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THERMODYNAMIC SIMULATION OF AMMONIA-WATER DOUBLE EFFECT ABSORPTION CHILLER

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ABSTRACT

This paper deals with the modelling and the thermodynamic simulation of an ammonia/water double effect-double generator absorption chiller. Simulation results are used to study the influence of the various operating parameters on the *COP*. Two configurations of the double effect-double generator absorption chiller are investigated and compared.

1. INTRODUCTION

Absorption refrigerators and heat pumps have been for a long time limited to very specific and marginal uses [1] because of their low *COP* compared to that of vapor compression machines. Over the last few decades however, the field has experienced a resurgence of interest. Unfortunately the *COP*s of simple effect cycles are low in comparison with those of vapor compression cycles even when the differential cost for heat and electric power is considered [2]. This means that the energetic advantage of this kind of chillers remains insignificant unless enhanced structures are developed. This is why the research interest shifted to multiple effect structures designed by combining single stage components and cycles [2,3]. The present work is a part of the investigations of our research unit on absorption chilling and solar refrigeration [4,5,6]. It deals with a particular structure : the double effect-double generator chiller operating with ammonia/water mixture.

Its purpose is the simulation of the double effect cycle and the investigation of the effect of various operating parameters. Two configurations of the double generator cycle are studied.

2. CYCLE DESCRIPTION

Figure 1 shows schematically a double effect-double generator configuration. Such a machine is composed essentially of two condensers, an evaporator, two steam generators each one provided with a boiler and distillation column, an absorber, four expansion valves, two pumps and an adjustable three-way valve. The steam (3) flowing from the evaporator to the absorber, is absorbed by the weak solution (9). The absorption heat is rejected towards the environment (cooling water or air) which also receives the energy released by the first condenser. The strong solution (4) is distributed, by an adjustable three-way valve, between the two stages of the machine comprising each a generator, a condenser and a solution heat exchanger. A fraction a of the strong solution (4) is sent to the first generator. The separation of the refrigerant is performed at two temperature levels : the energy needed in the first boiler is supplied by the second condenser (Cond2). Large temperature differences between heat exchanging streams are so avoided. The first condenser (Cond1) receives the throtted condensate from the second condenser and the refrigerant vapor from the distillation column over the first boiler. The strong solution (14) supplying the second generator is a cold source for the deparation column over the 2nd boiler and determines thus the purity limit of refrigerant (23) leaving the last stage of this column. For a rational use of the energy supplied, three counter-current heat exchangers (HEX1, HEX2, HEX3) are used.

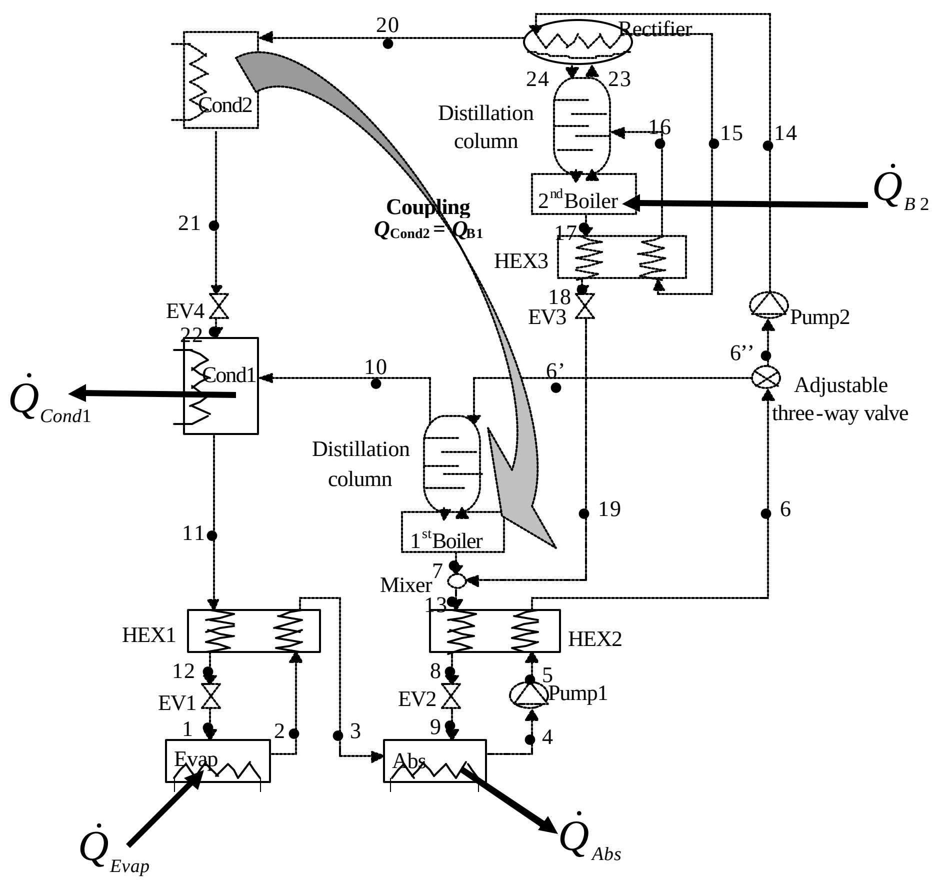

Figure 1 : Parallel flow double effect double generator absorption chiller

3. METHODOLOGY OF SIMULATION

The formulation of the simulation model for the chiller proceeds by the following steps:

- Determination of the chiller variance (number degrees of freedom),
- Specification of the fundamental operating conditions : the cooling capacity, the driving heat source temperature, the useful cold temperature and that of the environment,
- Formulation of the mass and energy balances governing the various chiller components.
- Characterisation of the heat transfer in the various heat exchangers (pinch method).

Independent equations expressing mass and energy balances in each component are developed:

$$m_f(=m_{11}) = m_{f1}(=m_{10}) + m_{f2}(=m_{20}) \quad (1)$$

$$(1 - a).m_r = m_{f2} + m_{p2}(=m_{17}) \quad (2)$$

$$\alpha.m_r = m_{f1} + m_{p1}(=m_7) \quad (3)$$

$$m_p(=m_{13}) = m_{p1} + m_{p2} \quad (4)$$

$$x_f.m_f = x_{f1}.m_{f1} + x_{f2}.m_{f2} \quad (5)$$

$$(1 - a).x_r.m_r = x_{f2}.m_{f2} + x_{p2}.m_{p2} \quad (6)$$

$$\alpha.x_r.m_r = x_{f1}.m_{f1} + x_{p1}.m_{p1} \quad (7)$$

$$x_p.m_p = x_{p1}.m_{p1} + x_{p2}.m_{p2} \quad (8)$$

$$Q_{Evap} = m_f.(h_2 - h_1) \quad (9)$$

$$h_3 - h_2 = h_{11} - h_{12} \quad (10)$$

$$h_{12} = h_1 \quad (11)$$

$$h_8 = h_9 \quad (12)$$

$$h_{18} = h_{19} \quad (13)$$

$$h_{21} = h_{22} \quad (14)$$

$$m_r.(h_6 - h_5) = m_p.(h_{13} - h_8) \quad (15)$$

$$m_p.h_{13} = m_{p1}.h_7 + m_{p2}.h_{19} \quad (16)$$

$$(1 - a).m_r.(h_{16} - h_{15}) = m_{p2}.(h_{17} - h_{18}) \quad (17)$$

$$m_{f1}.h_{10} + m_{p1}.h_7 - a.m_r.h_6 = m_{f2}.(h_{23} - h_{20}) \quad (18)$$

$$(1 - a).m_r.(h_{15} - h_{14}) = -m_{f2}.[h_g(T_{20},x_{f2},P_2) + R.h_f.(T_{20},x_{ref},P_2) - (1+R).\ h_g(T_{23},x,P_2) \quad (19)$$

$$R.(x_{f2} - x_{ref}) = x_{f2} - x \quad (20)$$

Based on practical considerations [4,5], some parameters are fixed as follows :

- 2nd Boiler : the temperature T_{17} must be fixed since it's the highest temperature in the cycle. It is the temperature of the weak saturated solution leaving the generator. In addition, we consider that the refrigerant steam leaves the rectifier, point 20, in a saturated state,
- 1st Condenser : the refrigerant leaves the first condenser, point 11, in sub-cooled state. The refrigerant temperature T_{11} and the sub-cooling *SRC* are fixed,
- Evaporator : the cooling capacity Q_E and the temperature T_2 are fixed,
- Absorber : The temperature T_4 and the sub-cooling *SRA* of the strong solution leaving the absorber are fixed,
- Pumps : the pumping operation is adiabatic,
- Condenser pressures are fixed so as to have a refrigerant as concentrated as possible,
- The evaporator pressure P_E is bounded by two limits [4,5]. The inferior limit corresponds to the case where it's not possible to absorb the refrigerant by the weak solution at the lowest absorber temperature. The superior limit is that when it's not more possible to evaporate the refrigerant at the highest temperature of the evaporator,
- Solution heat exchangers : the heat transfer is characterized by the minimal temperature approach in the exchanger [10,11]. The pinchs, at the cold ends of the exchangers, are fixed :

$$T_{12} = T_2 + pinch_{HEX1} \quad (21)$$

$$T_8 = T_5 + pinch_{HEX2} \quad (22)$$

$$T_{18} = T_{15} + pinch_{HEX3} \quad (23)$$

- The energy needed in the first boiler is supplied by the second condenser (Cond2):

$$m_{f1}.h_{10} + m_{p1}.h_7 - a.m_r.h_6 = m_{f2}.\,(h_{20} - h_{21}) \qquad (24)$$

- The coupling system between the second condenser and the first boiler is a counter-flow heat exchanger, also characterized by a pinch :

$$T_{21} = T_6 + pinch_{\mathrm{coupling}} \qquad (25)$$

4. RESULTS OF SIMULATION

To solve the large set of nonlinear equations of the simulation model the program CONLES, available as a FORTRAN 77 code [11], is used. The fluid thermodynamic properties are calculated in a subroutine incorporated in the program. The coefficient of performance is given by :

$$COP = \frac{Q_{Evap}}{Q_{B2}} \qquad (26)$$

The caracteristics of the studied case are:

- Cooling capacity: 17.5 kW at 300-285 K,
- First condenser and absorber temperature: 312 K,
- Evaporator exit temperature: 275 K,
- Maximum second generator temperature, T_{17} = 443 K,
- Pinch in heat exchangers: 5 K,
- Sub-cooling of absorber and condenser: 3 K.

4.1 Effect of evaporator pressure

Figure 2 shows the evolution of the chiller COP with the evaporator pressure. We note the existence of a maximum at 0,76, corresponding to an evaporator pressure of 3,5 bar. Such a behaviour, encountered also in the case of single effect chillers, is due to the existence of two opposite effects [4,5].

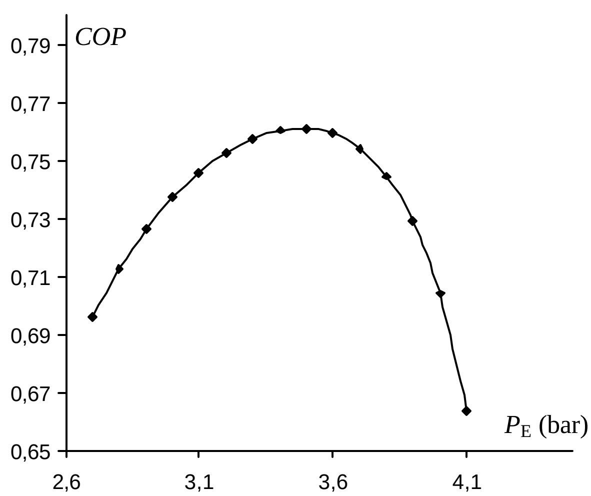

Figure 2 : COP vs. evaporator pressure P_E

4.2 Effect of partition coefficient *a*

Figure 4 shows the *COP* of the chiller as a function of the partition coefficient ($\alpha = m_{6'} / m_6$) for various evaporator pressure values P_E. For $a = 0$, the configuration is that of a single effect cycle. When the cooling capacity and the refrigerant flow rate m_{11} are fixed, the configuration is also equivalent to that of a single effect cycle for $a = 1$. In this case however, the total flowrate of the rich solution circulating in the first effect loop is very large (fig.5) compared to the refrigerant vapor flow rate m_{20} and no vapor is produced in the first boiler ($m_{10}=0$) because the energy supplied by condenser 2 to this boiler is not sufficient.
Hence, there is a maximum of *COP*, as seen in fig.4 for $a \sim 7\%$.

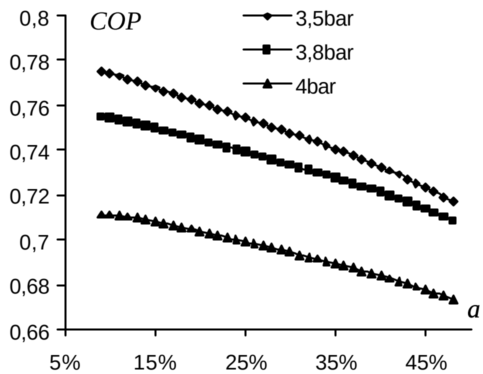

Figure 4 : *COP* vs. patition coefficient *a* for different evaporator pressures P_E

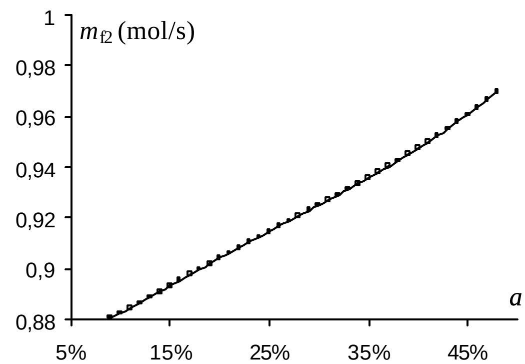

Figure 5 : Second stage refrigerant flow rate vs. partition coefficient, *a*

4.3 Effect of driving temperature T_{17}

The evolution of the *COP* as a function of the driving temperature for various values of P_E , is given in figure 6. With fixed cooling effect Q_E and refrigerant flow rate m_{11}, the increasing of this temperature, from a certain value on (450 K) is not necessary: it serves only to overheat the refrigerant steam.

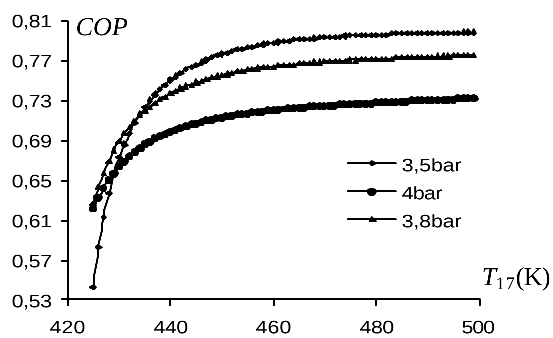

Figure 6 : *COP* vs. Chiller driving temperature for different values of P_E

5. SECOND CONFIGURATIONS

For the double generator-double effect absorption chiller (fig.1) studied previously, the strong solution (6) is distributed between the two generators. When the totality of this solution feeds the first generator and the weak solution of the first stage (7) feeds the second generator, a new configuration of the double effect double generator absorption chiller is obtained, where the two generators are assembled in series (fig.7).

Figure 8 shows the *COP* evolution of the two configuration for the same operating conditions. The parallel generator configuration of the first case is more performant than the present one. When the two generators are assembled in series the second generator is supplied by the weak solution of the first stage (7), whereas in the parallel configuration the two generators are supplied by the strong solution. In addition, in the first configuration (fig.1), an internal rectifier, where the energy of cooling is recovered by the strong solution, is used whereas in the second configuration the weak solution (7) can not be used to cool th rectifier.

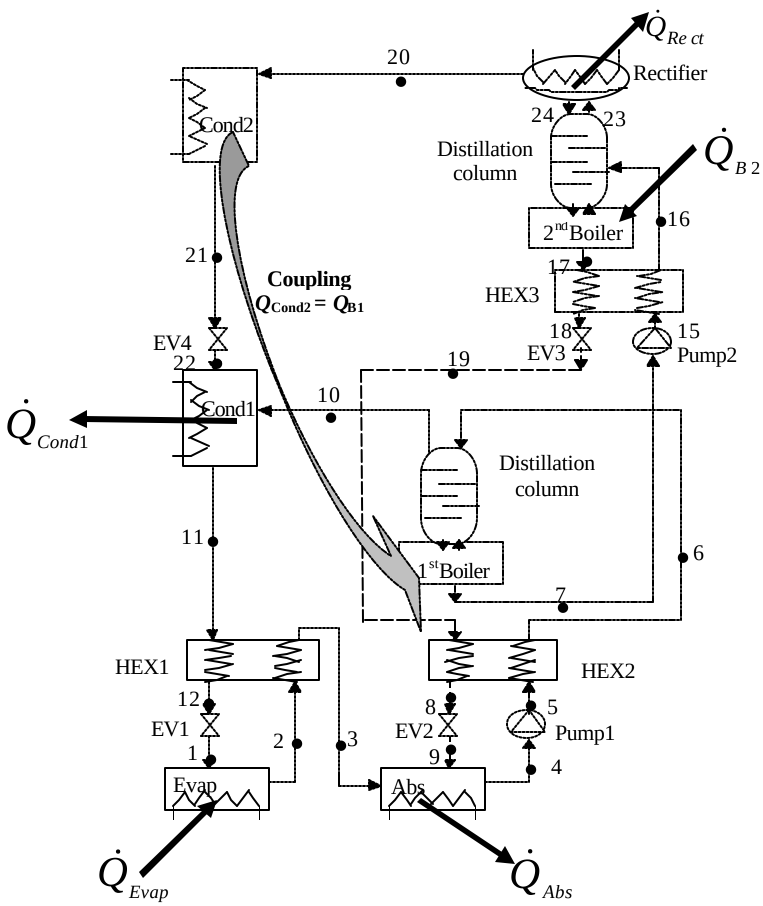

Figure 7 : Serie flow double effect double generator absorption chiller

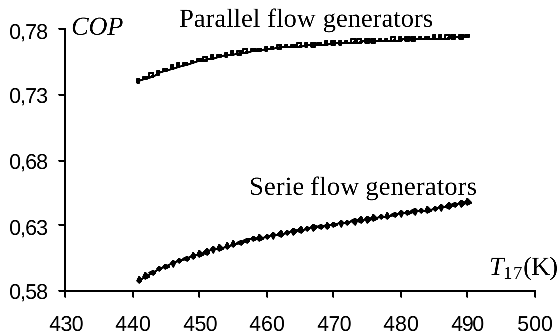

Figure 8 : *COP* of two configurations vs. Chiller driving temperature T_{17}

6. CONCLUSION

Simulation models for two ammonia-water double effect-double generator absorption chillers are developped in order to compare performances. Simulation enabled us to study the influence of the various operating parameters on the chiller performance. The parallel generator configuration of the double effect absorption chiller is more performant than the serie generator configuration.

NOMENCLATURE

COP	coefficient of performance	(–)
h	enthalpy	(J/mol)
m	molar flow rate	(mol/s)
P	pressure	(bar)
Q	heat transfer	(kW)
T	temperature	(°C,K)
W	mechanical work	(kW)
x	ammonia liquid concentration	(–)
y	ammonia vapor concentration	(–)
Greek		
a	distribution (partition) coefficient	(–)
DT	pinch	(°C, K)

Subscripts

Abs	absorber
B	boiler
Cond	condenser
Evap	evaporator
EV	expansion valve
P	pump
Rect	rectifier
HEX	heat exchanger
f	refrigerant
p	weak solution, poor
r	strong solution, rich

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