REAL TIME SIMULATION OF REGENERATORS WITH FLUID CHANGE OF PHASE FOR ABSORPTION REFRIGERATORS

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Abstract. *This paper introduces a general computational model for regenerators fed by a hot fluid stream on one side, whereas the other side is the liquid refrigerant, in which it undergoes a change of phase and a little amount of refrigerant remains in the liquid phase. A physical model, combines fundamental and empirical correlations, principles of classical thermodynamics, mass and heat transfer, is developed and the resulting three-dimensional differential equations are discretized in space using a three-dimensional cell centered finite volume scheme. The combination of the proposed simplified physical model with the adopted finite volume scheme for the numerical discretization of the differential equations is the so called volume element model. The model was based upon a general configuration of a regenerator. A fluid, in liquid phase, enters the core of the heat exchanger and experiences a phase change as it flows through it. The fraction of the fluid that is not vaporized is re-circulated through the inner layer of the heat exchanger. The results are a preliminary step by means of simplification (refrigerant as a pure substance instead a mixture) of the model for absorption system witch the refrigerant will be a mixture ammonia - water.*

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

1. Introduction

A large research effort has been brought to bear on absorption refrigeration systems in recent years, (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 energy 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 the change of phase of refrigerant (ammonia) in the innermost tube.

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. In absorption systems 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.

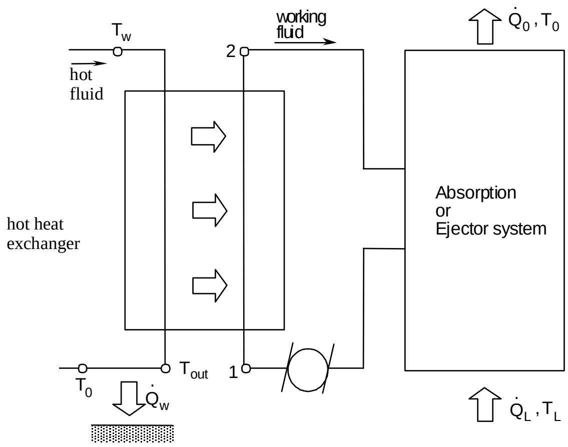

Figure1. Absorption refrigerator.

The hot heat exchanger is designed to store a certain amount of energy. The internal pipe where the cooling fluid flows is wrapped with a metallic grid. 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. The amount of metallic grid is measurement by porosity (ϕ).

$$\phi = \frac{V_v}{V_T} \tag{1}$$

where V_v is the empty volume in the external tube, m^3; V_T is the total volume of the external tube, m^3.

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. Porosity is an important parameter because it will provide more or less heat transfer area. If the porosity is high, the heat transfer area will be low, because high porosity means much empty space in external tube. If porosity is low then heat transfer area will be high. Low porosity means low empty space inside external tube.

Each cell is divided in three systems:

- System 1: Regenerator metallic matrix and pipe (solid material)
- System 2: Hot fluid
- System 3: Refrigerant in intermediate tube
- System 4: Refrigerant in innermost tube

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

Figure 3 represents a cell "m" of the heat exchanger and the heat and mass transfer interactions experienced by the four systems.

The regenerator heat exchanger is divided in volume elements (cells) according to Fig.2.

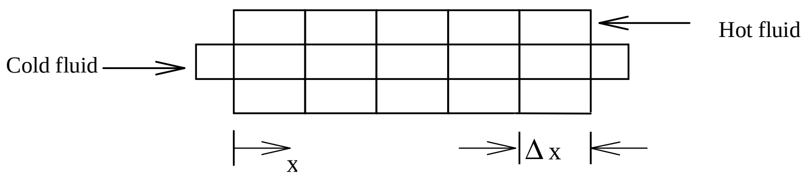

Figure 2. Heat exchanger divided in volume elements.

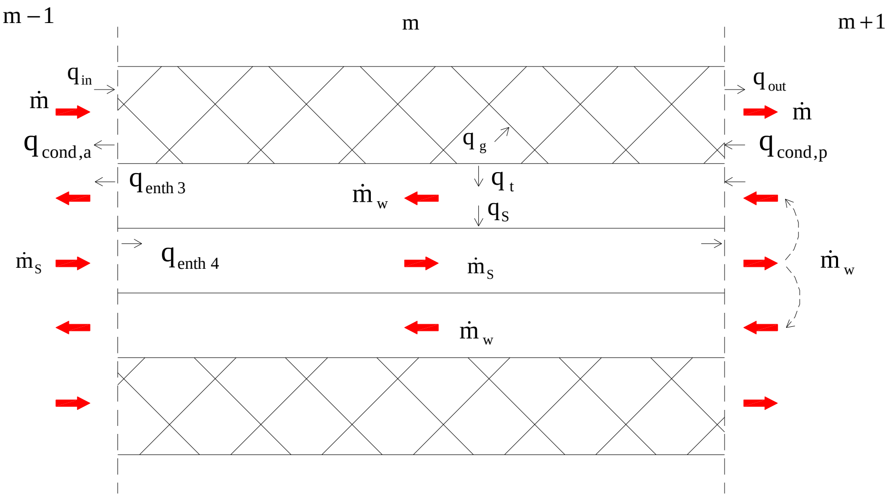

Figure 3. Heat and mass transfer interactions in cell "m"

System 1: *Regenerator metallic matrix and pipe (solid material)*

The energy balance in system 1 is written as follows:

$$q_g + q_{cond,p} + q_{cond,a} + q_t = m_g^m c_g \frac{dT_g^m}{dt} \tag{2}$$

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

From Figure 3, it is also written:

$$q_{in} = \dot{m} c_p T^{m-1} \tag{3}$$

$$q_{out} = \dot{m} c_p T^m \tag{4}$$

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.

$$q_g = h_g A_{gl}^m (T^m - T_g^m) \tag{5}$$

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.

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

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.

$$q_{cond,p} = -kA_{gs} \frac{(T_g^m - T_g^{m+1})}{\Delta x} \tag{7}$$

$$q_t = h_t A_{tl}^m (T_r^m - T_g^m) \tag{8}$$

where T_g^{m+1} is the temperature of system 1 in cell "m+1", K, h_t is the convective heat transfer coefficient between the intermediate 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

The energy balance in system 2 is written as follows:

$$q_{in} - q_{out} - q_g = m^m c_v \frac{dT^m}{dt} \tag{9}$$

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}.

System 3: Refrigerant in intermediate tube

The energy balance in system 3 is written as follows:

$$-q_t + q_s + q_{enth3} = (m_{NH_3,w} c_{NH_3,l} + m_{H_2O,w} c_{fi}) \frac{dT_w^m}{dt} \tag{10}$$

where $m_{H_2O,w}$ is the mass of water in ammonia, kg; $c_{NH_3,l}$ is the specific heat of liquid ammonia, $Jkg^{-1}K^{-1}$; $m_{NH_3,w}$ is the mass of ammonia in intermediate tube, kg; c_{fi} is the specific heat of water, $Jkg^{-1}K^{-1}$; T_w^m is the temperature of ammoia in intermediate tube in cell "m", K; q_S is the heat transfer by convection to the liquid ammonia in innermost tube, W.

Let's name α as total mass of steam if ammonia present in a cell "m". If $\alpha = 0$ then ammonia is whole liquid, $\alpha = m_{NH_3,l}$ then all refrigerant is steam ($m_{NH_3,l}$ is the ammonia liquid mass in innermost tube, kg). To $\alpha = 0$ or $\alpha = m_{NH_3,l}$, q_S is given by:

$$q_S = U_S A_{tl4}(T_S^m - T_w^m) \quad (11)$$

For $0 < \alpha < m_{NH_3,l}$:

$$q_S = U_b U_S A_{tl4}(T_S^m - T_w^m) \quad (12)$$

where U_b is the rate between heat transfer coefficient during boiling and heat transfer coefficient when one phase is present; U_S is defined as global heat transfer coefficient between system 3 and 4, W/m^2K; A_{tl4} is the heat transfer area of liquid ammonia, m²; T_S^m is the temperature of liquid ammonia in cell “m”, K.

$$q_{enth3} = \left(\dot{m}_{NH_3,w} c_{NH_3,l} + \dot{m}_{H_2O,w} c_{fi}\right)\left(T_{w,in} - T_w^m\right) \quad (13)$$

where $\dot{m}_{NH_3,w}$ is the ammonia mass flow rate in intermediate tube, kgs^{-1}; $\dot{m}_{H_2O,w}$ is the water mass flow rate in intermediate tube kgs^{-1}; $T_{w,in}$ is the inlet temperature of fluid in intermediate tube, K.

System 4: *Refrigerant in innermost tube*

The energy balance in system 4 is written as follows:

$$q_{enth4} - q_s = (m_{NH_3,l} c_{NH_3,l} + m_{H_2O,w} c_{fi})\frac{dT_s^m}{dt} \quad (14)$$

$$q_{enth4} = \left(\dot{m}_{NH_3,l} c_{NH_3,l} + \dot{m}_{H_2O} c_{fi}\right)\left(T_{s,in} - T_s^m\right) \quad (15)$$

where $T_{s,in}$ is the inlet temperature of ammonia in innermost tube, K; $\dot{m}_{H_2O}$ is the water mass flow rate in intermediate tube, kgs^{-1}; $\dot{m}_{NH_3,l}$ is the liquid ammonia mass flow rate in innermost tube, kgs^{-1}.

Equation (13) isn’t true to temperatures above saturation temperature (T_{sat}, K) of ammonia. Then, it is necessary to write other equation that models the behavior of system 4 when its temperature reaches T_{sat}. The relation is:

$$\frac{dT_S^m}{dt} = 0 \quad (16)$$

It means that in change of phase the temperature is constant.

2. 1 Efficiency (η)

Efficiency is defined as follows:

$$\eta = \frac{q}{q_{max}} \quad (17)$$

where q is the amount of heat transferred to system 4 , W; and q_{max} is the maximum amount o heat transfer possible, W.

$$q_{max} = C_q\left(T_{in,2} - T_{in,4}\right) \quad (18)$$

$$q = (1-r)x_{rl}\dot{m}_s\left(c_{NH_3,l}\left(T_{sat} - T_{in,4}\right) + h_{fg} + c_{NH_3,l}\left(T_S^m - T_{sat}\right)\right) \quad (19)$$

where $T_{in,2}$ is the inlet temperature of system 2, K; $T_{in,4}$ is the inlet temperature os system 4, K; C_q is the least thermal capacity between systems 2 and 4, JK^{-1}; r is the mass fraction of water in ammonia, x_{rl} is the quality of ammonia in the end of regenerator, h_{fg} enthalpy of vaporization, JK^{-1}, T_S^m is the temperature of system 4 in cell “m”, K.

The inlet parameters must be known. The analysis covered the four systems, delivering four ordinary differential equations (ODE´s), i.e., Eqs. (2), (9) and (10), (14) or (16) for each cell.

The system of equations, therefore, is composed by 4 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 for m = 1,... n_{cel}.

3. Result and Discussion

The results were normalized as shown:

$$\tilde{T}_i = \frac{T_i}{T_\infty} \tag{20}$$

$$\tilde{m} = \frac{\dot{m}}{\dot{m}_{ref}} \tag{21}$$

where $\tilde{T}_i$ is the admensional temperature of system "i"; T_i is the temperature of system "i", K; $T_\infty = 298.15K$ is the temperature of referential; $\tilde{m}$ is the admensional of mass flow rate of refrigerant, $\dot{m}$ is the mass flow rate of liquid ammonia in innermost tube, kgs^{-1}; $\dot{m}_{ref} = 0.1$ kgs^{-1} is the mass flow rate of referential.

The developed computational program was used to generate data shown in graphs below. Analyzing Fig.4 it is possible to determinate which is the optimal porosity (ϕ) for specific configuration of regenerator (c_1 and c_2). Notice that to $\phi = 0.5$ the system responded with higher efficiency than other values of porosity.

$$c_1 = \frac{d_1}{d_3} \tag{22}$$

$$c_2 = \frac{d_2}{d_3} \tag{23}$$

where d_1 is the diameter of innermost tube, m; d_2 is the diameter of intermediate tube, m; and d_3 is the diameter of external tube, m.

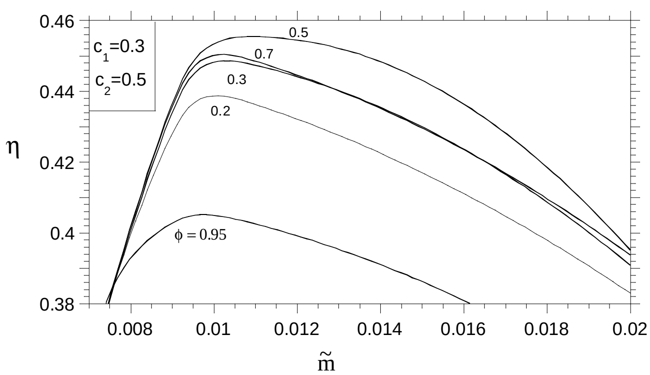

Figure 4. Efficiency as function of $\tilde{m}$ for different values of porosity (ϕ).

When $\phi \to 0$ the empty area in external tube is reduced, then the inner hot fluid speed is approximately equal to outlet hot fluid speed. This is the cause of the low efficiency when $\phi \to 0$. If $\phi \to 1$ the heat transfer area between metallic grade and the hot fluid is reduced, then the efficiency is reduced too. It is possible to assure that exist a maximum between these two points of porosity. The optimal point is $\phi = 0.5$, $\tilde{m} = 0.01$ e $\eta = 45.5\%$.

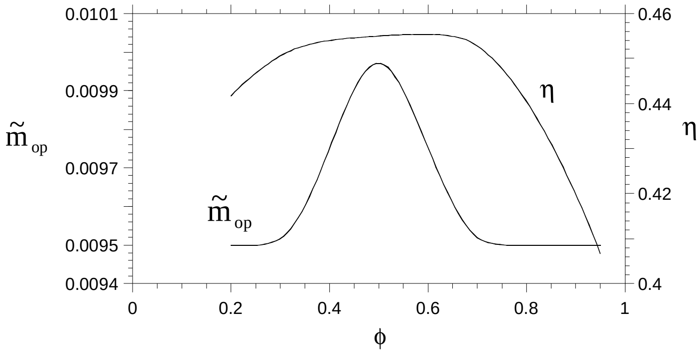

Figure 5. Efficiency and admensional optimal mass flow rate as function of porosity (ϕ) for $c_1 = 0.3$ and $c_2 = 0.5$.

Figure 5 express the efficiency and optimal admensional mass flow rate of refrigerant $\left(\tilde{m}_{op}\right)$ as function of porosity (ϕ).

3.1 Parametric Analysis

Figure 6 shows the results generated when c_2=0.5 and $\phi = 0.5$ remains constant and alter values of c_1.

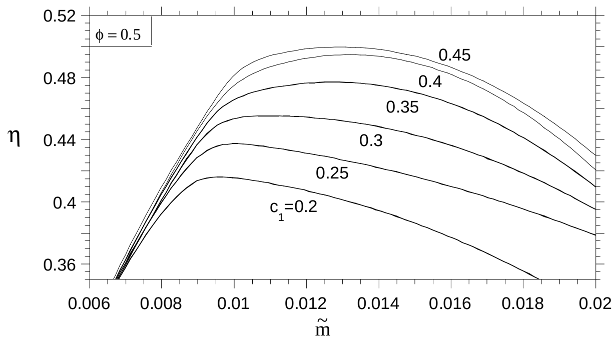

Figure 6. Efficiency as function of $\tilde{m}$ for different values of $c_{1.}$

Figure 7 express value of efficiency and admensional optimal mass flow rate to many value of c_1.

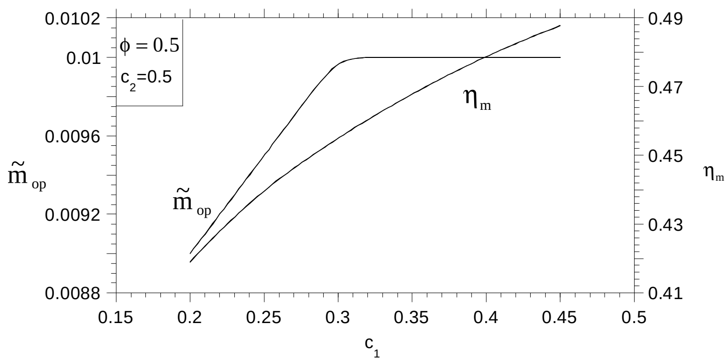

Figure 7. Efficiency and admensional optimal mass flow rate as function of c_1.

It was observed that there is no optimal point to values of c_1, the efficiency grows up indefinitely as c_1 increase. To find this optimal value of constant it would be necessary to consider the pressure drop above the hot flow rate although it is not aborded in this model.

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 fluid in inner tube was considered.

The modeling shown itself efficient cause its results were used to determinate what is the maximum efficiency to specific configuration like porosity, mass flow rate of refrigerant. Analyzing the responses of system face variance of parameters it is possible to determinate the optimal point of regenerator.

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. Acknowledgements

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