

Modeling of a Von Platen-Munters diffusion absorption refrigeration cycle

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Abstract. This article presents a thermodynamical model of a Von-Platen diffusion absorption refrigeration cycle for power electronics applications. It is first validated by comparison with data available in the literature for the classical water-ammonia-helium cycle for commercial absorption fridges. Then new operating conditions corresponding to specific ABB applications, namely high ambient temperature and new organic fluids combinations compatible with aluminium are simulated and discussed. The target application is to cool power electronics converters in harsh environments with high ambient temperature by providing refrigeration without compressor, for passive components losses of about 500 W, with a compact and low cost solution.

1. Introduction

Conventional refrigeration systems are dual pressure cycles where the saturation temperature difference between the condenser and the evaporator is produced by a system pressure difference. This requires a mechanical input to drive the compressor or pump needed to generate this change in pressure, which adds significantly to the noise level and cost of the system while reducing the reliability. On the other hand, single pressure absorption refrigeration systems use three working fluids to create temperature changes by imposing partial pressures on the refrigerant and a heat source to circulate and separate the fluids. A patent of an absorption cycle was filed in 1922 by Baltzer Von Platen and Carl Munters [1], using water, ammonia and helium. The electrical power market is pushing towards high ambient temperature applications (mining, solar, marine). In these harsh conditions, conventional cooling is not suitable anymore and active cooling systems are used in nowadays commercial products. Absorption cooling is an elegant potential alternative solution to provide at the same time refrigeration and energy harvesting with a passive system.

2. The Von Platen-Munters diffusion absorption refrigeration cycle

The diffusion-absorption refrigeration cycle uses the thermosyphon principle to drive the fluid through the circuit. Three heat sources are needed to drive the cooling process. A high power/high temperature heat source (typically a gas burner providing 70% of the total power), a low power/low temperature heat source (typically the refrigerator internal air at 4°C and 30% of the total power) and a medium temperature heat sink (external air at 40°C usually) to reject 100% of the power. The original Von Platen-Munters diffusion-absorption refrigeration cycle uses three different fluids, the absorbant (water), the refrigerant (ammonia) and the inert gas (helium). The Von Platen-Munters diffusion absorption refrigeration cycle, shown in figure 1, is



composed of seven heat exchangers (in the following the adjectives 'weak' and 'strong' refer to the amount of refrigerant fluid in the mixture).

The **generator**, where a pre-heated strong liquid solution of ammonia-water enters from the solution heat exchanger and is vaporized with a heat load Q_g , produces a strong vapor mixture of ammonia-water and a weak liquid solution of ammonia in water. A **bubble pump**, in the generator tank, sends the weak liquid solution of ammonia-water to the solution heat exchanger thanks to the growth of spatially confined bubbles created with a heat load Q_b . The **rectifier** condenses the small amount of steam still present in the vapor coming from the generator into liquid so that the ammonia vapor is as pure as possible. Rectification is an exothermic phenomena and reject a heat load Q_r into the ambient. The water condensate flows back into the generator by gravity. The **condenser** condenses the ammonia into liquid above atmospheric air temperature, which is possible since the partial pressure of ammonia without the helium is above the saturation pressure. Condensation is an exothermic phenomena and reject a heat load Q_c into the ambient. The **solution heat exchanger** (SHX) pre-cools the weak liquid solution of ammonia-water sent by the bubble pump with the strong liquid solution of ammonia-water coming out of the absorber. This pre-cooling increases the capacity of the water to absorb more ammonia in the absorber and therefore improves the effectiveness of the cycle. The **gas heat exchanger** (GHX) pre-cools the helium rich gas from the absorber and the liquid refrigerant from the condenser with the strong ammonia-helium vapour mixture coming out from the evaporator. This is a key component of the cycle because without this pre-cooling there would be no useful refrigeration power left in the evaporator. The **absorber** receives the pre-heated strong ammonia-helium vapour mixture coming out from the gas heat exchanger and showers it with the pre-cooled weak liquid solution of ammonia-water coming out from the solution heat exchanger in order to absorb the ammonia into the water, thus forming a strong liquid solution of ammonia-water at the exit of the absorber. Absorption is an exothermic phenomena and reject a heat load Q_a into the ambient. The **evaporator** receives liquid refrigerant from the condenser and pre-cooled helium rich gas from the absorber via the gas heat exchanger. In the presence of helium, the ammonia will evaporate at low partial pressure and therefore at the low temperature the refrigerator is intended to work at. This phenomena is endothermic and absorbs a heat load Q_e from inside the enclosure, Q_e being the evaporation heat. After the bubble pump a small flow of strong vapor mixture of ammonia-water is present and needs to be separated from the liquid flow to be sent into the condenser together with the main flow of strong vapor mixture of ammonia-water produced in the generator. This is achieved by gravity in the **reservoir**.

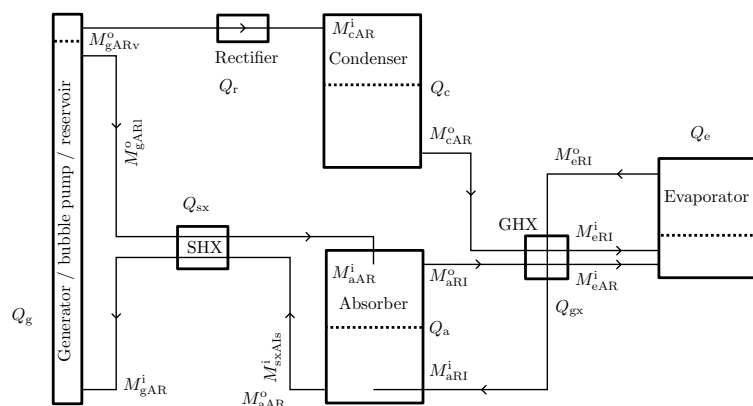


Figure 1. Model of the Von Platen-Munters diffusion absorption refrigeration cycle.

3. New fluid mixtures selection

The traditional Von Platen-Munters diffusion-absorption refrigeration cycle uses water, ammonia and helium as working fluids so that the heat exchangers have to be made from steel because water is not compatible with aluminium and ammonia is not compatible with copper. But steel has poor heat transfer performances compared to aluminium. This cycle was designed for small power refrigerators (about 60 W cooling power and 32°C air temperature) cooled by natural convection so that steel could be used with no adverse effect on the performances. However an electrical converter will require about 500 W of refrigeration power and more than 3000 W total cooling power at air temperatures greater than 70°C with a compact and low cost solution. This can be achieved only by using compact aluminium heat exchangers, therefore a new combination of fluids compatible with aluminium have to be found as well as a new cycle design compatible with compactness and low cost requirements. This new combination of fluids has to respect the following constraints: the absorbant-refrigerant fluid mixture should have a large boiling temperature glide to allow a good separation of absorbant and refrigerant fluid in the generator, the inert gas should not be able to condense in any condition of temperature and pressure present in the cycle, the refrigerant vapor must be strongly absorbed by the absorbing fluid at the refrigerant partial pressure, the inert gas must not be absorbed by the absorbing fluid at the inert gas partial pressure and the three fluids cannot chemically react with each other or their surroundings.

Zohar et al. [8] examined numerically the performance of the DAR cycle using an organic absorbent (DMAC), five different refrigerants (R22, R32, R124, R125 and R134a) and helium as inert gas. Target was to lower the generator temperature and system pressure along with a non-toxic refrigerant. Among the environmentally accepted working fluids, the DMAC-R134a system provides the highest COP, up to 40%.

One alternative new fluid combination for the Von Platen cycle is DMC (Dimethyl-carbonate) as absorbant, R134a as refrigerant and helium as inert gas, whose temperature-concentration phase diagram, obtained with REFPROP 9.0 [2], is shown in figure 2. For example, at 26 bar total pressure, a solution of 50% mass fraction of DMC at 353 K coming from the absorber (18 bar partial pressure), can be separated into 83% DMC liquid solution / 60% strong R134a vapour mixture by boiling at 451 K. This is an example with 100% rectification so that the absorption and condensation temperatures lie on the same line, but this does not yield the best performance. Figure 2 also shows the R134a-helium mixture phase diagram at 26 bar total pressure. At 26 bar total pressure, a mixture of 25% mole fraction of R134a starts boiling at 297 K while the pure R134a will condense at 353 K at this pressure. As illustrated in this figure, the working pressure is selected so that pure refrigerant in the condenser will condense at 10 K higher temperature than external air and the R134a-helium mixture will start evaporating at 26 K cooler temperature than the enclosure air. At 26 bar total pressure, the evaporator could work between 24°C and 50°C (enclosure cooling air at 50°C), the absorber and the condenser at 80°C (external cooling air at 70°C) and the generator between 127°C and 178°C, which is too hot for semiconductors operating at 150°C, but this temperature range can be lowered easily by 60 K if the rectification is limited to 90%. The thermodynamic properties of the DMC-R134a-helium fluid combination seem therefore adequate for making the Von Platen-Munters diffusion-absorption refrigeration cycle working at high ambient temperatures.

4. Mixtures thermodynamic properties

All thermodynamic properties mixtures were calculated with REFPROP 9.0 [2], which uses a new Helmholtz free energy formulation of the thermodynamic properties of the mixture water + ammonia published by Tillner-Roth and Friend in 1998 [3]. The thermodynamic state of a mixture are determined by the total pressure, the bulk composition and the temperature or the enthalpy. Therefore, knowing these, the following variables can be calculated: vapor phase

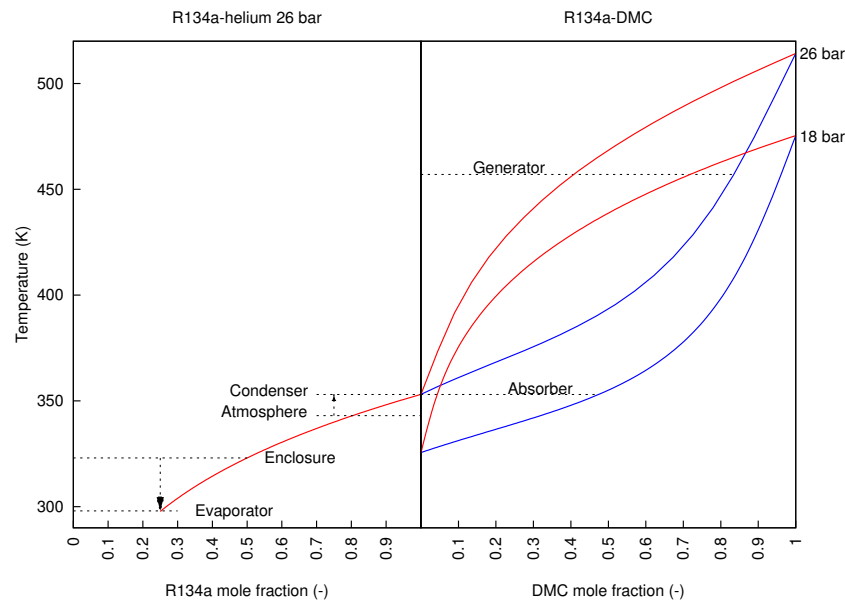


Figure 2. Phase diagrams of R134a-He and DMC-R134a mixtures at 26 bar.

composition y_v , liquid phase composition y_l , vapor quality x , enthalpy h , liquid phase enthalpy h_l , vapor phase enthalpy h_v , latent heat of vaporization Δh_{lv} , bulk temperature T and specific heat capacity c_p .

5. Modeling of the cycle

Zohar et al. [4] proposed a complete thermodynamic model and found the maximum COP when ammonia concentrations in strong and weak mixtures were respectively 0.25-0.30 and 0.10, and the cycle activation temperature was approximately 473 K. Starace and De Pascalis [5, 6] provided a thermodynamic model of the diffusion absorption cycle with not pure ammonia as refrigerant assumptions and taking into account the thermal pump heat losses towards the ambient. This model resulted in a more flexible and more realistic thermodynamic simulation of the cycle as well as in a better description of the real operation of the cycle than that the model of Zohar et al. Almèn [7] has written the most comprehensive report on the modeling and design of the Von Platen-Munters diffusion absorption refrigeration cycle. His study was not published in any journal but freely available on internet [7]. The model presented hereafter is inspired from his work.

The variables that were fixed in our model are the following: the desired vapor composition of absorbant-refrigerant after the rectifier y_{rARv}^o , the temperature at condenser exit T_{cAR}^o , the vapor quality at condenser exit (ideally 0%), the total pressure p_{tot} calculated to have 100% of liquid at the condenser exit with the composition y_{rARv}^o and the temperature T_{cAR}^o , the GHX temperature pinches $T_{eAR}^i - T_{eRI}^o$ and $T_{aRI}^i - T_{eRI}^o$, the SHX temperature pinch $T_{aAI}^i - T_{aAI}^o$, the generator heat load Q_g fixed by the power module heat load, the bubble pump heat load Q_b adjusted to satisfy the cycle energy balance, the bubble pump efficiency $\epsilon_b = M_{gAI}^o / M_{gAIv}^o$, the temperature of refrigerant-inert at evaporator inlet T_{eRI}^i , the temperature of refrigerant-inert at evaporator outlet T_{eRI}^o , the vapor quality at evaporator exit $x_{eRI}^o = 1$, the bulk composition of refrigerant-inert at evaporator outlet y_{eRI}^o calculated to have 100% of vapor at the prescribed temperature T_{eRI}^o , the vapor phase composition of refrigerant-inert at evaporator inlet y_{eRI}^i calculated so that it corresponds to the prescribed inlet saturation temperature T_{eRI}^i , the temperature of absorbant-refrigerant at absorber outlet T_{aAR}^o , the bulk composition of absorbant-refrigerant at absorber

outlet y_{aAR}^o (set at least to the minimum possible value to have 0% vapor at the prescribed temperature outlet T_{aAR}^o), the bulk composition of liquid absorbant-refrigerant at evaporator outlet $y_{eAR}^o = 1$ since the absorbant will always be liquid or solid at the evaporation temperature and all the refrigerant is vaporized.

Some variables that need to be initialized before their final value is determined by the solving procedure: the temperature of weak absorbant at generator outlet T_{gARl}^o initialized as the saturation temperature of the mixture at y_{gAR}^i composition and 30% vapor quality, the temperature of strong absorbant at generator inlet $T_{gAR}^i = T_{aAR}^o$, the composition of liquid refrigerant at evaporator inlet $y_{eAR}^i = y_{rARv}^o$, the composition of absorbant-refrigerant at absorber inlet y_{aAR}^i initialized at 0.85 (typical weak solution composition), and the mass flow of absorbant+refrigerant solution mixture at the exit of the evaporator $M_{eAR}^o = 0$ (ideal case). The procedure chosen to implement this model was to write the different equations through the cycle, starting from the generator and ending by the gas heat exchanger, as follows. First, the mass flow rate of vapor produced in the generator is completely determined by the generator heat load by the formula

$$M_{gARv}^o = x_{gAR}^o \cdot M_{gAR}^o, \quad x_{gAR}^o = \frac{Q_g}{M_{gAR}^i \cdot c_{p,gAR}^i \cdot \Delta T_{sat} + \Delta h_{lv,gAR}^i \cdot M_{gAR}^i}, \quad (1)$$

Then the bubble pump power determines the liquid mass flow rate sent to the absorber after the reservoir with

$$M_{gARl}^o = \epsilon_b \cdot M_{gARv}^o, \quad (2)$$

Finally the total mass flow rate exiting the generator is simply

$$M_{gAR}^o = M_{gARv}^o + M_{gARl}^o. \quad (3)$$

and mass conservation gives the mass flow rate entering the generator as

$$M_{gAR}^i = M_{gAR}^o + \Delta M_{rARl}^o, \quad (4)$$

ΔM_{rARl}^o being the mass flow rate of absorbant rich condensate coming from the rectifier back into the generator. The mass conservation in the rectifier is written

$$M_{rAR}^i = M_{gARv}^o, \quad M_{rAR}^o = M_{rAR}^i, \quad \Delta M_{rAR}^o = -M_{rAR}^i \cdot (y_{rARv}^o - y_{rAR}^i), \quad (5)$$

then T_{rAR}^o is the saturation temperature with the vapor composition y_{rARv}^o and the condensate composition y_{rARl}^o is fixed by the saturation temperature and vapor composition. The heat transferred in the rectifier is written

$$Q_r = (M_{rAR}^o - \Delta M_{rAR}^o) \cdot h_{rAR}^o - M_{rAR}^i \cdot h_{rAR}^i \quad (6)$$

The mass conservation in the condenser is written

$$M_{cAR}^i = M_{rAR}^o - \Delta M_{rARl}^o, \quad M_{cAR}^o = M_{cAR}^i, \quad (7)$$

The heat transferred in the condenser is written

$$Q_c = M_{cAR}^o \cdot h_{cAR}^o - M_{cAR}^i \cdot h_{cAR}^i \quad (8)$$

The heat transferred in the SHX is written

$$Q_{sx} = M_{aAR}^i \cdot c_{p,aAR}^i \cdot (T_{aAR}^i - T_{gARl}^o), \quad M_{aAR}^i = M_{gARl}^o + M_{eAR}^o. \quad (9)$$

The temperature of the strong absorbant entering the generator is solved using the modified Powell root solving numerical method so that the following equation is respected

$$Q_{sx} = M_{gAR}^i \cdot [c_{p,gAR}^i \cdot (T_{aAR}^o - T_{gAR}^i) + \Delta h_{lv,gAR}^i \cdot (1 - x_{gAR}^i) \cdot y_{gAR}^i]. \quad (10)$$

Then the enthalpy h_{gAR}^i of the rich absorbant entering the generator is then calculated from the temperature T_{gAR}^i and composition y_{gAR}^i . The generator heat load being fixed, the enthalpy h_{gAR}^o at the exit of the generator is

$$h_{gAR}^o = h_{gAR}^i + Q_g/M_{gAR}^i. \quad (11)$$

In the evaporator, some absorbant can still be present in the refrigerant liquid flow and will remain liquid at the evaporating temperature. Therefore this temperature T_{eAR}^o is set to the exit evaporation temperature T_{eRI}^o , which is one of the fixed parameters of the model. It can be easily demonstrated that the mass conservation in the evaporator yields:

$$M_{eAR}^i = M_{cAR}^o, \quad M_{eRV}^i = \alpha/(1 + \alpha \cdot \epsilon) \cdot (M_{gARl}^o + M_{cAR}^o) \quad (12)$$

$$M_{eIV}^i = (1 - y_{eRI}^i)/y_{eRI}^i \cdot M_{eRV}^i, \quad M_{eRI}^i = M_{eRV}^i + M_{eIV}^i \quad (13)$$

$$M_{eRI}^o = M_{eRI}^i \cdot \beta, \quad M_{eIV}^o = M_{eIV}^i \quad (14)$$

$$M_{eRV}^o = M_{eRI}^o - M_{eIV}^o, \quad M_{eAR}^o = (M_{eRI}^i + M_{eAR}^i) - M_{eRI}^o \quad (15)$$

with

$$\alpha = -(1 - y_{gARl}^o) \cdot \left(1 - \frac{1 - y_{gAR}^i}{1 - y_{gARl}^o} \cdot \frac{y_{gARl}^o}{y_{gAR}^i} \right), \quad \beta = \frac{1 - y_{eRI}^i}{1 - y_{eRI}^o}, \quad \epsilon = \frac{y_{eRI}^o - y_{eRI}^i}{y_{eRI}^i \cdot (1 - y_{eRI}^o)}. \quad (16)$$

Then the heat absorbed in the evaporator is calculated by the mixture enthalpies and the mass flow rates,

$$Q_e = (h_{eAR}^o \cdot M_{eAR}^o + h_{eRI}^o \cdot M_{eRI}^o) - (h_{eAR}^i \cdot M_{eAR}^i + h_{eRI}^i \cdot M_{eRI}^i). \quad (17)$$

The three heat loads transfered through the GHX are written

$$Q_{gxARl} = M_{cAR}^o \cdot c_{p,cAR}^o \cdot (T_{cAR}^o - T_{eAR}^i), \quad Q_{gxRIv} = M_{eRI}^o \cdot c_{p,eRI}^o \cdot (T_{aRI}^i - T_{eRI}^o) \quad (18)$$

$$Q_{gxIRv} = c_{p,gxIR}^i \cdot M_{gxIR}^i \cdot (T_{gxIR}^i - T_{gxRI}^m), \quad Q_{gxRIv} = \min(Q_{gxRIv}, Q_{gxARl} + Q_{gxIRv}) \quad (19)$$

$$T_{gxRI}^m = T_{eRI}^o + Q_{gxARl}/(M_{eRI}^o \cdot c_{p,eRI}^o). \quad (20)$$

The refrigerant+inert cold vapor stream exiting from evaporator is used to cool successively the hot liquid refrigerant stream coming from the condenser and then the inert rich hot gas stream coming from the absorber. Note that the pinch $T_{aRI}^i - T_{aAR}^o$ is fixed and T_{eRI}^i is then calculated with

$$T_{eRI}^i = T_{aRI}^o - (Q_{gxRIv} - Q_{gxARl})/(M_{gxIR}^o \cdot c_{p,gxIR}^i). \quad (21)$$

However, if $(Q_{gxRIv} - Q_{gxARl}) \geq Q_{gxIRv}$, this will result in a inert rich gas stream entering the evaporator at a temperature lower than the evaporation temperature, which is of course impossible. In this case the pinch $T_{eAR}^i - T_{eRI}^o$ is fixed instead. The mass conservation in the absorber is written

$$M_{aAR}^i = M_{gARl}^o + M_{eAR}^o, \quad M_{aRI}^i = M_{eRI}^o \quad (22)$$

$$M_{aRI}^o = M_{eRI}^i, \quad M_{aAR}^o = (M_{aAR}^i + M_{aRI}^i) - M_{aRI}^o, \quad (23)$$

and the heat load released due to absorption phenomena is written

$$Q_a = (h_{aRI}^o \cdot M_{aRI}^o + h_{aAR}^o \cdot M_{aAR}^o) - (h_{aAR}^i \cdot M_{aAR}^i + h_{aRI}^i \cdot M_{aRI}^i). \quad (24)$$

Finally the energy balance for the whole cycle is

$$Q_a + Q_r + Q_c + Q_e + Q_{sx} + Q_g = 0. \quad (25)$$

In order to respect the energy balance given by equation (25) either the bubble pump effectiveness ϵ_b or the bulk concentration of the strong solution y_{aAR}^o can be tuned, for given evaporation and condensation temperatures. An optimization algorithm can be used for this purpose, but the solution is actually more easily found by manual setting. The coefficient of performance (COP) of this cycle is then

$$COP = Q_e / Q_g. \quad (26)$$

The values of ϵ_b and y_{aAR}^o were judged satisfactory when a heat balance better than 1% was obtained. However the best performances are obtained when y_{aAR}^o is set at its minimum to have 0% vapor at the exit of the absorber and ϵ_b only was adjusted to satisfy the cycle energy balance.

6. Model validation

Table 1 first two columns show a comparison between the data published by Almèn [7] and the our model detailed in section 5 with ideal heat exchangers. This table shows the main parameters of the cycle simulated with water, ammonia and helium as working fluids at 23 bar pressure (working pressure for home refrigerators with ambient temperature up to 32° C). Comparison between the columns 'data' and 'model' shows that the present model gives numerical results close to that published by Almèn [7]. The numerical values are never exactly equal, which in our opinion comes from the differing evaluation methods of the mixtures properties since Almèn [7] used data from 1964 to calculate the mixtures properties. For the same reason, the fixed parameters are also slightly different in order to obtain convergence of our model on the energy balance. However the numerical values are always reasonably close so that our implementation of the model seems valid. In particular the coefficient of performance is accurately predicted with only 3 percentage point difference.

7. High ambient temperature simulations

Table 1 last column shows a simulation of the cycle with ambient temperature of 70°C, high power and non ideal heat exchangers (effectiveness ≈ 0.6). If DMC, R134a and helium at 19 bar are used as working fluids the simulation yields a COP of 32%, like water-ammonia-helium but with a much lower generator temperature of 131°C, which is beneficial for semiconductors cooling.

8. Conclusions

A theoretical model of the Von Platen-Munters diffusion absorption refrigeration cycle was made and validated with available data in the open literature. Secondly a new combination of organic fluids were identified in order to operate this cycle with aluminium heat exchangers and at high ambient temperature (70°C). For example DMC-R134a-helium is a possible combination, yielding a COP of 32% according to simulations of the cycle, with a maximum generator temperature of 136°C at 70°C ambient. The thermodynamic simulation of the cycle gives us the different stream flow rates, temperatures and enthalpies at the working point. The next step is to build the cycle into a prototype. To this end, pressure losses and heat transfer rates should be simulated as well with the thermodynamic data as input in order to determine piping diameters and heat exchangers sizes.

Table 1. Comparison between Almèn data and model. (*) = fixed parameters.

Source	data	model	model
Fluids	H2O-NH3-H2	H2O-NH3-H2	DMC-R134a-He
Pressure (bar)	23	22	19
(*) Bubble pump effectiveness	1.8	2	2.5
(*) Heat generator (W)	306	325	3000
Heat rectifier (W)	-91	-112	-345
Heat condenser (W)	-133	-163	-1568
Heat absorber (W)	-196	-153	-2042
Heat evaporator (W)	114	104	958
Absorber			
Weak sol. in flow (g/s)	0.40	0.35	21.2
R+I in flow (g/s)	0.39	0.35	13.6
Strong solution out (g/s)	0.48	0.44	28.6
Weak sol. in temp. (°C)	50	50	103
R+I in temp. (°C)	n.a.	48.7	69
Strong sol. out temp. (°C)	50	50	76
Weak sol. in comp. (mass)	88%	81%	73%
R+I in comp. (mole)	61%	50%	39%
(*) strong sol. out comp. (mass)	70%	70%	56.5%
Generator			
R vapor out flow (g/s)	0.22	0.17	8.3
Strong sol. in temp. (°C)	140	143	97
Weak sol. out temp. (°C)	187	178	131
Condenser			
R vapor in flow (g/s)	0.12	0.11	7.9
(*) R+A comp. (-)	90%	90%	88%
(*) R+A out temp. (°C)	57	57	73
Evaporator			
I gas in temp. (°C)	n.a.	-9	68
R in temp. (°C)	-17	-16	52
(*) R+I in boiling temp. (°C)	-39	-31	29
(*) R+I out boiling temp. (°C)	-17	-16	44
COP	35%	32%	32%

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