

MODELING AND SIMULATION OF A SINGLE STAGE ABSORPTION REFRIGERATOR

Luciana Cristina dos Santos Martinho, lesmartinho@gmail.com

José Viriato Coelho Vargas, jvargas@demec.ufpr.br

Department of Mechanical Engineering, Universidade Federal do Paraná – UFPR, Bloco IV do Setor de Tecnologia, Centro Politécnico (Campus II), Bairro: Jardim das Américas, Cx.: 19011, CEP: 81531-990-Curitiba – PR-Brasil.

Abstract. *This work develops a simplified transient mathematical model, which applies the mass, energy and species conservation principles to the components of a single stage ammonia-water absorption refrigeration system. A computer application was then developed to predict the response of the system as a whole and all components of the absorption cycle individually, i.e., condenser, evaporator, absorber, generator and rectifier. The model assigns thermodynamic control volumes to each component, therefore uniform properties are assumed within them, which yields a fourteen ordinary differential equations system, that are integrated in time using the adaptive time step Runge-Kutta Fehlberg fourth-fifth order method. An investigation of the model sensitivity was conducted, so that a qualitative assessment of how the model numerical results agree with the system expected response could be made, and the results showed that the physically expected trends were captured by the model. No experimental validation of the numerical results is presented at this initial stage. However, an absorption refrigerator system prototype is available at UFPR from a previous work, so that after future experimental validation, which is the next step of this ongoing investigation, it is expected that the proposed model becomes a useful tool for simulation, control, design and optimization of single-stage absorption refrigeration systems.*

Keywords: *absorption refrigeration; heat transfer; transient modeling*

1. INTRODUCTION

With the advent of electrical machinery, the use of absorption refrigeration systems (which have as energy source any fuel such as coal, wood or gas) became unattractive for a long period, since electrically driven vapor compression refrigeration systems are much more efficient. However, when the problems caused by prolonged use of Chlorofluorocarbon (CFC) and Hydrochlorofluorocarbon (HCFC) refrigerants, such as the destruction of the ozone layer, and possible global warming due to excessive fossil fuel burning, mainly in electrical energy production and transportation, became important, alternatives to vapor compression refrigeration systems started to be of interest again, since HVAC-R systems are responsible for a considerable part of the world electric grid consumption. Utility companies worldwide acknowledge that refrigeration and air conditioning systems are responsible for roughly 30% of total energy consumption, therefore unquestionably with a major impact on energy demand (Buzelin *et al.*, 2005).

In that context, the possibility of heat recovery from existing heat sources at no cost, such as waste heat and solar energy motivated the development of new technologies for absorption refrigeration systems. Although refrigeration systems with refrigeration capacity rates of 100 TR and above are currently being manufactured for commercial distribution, systems with low capacity are rare. Most of refrigeration and air-conditioning systems of low capacity rates are still based on the vapor compression cycle, which encourages scientific research on low capacity rate absorption refrigeration systems.

In absorption systems, a physicochemical process replaces the mechanical process of the vapor compression system by using energy in the form of heat rather than mechanical work (Moran and Shapiro, 2000). The absorption refrigeration ammonia-water cycle has some attractive features when compared to conventional compression cycle, e.g. no use of oil; if there is waste heat available, operating costs drop considerably; little consumption of electric energy and it produces no noise other than pumps (Hudson, 2002). The disadvantages of the absorption refrigeration systems include: higher capital investment is needed, the hardware is more complex and more space required. In Brazil the installed absorption refrigeration capacity is relatively small, despite the potential for it in different economy sectors (Cortez and Muhle, 1994).

Regarding modeling and simulation of absorption refrigeration systems, much studies have been published in the technical literature. However, most of them deal with steady state models (Herold *et al.*, 1996). Among dynamic simulation models, recently, Kim and Park (2007) presented a model for a particular type of absorption chiller, therefore not general for all types of single effect absorption refrigeration systems.

In the present study, motivated by the lack of general transient mathematical models for absorption refrigeration systems, a dynamic model for a single-effect ammonia-water absorption refrigeration system is developed. Ordinary differential equations emerge from mass, energy and species conservation principles applied to each component, and auxiliary algebraic equations (e.g., energy interactions, properties evaluation) are solved in parallel by numerical integration. The main objective is the development of a methodology to analyze the dynamic response of absorption refrigeration systems for simulation, design, control and optimization purposes.

2. THEORY

2.1. Description of a single-effect absorption chiller and components

The single stage absorption refrigeration system working with ammonia and water, illustrated by Fig. 1, consists of a condenser, an expansion valve, an evaporator and a thermal compressor. In this cycle, ammonia condenses (rejecting heat) and evaporates (extracting heat from the thermal load) similarly to a refrigeration cycle by mechanical compression. However, the thermal compressor located between the evaporator and condenser, performs vapor compression by using energy in the form of heat.

The thermal compressor consists of two unit operations: absorption and distillation. In the absorption, ammonia vapor is absorbed by water in a weak ammonia-water solution (absorber) rejecting heat and in the distillation process (generator and rectifier) the ammonia is separated from the strong ammonia-water solution consuming heat. For a better cycle efficiency, separation must be almost complete, and ammonia quality must be close to 1 (~0.9999) in the rectifier output (Herold *et.al.*, 1996). There are three components for performing the absorption and distillation processes: a pump, an expansion valve and a regenerative heat exchanger. The pump increases the pressure of the rich or strong solution formed in absorber, while the expansion valve reduces the pressure of the poor or weak solution that comes from the heat exchanger and generator. For improvement of the efficiency of the system a regenerative heat exchanger is used between the absorber and generator.

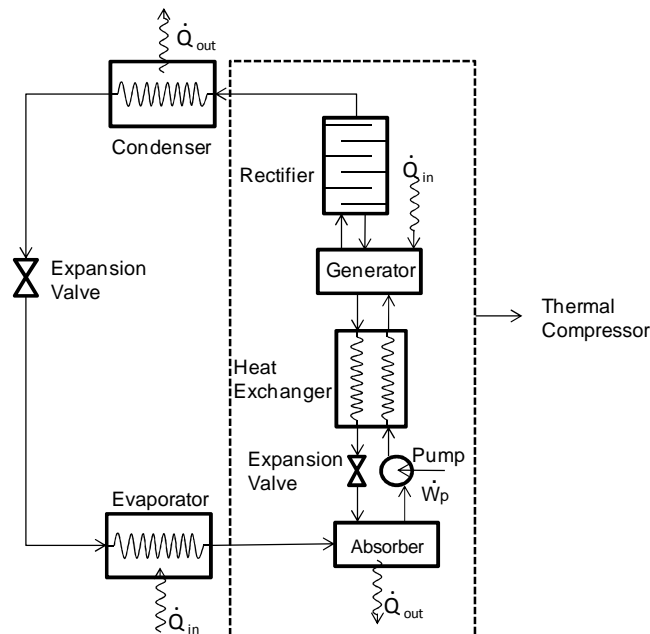


Figure 1. Schematic illustration of ammonia-water absorption refrigerator [adapted from Shapiro and Moran (2000)].

Electrical energy consumption in an absorption cycle is minimal when compared to a compression cycle, since only the pump uses this energy to raise the pressure of the liquid solution formed in the absorber. In a vapor compression cycle, the compressor consumes much more electrical energy to raise the pressure of the refrigerant vapor that comes out of the evaporator.

2.2. Mathematical model

A simplified model to predict the dynamic behavior of absorption chillers is proposed. Mass, energy, and species conservation principles are applied to each component. The following assumptions were considered:

1. Refrigerant and solution in each component are homogeneous, thus, each component is treated as a lumped system;
2. The refrigerant in the condenser and evaporator is considered as a pure and simple substance;
3. The fluid in the thermal compressor is treated as a solution of ammonia and water;
4. Kinetic and potential energy variations of the refrigerant are negligible with respect to internal energy variations;
5. The pressure drops in the components were assumed negligible and the pressure in each component is equal to the vapor pressure of the working solution, i.e., the condenser pressure is equal to the generator-rectifier pressure and the absorber pressure is equal to the evaporator pressure (Matsushima *et al.*, 2010);
6. The expansion valves operate as throttling thermodynamic processes;

7. The refrigerant in the condenser outlet is taken as a subcooled liquid;
8. The refrigerant in the evaporator outlet is taken as superheated vapor;
9. The solution leaving the absorber is at the same temperature and concentration as in the absorber;
10. The refrigerant leaving the rectifier is at the same temperature and concentration as in the rectifier;
11. Refrigerant mass in vapor or liquid phase are negligible with respect to mass in the region of phase change in the condenser and evaporator (Vargas and Parise, 1995);
12. The mass of vapor is negligible compared to the mass of the liquid phase in the thermal compressor;
13. Liquid phase is assumed to behave as an incompressible fluid, and
14. The pump is adiabatic and isenthalpic.

In order to apply the mass and energy conservation equations the absorption refrigeration cycle is divided into fourteen different control volumes. On the water side, four control volumes were established as follows: VC1, water side of the condenser; VC4, water side of the evaporator; VC6, water side of the absorber; VC13, water side of the rectifier. On the pure refrigerant side: VC2, refrigerant side of the condenser; VC3, expansion valve; VC5, refrigerant side of the evaporator. On the solution refrigerant-water side(thermal compressor): VC7, solution side of the absorber; VC8, expansion valve; VC9, pump; VC10, solution side of the heat exchanger (strong solution); VC11, solution side of the heat exchanger (weak solution); VC12, generator and VC14, solution side of the rectifier. A schematic of the absorption cycle, illustrating all control volumes, is seen in Fig. 2.

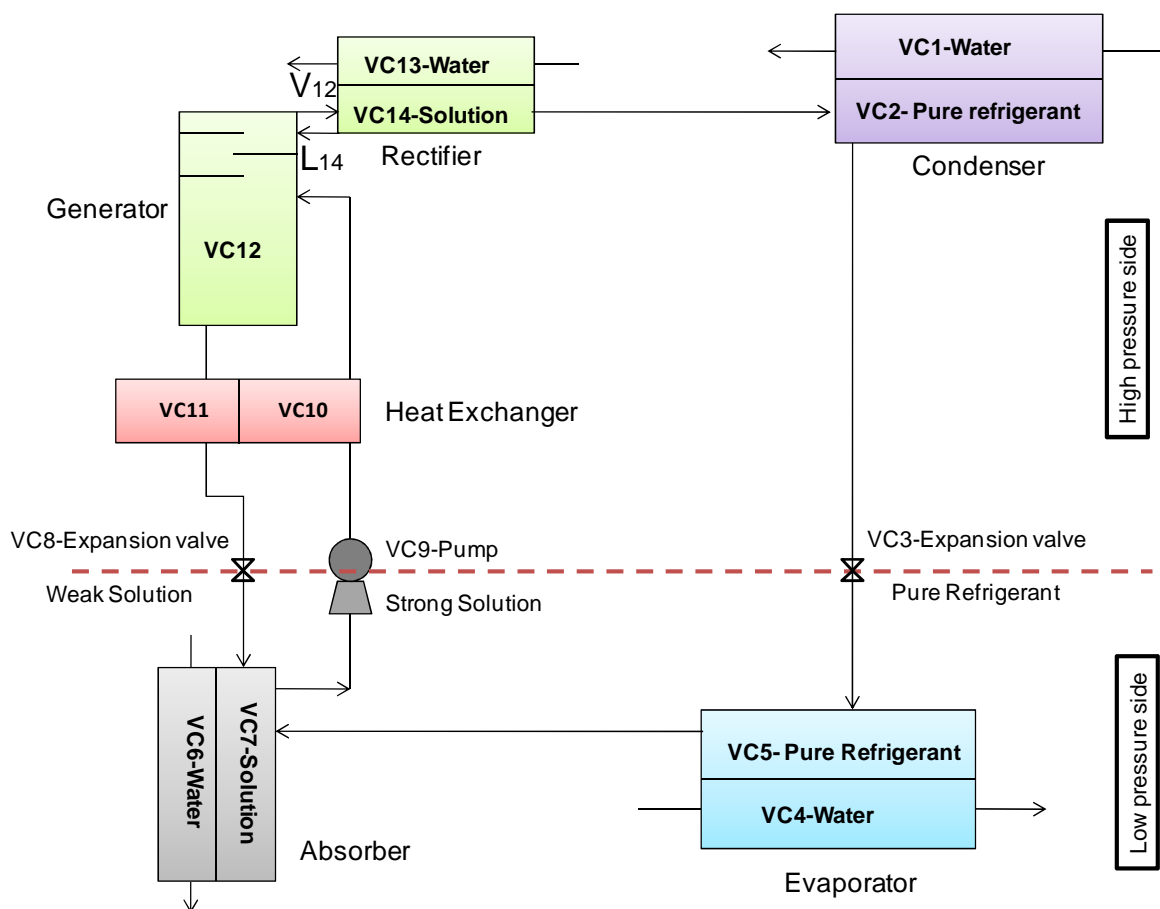


Figure 2. Schematic diagram of the absorption refrigeration system fourteen control volumes.

In Figure 2, V_{12} is the mass flow rate of vapor leaving the generator and L_{14} the mass flow rate of liquid leaving the rectifier.

The following subsections present the equations for each control volume.

2.2.1. Control volumes VC1 (condenser), VC4 (evaporator), VC6 (absorber), VC13 (rectifier) – water side

For the control volumes treated in this section, water is treated as an incompressible fluid. The energy balance reads as:

$$\frac{dT_i}{dt} = \frac{\dot{Q}_{VCi} + \dot{m}_{H_2O_i} \cdot C_{p_{H_2O}} \cdot (T_{in} - T_i)}{M_{H_2O_i} \cdot C_{p_{H_2O}}} \quad (1)$$

where:

T and t : temperature and time

$i = 1, 4, 6$ and 13 : control volume

M and \dot{m} : mass and mass flow rate

C_p : heat capacity

T_{in} : inlet temperature

\dot{Q}_{VCi} : net heat transfer rate in control volume i

2.2.2. Control volumes VC2 (condenser), VC3 (expansion valve), VC5 (evaporator) – pure refrigerant side

VC2 (condenser)

With the absorption refrigerator in operation, refrigerant enters in the same condition as the rectifier and exits as a subcooled liquid. A balance of energy (first law of thermodynamics), combined with mass conservation in the control volume provides the following equation:

$$\frac{dT_2}{dt} = \frac{-\dot{Q}_{VC2} + \dot{m}_{NH_3} (H_{14} - h_2)}{B \cdot M_{VC2}} \quad (2)$$

where:

T_2 and t : temperature in the condenser and time

M_{VC2} : refrigerant mass in the condenser

\dot{m}_{NH_3} : mass flow rate of the refrigerant

H_{14} : specific enthalpy of the refrigerant at the rectifier outlet

h_2 : specific enthalpy of refrigerant at the condenser outlet

$B(T_2)$: coefficient from the derivative of the internal energy with respect to time

\dot{Q}_{VC2} : net heat transfer rate in the condenser

The model considers uniform properties in the control volume, so for a complete phase change (quality of vapor is 1 at the input and 0 at the output) an average value for the refrigerant quality of 0.5 is assumed in the condenser. The refrigerant thermodynamic properties are approximated as the properties of a liquid-vapor mixture obtained from polynomial interpolations built with the thermodynamic table of saturated ammonia. For the subcooled liquid, the thermodynamic properties are assumed to be approximately the same as the saturated liquid at the same temperature of the compressed liquid at a known degree of subcooling. .

VC3 (expansion valve)

In the expansion valve refrigerant enters as subcooled liquid and exits as a liquid-vapor mixture. Applying the energy balance, the quality of the refrigerant at the evaporator inlet is given by:

$$y_3 = \frac{(h_2(T_{sc}) - h(T_5))}{(H(T_5) - h(T_5))} \quad (3)$$

where:

$T_{sc} = T_2 - \Delta T_{sr}$: temperature at the condenser outlet

ΔT_{sr} : degree of the subcooling

h_2 : specific enthalpy of refrigerant at the condenser outlet

h : specific enthalpy of saturated liquid at the evaporator

H : specific enthalpy of saturated vapor at the evaporator

T_5 : temperature in the evaporator

The mass flow rate of the refrigerant through the condenser and evaporator is given by mass conservation through the expansion valve orifice:

$$\dot{m}_{NH_3} = C_v \cdot A \cdot \left(\frac{2}{\gamma} (P_C - P_E) \right)^{1/2} \quad (4)$$

where:

v_2 : specific volume at the orifice

$P_C=P(T_2)$ and $P_E=P(T_5)$: pressure in the condenser and pressure in the evaporator, respectively

A : expansion valve area

C_V : valve coefficient

\dot{m}_{NH_3} : mass flow rate of refrigerant

VC5 (evaporator)

With the absorption refrigerator in operation, refrigerant enters as a liquid-vapor mixture and exits as superheated vapor. The resulting balance of energy leads to:

$$\frac{dT_5}{dt} = \frac{\dot{Q}_{VC5} + \dot{m}_{NH_3}(h_2 - H_5)}{C \cdot M_{VC5}} \quad (5)$$

where:

T_5 and t : temperature in the evaporator and time

$C(T_5)$: coefficient from the derivative of the internal energy with respect to time

H_5 : specific enthalpy of the refrigerant at the evaporator outlet

h_2 : specific enthalpy of the refrigerant at the condenser outlet

M_{VC5} : refrigerant mass in the evaporator

\dot{m}_{NH_3} : refrigerant mass flow rate

\dot{Q}_{VC5} : net heat transfer rate in the evaporator

In order to determine the specific enthalpy of refrigerant at the evaporator outlet as a superheated vapor, a *quasi* steady balance of energy in the thermal compressor states that, i.e.:

$$H_5 = \frac{\dot{Q}_{CT} + \dot{m}_{NH_3} H_{14}}{\dot{m}_{NH_3}} \quad (6)$$

where:

\dot{Q}_{CT} : net heat transfer rate in the thermal compressor

H_{14} : specific enthalpy of the refrigerant at the rectifier outlet

Similarly to the condenser, the refrigerant thermodynamic properties are approximated as the properties of a liquid-vapor mixture obtained from polynomial interpolations built with the thermodynamic table of saturated ammonia. The variation of the refrigerant quality across the control volume is taken as the arithmetic mean between inlet and outlet conditions of the refrigerant, given by:

$$y_5 = \frac{y_3 + 1}{2} \quad (7)$$

where:

y_5 : refrigerant quality in the evaporator

y_3 : refrigerant quality at the evaporator inlet

2.2.3. Control volumes VC7 (absorber), VC8 (expansion valve), VC9 (pump), VC10 (heat exchanger “strong solution”), VC11 (heat exchanger “weak solution”), VC12 (generator), VC14 (rectifier) - solution ammonia-water side (thermal compressor)

VC7 (absorber)

The differential equations for the control volume are established using the simplifying assumptions described in item 2.2. The following equations are obtained from mass and energy conservation principles applied to the control volume:

$$\frac{dx_7}{dt} = \frac{\dot{m}_{SFR} x_{12} + \dot{m}_{NH_3} - \dot{m}_{SFO} x_7}{M_{VC7}} \quad (8)$$

$$\frac{dT_7}{dt} = \frac{-\dot{Q}_{VC7} + \dot{m}_{SFR} h_{11} + \dot{m}_{NH_3} H_5 - \dot{m}_{SFO} h_7}{D \cdot M_{VC7}} \quad (9)$$

where:

x_7 and x_{12} : mass fraction of liquid ammonia in the absorber and generator

M_{VC7} : liquid mass in the absorber

\dot{m}_{NH_3} : refrigerant mass flow rate

T_7 : Temperature in the absorber

$D(T_7, x_7)$: coefficient from the derivative of the internal energy with respect to time

H_5 : specific enthalpy of refrigerant at the evaporator outlet

h_7 and h_{11} : specific enthalpies of the strong solution in the absorber and the weak solution in the heat exchanger

\dot{Q}_{VC7} : net heat transfer rate in the absorber

VC8 (expansion valve)

The mass flow rate of weak solution is calculated by mass conservation through the expansion valve orifice:

$$\dot{m}_{SFR} = C_V \cdot A \cdot \left(\frac{2}{\gamma_{SFR11}} (P_C - P_E) \right)^{1/2} \quad (10)$$

where:

v_{SFR11} : specific volume at the orifice

$P_C = P(T_2)$ and $P_E = P(T_5)$: pressure in the condenser and pressure in the evaporator

A : expansion valve area

C_V : valve coefficient

\dot{m}_{SFR} : mass flow rate of weak solution

VC9 (pump)

The energy balance provides the following equation to evaluate the strong solution mass flow rate:

$$\dot{m}_{SFO} = \frac{\rho_{SFO7} \cdot \dot{W}}{\eta_p \cdot (P_C - P_E)} \quad (11)$$

where:

ρ_{SFO7} : density of the strong solution (absorber)

$P_C = P(T_2)$ and $P_E = P(T_5)$: pressure in the condenser and pressure in the evaporator

\dot{W} and η_p : work rate and efficiency of pump

\dot{m}_{SFO} : mass flow rate of the strong solution

VC10 (regenerative heat exchanger – “strong solution” side) and VC11 (regenerative heat exchanger – “weak solution” side)

The balance of energy in the control volumes 10 and 11 are given by:

$$\frac{dT_{10}}{dt} = \frac{\dot{Q}_{VC10} + \dot{m}_{SFO} h_7 - \dot{m}_{SFO} h_{10}}{F \cdot M_{VC10}} \quad (12)$$

$$\frac{dT_{11}}{dt} = \frac{-\dot{Q}_{VC11} + \dot{m}_{SFR} h_{12} - \dot{m}_{SFR} h_{11}}{G \cdot M_{VC11}} \quad (13)$$

where:

T_{10} and T_{11} : temperature in the heat exchanger of strong and weak solution, respectively

M_{VC10} and M_{VC11} : liquid mass of strong and weak solution in the heat exchanger, respectively

\dot{m}_{SFO} and \dot{m}_{SFR} : mass flow rates of strong and weak solution, respectively

$F(T_{10}, x_7)$ and $G(T_{11}, x_{12})$: coefficients from the derivative of the internal energy with respect to time

h_7 and h_{11} : specific enthalpies of the strong solution in the absorber and weak solution heat exchanger, respectively

h_{10} and h_{12} : specific enthalpies of the weak solution in the heat exchanger and generator, respectively

\dot{Q}_{VC10} and \dot{Q}_{VC11} : heat transfer rate in the heat exchanger strong and weak solution sides, respectively

VC12 (generator)

In VC12, the refrigerant (ammonia) is separated from the water by heat input either from burning LPG (liquefied petroleum gas) or available waste heat (exhaust gases). This process results in two products: vapor (rich in ammonia) at the top of the component and weak solution (poor in ammonia) at the bottom. The mass and energy conservation principles applied to VC12 state that:

$$\frac{dx_{12}}{dt} = \frac{\dot{m}_{SFO}x_7 + L_{14}x_{14} - \dot{m}_{SFR}x_{12} - V_{12}y_{12}}{M_{VC12}} \quad (14)$$

$$\frac{dT_{12}}{dt} = \frac{\dot{Q}_{VC12} + \dot{m}_{SFO}h_{10} + L_{14}h_{14} - \dot{m}_{SFR}h_{12} - V_{12}H_{12}}{G \cdot M_{VC2}} \quad (15)$$

where:

x_{12} , x_7 and x_{14} : mass fraction of liquid ammonia in the generator, absorber and rectifier, respectively

y_{12} : quality of refrigerant in the generator

M_{VC12} : liquid mass in the generator

\dot{m}_{SFO} and \dot{m}_{SFR} : mass flow rate of strong solution and weak solution, respectively

L_{14} and V_{12} : mass flow rate of liquid in the rectifier and vapor in the generator, respectively

T_{12} : temperature in the generator

$G(T_{12}, x_{12})$: coefficient from derivative of the internal energy with respect to time

H_{12} : specific enthalpy of refrigerant vapor in the generator

h_{10} , h_{12} and h_{14} : specific enthalpies of strong solution in the heat exchanger, weak solution in the generator and liquid solution in the rectifier, respectively

\dot{Q}_{VC12} : heat transfer rate in the generator

VC14 (rectifier)

In VC14 vapor from generator is additionally cooled, so that practically all of the remaining water still present in the vapor mixture condenses returning to the generator. This condensate is expected to have a much higher concentration of water than the vapor phase. The mass and energy balances applied to VC14 state that:

$$\frac{dx_{14}}{dt} = \frac{V_{12}y_{12} - \dot{m}_{NH_3}y_{14} - L_{14}x_{14}}{M_{VC14}} \quad (16)$$

$$\frac{dT_{14}}{dt} = \frac{-\dot{Q}_{VC14} + V_{12}H_{12} - \dot{m}_{NH_3}H_{14} - L_{14}h_{14}}{J \cdot M_{VC14}} \quad (17)$$

where:

x_{14} : mass fraction for liquid ammonia in the rectifier

y_{14} and y_{12} : quality of refrigerant in the rectifier and generator, respectively

M_{VC14} : liquid mass in the rectifier

\dot{m}_{NH_3} : mass flow rate of refrigerant

T_{14} : temperature in the rectifier

$J(T_{14}, x_{14})$: coefficient from the derivative of the internal energy with respect to time

H_{12} and H_{14} : specific enthalpy of vapor in the generator and rectifier, respectively

\dot{Q}_{VC14} : heat transfer rate in the rectifier

2.2.4. Global energy balance

The global system is defined considering a control surface surrounding the entire refrigerator. For such system, the model assumes that the sum of the global system energy inputs is approximately equal to the sum of energy outputs at any instant of time, i.e., *quasi* steady operation, therefore a global energy balance states that:

$$\dot{Q}_{VC12} + \dot{W} + \dot{Q}_{VC5} = \dot{Q}_{VC2} + \dot{Q}_{VC7} + \dot{Q}_{VC14} \quad (18)$$

where:

\dot{Q}_{vc} and \dot{W} : heat and pump work

Equation (18) therefore assumes negligible time variation of the refrigerant internal energy, when compared with the other terms in the energy equation. Considering that the temperature time derivatives usually show opposite signs in the system components: condenser/evaporator and rectifier+absorber/generator, the time derivative terms of the internal energy tend to cancel out in the global system energy balance at any instant of time. It should also be noted that transient terms were, of course, kept in the energy balance of each individual heat exchanger.

The heat transfer rate for each control volume is given by:

$$\dot{Q}_{vc} = U \cdot A \cdot LMTD \tag{19}$$

where:

U , A and $LMTD$: global heat transfer coefficient, heat transfer area and Logarithmic Mean Temperature Difference

3. RESULTS AND DISCUSSION

The system of differential and algebraic equations, provided by the mathematical model, was integrated numerically in time using a Runge-Kutta Fehlberg fourth-fifth order method. An initial condition had to be specified to solve this initial value problem.

The physical parameters for the mathematical model are listed in Tables 1 and 2. For the calculation of thermodynamic properties of the mixture, both in the vapor and liquid phases, as well as the vapor-liquid equilibrium, the simplified formulation proposed by Pátek and Klomfar (1995) was used. The coefficient functions B, C, D, F, G, and J required by the model resulted from the differentiation with respect to time of the algebraic correlations used to determine thermodynamic properties.

Table 1. Data for water side of heat exchangers.

	Condenser	evaporator	absorber	Rectifier
Mass flow rate (kg/s)	2.5	1.5	2.5	1.0
Mass (kg)	40.0	35.0	40.0	10.0
heat capacity (kJ/kg K)		4.818		
Inlet temperature(K)		290.15		
Initial value (K)		298.15		

Table 2. Data for refrigerant.

	Condenser	evaporator	absorber	Generator	rectifier
Mass (kg)	35.0	40.0	40.0	40.0	20.0
Initial value temperature(K)	300.15	277.15	340.0	375.15	317.15
Initial value mass fraction	—	—	0.37	0.10	0.70

As design parameters, the following data were used: 0.0016 m of diameter for expansion valves and 3.0 kW pump power. Figures 3, 4 and 5 show the results obtained to simulate system operation up to 1000 seconds.

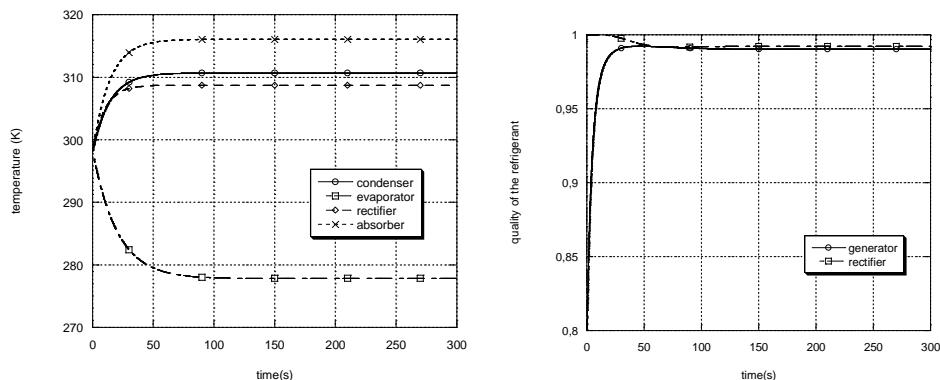


Figure 3. Numerical simulation for water side heat exchanger (left) and quality of the refrigerant in the generator and rectifier outputs (right).

Figure 3 shows the behavior for the temperatures of the: condenser, evaporator, rectifier and absorber (water side) and refrigerant quality at the outlets of the generator and rectifier. The steady state is established in about 100 seconds of operation in both cases. This is so fast probably due to the conditions fixed for the water (mass of control volume and mass flow rate) and assumption of *quasi* steady state for the generator and rectifier mass flow rate. Note that the quality of refrigerant from the rectifier is purer than the generator, indicating the condensation of water.

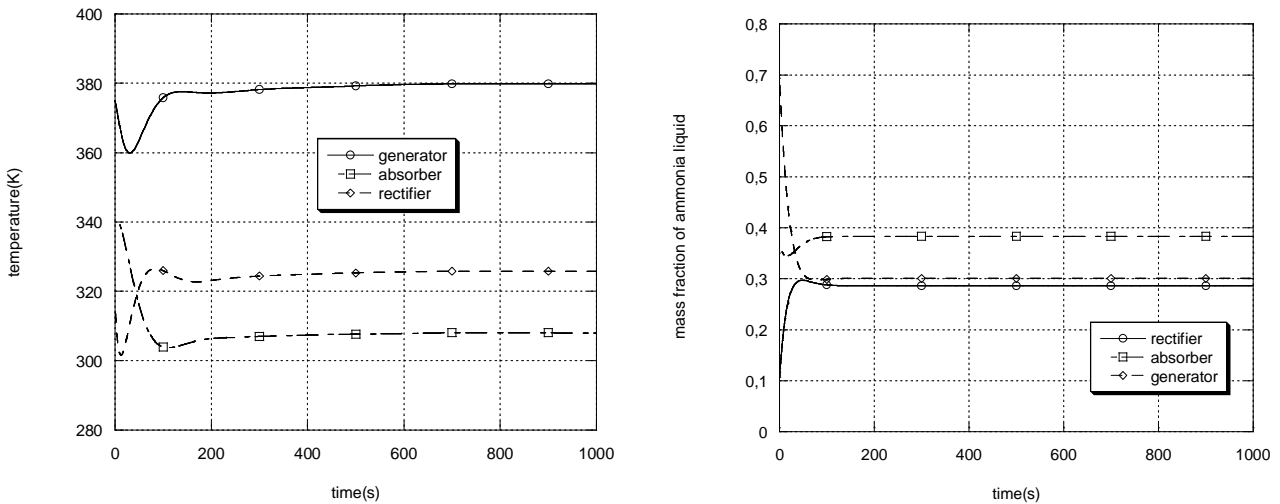


Figure 4. Numerical simulation of the thermal compressor: temperature (left) and composition of ammonia in the liquid phase (right).

For the thermal compressor results shown in Fig. 4, the steady state of temperature was reached in about 700 seconds of operation (380 K to the generator, 325 K to the rectifier and 308 K to the absorber) and for the mass fraction of liquid phase, steady state was established in about 100 seconds of operation (0.29 in the generator, 0.30 in the rectifier and 0.38 in the absorber).

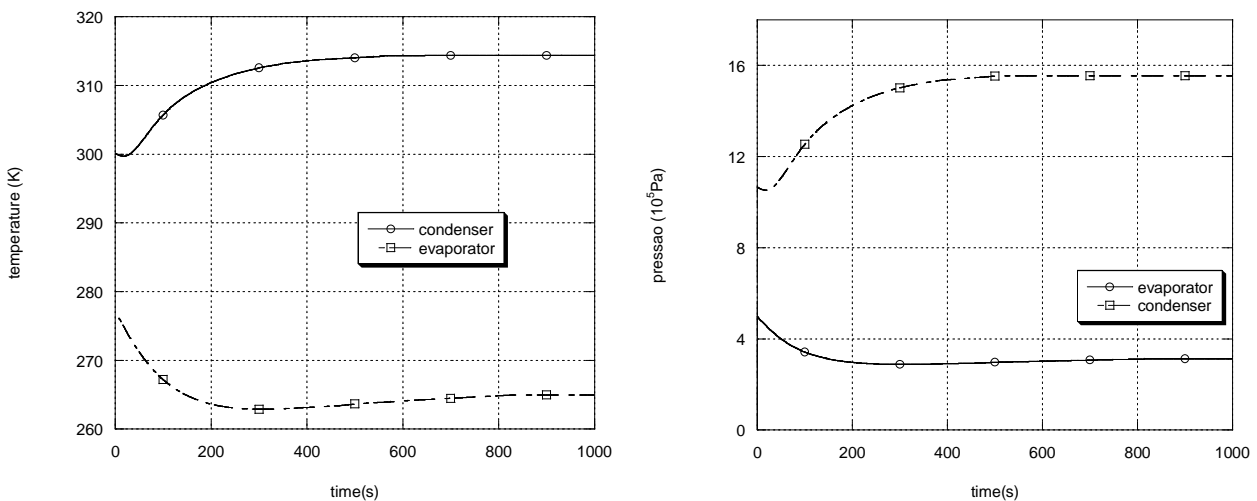


Figure 5. Numerical simulation for the refrigerant in the condenser and evaporator: temperature (left) and pressure (right).

Figure 5 shows the results for the region assumed to be pure refrigerant. The steady state regime of the absorption cycle happens after 900 seconds of operation, reaching temperatures and pressures: 314 K - 15 10⁵Pa to the condenser and 265 K - 3 10⁵Pa to the evaporator.

4. CONCLUSIONS

This paper presented a simplified transient mathematical model for a single stage absorption refrigeration cycle working with ammonia (refrigerant) and water (absorbent). The key conclusions of the simulation are:

- The heat exchangers water sides reach steady state quickly, since the mass flow rates and mass were fixed;
- The vapor quality at the rectifier output was approximately 0.999 indicating a good approximation to the hypothesis of pure refrigerant for the condenser and evaporator;

- The refrigerant composition reaches steady state before the thermal compressor temperature. This probably happens due to the *quasi* steady hypothesis used for the refrigerant mass flow rate;
- The thermal compressor start up transient is smaller than the pure refrigerant region start up transient, which indicates that Eq. (6) used to evaluate the enthalpy of superheated vapor at the evaporator outlet is appropriate to capture the actual system physical trend, and
- The temperature and pressure for the refrigerant obtained in the steady state are consistent with the case study presented by Herold *et al.* (1996).

Thus, it is expected that the proposed model, after experimental validation, could be a useful tool for simulation of single stage absorption refrigerators, and that subsequently could be used for system control, design and optimization.

5. ACKNOWLEDGEMENTS

The authors gratefully acknowledge support from CAPES, NILKO, NPDEAS(UFPR) and PG-Mec(UFPR).

6. REFERENCES

- Buzelin, L. O. S., Amico, S. C., Vargas, J. V. C., Parise, J. A. R., 2005, "Experimental development of an intelligent refrigeration system", *International Journal of Refrigeration*, Vol. 28, pp. 165–175.
- Cortez, L. A. B., Muhle, I. N., and Silva, A., 1994, "Refrigeração por absorção com o par água-amônia e seu potencial no caso brasileiro", *Revista ABRAVA*, pp. 33-38.
- Herold, K. E., Radermacher, R., and Klein, S. A., 1996, "Absorption Chillers and Heat Pumps", CRC Press, Florida, 182 p.
- Hudson, D. W., 2002, "Ammonia absorption refrigeration plant", *The Official Journal of AIRATH*, pp.26-30.
- Kim, B. and Park, J., 2007, "Dynamic simulation of a single-effect ammonia-water absorption chiller", *International Journal of Refrigeration*, Vol. 30, pp. 535-545.
- Moran, M. J., and Shapiro, H.N., 2000, "Fundamentals of engineering thermodynamics", John Wiley & Sons, 5th ed., 471 p.
- Matsushima, H., Fuji, T., Komatsu, T., and Nishiguchi, A., 2010, "Dynamic simulation program with object-oriented formulation for absorption chillers (modelling, verification, and application to triple-effect absorption chiller)", *International Journal of Refrigeration*, Vol. 33, pp. 259-268.
- Pátek, J. and Klomfar, J., 1995, "Simple functions for fast calculations of selected thermodynamic properties of the ammonia-water system, *International Journal of Refrigeration*, Vol 18, No. 4, pp. 228-234.
- Vargas, J. V. C., and Parise, J. A. R., 1995, "Simulation in transient regime of a heat pump with closed-loop and on-off control", *International Journal Refrigeration*, Vol. 18, pp. 235-243.

7. RESPONSIBILITY NOTICE

The authors are the only responsible for the printed material included in this paper.