution entering the absorber remains constant, the increase in the mixing temperature is quite low. In the other hand, the temperature of the rich solution going out of the absorber suffers a higher increase. The combination of these two effects, make the LMTD in the absorber to be increased when the pinch temperature in the evaporator is decreased.

Changes in the temperatures involved in the absorber make its correction factor changes. The result of the behaviour of the temperatures mentioned above is an increase in the correction factor of the absorber when the mentioned pinch temperature is reduced.

Taking in account the increases of the LMTD and the correction factor, and the reduction of the heat capacity of the absorber, and considering that the overall heat transfer coefficient remains constant as in all cases, the area of the absorber has to decrease to satisfy the equation above.

To conclude the analysis of the absorber, the area of this component and thus its volume is reduced when the pinch temperature of the evaporator does so.

Total:

The total volume of the system is the addition of the volumes of each component, satisfying:

$$V_{total} = V_{evap} + V_{cond} + V_{gen} + V_{abs} + V_{HEx}$$

Considering that the volume of the heat exchanger is considerably lower than the volumes of the rest of the components, how the total volume of the system reacts to changes in the pinch temperature of the evaporator depends on how the volumes of the other four components react.

As seen before, the volumes of the condenser, the generator and the absorber are reduced when the mentioned pinch temperature is reduced, while the volume of the evaporator is increased in that same situation. Consequently, the total volume is increased or decreased depending on which of the two effects is predominant.

In the case of 49 °C ambient temperature, the logarithmic mean temperature difference in the absorber is highly increased when the pinch temperature in the evaporator is reduced (specially between 7 K and 5 K), and the same occurs to the correction factor. This results in a considerably high reduction of the volume of the absorber, and its effect is predominant over the changes in the volumes of the other three components, making the total volume of the system to be reduced.

In the case of 40 °C and 35 °C ambient temperature, changes in the absorber are moderated, and is the increase in the volume of the evaporator which predominates over the rest, resulting in an increase of the total volume.

To sum up, the behaviour of the total volume as a function of the evaporating temperature (or what is the same the pinch temperature in this component), depends on the behaviour of the volume of the different components of the system as a function of the same parameter, and how important the changes each of them suffer depending on various parameters as the ambient temperature are.

This explains why the total volume of the system reacts in a different way for the case in which the ambient temperature is 49 °C than in the other two cases.

In the appendix, table A15 shows the variation of some parameters when modifying the pinch temperature of the evaporator in the case of the three different environmental temperatures, so it is easier to understand the explanation given.

8. Conclusions and future work

An absorption heat pump has been modelled for automotive applications, with the aim of giving an alternative air conditioning system that could solve the existing problem of the electric vehicles' range when the air conditioning is activated.

As a first step, the thermodynamic properties of the water/LiBr mixture present in the cycle have been studied, analysing different ways for their calculation with various temperature references, and validating them in the range of temperatures and mass fractions considered. This study allows anyone interested in knowing the properties of the mixture to have an easy access to them without the necessity of disposing any kind of software.

With the purpose of knowing the maximum cooling effect that the absorption heat pump has to provide, the worst ambient conditions in which the vehicle could be driven are employed as inputs in the model. Furthermore, two different vehicles are studied, each of them has a different colour, black and white respectively. It is observed that the cooling capacity that a black vehicle needs under extreme conditions is quite higher than that needed by the white car.

Taking the cooling requirements of the white car as input, the model of the absorption heat pump is developed. EES has been used with this purpose, and the model of the system has been validated, obtaining satisfactory results.

Considering that the application of the system is air conditioning of vehicles, is in this point where the size takes a high importance. The volume that the air conditioning system can occupy in a vehicle is limited, and therefore the volume necessities of the modelled absorption heat pump must be studied.

The results obtained for the required volume of the system depending on the conditions under which it will be used and the minimum required overall heat transfer coefficients in each case are reasonable enough to affirm that the system seems feasible for the considered application.

Not only the volume occupied by the system seems reasonable, but also the minimum required U values are, in most cases, small enough to be reachable. Moreover, if the U values of the different components are higher than those given as minimum, the volume occupied by the system will be lower, making it even more feasible.

As future work, it could be useful to check the minimum U values required when different combinations of volumes of the different components are tried.

It could be also worth it to study which are the actual values of the overall heat transfer coefficients that each component of the absorption heat pump can work with. A study in this field would facilitate the volume analysis and determine which the best pinch temperature in the evaporator is in order to reduce the size of the system.

Appendix

Principles of the batteries:

A more technical explanation about the working principles of the batteries than that given in the specific section could be as follows: the anode undergoes an oxidation reaction (the anode combines with two or more ions from the electrolyte during the discharge to release one or more electrons forming a compound), while the cathode, simultaneously, undergoes a reduction reaction (the free electrons released from the anode and the ions, which is the material the cathode is made of, combine to form compounds) [4].

The flow of electrons gradually slows down, and consequently the electricity produced drops, until one (or both) of the electrodes run out of reagents that make their respective reactions possible. This can also be explained as until there is the same amount of electrons in the two sides, positive and negative. In that moment the battery is no longer capable of producing an electric flow unless it is recharged. Rechargeable batteries use electrical energy to reverse the positive and negative electrodes of the electrochemical cells, so that the electron flow can be restarted [4].

The working principals of a cell are shown in the figure below [2]:

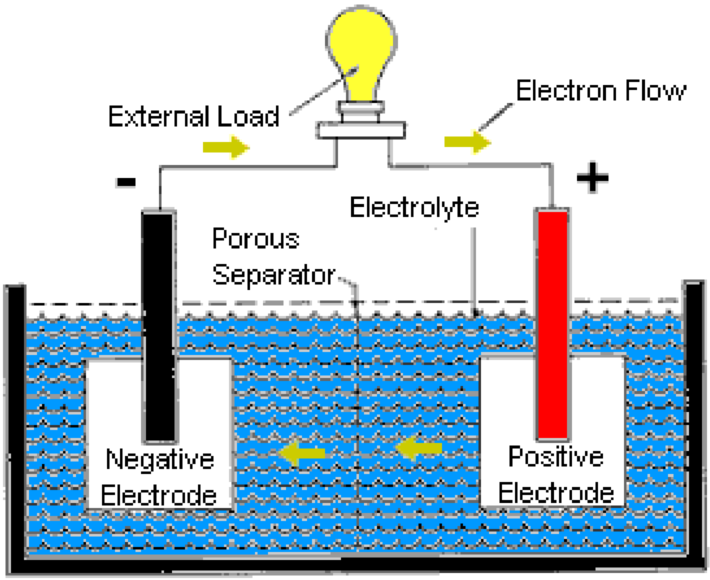


Fig 25. Electron flow in the external circuit from the negative to the positive electrode [2].

EV applications require an amount of energy that can only be achieved by connecting a considerably high number of cells. The connection can be done in series or parallel configuration [2].

The series configuration is characterized because the negative side of a cell is directly connected to the positive side of the following cell and so on until the end of the pack, which is determined by the voltage and energy capacity desired. The total voltage corresponds to the multiplication of the number of cells by the individual voltage of each of them [2].

Opposite to the series configuration, in the parallel, the positive side of each cell is connected to the correspondent positive side in the following cell, and the same happens to the negative electrodes. In this configuration, the total voltage is the same as the individual cell's voltage, so what determines the number of cells is the desired energy storage [2].

Types of batteries:

For electric car applications, there are three suitable main types of rechargeable batteries, which are lead-acid batteries, nickel metal hydride batteries and lithium-ion [11].

The oldest form of rechargeable battery that are still in use are the lead-acid batteries, which were invented in 1859 [11]. Reduction and oxidation reactions that occur at each electrode are responsible of the electron flow. In the negative electrode an oxidation reaction takes place and an electron is originated as consequence. For every electron generated in that side, there is an electron consumed in the positive side, due to a reduction reaction. The two reactions are given by [2]:

Reaction at the negative electrode: $Pb + SO_4^{-2} \rightarrow PbSO_4 + 2e^-$

Discharge reaction at the positive electrode: $PbO_2 + SO_4^{-2} + 4H^+ + 2e^- \rightarrow PbSO_4 + 2H_2O$

The main advantage of this kind of batteries is that they have been used for many years, what makes them to be cheap and well understood. However, they have a major disadvantage; if they are overcharged there is a risk of explosion due to the production of dangerous gases while being used [11].

Nickel metal hydride batteries started to be commercialized in the late 1980s. The most important aspects of this type of batteries are that their energy density is very high and that they contain no toxic metals [11].

The most recent type of the three mentioned before are the lithium-ion batteries, which came into commercial use some years after the nickel metal hydride ones. They have a very high energy density as well, and the self-discharge is lower than in most of the batteries. Self-discharge is a property referred to the loss of charge when not being used. Furthermore, they have light weight and require low maintenance, what make them to be widely used in electronic devices. Lithium-ion polymer batteries are a variation on this kind of batteries that could also be suitable for EV application. Nowadays the most important disadvantage they present is their expensive price [11].

Fuel Cells:

PEM fuel cells are fed with hydrogen and oxygen to produce heat and electricity. They are based on an electrolyte membrane that is able to conduct protons but not electrons, placed between two electrodes known as anode and cathode. As in batteries, an oxidation reaction takes place in the anode, while in the cathode the reaction corresponds to a reduction reaction. Both are porous so that hydrogen and oxygen can, respectively, pass through them [27].

Protons through the electrolyte reach the cathode, where they combine with oxygen and electrons in the reduction reaction mentioned before and thus, produce water and heat [28].

The fact that the proton exchange membrane does not allow the flow of electrons pass through it is what makes possible to obtain electricity. Electrons must flow through an external circuit around the membrane, what creates an electrical current. The power produced by a single fuel cell as a consequence of this current is lower than 1.16 V, which is not enough power to run almost any application. Usually a great amount of fuel cells, constituting a fuel cell stack, are needed [27].

Thermodynamic properties:

Table A1. Coefficients and exponents of the pressure equation [22].

I	**m_i**	**n_i**	**t_i**	**a_i**
1	3	0	0	$-2.41303*10^{2}$
2	4	5	0	$1.91750*10^{7}$
3	4	6	0	$-1.75521*10^{8}$
4	8	3	0	$3.25430*10^{7}$
5	1	0	1	$3.92571*10^{2}$
6	1	2	1	$-2.12626*10^{3}$
7	4	6	1	$1.85127*10^{8}$
8	6	0	1	$1.91216*10^{3}$

T_c = 647.096 K

Table A2. Coefficients and exponents of the enthalpy equation [22].

i	**m_i**	**n_i**	**t_i**	**a_i**
1	1	0	0	$2.27431*10^{0}$
2	1	1	0	$-7.99511*10^{0}$
3	2	6	0	$3.85239*10^{2}$
4	3	6	0	$-1.63940*10^{4}$
5	6	2	0	$-4.22562*10^{2}$
6	1	0	1	$1.13314*10^{-1}$
7	3	0	1	$-8.33474*10^{0}$
8	5	4	1	$-1.73833*10^{4}$
9	4	0	2	$6.49763*10^{0}$
10	5	4	2	$3.24552*10^{3}$
11	5	5	2	$-1.34643*10^{4}$
12	6	5	2	$3.99322*10^{4}$
13	6	6	2	$2.58877*10^{5}$
14	1	0	3	$-1.93046*10^{-3}$
15	2	3	3	$2.80616*10^{0}$
16	2	5	3	$-4.04479*10^{1}$
17	2	7	3	$1.45342*10^{2}$
18	5	0	3	$-2.74873*10^{0}$
19	6	3	3	$-4.49743*10^{2}$
20	7	1	3	$-1.21794*10^{1}$
21	1	0	4	$-5.83739*10^{-3}$
22	1	4	4	$2.33910*10^{-1}$
23	2	2	4	$3.41888*10^{-1}$
24	2	6	4	$8.85259*10^{0}$
25	2	7	4	$-1.78731*10^{1}$
26	3	0	4	$7.35179*10^{-2}$
27	1	0	5	$-1.79430*10^{-4}$
28	1	1	5	$1.84261*10^{-3}$
29	1	2	5	$-6.24282*10^{-3}$
30	1	3	5	$6.84765*10^{-3}$

T_c = 647.096 K; T_p = 221 K; h_c = 37548.5 J/mol

Table A3. Values of the enthalpies for the different methods when x=0.2 and differences with the results obtained using EES.

x=0.2 (w=0.5465)						
Pressure		1 kPa (T=309 K)	2 kPa (T=321 K)	15 kPa (T=363 K)	30 kPa (T=380 K)	45 kPa (T=391 K)
Method	EES	85.170	109.700	197.100	234.100	257.800
	[22] Ref. 273 K (Experimental)	85.551	109.951	197.010	233,9718	257.650
	[19] Reference 298 K	-216.389	-192.302	-108.513	-73,6913	-51.539
	[23] Ref. 273 K (Theoretical)	-32.915	-6.938	85.046	124.034	149.080
Diff with EES	Ref. 273 K (Experimental)	-0.381	-0.251	0.090	0.128	0.149
	Reference 298 K	301	302	306	308	309
	Ref. 273 K (Theoretical)	118	117	112	110	109

Table A4. Values of the enthalpies for the different methods when x=0.3 and differences with the results obtained using EES.

x=0.3 (w=0.6739)						
Pressure		1 kPa (T=335 K)	2 kPa (T=348 K)	15 kPa (T=392 K)	30 kPa (T=411 K)	45 kPa (T=422 K)
Method	EES	209.3	231	307.5	340	360.9
	[22] Ref. 273 K (Experimental)	209.490	231.042	307.317	339.795	360.694
	[19] Reference 298 K	-149.37	-128.126	-54.230	-23.519	-3.983
	[23] Ref. 273 K (Theoretical)	-75.811	-53.486	26.169	60.179	82.097
Diff with EES	Ref. 273 K (Experimental)	-0.190	-0.042	0.183	0.204	0.206
	Reference 298 K	359	359	362	364	365
	Ref. 273 K (Theoretical)	285	284	281	280	279

Size Optimisation:

Table A5. COP and volume of the system as a function of T_{pinch} in the evaporator. The minimum U values employed are those correspondent to 49 °C ambient temperature and 77 L of the system when T_{pinch} = 5 K.

T_{env} = 49 °C	T_{pinch} [K]	COP	V_{total} (49 °C) [L]	V_{total} (40 °C) [L]	V_{total} (35 °C) [L]
Q_{evap} = 9.5 kW U_{abs} = 0.1912 kW/m^{2}K U_{cond} = 0.04304 kW/m^{2}K U_{evap} = 0.02825 kW/m^{2}K U_{gen} = 0.09459 kW/m^{2}K U_{HEx} = 0.1 kW/m^{2}K	7	0.6820	83.01	60.79	59.35
	6	0.6849	72.51	61.47	60.54
	5	0.7116	76.99	64.30	63.11
	4	0.6929	68.54	64.38	64.51
	3	0.6955	69.32	66.99	67.84
	2	0.6990	71.75	71.16	72.97

Table A6. COP and volume of the system as a function of T_{pinch} in the evaporator. The minimum U values employed are those correspondent to 49 °C ambient temperature and 77 L of the system when T_{pinch} = 5 K.

T_{env} = 49 °C	T_{pinch} [K]	COP	V_{total} (49 °C) [L]	V_{total} (40 °C) [L]	V_{total} (35 °C) [L]
Q_{evap} = 9.5 kW U_{abs} = 0.1912 kW/m^{2}K U_{cond} = 0.04304 kW/m^{2}K U_{evap} = 0.02825 kW/m^{2}K U_{gen} = 0.09459 kW/m^{2}K U_{HEx} = 0.1 kW/m^{2}K	7	0.6820	83.01	60.79	59.35
	6	0.6849	72.51	61.47	60.54
	5.001	0.6889	69.11	62.61	62.20
	4	0.6929	68.54	64.38	64.51
	3	0.6955	69.32	66.99	67.84
	2	0.6990	71.75	71.16	72.97

Table A7. COP and volume of the system as a function of T_{pinch} in the evaporator. The minimum U values employed are those correspondent to 40 °C ambient temperature and 77 L of the system when T_{pinch} = 5.001 K.

T_{env} = 40 °C	T_{pinch} [K]	COP	V_{total} (49 °C) [L]	V_{total} (40 °C) [L]	V_{total} (35 °C) [L]
Q_{evap} = 9.5 kW U_{abs} = 0.05585 kW/m^{2}K U_{cond} = 0.03488 kW/m^{2}K U_{evap} = 0.03236 kW/m^{2}K U_{gen} = 0.1102 kW/m^{2}K U_{HEx} = 0.1 kW/m^{2}K	7	0.6820	152.6	78.25	72.56
	6	0.6849	114.7	77.28	72.7
	5	0.6889	99.83	77	73.35
	4	0.6929	93.76	77.59	74.79
	3	0.6955	90.42	78.8	76.94
	2	0.6990	89.92	81.59	80.82

Table A8. COP and volume of the system as a function of T_{pinch} in the evaporator. The minimum U values employed are those correspondent to 35 °C ambient temperature and 77 L of the system when T_{pinch} = 5.001 K.

T_{env} = 35 °C	T_{pinch} [K]	COP	V_{total} (49 °C) [L]	V_{total} (40 °C) [L]	V_{total} (35 °C) [L]
Q_{evap} = 9.5 kW U_{abs} = 0.04691 kW/m^{2}K U_{cond} = 0.03087 kW/m^{2}K U_{evap} = 0.03576 kW/m^{2}K U_{gen} = 0.1102 kW/m^{2}K U_{HEx} = 0.1 kW/m^{2}K	7	0.6820	173	84.12	77
	6	0.6849	127.6	82.62	76.74
	5	0.6889	109.6	81.84	77
	4	0.6929	101.9	81.99	78.05
	3	0.6955	97.36	82.64	79.68
	2	0.6990	95.87	84.83	82.95

Table A9. COP and volume of the system as a function of T_{pinch} in the evaporator. The minimum U values employed are those correspondent to 49 °C ambient temperature and 70 L of the system when T_{pinch} = 5.001 K.

T_{env} = 49 °C	T_{pinch} [K]	COP	V_{total} (49 °C) [L]	V_{total} (40 °C) [L]	V_{total} (35 °C) [L]
Q_{evap} = 9.5 kW	7	0.6820	92.92	58.98	56.72
U_{abs} = 0.1224 kW/m^{2}K	6	0.6849	76.07	59.26	57.64
U_{cond} = 0.04674 kW/m^{2}K	5	0.6889	70	60.04	59.01
U_{evap} = 0.03068 kW/m^{2}K	4	0.6929	68.13	61.48	61.06
U_{gen} = 0.1215 kW/m^{2}K	3	0.6955	67.86	63.65	63.96
U_{HEx} = 0.1 kW/m^{2}K	2	0.6990	69.48	67.31	68.57

Table A10. COP and volume of the system as a function of T_{pinch} in the evaporator. The minimum U values employed are those correspondent to 40 °C ambient temperature and 70 L of the system when T_{pinch} = 5.001 K.

T_{env} = 40 °C	T_{pinch} [K]	COP	V_{total} (49 °C) [L]	V_{total} (40 °C) [L]	V_{total} (35 °C) [L]
Q_{evap} = 9.5 kW	7	0.6820	138.6	71.12	65.97
U_{abs} = 0.06158 kW/m^{2}K	6	0.6849	104.2	70.26	66.09
U_{cond} = 0.03846 kW/m^{2}K	5	0.6889	90.71	70	66.69
U_{evap} = 0.03568 kW/m^{2}K	4	0.6929	85.21	70.55	68
U_{gen} = 0.1215 kW/m^{2}K	3	0.6955	82.18	71.64	69.95
U_{HEx} = 0.1 kW/m^{2}K	2	0.6990	81.72	74.18	73.48

Table A11. COP and volume of the system as a function of T_{pinch} in the evaporator. The minimum U values employed are those correspondent to 35 °C ambient temperature and 70 L of the system when T_{pinch} = 5.001 K.

T_{env} = 35 °C	T_{pinch} [K]	COP	V_{total} (49 °C) [L]	V_{total} (40 °C) [L]	V_{total} (35 °C) [L]
Q_{evap} = 9.5 kW	7	0.6820	157	76.44	69.99
U_{abs} = 0.05173 kW/m^{2}K	6	0.6849	115.9	75.09	69.76
U_{cond} = 0.03404 kW/m^{2}K	5	0.6889	99.56	74.39	70
U_{evap} = 0.03943 kW/m^{2}K	4	0.6929	92.62	74.52	70.95
U_{gen} = 0.1215 kW/m^{2}K	3	0.6955	88.46	75.12	72.44
U_{HEx} = 0.1 kW/m^{2}K	2	0.6990	87.12	77.1	75.4

Table A12. COP and volume of the system as a function of T_{pinch} in the evaporator. The minimum U values employed are those correspondent to 49 °C ambient temperature and 60 L of the system when T_{pinch} = 5.001 K.

T_{env} = 49 °C	T_{pinch} [K]	COP	V_{total} (49 °C) [L]	V_{total} (40 °C) [L]	V_{total} (35 °C) [L]
Q_{evap} = 9.5 kW	7	0.6820	79.56	50.59	48.66
U_{abs} = 0.1434 kW/m^{2}K	6	0.6849	65.18	50.83	49.44
U_{cond} = 0.05477 kW/m^{2}K	5	0.6889	60	51.5	50.62
U_{evap} = 0.03595 kW/m^{2}K	4	0.6929	58.41	52.74	52.38
U_{gen} = 0.1424 kW/m^{2}K	3	0.6955	58.19	54.59	54.85
U_{HEx} = 0.1 kW/m^{2}K	2	0.6990	59.57	57.72	58.79

Table A13. COP and volume of the system as a function of T_{pinch} in the evaporator. The minimum U values employed are those correspondent to 40 °C ambient temperature and 60 L of the system when T_{pinch} = 5.001 K.

T_{env} = 40 °C	T_{pinch} [K]	COP	V_{total} (49 °C) [L]	V_{total} (40 °C) [L]	V_{total} (35 °C) [L]
Q_{evap} = 9.5 kW U_{abs} = 0.07216 kW/m^{2}K U_{cond} = 0.04506 kW/m^{2}K U_{evap} = 0.04181 kW/m^{2}K U_{gen} = 0.1424 kW/m^{2}K U_{HEx} = 0.1 kW/m^{2}K	7	0.6820	118.5	60.95	56.55
	6	0.6849	89.15	60.21	56.66
	5	0.6889	77.67	60	57.18
	4	0.6929	72.99	60.48	58.3
	3	0.6955	70.4	61.41	59.97
	2	0.6990	70.02	63.58	62.98

Table A14. COP and volume of the system as a function of T_{pinch} in the evaporator. The minimum U values employed are those correspondent to 35 °C ambient temperature and 60 L of the system when T_{pinch} = 5.001 K.

T_{env} = 35 °C	T_{pinch} [K]	COP	V_{total} (49 °C) [L]	V_{total} (40 °C) [L]	V_{total} (35 °C) [L]
Q_{evap} = 9.5 kW U_{abs} = 0.06061 kW/m^{2}K U_{cond} = 0.03989 kW/m^{2}K U_{evap} = 0.0462 kW/m^{2}K U_{gen} = 0.1424 kW/m^{2}K U_{HEx} = 0.1 kW/m^{2}K	7	0.6820	134.3	65.49	59.98
	6	0.6849	99.17	64.34	59.79
	5	0.6889	85.23	63.75	60
	4	0.6929	79.31	63.87	60.82
	3	0.6955	75.77	64.38	62.09
	2	0.6990	74.62	66.07	64.62

Table A15. Variation of some parameters of the absorber and of the volumes of the different components as a consequence of modifying the pinch temperature in the evaporator. Numbers correspond to the case in which U values were calculated for 49 °C ambient temperature and 77 L of the system when T_{pinch} = 5.001 K.

T_{env}	T_{pinch} (K)	Q_{abs} (kW)	$\Delta T_{LMTD_{abs}}$ (K)	F_{abs}	V_{abs} (L)	V_{cond} (L)	V_{evap} (L)	V_{gen} (L)
49 °C	7	13.45	6.195	0.4509	48.94	21.52	18.85	11.26
	6	13.39	7.682	0.6006	29.51	21.50	19.91	11.02
	5	13.33	9.113	0.6920	21.49	21.49	21.49	10.75
	4	13.25	10.23	0.7436	17.72	21.47	23.47	10.42
	3	13.20	11.44	0.7825	15.01	21.45	26.08	10.25
40 °C	7	13.45	14.34	0.7539	12.48	17.71	21.69	11.26
	6	13.39	15.15	0.7694	11.68	17.69	23.02	11.02
	5	13.33	15.94	0.7863	10.81	17.68	25.00	10.75
	4	13.25	16.58	0.7992	10.17	17.67	27.49	10.42
	3	13.20	17.50	0.8127	9.438	17.65	30.81	10.25
35 °C	7	13.45	17.35	0.7737	10.08	15.67	23.78	11.26
	6	13.39	18.06	0.7835	9.621	15.66	25.32	11.02
	5	13.33	18.78	0.7946	9.082	15.65	27.62	10.75
	4	13.25	19.33	0.8030	8.677	15.64	30.54	10.42
	3	13.20	20.22	0.8127	8.171	15.63	34.43	10.25

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