

Comparison of a Butane/Octane Absorption Machine with an Ammonia/Water One

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Abstract: We propose in this study, the use of the butane/octane mixture as working fluids for the absorption chiller and the comparison of its performance with the classical ammonia/water mixture. The efficiencies of the considered mixtures are estimated in the same operating conditions of a simple effect machine. When the condenser and the absorber of the machine are cooled with the ambient air the COP of the first system is 0.37, while the performance of the ammonia solution is about 0.52. For a machine cooled with the water the two mixtures lead to comparable coefficients of performance: 0.62 for the butane/octane and 0.67 for the ammonia/water mixture. This harmless natural combination is an interesting alternative since the machine working with this mixture can use a heat driven source at moderate temperature such as a thermal waste or vacuum solar panel. The two working pressures are also moderate (<5 bar) reducing the necessity of high airtightness in the construction of the machine.

Key words: Absorption, butane, octane, ammonia, water, COP

INTRODUCTION

Searching for new working fluids for the absorption machines is an old field knowing a renaissance since the interdiction of the use of the CFC and the HCFC for the compression machines. In spite of the numerous combinations of fluids that have been under investigation, only two working mixtures are extensively used: water/lithium bromide and ammonia/water. However, some disadvantages limit these two classical mixtures.

Many binary mixtures of alkane were investigated to be used as working fluids for the absorption chillers (Chekir *et al.*, 2006). The most suitable combination in the case of a classical machine is the butane/octane mixture. We propose in this study, the simulation of the research of an absorption machine operating with the butane/octane mixture and the comparison with the classical ammonia/water system. The chiller configuration studied (Fig. 1) is a classical simple effect machine composed of a condenser, an evaporator, an absorber, a generator, a pre-cooler and a solution heat exchanger. The proposed mixtures require a multistage separation in the generator, ensured by a distillation column (King, 1981; Wuithier, 1972). Two alternative cooling mediums for the condenser and the absorber are considered: the ambient air at 35°C and the water at 25°C.

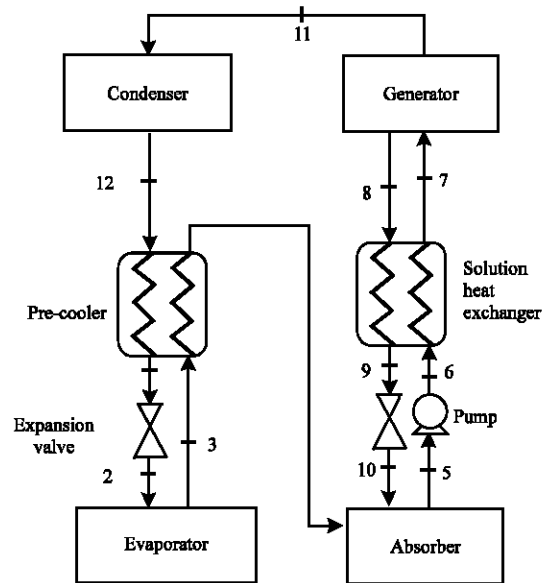


Fig. 1: Simple effect absorption chiller

MATERIALS AND METHODS

Machine modeling: A model of simulation of the research of an absorption machine is elaborated using the different equations of mass and energy balances of each

component of the cycle. The thermodynamic properties of the working fluids at different states in each part of the machine as well as the liquid-vapor equilibrium of the mixtures are estimated using the PC-SAFT equation of state (Mejbri and Bellagi, 2005).

Hence, the simulation of the machine requires the simultaneous resolution of the system of non-linear equations obtained. A numeric code is developed using the NS11 subroutine of the Harwell Library (HSL) and the algorithms of Newton-Raphson and Steepest-Descent.

Operating conditions: The proposed installation is a chilled water machine having a cooling capacity of 17.5 kW. The temperature of the chilled water is 7°C. The refrigerant output temperature of the evaporator is fixed at 2°C and the maximal hot temperature of the driving heat source in the generator is chosen 150°C.

The condenser and the absorber are cooled either with the ambient air at 35°C, either with the water at 25°C. The temperatures of the end of condensation and of the end of absorption are supposed higher of 12-15°C of the medium temperature in the case of the air and of 5°C in that of the water.

The purity of the refrigerant getting out of the generator is taken between 95 and 99%. The condensation pressure P_C is that of the saturated liquid at the end of condensation temperature with a composition equal to the fixed purity. The evaporation pressure varies between two limits; the higher one corresponds to the pressure of the saturated liquid leaving the evaporator having a composition equal to the purity of the refrigerant. Above this value, the refrigerant can no more evaporate. The lower limit is the pressure of saturation of the liquid having the molar composition of the poor solution at the absorption temperature.

Below this limit the refrigerant steam can not be absorbed by the poor solution. The vapor of refrigerant leaving the generator to the condenser is supposed saturated. The poor solution at the bottom of the generator is taken saturated too at the heat driving temperature of the cycle, T_8 .

The rich solution is pumped using an isentropic solution pump. The liquids getting out of the condenser and the absorber are supposed subcooled by 4°C. The two heat exchangers used here are supposed adiabatic and characterized each by a fixed temperature pinch: 10°C for the precooler and 5°C for the solution heat exchanger. All these inputs and assumptions are resumed in Table 1.

Table 1: Input data and assumptions

| Data | Designation | Value |
|---------------------------------------|-------------------------------------|--|
| Condensation temperature | T_C | 50°C for the air 30°C for the water |
| Evaporation pressure | P_E | Optimal value |
| Cooling capacity | \dot{Q}_E | 17.5 kW |
| Outlet evaporator temperature | T_3 | 2°C |
| Cooling medium temperature | T_a | 35°C for the air 25°C for the water |
| Subcooling in the condenser | ΔT_{SRC} | 4°C |
| Subcooling in the absorber | ΔT_{SRA} | 4°C |
| Molar fraction of refrigerant | y_{11} | ≥ 0.95 |
| Heat driven source temperature | T_8 | $\leq 150^\circ\text{C}$ |
| Thermal pinches in the two exchangers | ΔT_{SHE} ΔT_{PR} | 5°C 10°C |

Assumptions: The refrigerant vapor getting out of the generator is saturated. The poor solution getting out of the generator is saturated. The pumping is supposed isentropic

RESULTS AND DISCUSSION

The performance of the studied machine is estimated by calculating its performance coefficient COP. In the generator, a thermally driving power, Q_B is provided to the reboiler of the distillation column to extract the refrigerant from the absorbent. The COP is giving by:

$$COP = \frac{\dot{Q}_E}{\dot{Q}_B + \dot{W}_P}$$

For the two studied mixtures, the problem is approached according the chart presented in the Fig. 2.

The variation of the COP versus the evaporation pressure is having the form of a curve with a maximum situated between the two limits of P_E . The optimal value leading to a maximum COP is retained for the research with the two proposed mixtures. The Fig. 3 and 4 illustrate the variation of the COP versus the evaporation pressure P_E , respectively for the butane/octane system and the ammonia/water one in the case of an air cooling of the absorber and the condenser. Similarly, taking the water as cooling medium we represent in Fig. 5 and 6 this variation for the couple of mixtures. In Table 2 and 3 are presented the different parameters of the machine working with the two mixtures and the two cooling mediums. These parameters are the molar fraction respectively of the refrigerant, of the rich solution and of the poor one as well as their molar liquid rate. The two working pressures and the different thermal exchanged powers are also given. The last presented parameter is the COP. We can see here that the evaporation pressure for the C_4/C_8 mixture (1 bar) is lower than that of the NH_3/H_2O one (3.4 bar) with the first cooling medium. The same remark is observed in the case of water cooling. The observation of the curves

shows

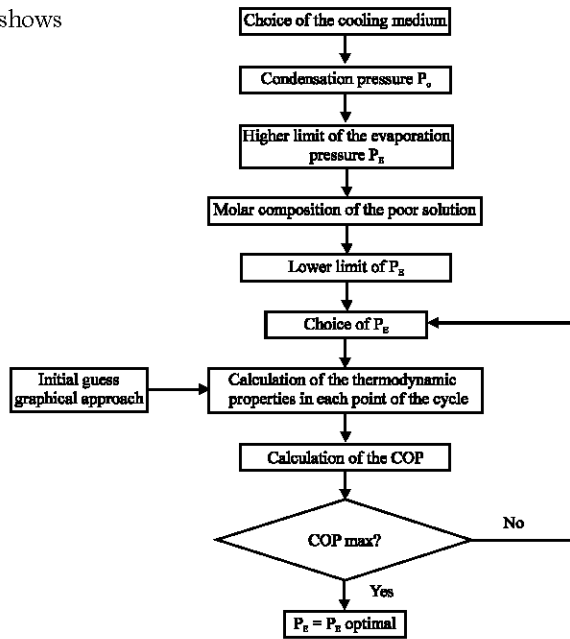


Fig. 2: Chart of numerical resolution

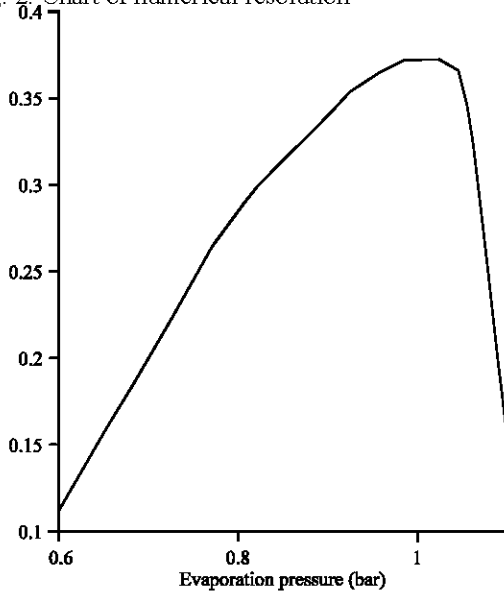


Fig. 3: Variation of the COP versus P_E for the C₄/C₈ system (air cooling)

that for the NH₃/H₂O machine the COP increases slowly and the curve is flattened, while for the second system it is sharp.

This may make the work of the butane machine sensitive to the evaporation pressure and even a minor variation of this pressure may affect the COP. When the

condenser and the absorber reject their heat to the water

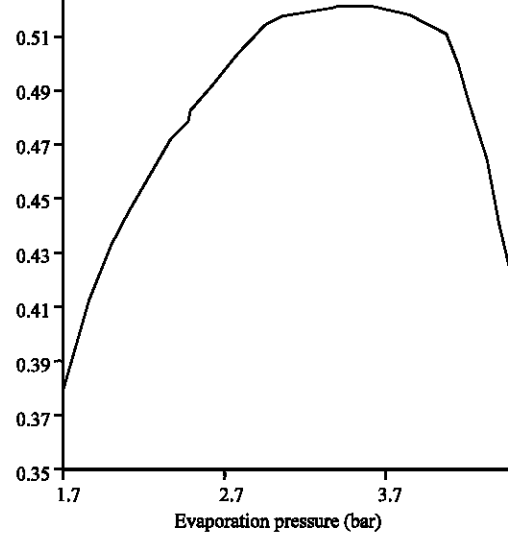


Fig. 4: Variation of the COP versus P_E for the NH₃/H₂O system (air cooling)

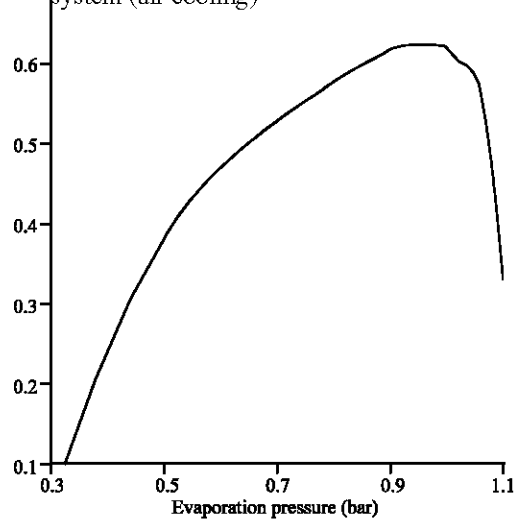


Fig. 5: Variation of the COP versus P_E for the C₄/C₈ system (water cooling)

the performances of the two mixtures are quite similar (0.67 and 0.62). The performance of an air-cooled machine operating with the C₄/C₈ system and air-cooled is less about 28.6% than that working with the ammonia solution at the same operating conditions.

Despite the enthalpy of vaporization that determines the quantity of heat required to evaporate the refrigerant

and effects directly the COP, many other parameters occur too such as the rich solution flow rate and the rate of degasification. For the first working mode of the machine, the degasification rate, $x_r - x_p$ is 0.09 for the C_4/C_8 mixture and 0.14 for the NH_3/H_2O one. To compensate this low value the rich solution flow rate for

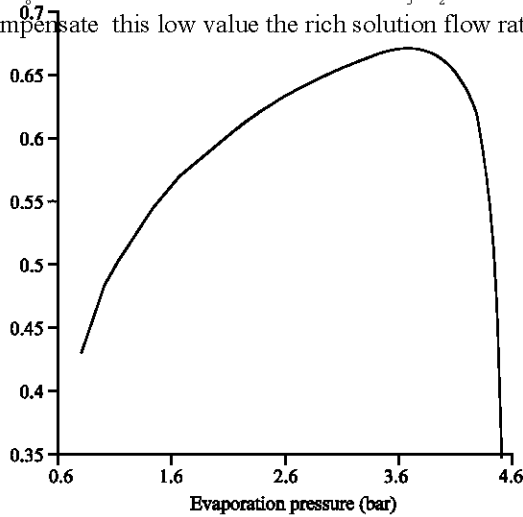


Fig. 6: Variation of the COP versus P_E for the NH_3/H_2O system (water cooling)

Table 2: Simulated working parameters for the two mixtures (air cooling)

| Parameters | C_4/C_8 | NH_3/H_2O |
|------------------------------------|-----------|-------------|
| x_r (mol) | 0.990 | 0.990 |
| x_c (mol) | 0.220 | 0.380 |
| x_p (mol) | 0.130 | 0.240 |
| P_E (bar) | 1.000 | 3.400 |
| P_C (bar) | 4.800 | 19.820 |
| \dot{L}_r (mol s ⁻¹) | 0.920 | 0.880 |
| \dot{L}_c (mol s ⁻¹) | 9.010 | 4.750 |
| \dot{L}_p (mol s ⁻¹) | 8.100 | 3.870 |
| \dot{Q}_E (kW) | 17.500 | 17.500 |
| \dot{Q}_B (kW) | 46.770 | 33.610 |
| \dot{Q}_{CP} (kW) | 16.670 | 7.000 |
| \dot{Q}_C (kW) | 20.260 | 17.910 |
| \dot{Q}_A (kW) | 27.650 | 26.290 |
| \dot{W} (kW) | 0.314 | 0.099 |
| COP | 0.372 | 0.521 |

Table 3: Simulated working parameters for the two mixtures (water cooling)

| Parameters | C_4/C_8 | NH_3/H_2O |
|------------------------------------|-----------|-------------|
| x_r (mol) | 0.990 | 0.990 |
| x_c (mol) | 0.360 | 0.520 |
| x_p (mol) | 0.130 | 0.210 |
| P_E (bar) | 0.950 | 3.700 |
| P_C (bar) | 2.820 | 11.700 |
| \dot{L}_r (mol s ⁻¹) | 0.880 | 0.900 |
| \dot{L}_c (mol s ⁻¹) | 3.320 | 2.270 |
| \dot{L}_p (mol s ⁻¹) | 2.430 | 1.370 |
| \dot{Q}_E (kW) | 17.500 | 17.500 |
| \dot{Q}_B (kW) | 28.040 | 26.000 |
| \dot{Q}_{CP} (kW) | 4.310 | 0.020 |
| \dot{Q}_C (kW) | 20.880 | 19.740 |
| \dot{Q}_A (kW) | 20.400 | 23.770 |

| | | |
|----------------|-------|-------|
| \dot{W} (kW) | 0.050 | 0.020 |
| COP | 0.623 | 0.672 |

the butane is higher than that of the ammonia what makes the quantity of heat required at the reboiler Q_B more important and consequently the COP lower.

Concerning the second working mode the difference of the two degasification rates are not very important, so the rich solution flow rates have comparable values as well as the quantity of heat Q_B . This leads to similar performances (0.62 and 0.67). Even though, the COP of this new refrigerant mixture is lower than that of the classical ammonia solution one, it offers some advantages such as working under moderate pressure. The evaporation pressure is 1 bar for the alkane pair, while it is 3.4 bar for the second. The condensation pressure is respectively 4.8 bar for the first mixture and 19.8 bar for the second. Operating at moderate pressures doesn't require high technology of airtightness, making easier the construction of the machine and reducing its price. This harmless natural combination is an interesting alternative since the machine working with this mixture can use a heat driven source at moderate temperature such as a thermal waste or a vacuum solar panel (Kim *et al.*, 2005; Zetzsche *et al.*, 2007).

The alkanes are also not as corrosive as the ammonia making possible the construction of the machine with the copper, having better thermal conductivity compared to the steel.

CONCLUSION

The purpose of this study is to compare the research of an absorption machine with two mixtures: the C_4/C_8 and the NH_3/H_2O . The proposed machine is a classical simple effect chiller simulated with two cooling mediums for the condenser and the absorber: the ambient air and the water. The PC-SAFT equation of state is used to evaluate the thermodynamic properties of the working fluids in the different components of the machine as well as the liquid-vapor equilibrium. We have simulate the research of the machine first with the butane/octane mixture and then with the ammonia/water one at the same operating conditions and successively with the two cooling modes. The performance is determined by calculating the COP. For an air-cooled machine, the COP of the first pair is estimated to be 0.37. This performance is 28.6% lower than that of the second pair having a COP of 0.52. This is due to the differences in the enthalpy of vaporization in the degasification rate and in the solution flow rates. When the water is the cooling medium, the two mixtures lead to comparable performances. The performance of this mixture can be improved by a double effect machine or by

modification of the initial configuration to reduce the loss of the energy.

NOMENCLATURE

| | | |
|-----------------|---|---|
| COP | = | Coefficient of performance |
| \dot{L} | = | Molar liquid flow rate (mol s^{-1}) |
| P_C | = | Condensation pressure (bar) |
| P_E | = | Evaporation pressure (bar) |
| \dot{Q} | = | Thermal power (W) |
| ΔT_{SR} | = | Sub-cooling ($^{\circ}\text{C}$) |
| T_a | = | Ambient temperature ($^{\circ}\text{C}$) |
| T_C | = | Condensation temperature ($^{\circ}\text{C}$) |
| T_i | = | Temperature in point i ($^{\circ}\text{C}$) |
| ΔT | = | Thermal pinch ($^{\circ}\text{C}$) |
| \dot{V} | = | Molar vapor flow rate (mol s^{-1}) |
| \dot{W}_p | = | Mechanical power of the pump (W) |
| x | = | Liquid molar fraction of refrigerant |
| y | = | Vapor molar fraction of refrigerant |

Indices

| | | |
|----|---|-------------------|
| A | = | Absorber |
| B | = | Reboiler |
| C | = | Condenser |
| CP | = | Partial condenser |
| E | = | Evaporator |
| G | = | Generator |
| f | = | Refrigerant |
| p | = | Poor |
| r | = | Rich |

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