SIMULATION OF AN AMMONIA WATER COOLED CHILLER

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Abstract: The present study aims to develop a simple simulation method for an ammonia water cooled chiller. This method is based on the thermodynamic principals and heat transfer fundamental laws. The performances of the chiller have been looked at as function of the water temperatures at the entry of the evaporator and the condenser. These temperatures are considered as the main parameters of the simulation. The chiller uses ammonia, a natural environment friendly refrigerant, under study to replace R22 for these applications. The calculation of thermodynamic properties of the refrigerant, necessary for the simulation, is carried out by simple and reliable local equations of state. The results show that the COP of the system increase slightly when increasing the evaporator water inlet temperature and decrease when increasing the condenser inlet water temperature.

Keywords: ***Chiller, Simulation, Ammonia, Equations of State.***

1 INTRODUCTION

The increasing concerns of energy saving and environmental issues, as a consequence of the continuous industry growing, is by a large part due to the energy consume of the refrigeration and air-conditioning systems. The sustainable development, permitting to satisfy of the present necessities without putting at risk the life on the earth, has forced the refrigeration and air-conditioning industries to focus their efforts to develop more efficient ways to use energy with respect to the environment. Hence, ammonia chillers can represent a viable alternative. Ammonia systems can significantly improve reliability and energy efficiency compared with standard chiller arrangements.
All chillers require a working fluid to transfer excess energy from one heat source to another. Traditionally, R22 was generally accepted as the more adapted refrigerant in the applications of refrigeration, heat pumps and air conditioning. Because of the harmful effects of R22 on the environment, it has been gradually phased out of production. A big attention is attracted to natural refrigerants to encourage and promote measures to sustainable development. Among these refrigerants, ammonia was commonly accepted as the refrigerant of choice in large commercial and industrial refrigeration units. The importance of ammonia as refrigerant in food refrigeration applications is well known (Korfitsen and Kristensen, 1998). Nowadays, several research groups activate to use ammonia in small capacity refrigeration and heat pump systems. However, as ammonia is toxic and flammable, a number of design and safety issues need to be considered before incorporating the chillers into the building.
Simulation is a powerful and economic tool to carry out research in the area of refrigeration and heat pumps. It is possible to avoid the recourse to experimental studies on model tests which are rarely available for researchers. There are also a lot of variables to be measured in different configurations, which are consuming time and money.

It is necessary to be able to model and simulate chillers in order to get better understanding of their behaviour and to build tools to optimise their use. Recently, heat pumps and chillers have been a topic of a number of simulation models. Gordon and Choon (1994) have modelled thermodynamically reciprocating chillers by a simple model. Jin and Spliter (2002) have presented a steady-state simulation model for a water-to-water reciprocating vapour compression heat pump. The model includes several unspecified parameters that are estimated from manufacturers' catalogue data using a multi-variable optimization procedure. Zhao *et al.* (2003) have simulated a geothermal heat pump with a non-azeotropic mixture. The model is modified and verified with experimental data. Rajapaksha and Suen (2004) have analysed the steady state performance of a reversible water-to-water heat pump using a computer simulation. The simulation adopts a distributed parameter modelling approach using two refrigerants, R407C and R134a. Youbi-Idrissi *et al.* (2005) have proposed a local simulation model of a water-to-water heat pump by adopting a modular approach. The model consists of the association of an elementary model for each basic component using a ternary zeotropic blend R-134a/R-125/R-32. The models used require either experimental data or manufacturer's data. They require also more computing time.
The aim of this paper is to develop a simple thermodynamic model for the simulation of an ammonia water cooled chiller using ammonia as refrigerant. This model is based on the thermodynamic principals and heat transfer fundamental laws. This model is simple and it requires few experimental data. The parameters of the analysis are the inlet water temperatures of the evaporator and the condenser. The circuit of this heat pump uses ammonia which is a refrigerant with zero ODP and GWP (Table 1) compared to R22.
The use of reliable thermodynamic data of the refrigerant is integrated in the thermodynamic model through a set of local equations of state.

Table 1: Principal characteristics of R22 and R717

	R22	**R717**
ODP	0.055	0
GWP	1300	0
Molecular mass (kg/kmol)	86.5	17.03
Normal boiling point (°C)	-40.8	-33.3
Critical temperature (°C)	96.0	132.4
COP^{*}	5.93	6.18

*for an ideal single vapour compression chiller operating between evaporation and condensation temperatures of 2° C and 40° C respectively.

2 REFRIGERANT PROPERTIES

Equations of state are particularly convenient to make property and phase equilibrium calculations for pure refrigerants and their mixtures. The computational model adopted in this study is based on four local equations of state presented below (Ouadha *et al.,* 2005):

- an equation of state for the gas state,
  $$Z = Z(\theta, \omega) \tag{1}$$
- a correlation for the saturated vapour pressure,
  $$p_s = p_s(T_r) \tag{2}$$
- a correlation for the saturated liquid density,
  $$\rho_L = \rho_L(T_r) \tag{3}$$
- an equation of the specific heat capacity at constant pressure in the ideal gas state.
  $$c_p^0 = c_p^0(T_r) \tag{4}$$

Using the four basic equations (1) to (4) and the differential equations of thermodynamics (Syechev, 1983), it is possible to calculate the other essential thermodynamic functions necessary for the simulation, namely, the enthalpy and the entropy.

3 SIMULATION METHOD

In order to improve the design of efficient chillers, it is important to develop simulation methods enabling computation of the main performance as function of parameters imposed by the environment. For this purpose, a simulation method of an ammonia water cooled chiller is developed. The details of this method are presented in the following sections.

3.1 System description

A chiller is a system that removes heat from a liquid via a vapour compression or absorption refrigeration cycle. Most often water is chilled. Chilled water can be used to cool and dehumidify air in mid to large size commercial, industrial, and institutional facilities. Chillers are a sustainable method of cooling. Their use reduces air pollution by the reduction of the consumption of the fossil energy.
Figure 1 shows the system considered in the present study. It consists of a single stage vapour compression water cooled chiller comprising a compressor, a condenser, an expansion device and an evaporator.
The refrigerant, at superheated vapour state (1), enters the compressor where it is compressed to reach the condenser pressure (1-2). The high pressure superheated vapour enters in the condenser, where it is de-superheated and condensed (2-4). The refrigerant is subcooled (4-5) before being throttled to the evaporator (5-6). Finally, the refrigerant enters the evaporator, where it absorbs heat (6-7). Before entering again in the compressor, the refrigerant is superheated by absorbing more heat in the suction line (7-1). An internal heat exchanger (IHEX, Liquid Line/Suction Line Heat Exchanger) is used to superheat the suction gas while subcooling the condensate upstream of the expansion valve. The incorporation of this heat exchanger generally ameliorates the performances of the system (Domanski (1995); Klein *et al.* (2000)).
The refrigeration effect is $h_1 - h_6$ and the heat rejected to the heat sink is $h_2 - h_5$. While the work supplied to the compressor is $h_2 - h_1$.

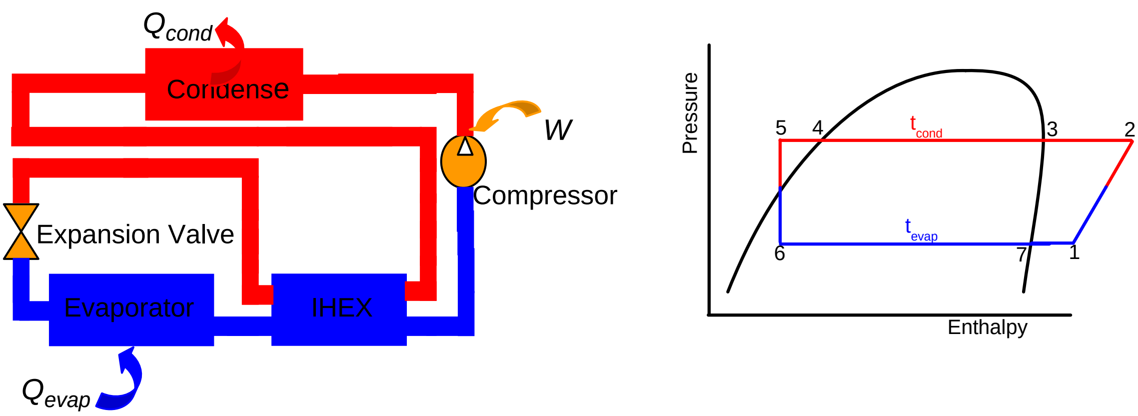

Figure 1: Scheme and cycle of a water cooled chiller with an internal heat exchanger.

3.2 Energy balance of the chiller

The entire system has been simulated on the basis of energy balance and heat transfer relationships. The energy balance of the water cooled chiller is established using the following assumptions:

- each component of the system is analyzed as a control volume at steady state.
- heat losses in the liquid line/suction line heat exchanger are neglected (all the heat used for the superheating of the vapour before the compressor is taken from subcooling the liquid after the condenser). Thus, only the compressor thermal dissipations are included.
- pressure drops in the suction line and discharge line are ignored.
- refrigerant flows at constant pressure in the evaporator and condenser .

The thermal load of the condenser is:

$$Q_{cond} = \left(\dot{W} - Q_{loss}\right) + Q_{evap} \tag{5}$$

The condenser and evaporator loads can be calculated by the following expressions:

$$Q_{cond} = K_{w,cond} \cdot S_{cond} \cdot LMTD_{cond} \tag{6}$$

$$Q_{evap} = K_{w,evap} \cdot S_{evap} \cdot LMTD_{evap} \tag{7}$$

Where $LMTD_{cond}$ and $LMTD_{evap}$ are the logarithmic mean temperature differences in condenser and evaporator, respectively. They are given by:

$$LMTD_{cond} = \frac{T_{o,cond} - T_{i,cond}}{\log\left(\frac{T_{cond} - T_{i,cond}}{T_{cond} - T_{o,cond}}\right)} \tag{8}$$

$$LMTD_{evap} = \frac{T_{o,evap} - T_{i,evap}}{\log\left(\frac{T_{i,evap} - T_{evap}}{T_{o,evap} - T_{evap}}\right)} \tag{9}$$

The condenser and evaporator loads can be also calculated from the temperature change of the secondary heat transfer fluid (water), the mass flow rate and the specific heat capacity of this heat transfer fluid. They are given by:

$$Q_{cond} = \dot{m}_{w,cond} \cdot C_{w,cond} \cdot \left(T_{o,cond} - T_{i,cond}\right) \tag{10}$$

$$Q_{evap} = \dot{m}_{w,evap} \cdot C_{w,evap} \cdot \left(T_{o,evap} - T_{i,evap}\right) \tag{11}$$

Using the enthalpy change of the refrigerant:

$$Q_{cond} = \dot{m} \cdot \left(h_2 - h_5\right) \tag{12}$$

$$Q_{evap} = \dot{m} \cdot \left(h_1 - h_6\right) \tag{13}$$

The overall heat transfer coefficient of the condenser is given by:

$$K_{w,cond} = \frac{Q_{cond}}{S_{cond} \cdot LMTD_{cond}} \tag{14}$$

3.3 Compressor model

As the compressor is the most important element of the chiller, it is imperative to pay closer attention to its volumetric characteristics calculation.
The mass flow rate of the refrigerant can be calculated as:

$$\dot{m} = \rho_1 \cdot \dot{V}_s \cdot \eta_{vol} \tag{15}$$

where ρ_1 is the inlet density, $\dot{V}_s$ is the displacement of the compressor in m^3/s and η_{vol} is the volumetric efficiency.
The power consumed by the compressor:

$$\dot{W} = \frac{\dot{m} \cdot \left(h_{2s} - h_1\right)}{\eta_{comp}} \tag{16}$$

where η_{comp} is the compressor efficiency, h_1 is the inlet enthalpy and h_{2s} is the isentropic outlet enthalpy.

The outlet enthalpy of the compressor:

$$h_2 = h_1 + \frac{h_{2s} - h_1}{\eta_{comp}}(1-\xi) \quad (17)$$

where ξ is the heat losses ratio to the environment which is imposed and equal to 10%. η_{comp} and η_{vol} standing for volumetric and compressor efficiencies, calculated using the following expressions (Lee et al., 2006):

$$\eta_{vol} = 1.03231 - 0.05080\,\tau - 0.00076\tau^2 \quad (18)$$

$$\eta_{vol} = 0.83955 - 0.01026\,\tau - 0.00097\,\tau^2 \quad (19)$$

where, τ is the pressure ratio which is the ratio of condenser pressure to evaporator pressure.

4 NUMERICAL PROCEDURE

The simulation of the water cooled chiller is carried out by combining and solving equations (6) and (7). Starting from the given values of $T_{i,evap}$ and $T_{i,cond}$ and an initial value of T_{evap}, parameters at states 1 and 2 are completely determined. The temperature of condensation is calculated using a simple transformation of the equations (5), (6) and (7):

$$T_{cond} = \frac{T_{i,cond} - T_{o,evap} \cdot e^{\frac{K_{w,evap} \cdot S_{cond} \cdot (T_{o,cond} - T_{i,cond})}{\dot{W} + Q_{evap}}}}{1 - e^{\frac{K_{w,evap} \cdot S_{cond} \cdot (T_{o,cond} - T_{i,cond})}{\dot{W} + Q_{evap}}}} \quad (20)$$

Temperatures $T_{i,evap}$ and $T_{o,cond}$ are determined using:

$$\dot{m}_{w,evap} \cdot C_{w,evap} \cdot (T_{evap} - T_{i,evap}) = K_{w,evap} \cdot S_{evap} \cdot LMTD_{evap} \quad (21)$$

$$T_{o,cond} = \frac{Q_{cond}}{C_{w,cond}\,\dot{m}_{w,cond}} + T_{i,cond} \quad (22)$$

5 RESULTS AND DISCUSSION

A computer programme was developed to calculate the temperature of condensation T_{cond} and the performances of the heat pump.
The analysis was carried out for a water cooled chiller using ammonia as refrigerant. Table 2 presents the design parameters of the refrigerants and the system.

Table 2: Design parameters of the system

Condenser	
area (m^2)	9.4
Mass flow rate of water (kg/s)	1.2
Specific heat capacity of water (kJ/kg.K)	4.185
Evaporator	
area (m^2)	4.0
Mass flow rate of water (kg/s)	1.9
Heat transfer coefficient ($kW/m^2.K$)	1.8

5.1 Influence of the evaporator inlet water temperature

In this case, the inlet temperature of water entering the condenser was maintained at 298 K, while the evaporator inlet water temperature varies from 285 to 293 K.

Figure 2 shows the variation of condensation and evaporation temperatures with the evaporator inlet water temperature $T_{i,evap}$. The increase of the evaporator inlet water temperature yields in a linear increase in the evaporation and condensation temperatures. This increase is more significant for evaporation temperature than condensation temperature. Evaporation temperature varies from 279.5 K for an evaporator inlet water temperature equals to 285 K to 286 K for an evaporator inlet water temperature of 293 K while the condensation temperature ranging from 311 K to 314 K.

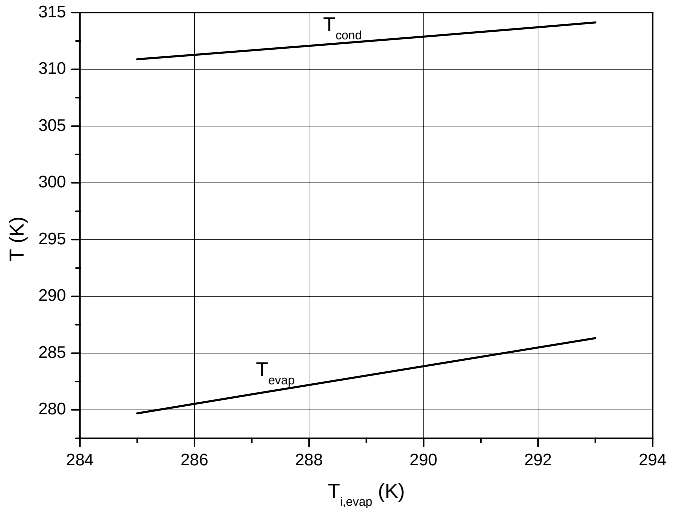

Figure 2: Influence of $T_{i,evap}$ on evaporator and condenser temperatures.

Condenser capacity (Q_{cond}), evaporator capacity (Q_{evap}) and power consumed by the compressor ($\dot{W}$) are linearly proportional to the evaporator inlet water temperature. The power consumed by the compressor varies slightly with the variation of the evaporator inlet water temperature. Condenser and evaporator capacities increase with the increase of the evaporator inlet water temperature (Figure 3). It is obvious that the variation of the evaporator inlet water temperature affects directly the evaporator capacity Q_{evap}. The increase in Q_{evap} involves an increase in condensation temperature and by the way an increase in the condenser capacity (Q_{cond}).

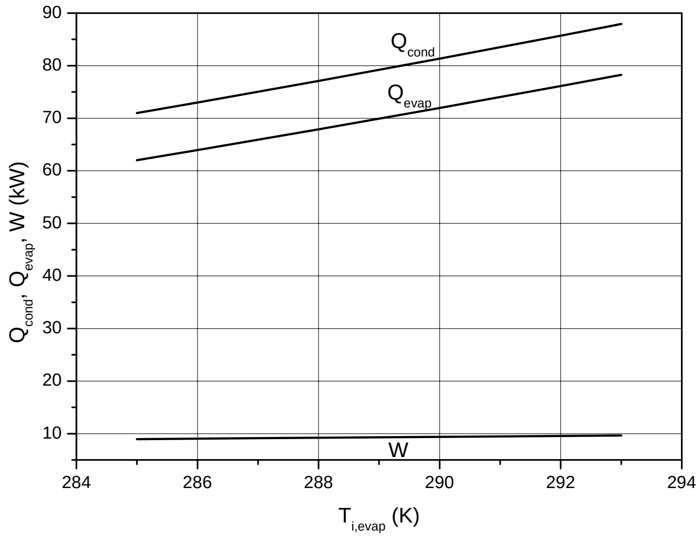

Figure 3: Influence of $T_{i,evap}$ on evaporator and condenser capacities and the power consumed by the compressor.

The system coefficient of performance (COP) is defined as the ratio between the cooling effect and the driving energy. It is plotted against the evaporator inlet water temperature in Figure 4. The increase in the coefficient of performance is mainly due to the increase in the cooling effect. This is because the variation in the power consumed by the compressor is minor. It passes from 6.93 to 8.12 in the same interval of variation of the evaporator inlet water temperature.

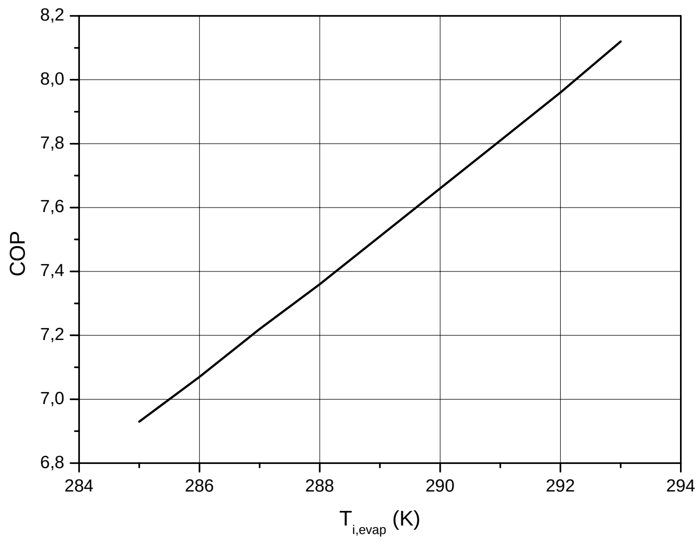

Figure 4: Influence of $T_{i,evap}$ on the coefficient of performance (COP).

5.2 Influence of the condenser inlet water temperature

Here, the inlet temperature of water entering the evaporator was maintained at 285 K, whereas the condenser inlet water temperature varies from 298 to 308 K.

The variation of the condensation and evaporation temperatures is plotted as function of the condenser inlet water temperature on Figure 5. The increase of the condenser inlet water temperature yields a linear increase in the condensation temperature, whereas, the evaporation temperature is almost constant.

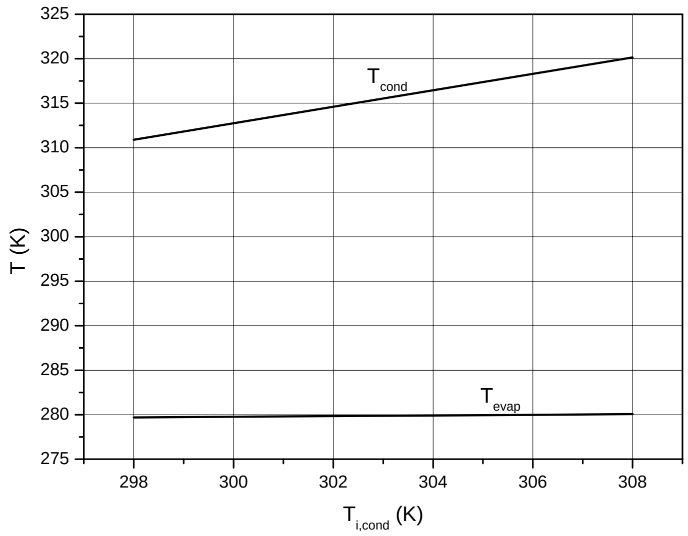

Figure 5: Influence of $T_{i,cond}$ on evaporator and condenser temperatures.

Condenser and evaporator capacities decrease with the increase of the condenser inlet water temperature, while power consumed by the compressor increases slightly (Figure 6). It varies from 8.92 to 11.25 kW in the interval of variation of the condenser inlet water temperature.

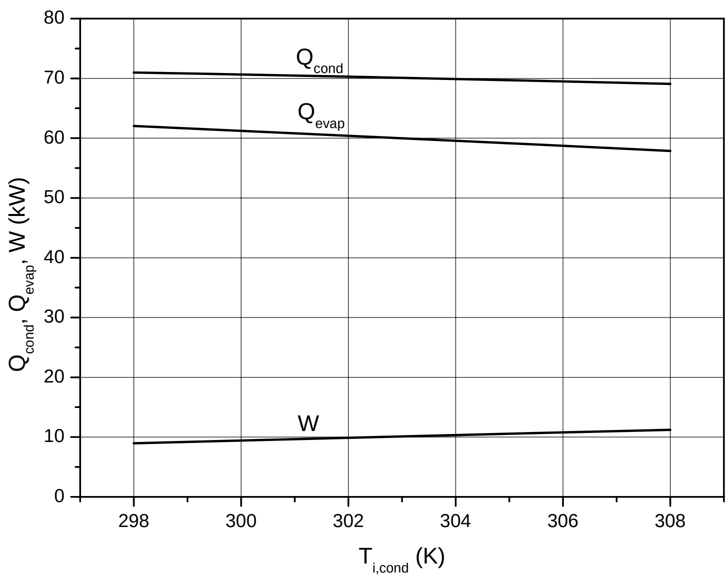

Figure 6: Influence of $T_{i,cond}$ on evaporator and condenser capacities and the power consumed by the compressor.

The decrease of the evaporator capacity produces a decrease in the coefficient of performance. Figure 7 shows the evolution of the coefficient of performance versus the condenser inlet water temperature. It drops from 6.92 to 5.16 in the same interval of variation of the condenser inlet water temperature.

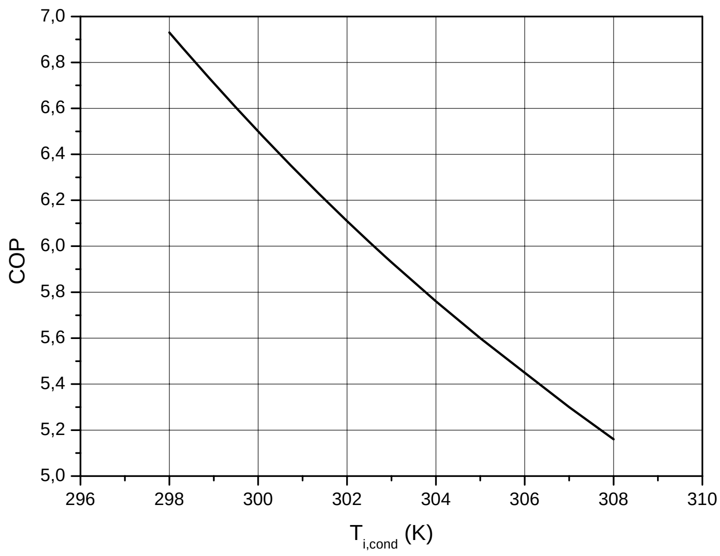

Figure 7: Influence of $T_{i,cond}$ on the coefficient of performance (COP).

The linearity is fully justified by the equations presented above. For example, equations (10) and (11) show clearly the linearity variation of Q_{cond} and Q_{evap} versus $T_{i,cond}$ and $T_{i,evap}$, respectively. As the COP, the evaporation and condensation temperatures result from combination of these parameters, theirs variations versus $T_{i,cond}$ and $T_{i,evap}$ should be linear.

6 CONCLUSIONS

A simulation method of an ammonia water cooled chiller is presented. It includes a thermodynamic model based on the heat transfer fundamental laws and energy balances and a parametric analysis.

The performances of the system are looked at as function of the water temperatures at the entry of the evaporator and the condenser. The calculation of thermodynamic properties of the refrigerant, necessary for the thermodynamic modelling, is carried out by simple and reliable local equations of state. The results show that the COP of the system increase slightly when increasing the evaporator water inlet temperature and decrease when increasing the condenser inlet water temperature.

NOMENCLATURE

C	Specific heat (kJ/kg K)
COP	Coefficient of Performance
GWP	Global Warming Potential
h	Specific enthalpy (kJ/kg)
K	Overall heat transfer Coefficient (kW/m^2.K)
$\dot{m}$	Mass flow rate (kg/s)
ODP	Ozone Depletion Potential
p	Pressure (MPa)
Q	Heat load (kW)
S	Area (m^2)
T	Temperature (K)
$\dot{V}_s$	Volume swept by the compressor (m^3)
v	Specific volume (m^3/kg)
w	Compressor work (kJ/kg)
$\dot{W}$	Power consumed by the compressor (kW)
Z	Coefficient of compressibility
Greek Symbols	
η	Efficiency (%)
ρ	Density (kg/m^3)
θ	Inverse of the reduced temperature
ω	Reduced density
Indices	
c	Cooling mode
$cond$	Condensation /Condenser
$comp$	Compressor
$evap$	Evaporation / Evaporator
h	Heating mode
$i,cond$	Inlet condenser
$i,evap$	Inlet evaporator
$o,cond$	Outlet condenser
$o,evap$	Outlet evaporator
r	Reduced
vol	Volumetric
w	Water

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