



## DYNAMIC SIMULATION OF A SINGLE STAGE ABSORPTION REFRIGERATOR

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**Abstract.** *This work proposes the development of a simplified mathematical model, which combines empirical correlations with fundamental equations of mass and energy balance with the purpose to represent the behavior of refrigeration by absorption of a single stage, working with ammonia-water as fluids. The Simulator developed is used to predict the operation of key components of the absorption cycle: condenser, evaporator and thermal compressor ( absorber, generator and rectifier). The model considers the uniformity of the variables to the condenser and evaporator, which yields a eight ordinary differential equations system, which are integrated in time using the Runge-Kutta Fehlberg fourth-fifth order method, interacting with a thermal compressor modeled on the condition of “local steady-state” whose solution is obtained by the Newton-Raphson method. Appropriate dimensionless groups are identified and the results presented in normalized charts for general application. An investigation of the model sensitivity was conducted, so that a qualitative assessment of how the numerical model results agree with the system expected response could be made, and the results showed that the physically expected trends were captured by the model. As a result, it is expected that the proposed model is a useful tool for simulation of single-stage absorption chillers in operation, and that subsequently can be used for testing different conditions of operation and optimized procedures.*

**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).

Historically, chiller systems have been designed against fixed operating conditions, and many steady-state models have been developed as design aids or as tools to estimate energetic annual performance. However, chiller systems will typically operate under dynamic conditions, due to (for example) ‘On’/‘Off’ or modulating behaviour, start-up and shut down or other temporal fluctuations in operating conditions. Dynamic models, whilst intrinsically more complex, are

therefore a more appropriate means to assess the actual performance of absorption chillers within a trigeneration system reliant on transient heat input such as that supplied by solar energy and subject to a fluctuating load. Further, in integrated building-plant models dynamic modelling is the only accurate way of producing a true representation of the behaviour of the chiller (Borg *et. al.*, 2012).

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. From mass, energy and species conservation principles to applied to condenser, evaporator, refrigerated space and compressor thermal, and auxiliary algebraic equations (e.g., energy interactions, properties evaluation) eight ordinary differential equations and a system of nonlinear equations (3x3) emerge. The obtained numerical problem is solved by integrating the differential equations in the time and solving the nonlinear system at each time step. Appropriate dimensionless groups are identified and the results presented in normalized charts for general application. 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

### Description of a single-effect absorption chiller and components

The single stage absorption refrigeration system working with ammonia and water, illustrated by Fig. 1 (left), 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 ( $\sim 0.9999$ ) 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.

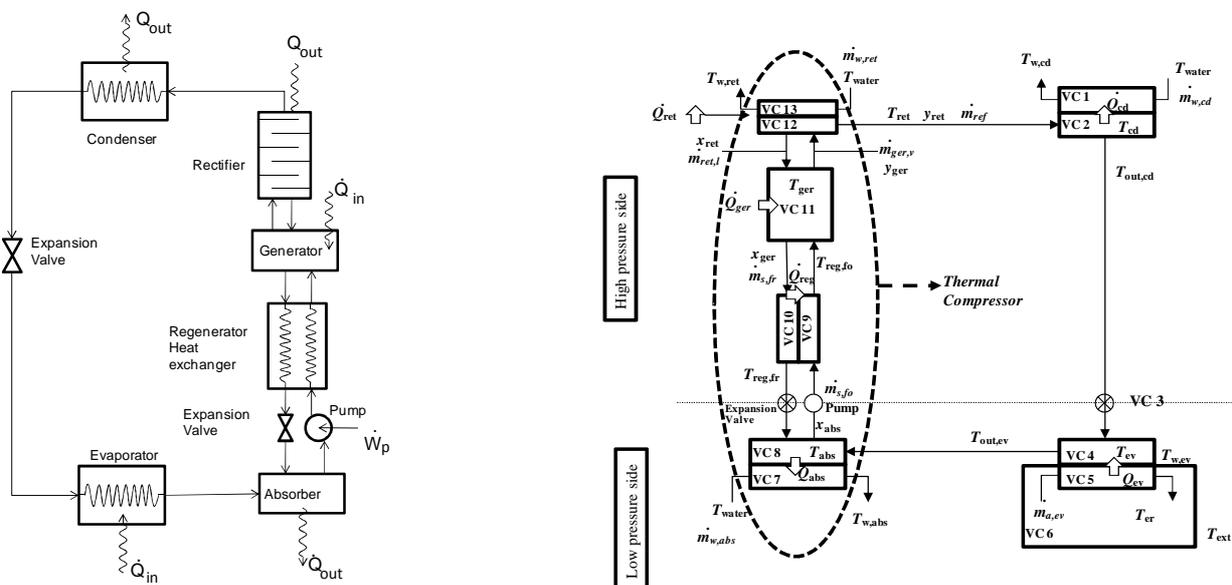


Figure 1. Schematic illustration of ammonia-water absorption refrigerator [adapted from Shapiro and Moran (2000)] (left), and Schematic diagram of the absorption refrigeration system thirteen control volumes.(right)

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.

## 2.1 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 model proposes two regions: region of refrigerant (condenser, expansion valve and evaporator) and solution region (thermal compressor).

In order to simplify the equations of mass and energy balances from region refrigerant, It is assumed that :

- (i) The refrigerant in the condenser and evaporator is considered as a pure and simple substance and each component is treated as a lumped system;
- (ii) Kinetic and potential energy variations of the refrigerant are negligible with respect to internal energy variations;
- (iii) The pressure drops in the components were assumed negligible. It is assumed that the cycle operates at two pressure levels: high pressure (saturation pressure of the condenser) and low pressure (saturation pressure in the evaporator). The condenser pressure is equal to the generator-rectifier pressure and the absorber pressure is equal to the evaporator pressure (Matsushima et al., 2010);
- (iv) The expansion valves operate as throttling thermodynamic processes;
- (v) The refrigerant in the condenser outlet is taken as a subcooled liquid ;
- (vi) The refrigerant in the evaporator outlet is taken as superheated vapor;
- (vii) Refrigerant mass in vapor or liquid phase are negligible with respect to mass in the region of phase change in the condenser and evaporator for the calculation of the internal energy (Vargas and Parise, 1995);
- (viii) Derived from the mass is negligible in relation to the derivative of the internal energy, both in relation to time.
- (ix) Negligible transients resulting from the filling of the cycle regarding the thermal inertia.

In order to apply the mass and energy conservation equations the absorption refrigeration cycle is divided into thirteen different control volumes. On the water side, five control volumes were established as follows: VC1, water side of the condenser; VC 5, water side of the evaporator; VC 6, refrigerated space, VC 7 water side of the absorber; VC 13, water side of the rectifier. On the pure refrigerant side: VC 2, refrigerant side of the condenser; VC 3, expansion valve; VC 4, refrigerant side of the evaporator. On the solution side(thermal compressor): VC 8, solution side of the absorber; VC 9, solution side of the regenerator heat exchanger (strong solution);VC 10, solution side of the regenerator heat exchanger (weak solution); VC 11, generator and VC 12, solution side of the rectifier. A schematic of the absorption cycle, illustrating all control volumes, is seen in Fig. 1 (right).

The following subsections present the equations for each control volume.

### 2.2.1. Dimensionless groups

In order to present general results for the system configuration proposed in Fig. 2, dimensionless variables are needed. First, since the thermal conductances ( $UA$ ) present in almost all control volumes are commodities in short supply, it makes sense to recognize the thermal conductance inventory (hardware) as a constraint. Therefore, one may define the total external conductance inventory,  $UA$  (hardware), as a design constraint:

$$UA = UA_{ev} + UA_{abs} + UA_{ger} + UA_{ret} + UA_{reg} \quad (1)$$

where: *ev*: evaporator; *abs*: absorber; *ger*: generator; *ret*: rectifier and *reg*: regenerator heat exchanger

As a result, for the whole system, dimensionless thermal conductances are defined by:

$$\gamma_i = \frac{(UA)_i}{UA} \quad (2)$$

Dimensionless heat transfer rate, temperatures, time, mass, pressure, enthalpy, internal energy, heat capacity rates and heat capacity are defined by:

where the subscript  $i$  indicates the control volume considered.

$$\tilde{Q}_i = \frac{\dot{Q}_i}{UAT_o}, \theta_i = \frac{T_i}{T_o}, \tilde{t} = \frac{t}{(m_{er}cp_a)/(UA)}, \tilde{m}_i = \frac{m_i}{m_{er}}, \tilde{P}_i = \frac{P_i}{P_o}, \tilde{h}_i = \frac{\bar{h}_i}{\bar{h}_o}, \tilde{U}_i = \frac{\bar{U}_i}{\bar{U}_o}, \psi_i = \frac{\dot{m}_i cp_i}{UA}, c\tilde{p}_i = \frac{cp_i}{cp_o} \quad (3)$$

$$T_o = 273.15 \text{ K}, UA = 3.006 \text{ kW/K}, m_{er} = 250 \text{ kg}, cp_o = 4.18 \text{ kJ/kg K}, P_o = 1.01310^5 \text{ Pa}, \bar{h}_o = 1011 \text{ kJ/kg}, \bar{U}_o = 1000 \text{ kJ/kg}$$

### 2.2.2. Control volumes VC1 (condenser), VC5 (evaporator), VC6 (refrigerated space) – water side

For these control volumes the water no phase change and acts as incompressible. Thus the balance of mass and energy combined result in:

$$\frac{d\theta_{w,cd}}{d\tilde{t}} = \frac{\tilde{Q}_{cd} + \psi_{cd}(\theta_{cist} - \theta_{w,cd})}{\tilde{m}_{w,cd} c\tilde{p}_w} \quad (4)$$

-VC 5

$$\frac{d\theta_{w,ev}}{d\tilde{t}} = \frac{-\tilde{Q}_{ev} + \psi_{ev}(\theta_{er} - \theta_{w,ev})}{\tilde{m}_{w,ev} c\tilde{p}_w} \quad (5)$$

-VC 6

$$\frac{d\theta_{er}}{d\tilde{t}} = \frac{\gamma_{parede}(\theta_{ext} - \theta_{er}) + \psi_{ev}(\theta_{ev} - \theta_{er})}{\tilde{m}_{er} c\tilde{p}_w} \quad (6)$$

### 2.2.3. Control volumes VC 2 (condenser), VC 3 (expansion valve), VC 4 (evaporator) – pure refrigerant side

-VC 2 (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:

$$\frac{d\tilde{U}_{cd}}{dt} = \frac{-\tilde{Q}_{cd} + \tilde{m}_{ref}(\tilde{h}_{ret,v} - \tilde{h}_{lsr})}{\tilde{m}_{cd}} \quad (7)$$

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.

The refrigerant internal energy of a liquid-vapor mixture is given by:

$$\tilde{U}_{cd} = y_{cd,vs} \tilde{U}_{vs}(\theta_{cd}) + (1 - y_{cd}) \tilde{U}_{ls}(\theta_{cd}) \quad (8)$$

$vs$ =saturated vapor;  $ls$ =saturated liquid ;  $y_{cd} = 0.5$ ,  $\tilde{U}_{vs} = a_1\theta_{cd}^2 + b_1\theta_{cd} + c_1$ ,  $\tilde{U}_{ls} = a_2\theta_{cd}^2 + b_2\theta_{cd} + c_2$

As the internal energies of saturated liquid and vapor are defined by polynomial interpolation (by way of example be considered polynomials of degree 2) and deriving the equation (8) with respect to time, we arrive at:

$$\frac{d\tilde{U}_{cd}}{d\tilde{t}} = (y_{cd}(2a_1\theta_{cd} + b_1) + (1 - y_{cd})(2a_2\theta_{cd} + b_2)) \frac{d\theta_{cd}}{d\tilde{t}} \quad (9)$$

Thus, the change in saturation temperature in the condenser in time as a function of temperature for saturation and the rate of heat transfer is defined as:

$$\frac{d\theta_{cd}}{dt} = \frac{d\tilde{U}_{cd} / d\tilde{t}}{(y_{cd}(2a_1\theta_{cd} + b_1) + (1 - y_{cd})(2a_2\theta_{cd} + b_2))} \quad (10)$$

- VC 3 (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:

$$y_{ev,in} = \frac{\tilde{h}_{lsr} - \tilde{h}_{ev,ls}}{\tilde{h}_{ev,vs} - \tilde{h}_{ev,ls}} \quad (11)$$

- VC 4 (evaporator)

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

$$\frac{d\tilde{U}_{ev}}{dt} = \frac{\tilde{Q}_{ev} + m_{ref}(\tilde{h}_{lsr} - \tilde{h}_{vsa})}{\tilde{m}_{ev}} \quad (12)$$

Using the same procedure described in the previous section it is possible to conclude that:

$$\frac{d\theta_{ev}}{dt} = \frac{d\tilde{U}_{ev} / d\tilde{t}}{(y_{ev}(2a_1\theta_{ev} + b_1) + (1-y_{ev})(2a_2\theta_{ev} + b_2))} \quad (13)$$

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 refrigerant. 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:

$$y_{ev} = \frac{y_{ev,in} + 1}{2} \quad (14)$$

The variation in time of the internal energy of the superheated refrigerant is assumed to be approximately the same as the variation in time of the internal energy of the saturated vapor in the evaporator. Thus the instantaneous internal energy of the refrigerant at exit from evaporator can be estimated from:

$$\frac{d\tilde{U}_{s,ev}}{d\tilde{t}} \approx \frac{d\tilde{U}_{vs,ev}}{d\tilde{t}} \quad (15)$$

Equation (15) neglects the pressure effect on the variation of internal energy in the superheated region, within a narrow range of temperatures that constitutes the degree of superheat at the evaporator outlet. In order to obtain the refrigerant temperature the evaporator outlet, the behavior of the refrigerant leaving the evaporator is approximated to that of an ideal gas. Therefore:

$$\frac{d\theta_{s,ev}}{d\tilde{t}} \approx \frac{d\tilde{U}_{vs,ev} / d\tilde{t}}{c\tilde{v}_{ev}} \quad (16)$$

## 2.2.4- Thermal Compressor

The strategy adopted in the modeling of the thermal compressor, control volumes 7-13, was to consider each component in the operating condition of a "steady state place". That is, the system of differential equations presented in the preceding sections interacts with this thermal compressor, making it also march in time, providing at each instant, the temperature and composition to the absorber, the generator and rectifier.

- Mass balances : Rectifier and Generator

Performing the mass balances, overall and by species, in the rectifier and working with these equations algebraically possible to obtain:

$$\dot{m}_{ret,l} = \frac{(y_{ret} - y_{ger})}{(y_{ger} - x_{ret})} \dot{m}_{ref} \quad (17)$$

Where:

$$rcr = \frac{(y_{ret} - y_{ger})}{(y_{ger} - x_{ret})} \quad (18)$$

$rcr$  circulation ratio from rectifier;  $x_{ret}$  liquid composition in rectifier

$$\dot{m}_{ret,l} = rcr \dot{m}_{ref} \quad (19)$$

$$\dot{m}_{ger,v} = (rcr + 1) \dot{m}_{ref} \quad (20)$$

Performing the mass balances, overall and by species, in the generator and working with these equations algebraically possible to obtain:

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$$\dot{m}_{s,fo} = rcg\dot{m}_{ref} \quad (21)$$

$$\dot{m}_{s,fr} = (rcg-1)\dot{m}_{ref} \quad (22)$$

Where:

$$rcg = \frac{(y_{ret}-x_{ret})}{(x_{abs}-x_{ger})} \quad (23)$$

$rcg$  circulation ratio from generator

- Energy balance : VC 8 (Absorber)

$$-\tilde{Q}_{abs} + \dot{m}_{ref}((\tilde{h}_{sev}-\tilde{h}_{reg,fr}) + rcg(\tilde{h}_{reg,fr}-\tilde{h}_{abs})) = 0 \quad (24)$$

-Controls volumes : VCs 9 and 10(regenerator heat exchanger)

For this control volume is assumed no phase and concentration changes. To calculate the outlet temperatures of the heat exchanger regenerator used the method of effectivity (Bejan, 1986). This method considers two dimensionless groups:  $NUT$  and  $\varepsilon$ . As a simplifying assumption will be assumed that the cps of warm and cold currents are approximately equal. Thus, we define the outlet temperatures of the heat exchanger regenerator as:

$$\theta_{reg,fo} = \theta_{abs} + \varepsilon((rcg-1)/rcg)(\theta_{ger}-\theta_{abs}) \quad (25)$$

$$\theta_{reg,fr} = \theta_{ger} + \varepsilon(\theta_{ger}-\theta_{abs}) \quad (26)$$

$$\varepsilon = \frac{1-\exp(-NUT(1-((1-rcg)/rcg)))}{1-((1-rcg)/rcg)\exp(-NUT(1-((1-rcg)/rcg)))} \quad (27)$$

$$NUT = \frac{UA_{reg}}{((rcg-1)\dot{m}_{ref}cp_{reg,fr})} \quad (28)$$

- Energy balance : -VC 11 (Generator)

$$\tilde{Q}_{ger} + \dot{m}_{ref}((\tilde{h}_{ger,l}-\tilde{h}_{ger,v}) + (rcr(\tilde{h}_{ret,l}-\tilde{h}_{ger,v})) + (rcg(\tilde{h}_{reg,fo}-\tilde{h}_{ger,l}))) = 0 \quad (29)$$

- Energy balance : VC 12 (Rectifier)

$$\tilde{Q}_{ret} + \dot{m}_{ref}((\tilde{h}_{ret,v}-\tilde{h}_{ger,v}) + rcr(\tilde{h}_{ret,l}-\tilde{h}_{ger,v})) = 0 \quad (30)$$

- Refrigerant mass flow rate

Applying an energy balance in steady state and neglecting the pumping work in the thermal compressor, in order to determine the refrigerant mass flow rate in the cycle, defined :

$$\dot{m}_{ref} = \frac{(\tilde{Q}_{ger}-\tilde{Q}_{ret}-\tilde{Q}_{abs})}{(\tilde{h}_{ret,v}-\tilde{h}_{vsa})} \quad (31)$$

## 2.2.5- Heat transfer rate

The rate of heat transfer dimensionless are defined for each control volume as:

$$\tilde{Q}_{ev} = \frac{(UA)_{ev}(T_{w,ev}-T_{ev})}{UAT_o} = \gamma_{ev}(\theta_{w,ev}-\theta_{ev}) \quad (32)$$

$$\tilde{Q}_{ger} = \gamma_{ger} (\theta_{gases} - \theta_{ger}) \tag{33}$$

$$\tilde{Q}_{abs} = \gamma_{abs} (\theta_{abs} - \theta_{w,abs}) \tag{34}$$

$$\tilde{Q}_{ret} = \gamma_{ret} (\theta_{ret} - \theta_{w,ret}) \tag{35}$$

$$\tilde{Q}_{cd} = \tilde{Q}_{ger} + \tilde{Q}_{ev} - \tilde{Q}_{abs} - \tilde{Q}_{ret} \tag{36}$$

### 3. RESULTS AND DISCUSSION

The numerical results were produced with a code written in FORTRAN -based mathematical model described in section 2.

The numerical problem consisted of integrating Eqs. 5,6,7,10,12,13 e 16 in time and solving the nonlinear system Eqs. 24, 29 e 30 at each time step.

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 system equations non-linear and algebraic equations were solved using the Newton-Raphson method.

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.

For the simulation of a case study were assumed the following values for the model parameters:  $\theta_{cist} = \theta_{ext} = 0.72$ ,  $\theta_{w,abs} = \theta_{w,ret} = 0.98$ ,  $\theta_{gases} = 2.66$ .

The physical parameters for the mathematical model are listed in Tables 1.

Table 1. Parameters of the mathematical model

	Condenser	Evaporator	Absorber	Rectifier	Generator	Refrigerated Space
$\psi$	1.015	1.279	—	—	—	—
$\gamma$	0.42	0.44	1.42	0.296	0.13	0.11
$\tilde{m}$	$1.41 \cdot 10^{-2}$	$7.88 \cdot 10^{-3}$	—	—	—	1.0
initial condition ( $\theta$ )*	0.97	0.97	—	—	—	0.97
initial condition ( $U$ )	0.66	1.0	—	—	—	—
Composition (initial guess)	—	—	0.6	0.4	0.3	—

\* refrigerant and water side

Figures 2, 3 and 4 show the results obtained to simulate system operation up to 4600 seconds, or dimensionless time, 13.7.

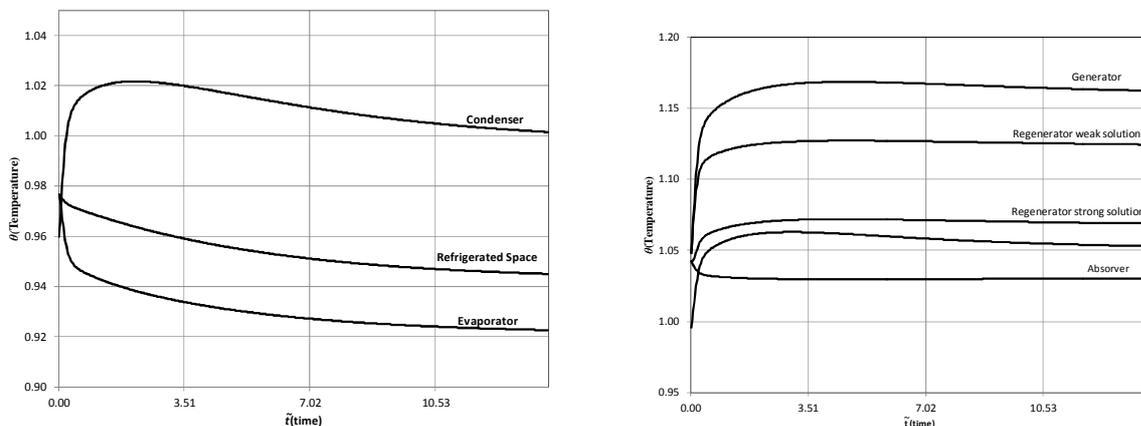


Figure 2- . Numerical simulation for temperatures

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Figure 2 shows the behavior for the temperatures of the: condenser, evaporator, refrigerated space and thermal compressor. Note that, the mathematical model was able to capture the expected behavior for the temperatures, with exception to the absorber, where the temperature drops initially.

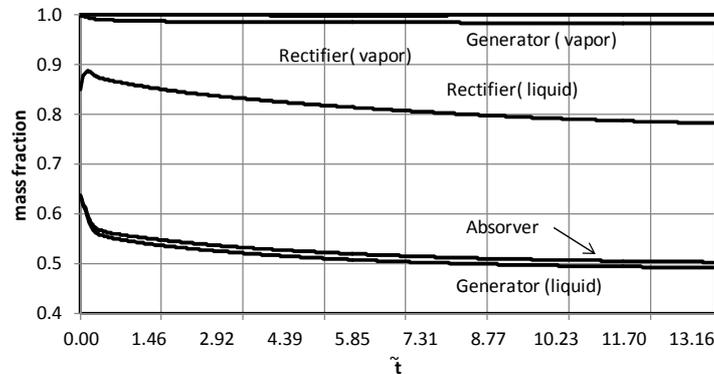


Figure 3-. Numerical simulation for ammonia composition

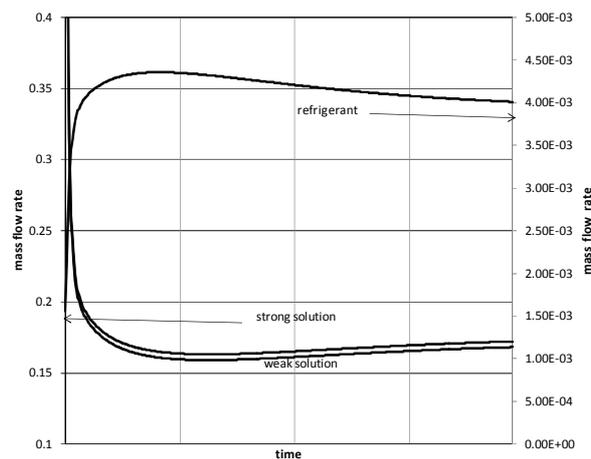


Figure 4- Numerical simulation for mass flow

Figure 3 shows the behavior for the ammonia composition of the thermal compressor. Note that, the results show the expected behavior, since the composition of the ammonia absorber is greater than the bottom product of the generator and the rectifier vapor concentration calculated by the model, presents a value close to one.

Figure 4 shows the values mass flows: refrigerant and solutions (weak and strong) for the refrigeration cycle calculated by the mathematical model. The order of magnitude of the variables are consistent with literature data (Herold *et al.*, 1996).

#### 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:

- (i) proposed model was able to capture the behavior of an absorption refrigerator using low computational time;
- (ii) although thermal compressor not provide mathematical equation considering its thermal inertia, the model was capable of capturing an increase in temperature with time for nearly all components, with the exception that the absorber converged to a temperature below the initial temperature since compressor thermal transients are artificial;
- (iii) calculated concentrations are consistent since the separation performed by the generator and rectifier produces refrigerant vapor with high purity.
- (iv) the thermal inertia of the condenser, evaporator and refrigerator space were slower when compared to the thermal compressor, indicating that adjustments need to be made.

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(v) the coefficient of performance (COP) of the cycle, is defined as the ratio between the rates of heat transfer from the evaporator generator, was about 0.2, which represents a low value for this type of system. Thus, studies of optimizations could be performed with the proposed model seeking to optimize the COP and the cooling rate.

No experimental validation of the numerical results is presented this work. 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.

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