Proceedings of the 10th Brazilian Congress of Thermal Sciences and Engineering -- ENCIT 2004
Braz. Soc. of Mechanical Sciences and Engineering -- ABCM, Rio de Janeiro, Brazil, Nov. 29 -- Dec. 03, 2004

Paper CIT04-0669

REAL TIME SIMULATION OF REGENERATORS FOR ABSORPTION REFRIGERATORS

Lauber de Souza Martins
Universidade Federal do Paraná, Departamento de Engenharia Mecânica, Centro Politécnico, Jardim das Américas C.P.: 19011, CEP: 81531-990,Curitiba, Paraná, Brasil.
lauber@demec.ufpr.br

José Viriato Coelho Vargas
Universidade Federal do Paraná, Departamento de Engenharia Mecânica, Centro Politécnico, Jardim das Américas C.P.: 19011, CEP: 81531-990,Curitiba, Paraná, Brasil.
jvargas@demec.ufpr.br

Abstract. *This paper introduces a general computational model for regenerators fed by a hot fluid stream on one side, whereas the other side is a fluid mixture, in which one of the components (refrigerant) undergoes a change of phase (vaporizes) and the other component (absorbent) remains in the liquid phase. A simplified physical model, which combines fundamental and empirical correlations, and principles of classical thermodynamics, mass and heat transfer, is developed. The combination of the proposed simplified physical model with a cell-centered finite volume scheme for the numerical discretization of the differential equations is the so called volume element model, VEM (Vargas, J. V. C. et al, 2001, A numerical model to predict the thermal and psychrometric response of electronic packages, ASME Journal of Electronic Packaging, Vol. 123, pp.200-210). The model was based upon a general configuration of a regenerator, i.e.,a heat exchanger where the external "hot" side is a metallic matrix and the internal "cold" side is composed of two concentric tubes. The first phase of the study is herein presented, and consists of the modeling and simulation of a regenerator with only one inner tube with a mixture of two fluids (refrigerant and absorbent) in a single phase.*

Keywords*. Temperatures, concentrations, entropy generation, effectivity.*

1. Introduction

A large research effort has been brought to bear on absorption refrigeration systems in recent years (e.g., Perez-Blanco, 1993; Sokolov and Hershgal, 1991; Stolk, 1980; Suri and Ayyash, 1982; Wijeysundera, 1996, Didion and Radermacher, 1984). These systems can make use of waste heat and renewable energy sources such as solar energy (Vargas et al., 1996, Vargas et al., 2000). In countries in which electricity and fossil fuel are available only intermittently or are very expensive, solar energy may be an alternative for the preservation of foodstuffs and medical supplies (Worsoe-Schmidt and Holm, 1989) since a 'reservoir of cold' can also be generated through formation of subcooled ice for use when the sun is obscured. However, when uninterrupted precisely controlled cooling is required, heat driven systems powered by gas, electricity or oil are needed either as an alternative (to solar thermal exergy and waste heat), or as supplementary thermal energy source (Ayyash et al., 1985). The improvement and optimization of the design and control of heat driven refrigerators is a crucial issue regardless of the energy source used. Heat driven refrigerators, with an effective control system, may well be able to combine economy with a low environmental impact. To be effective, a thermal system must be capable of capturing 'realistic' features of the heat transfer process, and the unit. Several studies have dealt with these issues by using the method of entropy generation minimization (Bejan et al., 1995, Bejan, 1989; Bejan, 1995; Bejan, 1988) (exergy optimization), which pursues realistic models and takes into account the inherent irreversibility of heat, mass and fluid flow.

In this paper, a mathematical model is introduced to simulate the transient behavior of a regenerator heat exchanger, which is a key component of a heat driven refrigerator that uses a hot fluid system (e.g., solar heated fluid, exhaust gases) as the high temperature source. Only the first phase of the study is presented in the sense that in actual operation, the cold stream undergoes a change of phase, whereas in this paper the cold stream is initially treated as a single phase fluid. For that, the model starts with only one inner tube. The second phase will be treated in a follow up study. It will consist of the introduction of the innermost tube with a mixture of two fluids (refrigerant and absorbent), with the refrigerant experiencing a change of phase from liquid to gas, and the absorbent remaining in the liquid phase.

2. Mathematic model

In most absorption refrigeration systems, two distinct fluids exist, that is, a hot fluid (heating system) and an absorbent/refrigerant solution (absorption system). The cooling side solution receives the heat from the hot fluid (gases or liquid) through a heat exchanger, called generator. This study considers a different type of heat exchanger as the system's heat source, i.e., a regenerator where the external "hot" side is a metallic matrix and the internal "cold" side is composed of two concentric tubes. The thermodynamic cycle of the 2 – fluid mixture in the absorption system

corresponds to a cycle of a vapor-compression refrigeration system, where the compressor is replaced by a set of equipments, that is: absorber, pump, valve and regenerator (or generator).

The focus of this study is the hot heat exchanger (regenerator), which is one component of the absorption refrigerator, as shown in Fig.1. The study is limited to a first phase, i.e., the modeling and simulation of a regenerator with only one inner tube with a 2-fluid mixture in a single phase.

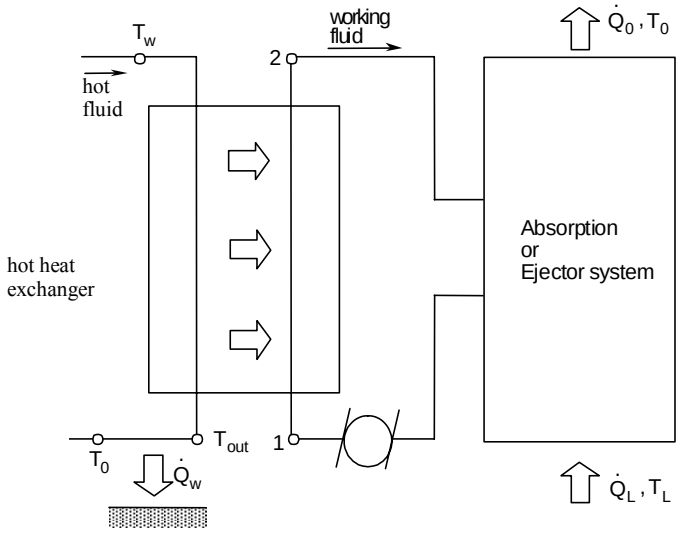

Figure1. Absorption refrigerator.

The hot heat exchanger is designed to store a certain amount of energy. This storage is accomplished by the following means. The internal pipe where the cooling fluid flows is wrapped with a metallic pure aluminum grid (Fig. 2). The result is the direct contact between the matrix and the hot fluid. Beyond the exchange of heat with the cold fluid, heat is stored in the metallic matrix according to the hot fluid mass flow rate, which may vary in time. This way, the regenerator stores heat when the hot fluid flow rate is high, releasing the necessary heat for the system to operate when the hot fluid flow rate is low.

Figure 2. The metallic aluminum grid utilized to assemble the metallic matrix.

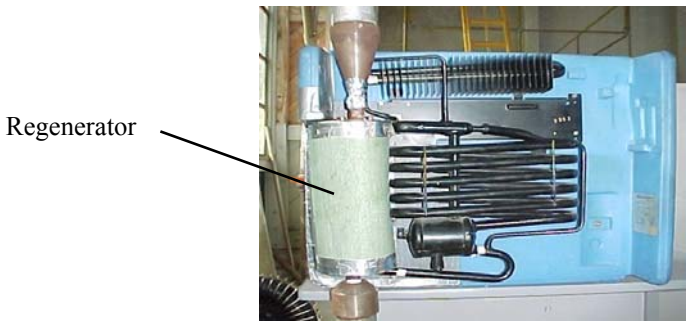

Figure 3. Location of the regenerator in the prototype.

The practical importance of a regenerative heat exchanger refers to its application in a cycle where the hot gases from cars or industrial gases are used as the hot fluid. The absorption system must operate with a periodic energy

supply, i.e., either with a low or high hot fluid mass flow rate. For example, in the case of an automotive absorption system in a traffic jam, or stopped at a traffic light, there will not be enough emissions of hot gases, that is, the flow of hot gases will decrease. However, the air conditioning has to continue its operation. With a regenerator, during such periods, the system continues to work due to the thermal energy that was stored in the metallic matrix during the high flow periods. The metallic matrix provides thermal inertia to the system.

The regenerator is schematically shown in Fig.4.

Figure 4. Counter-flow heat exchanger.

The regenerator heat exchanger is divided in volume elements (cells) according to Fig.5.

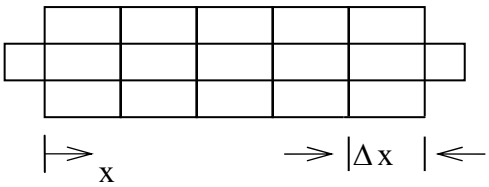

Figure 5. Heat exchanger divided in volume elements.

Each cell is divided in three systems:

- System 1: Regenerator metallic matrix and pipe (solid material)
- System 2: Hot fluid (external)
- System 3: Cold fluid (internal)

Each cell has its thermal behavior defined by these three systems. The first law of thermodynamics is applied to each system in each volume element.

Figure 6 represents a cell "m" of the heat exchanger and the heat and mass transfer interactions experienced by the three systems.

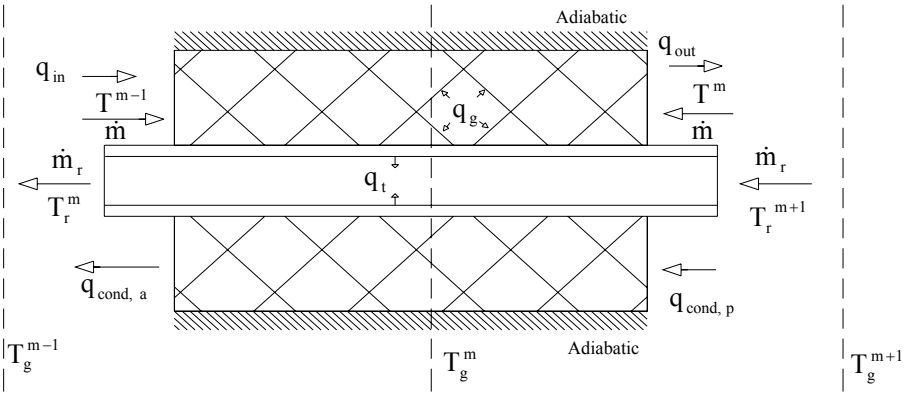

Figure 6. Heat and mass transfer interactions in cell "m"

<u>System 1:</u> Regenerator metallic matrix and pipe (solid material)

The energy balance in system 1 is written as follows:

$$q_g + q_{cond,p} + q_{cond,a} + q_t = m_g^m c_g \frac{dT_g^m}{dt} \tag{1}$$

Where:

q_g : heat transfer rate between the metallic matrix and the hot fluid, W.
$q_{cond,p}$: heat transfer rate by conduction through the matrix in cell m to the matrix in cell "m+1", W.
$q_{cond,a}$: heat transfer rate by conduction through the matrix in cell m to the matrix in cell "m-1", W.
q_t : heat transfer rate by convection between the tube wall and the internal cold fluid, W.
T_g^m : temperature of system 1 in cell "m", K.
m_g^m : mass of metallic matrix in cell "m", kg.
c_g : specific heat of the metallic matrix, $Jkg^{-1}K^{-1}$.
t : time, s.

From Figure 6, it is also written:

$$q_{in} = \dot{m} c_p T^{m-1} \tag{2}$$

$$q_{out} = \dot{m} c_p T^m \tag{3}$$

where q_{in} is the enthalpy transfer rate entering cell "m" from cell "m-1", W, q_{out} the enthalpy transfer rate leaving cell "m" to cell "m+1", $\dot{m}$ the hot fluid mass flow rate, kg/s, c_p the specific heat at constant pressure of the hot fluid, J/(kgK), T^{m-1} the temperature of the hot fluid in cell "m-1", T^m the temperature of the hot fluid in cell "m", K.

$$q_g = h_g A_{gl}^m (T^m - T_g^m) \tag{4}$$

where h_g is the convective heat transfer coefficient between the metallic matrix and the hot fluid, $Jkg^{-1}K^{-1}$, A_{gl}^m the heat transfer area of the metallic matrix in cell "m", m²; T^m is the temperature of system 2 in cell "m", K.

$$q_{cond,a} = -kA_{gs} \frac{(T_g^m - T_g^{m-1})}{\Delta x} \tag{5}$$

where k is the thermal conductivity of the metallic matrix, $Wm^{-1}K^{-1}$, A_{gs} the heat transfer area in the cross section of the cell, m², Δx the length of the cell, m, and T_g^{m-1} the temperature of system 1 in cell "m-1",K.

$$q_{cond,p} = -kA_{gs} \frac{(T_g^m - T_g^{m+1})}{\Delta x} \tag{6}$$

where T_g^{m+1} is the temperature of system 1 in cell "m+1", K.

$$q_t = h_t A_{tl}^m (T_r^m - T_g^m) \tag{7}$$

where h_t is the convective heat transfer coefficient between the cold fluid and the tube wall, $Wm^{-2}K^{-1}$, A_{tl}^m the heat transfer area of the tube in cell "m", m², T_r^m the temperature of system 3 in cell "m", K.

In the model, the convective heat transfer coefficients h_g and h_t are assumed constant. However, in a more detailed treatment, these coefficients can be taken as functions of the Reynolds (Re) number and the Prandtl number (Pr), by utilizing empirical correlations for the laminar and turbulent regimes, both for the internal and external flows (Kays and London, 1998).

System 2: Hot fluid (external)

The energy balance in system 2 is written as follows:

$$q_{in} - q_{out} - q_g = m^m c_v \frac{dT^m}{dt} \tag{8}$$

Where, m^m is the mass of hot fluid in cell "m", and c_v, the specific heat at constant volume of the hot fluid $Jkg^{-1} K^{-1}$.

In order to calculate the mass of hot fluid in the control volume, the hot fluid density is utilized as follows.

$$m^m = \rho V^m \tag{9}$$

where ρ is the density of the hot fluid, kg m^{-3}, and V^m the volume of hot fluid in cell "m", m^3.

Additionally, for the first cell it is assumed that:

$$q_{cond,a} = 0 \tag{10}$$

For the first cell, m = 1, it is also noted that:

$$T^{m-1} = T^0 = T_{in} \tag{11}$$

where the inlet temperature of the hot fluid (T_{in} , K) must be a known parameter.

For the last cell it is assumed that:

$$q_{cond,b} = 0 \tag{12}$$

System 3: Cold fluid (internal)

The energy balance in system 3 is written as follows:

$$-q_t + \dot{m}_r c_{pr}\left(T_r^{m+1} - T_r^m\right) = \dot{m}_r^m c_{vr} \frac{dT_r^m}{dt} \tag{13}$$

where $\dot{m}_r$ is the internal fluid mass flow rate, kg s^{-1}, c_{vr} the specific heat at constant volume of the internal fluid, J $kg^{-1} K^{-1}$, T_r^m the temperature of the internal fluid in cell "m", K, T_r^{m+1} is the temperature of the internal fluid in cell "m+1", K. In the last cell, it is noted that:

$$T_r^{ncel} = T_{r,\,in} \tag{14}$$

where n_{cel} is the total number of cells in the heat exchanger.

The inlet parameters must be known. The analysis covered the three systems, delivering three ordinary differential equations (ODE´s), i.e., Eqs. (1), (8) and (13), for each cell.

The system of equations, therefore, is composed by 3 n_{cel} ODE's. An adaptive time step Runge-Kutta-Fehlberg method (Kincaid and Cheney,1991) is utilized to integrate the system in time until steady state or a final simulation time is achieved, from a given set of initial conditions, $T_{g,0}^m$, T_0^m , and $T_{r,0}^m$, for m = 1,... n_{cel}.

3. Results and Discussion

In a first analysis, the numerical results of two different configurations of porosity, $\phi = \frac{(A_s - A_{gs})}{A_s}$, are compared, where A_s is the cross sectional area of the annular space. The program has the option to run until a final time of simulation that is defined by the user or until steady state is achieved. The four following graphs were obtained for the latter condition, which are shown in Figs. 7-10.

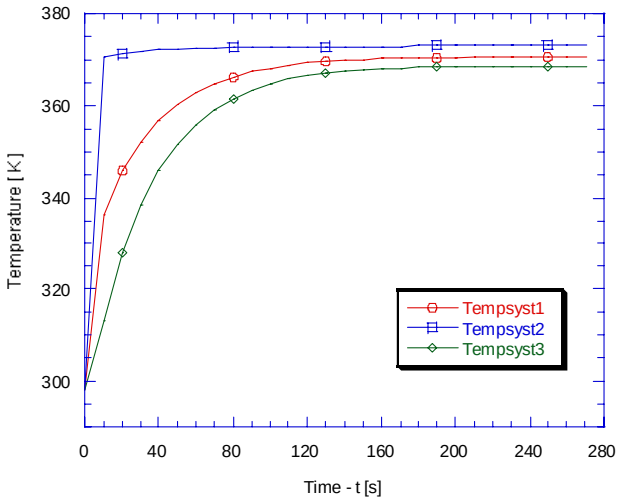

Figure 7. Temperature transient evolution in the first cell for $\phi = 0.95$; L = 1 m; $\dot{m} = 0.1$ kg/s.

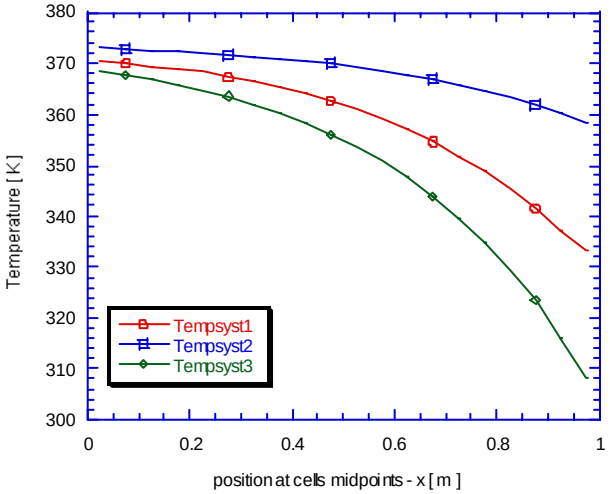

Figure 8. Steady state temperature values for $\phi = 0.95$; L = 1 m; $\dot{m} = 0.1$ kg/s.

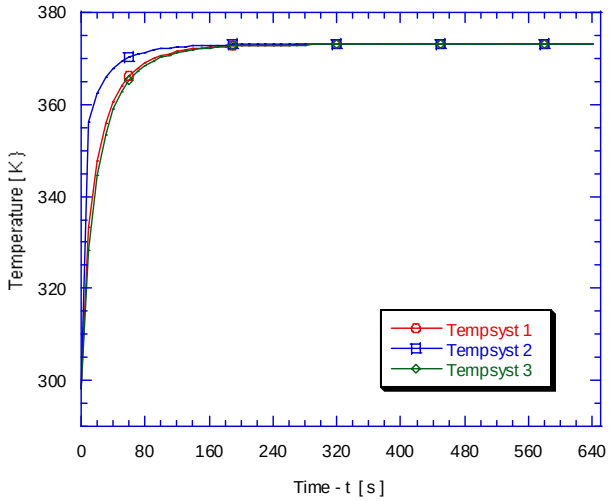

Figure 9. Temperature transient evolution in the first cell for $\phi = 0.5$; L = 1 m; $\dot{m} = 0.1$ kg/s.

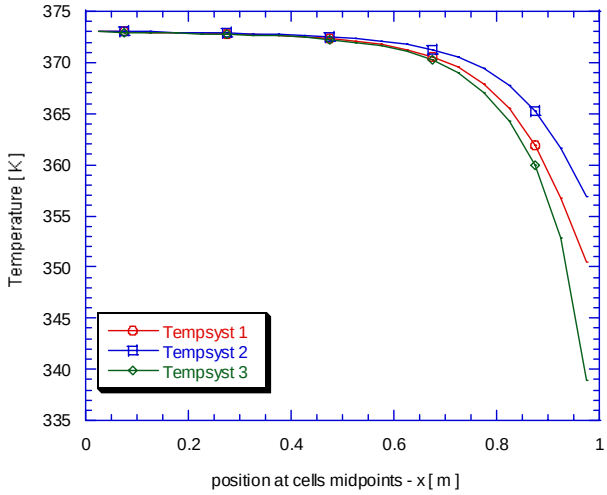

Figure 10. Steady state temperature values for $\phi = 0.5$; L = 1 m; $\dot{m} = 0.1\,kg/s$.

It is observed that the time to achieve steady state is larger in the exchanger with smaller porosity, i.e., in the heat exchanger with more metallic mass (more metallic matrix). We may conclude that the larger the mass of metallic matrix, the larger the thermal inertia, what will imply in a larger transient. On the other hand, it is verified that a better thermal contact is obtained with $\phi = 0.5$, i.e., with more mass of metallic matrix. This observation is corroborated by the effectiveness, ε, values computed for each heat exchanger, according to the porosity variation: $\varepsilon = 0.99$ for $\phi = 0.5$, and $\varepsilon = 0.93$ for $\phi = 0.95$, where the effectiveness was defined according to the well known ε-NTU relations for heat exchangers (Bejan, 1993).

It is important to stress that with lower porosities the free flow cross sectional area is reduced, therefore pressure drop is expected to increase as $\phi \to 0$. In the present model, pressure drop was not accounted for. However, the results are expected to be accurate down to the lowest porosity value considered in these simulations, i.e., $\phi = 0.5$. Therefore, for the cases considered in this study, pressure drop was assumed negligible.

In Figures 11-14 the length of the tubes was varied, for the same porosity, $\phi = 0.8$.

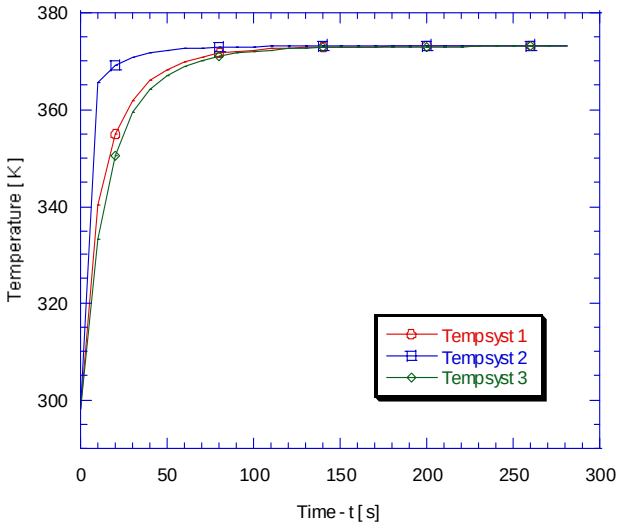

Figure 11. Temperature transient evolution in the first cell for $\phi = 0.8$; L = 0.5 m; $\dot{m} = 0.1\,kg/s$.

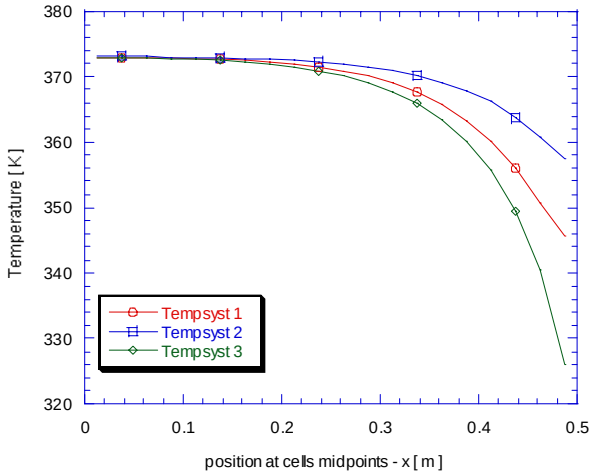

Figure 12. Steady state temperature values for $\phi = 0.8$; L = 0.5 m; $\dot{m} = 0.1$ kg/s.

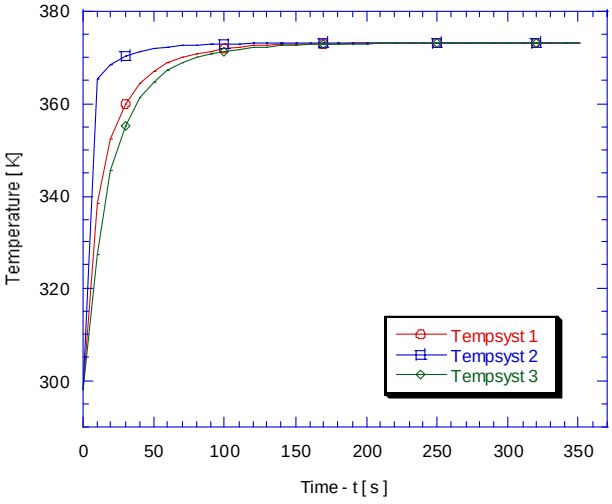

Figure 13. Temperature transient evolution in the first cell $\phi = 0.8$; L = 1 m; $\dot{m} = 0.1$ kg/s.

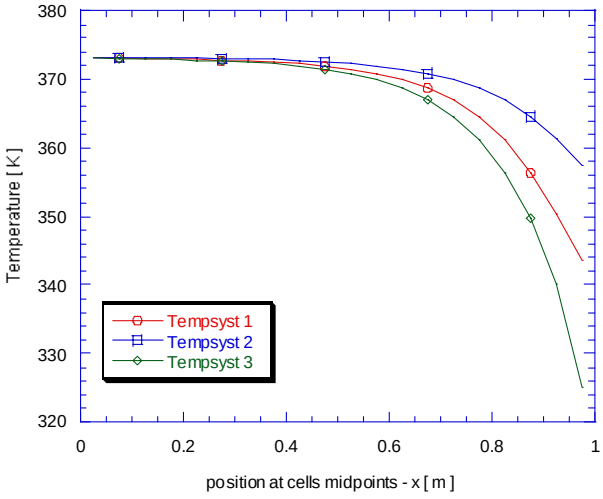

Figure 14. Steady state temperature values for $\phi = 0.8$; L = 1 m; $\dot{m} = 0.1$ kg/s.

It is observed that in Figs. 13 and 14 there is an unnecessary tube length, because for L = 0.5 m, the three systems have already reached thermal equilibrium with the inlet hot fluid temperature, therefore, for longer heat exchangers, no more heat transfer will occur between the two fluid streams. So, the model herein developed allows for the determination of the minimum necessary length of heat exchanger for maximum effectiveness. The procedure is to start with a simulation for a long heat exchanger, and then proceed with a comparison with the simulation results for shorter and shorter ones, stopping the process when the effectiveness starts to drop. At this point, it is determined the optimal heat exchanger length for that condition. This is a method of defining the appropriate heat exchanger length.

The main finality of the metallic matrix is to store energy, therefore at the moments where the hot fluid flow rate decreases, the regenerator must be able to supply enough energy to the absorption system to operate satisfactorily. Such situation is simulated by varying the hot fluid flow rate according to a periodic function, the cosine function, i.e.,

$$f(t) = \frac{\dot{m}_{max}}{2}\cos t + \frac{\dot{m}_{max}}{2}.$$

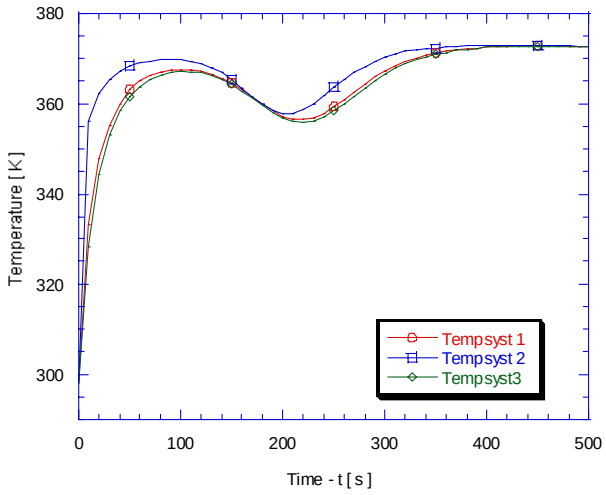

Figure 15. Temperature transient evolution in the first cell for $\phi = 0.5$; L = 1 m; $\dot{m} = f(t)$.

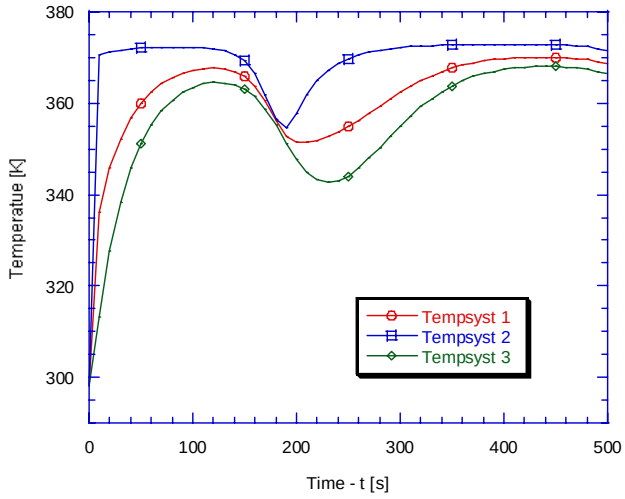

Figure 16. Steady state temperature values for $\phi = 0.95$; L = 1 m; $\dot{m} = f(t)$.

In Figure 15, it is seen that the metallic matrix performed according to the expectations. The oscillation in the hot fluid mass flow rate was not the same as in the temperatures, which showed a much smaller amplitude. Therefore, the system is able to continue its normal operation even with low hot fluid mass flow rates. In Figure 16, the regenerator had a smaller metallic mass, therefore the porosity was high $\phi = 0.95$. In this case, the system normal operation is compromised since the temperature oscillation showed a higher amplitude.

4. Conclusions

In this study, a general computational model for regenerative heat exchangers was developed based on a volume element methodology (Vargas *et al*.,2001). The treatment was introductory, presenting a first phase of a more complex study, i.e., for the configuration studied, only one inner tube was considered whereas the 2-fluid mixture that flows in it dose not experience a change of phase.

The results are shown to capture the expected physical trends of the equipment, therefore it is now possible to proceed to the second phase of the study, which will consider the actual configuration of the regenerator in an absorption system, i.e., with two internal concentric tubes, and with one of the fluids (the refrigerant) experiencing a change of phase. The analysis herein presented consisted of the investigation of the temperatures profiles response along the heat exchanger according to the variation of two key design parameters, i.e., the hot side porosity and heat exchanger total length. The analysis also investigated the transient response of the regenerator, showing the influence of the variation of the two parameters on the two fluid streams temperature profiles and, therefore, on heat exchanger effectiveness. The results also showed how the simulation model could be used to appropriately dimension the heat exchanger length. The main conclusion is that the model could be used in general for design and optimization of heat exchangers with a similar configuration to the one studied in this paper.

5. Acknowledgment

The authors acknowledge with gratitude the support of the Program of Human Resources for the Oil Sector and Natural Gas, of Brazilian Oil National Agency – PRH24/ANP/MCT.

6. References

Ayyash, S., Suri, R. K., Al Shami, H. “Performance results of a solar absorption cooling installation” International Journal of Refrigeration (1985) 8 177-183.

Bejan, A. “Heat transfer”, Willey, New York (1993).

Bejan, A. “Entropy Generation Minimization” CRC Press, Boca Raton, FL (1995).

Bejan, A. “Theory of heat transfer-irreversible refrigeration plants”. International Journal of Heat and Mass Transfer (1989) 32 1631-1639.

Bejan, A. “Theory of heat transfer-irreversible power plants.” International Journal of Heat and Mass Transfer (1988) 31 1211-1219.

Bejan, A., Vargas, J. V. C., Sokolov, M. “Optimal allocation of a heat exchanger inventory in heat driven refrigerators”. International Journal of Heat and Mass Transfer (1995) 38 2997-3005.

Didion, D., Radermacher, R. “Part-load performance characteristics of residential absorption chillers and heat pumps”. International Journal of Refrigeration (1984) 7 393-398.

Kays, W.M., London, A.L., "Compact Heat Exchangers, 3rd Edition, Krieger Publishing Company, Malabar, FL (1998).

Perez-Blanco, H. “Conceptual design of a high-efficiency absorption cooling cycle”. International Journal of Refrigeration (1993) 16 429-433.

Sokolov, M., Hershgal, D. “Operational envelope and performance curves for compression-enhanced ejector refrigeration system”. ASHRAE Transactions (1991) 97:2 394-402.

Stolk, A. L. “New possibilities for absorption refrigeration machines for energy saving”. International Journal of Refrigeration (1980) 3 78-82.

Suri, R. K., Ayyash, S. “Solar absorption cooling effect on operational parameters on power saving”. International Journal of Refrigeration (1982) 5 274-279.

Vargas, J. V. C., Sokolov, M., Bejan, A. “Thermodynamic optimization of solar-driven refrigerators”. Journal of Solar Energy Engineering (1996) 118 130-135.

Vargas, J.V.C., Fleming,J.S.,Parise, J.A.R, "Maximium exergy input rate from a hot stream in solar-driven refrigerators.” International Journal of energy Research (2000) 25 751-767.

Vargas, J.V.C., Stanescu, G., Florea,R.,Campos, M.C., “A numerical model to predict the thermal and psychometric response of electronic package”, ASME Journal of Electronic Packaging (2001) 123 200-210.

Wijeysundera, N. E. “Performance limits of absorption cycles with external heat transfer irreversibilities”. Applied Thermal Engineering (1996) 16:2 175-181.

Worsoe-Schmidt, P., Holm, F. “Development field testing of a solar-powered block-ice plant”. Jutland Technological Institute Teknologiparken DK-8000 Aarhus C Denmark FHO/VIM (1989) 1-17.