TRANSIENT MODEL OF A STATIC EVAPORATOR FOR AN AIR-WATER HEAT PUMP

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Abstract. *The increasing demand of electric energy in Brazil, allied to the great consumption in the rush hour, has stimulated the study of water heating systems that substitute the electric shower, such as gas warming, solar collectors and heat pump. One of these equipments, the solar collector, is the most viable, with the best cost-benefits relation, because Brazil is a tropical country. A heat pump can be used as a support to the solar collectors in places where the climatic conditions and/or the lack of available area of solar collection limit the use of the system. One way to perfect this heat pump would be the substitution of its conventional evaporator (air with forced ventilation) for a static evaporator (natural convection). This evaporator is constituted of a metallic plate with conformed canals, inside of which the coil is fixed through which the refrigerator cooling of the heat pump flows. The objective of this paper is the development of a mathematical model in transient regimen to simulate the static evaporator operation of an air-water heat pump. With the model, some simulations had been carried through, that had allowed to test geometric parameters of the system (diameter of the pipe, area of the plate, etc), materials for the pipes and plates and different weather conditions. These computational tests had indicated that the model represents a good tool to project static evaporators.*

Key words: *Heat Pump, Static Evaporator, Mathematical Model*

1. INTRODUCTION

In Brazil the water heating using electrical resistance is responsible for over than 25% of residential consume, according BEN (2009). So, an alternative way to obtain heat water saving some electrical energy would represent a significative reduction into national process, reducing the risks of an energetic collapse, like occurred in 2002.

Thus, much has been done to reduce the spending of energy with residential water heating, since water heating by electrical resistance method that is used almost exclusively in Brazil, is considered an outdated technology. Alternative equipment heating water for residential use is the heat pump.

The heat pump is a method that is currently diffused in Europe, USA and mainly in Japan. This is based on the use of a free energy which is present in ambient for water heating using equipment called heat pump. With this system we get a significant reduction on electrical energy expenditure when it compared with electrical shower

There are several configurations of heat pumps. One of these configurations uses an evaporator where the heat exchange is done by natural convection. This configuration brings a certain energy saving as electric fans will not be utilized for this exchange of heat. This configuration is called heat pump with static evaporator.

The function of the fans is to increase the coefficient of heat exchange in the evaporator. The static evaporator is a project that has an evaporator which exchanges heat surface is so great as to compensate for a deficiency in the coefficient of heat exchange caused by the absence of fans.

Many years ago, the researches in refrigeration and heating areas through vapor compress cycle were made by simple experiments and calculus. In sixth decade, the model technique started to be used. With the advance info in the seventies, the numerical models utilization consisted of an important tool to make studies about vapor compress machines behavior. The model of system is constituted by many components that have been adopted, in general, the modular structuration technique, where each modulus corresponds to each component system model. In refrigeration systems cases, one of the first advantages of the modular structuration is the possibility of a system project optimization through separated study of its components by comparative simulations. In particular, the refrigeration systems modeling come allowing to simulate the results of geometrical changes and traditional refrigerant fluids substitution by other ones not noxious to the ozone layer across refrigerant behavior simulations. To this, a simple database shift in model is made.

The objective of this work is the development of a mathematical model in non-standing regimen to simulate the static evaporator operation of an air-water heat pump. With the model, some simulations had been carried through, that had allowed to test geometric parameters of the system (diameter of the pipe, surface of the plate, etc), materials for the pipes and plates and different weather conditions. These computational tests had indicated that the model represents a good tool to project static evaporators.

2. THE VAPOR COMPRESSION CYCLE

It is possible to summarize the vapor compression cycle in four stages, Fig. 1. Initially the fluid in a state of superheated vapor and a low pressure (point 1) follows to the compressor which has increased the pressure and temperature, but remains in a state of superheated vapor (point 2). Then the refrigerant follows to its first heat exchanger where it passes through the condenser and exchanges heat with the heat source, keeping constant pressure and reducing the temperature. At the end of this process (point 3) the refrigerant is in a subcooling state. A decrease in pressure is then imposed by the expansion device, the refrigerant is replaced by lower pressure and temperature and at this point (point 4) its state is liquid-vapor mixture. To complete the cycle the refrigerant moves to the second heat exchange that happens in the evaporator, the refrigerant exchanges heat with the heat sink, changes phase and superheats, following to the point 1, completing the cycle.

Four components can be highlighted in this cycle. The compressor, the condenser, the expansion device and the evaporator. The latter is the main theme of this paper, as it was already explained that, this paper cares of the mathematical model of a static evaporator heat pump. However, it was observed that auxiliary models were needed to feed the model of the evaporator, thus it was part of this work to develop the mathematical model of the compressor and expansion device too.

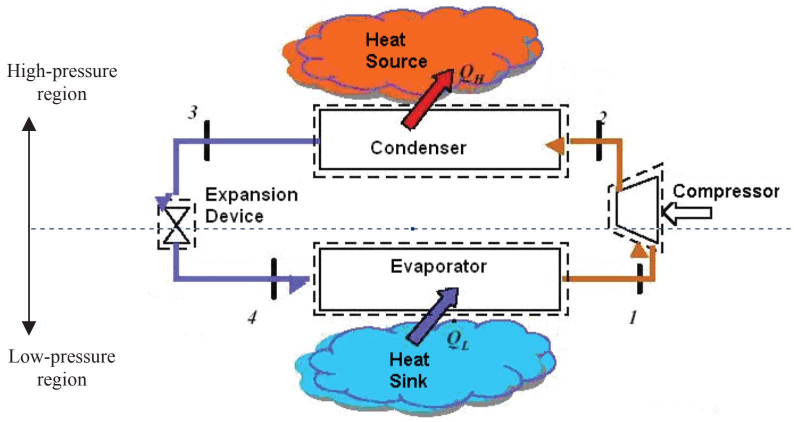

Figure 1. Schematic drawing of a cooling and heating system by vapor compression.

Because it is a heat pump air-water heat exchange in the heat sink is done with ambient air while the heat supplied to the heat source is used in the tank to heat water for residential use.

3. MODEL OF THE DEVICE OF EXPANSION

The evaporator model requires the value of the mass flow of refrigerant in its entry (m_{exp}). This value must be provided by the model of the expansion device. The expansion device chosen for this study was the orifice plate. This choice was made due to low cost and ease of fabrication of this device and it is a heat pump with small variations in operating conditions.

The hypothesis for this model are that the flow is one-dimension within the tube, the expansion process is adiabatic and the refrigerant is pure, uncontaminated by the compressor oil.

The Eq. 1 determines the format of the orifice plate, this equation can be found in Nunes (2010).

$$\dot{m}_{exp} = C_D \sqrt{\left(\frac{P_{cd} - P_{ev}}{v_{exp}}\right)} \tag{1}$$

C_D is the discharge coefficient which depends on the area of the orifice plate. The condensing pressure (P_{cd}) and specific volume at the entrance of the expansion device (v_{exp}) are obtained through a function that simulates the heating of water. The evaporation pressure (P_{ev}) is obtained by initial guess as it will be explained later.

4. MODEL OF COMPRESSION.

Another crucial variable for the model of the evaporator is the flow of refrigerant in its output (m_{comp}). This value is given by the model of the compressor. The compressor chosen for the heat pump was the Embraco FFI12HBX, hermetic with 11.14 cc and a rotation of 3500 rpm.

The hypothesis for this model were that the adiabatic compression process is irreversible, loss of pressure in the aspiration and discharge valves are negligible, the mass flow is constant during compression, the refrigerant is pure, uncontaminated by the compressor oil.

Maia (2005) brings the following equation to calculate the mass flow of compressor.

$$\dot{m}_{comp} = \frac{NV\eta_v}{v_{comp1}} \tag{2}$$

Where N is the rotation, V is the volume of the compressor, ν_{comp1} is the specific volume of the refrigerant at the compressor inlet and η_v is the volumetric efficiency of the compressor. The latter was calculated based on Eq (3) that was shown in Maia (2007).

$$\eta_v = 0,757 - 1,244\times10^{-7} P_{cd} + 7,791\times10^{-7} P_{ev} - 5,047\times10^{-15} P_{cd}^2 - 1,185\times10^{-12} P_{ev}^2 + 2,089\times10^{-13} P_{cd} P_{ev} \tag{3}$$

This equation was developed using multiple regression for this particular compressor, whose data were supplied by the manufacturer. If it changes the compressor model, new regressions will be required.

5. MODEL OF STATIC EVAPORATOR

In preparing the model of the static evaporator hypotheses have been raised. First hypothesis is that the relative magnitudes of the fluid refrigerant is evenly distributed in each cross section of the tube, in addition, the fluid flow was considered one-dimensional, pressure losses and heat in the evaporator return curves were disregarded, the refrigerant was considered pure, uncontaminated by the compressor oil, the fin acts as if it was perfectly fixed to the tube, with no contact resistance and temperature of the walls of the laboratory was approximated as been equals the room temperature.

In static evaporator's model are provided initial and boundary conditions. These are the variables of the system of differential equations that were used for the calculation. These variables are the mass flows into and out of the evaporator, provided by the models of the expansion device of the compressor, temperature and relative humidity of environment and the dimensions of the evaporator. Figure 2 shows these variables.

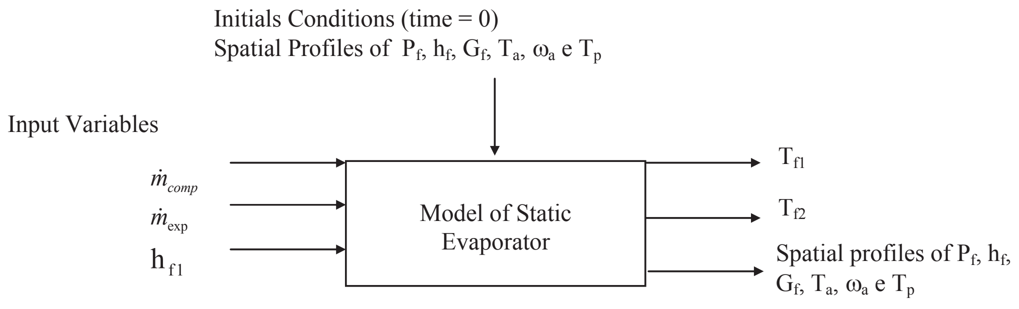

Figure 2. Block diagram of the model of the static evaporator

Where G_f is the mass velocity of the fluid, h_f is the fluid enthalpy, T_p is the temperature of the tube wall, T_a the environmental temperature, ω_a is the environment absolute humidity, P_f is the fluid pressure, T_{f1} is the fluid temperature at the evaporator inlet, T_{f2} is the fluid temperature at the evaporator outlet.

Once provided the system of equations input variables of the model, using the equations of energy, continuity and momentum, respectively shown in Eq.4, Eq.5 and Eq.6 is possible to trace the spatial profiles fluid temperature, mass flow and all the magnitudes resulting from these.

$$\frac{\partial \rho_f}{\partial t} + \frac{\partial G_f}{\partial z} = 0 \tag{4}$$

$$\frac{\partial}{\partial z}\left\{P_f + G_f^2\left[\frac{x^2 v_v}{\alpha} + \frac{(1-x)^2 v_l}{1-\alpha}\right]\right\} = -\frac{\partial G_f}{\partial t} - \left(\frac{dP}{dz}\right)_f - g\rho_f sen(\theta) \tag{5}$$

$$A_i \frac{\partial}{\partial t}\left[\rho_f\left(h_f - P_f v_f\right)\right] = H_i p_f\left(T_p - T_f\right) - A_i \frac{\partial}{\partial z}\left(G_f h_f\right) \tag{6}$$

Where A_i represents the cross-sectional area of the inner tube, T_f the temperature of the fluid, x the quality of the fluid, v_l the specific volume of liquid, v_v the specific volume of steam, ρ_f the specific mass of the fluid, α is the void fraction, H_i the convective coefficient internal, g the gravity acceleration and θ the inclination of the fluid flow. Their mathematical proofs of these equations can be found in Machado (1995).

The mathematical model to simulate the static evaporator operation was prepared in Fortran computer language. The system of equations was solved for each time step, and each time it arbitrates the value of evaporating pressure (P_{ev}). The model calculates all the spatial profile coming from the evaporator, and finally, the last control volume. The compressor model calculates the mass flow rate imposed by it, this flow is compared to the outflow from the last output volume control of the static evaporator. If the value of the flow imposed by the compressor doesn't coincide with the flow supplied by the model of the evaporator static a new evaporating pressure is estimated, and this is corrected by using the Newton Raphson. This process is iterative and is repeated until the value is converged. Then the model follows to calculate the external energy balance, the system of equations governing the balance is intended to find the temperature of the tube wall, if the temperature found doesn't match the temperature profile of the wall that was initially arbitrated the whole process restarts and the calculations are repeated until the temperatures stabilize.

The Fig. 3 shows a detailed flowchart of the static evaporator's model

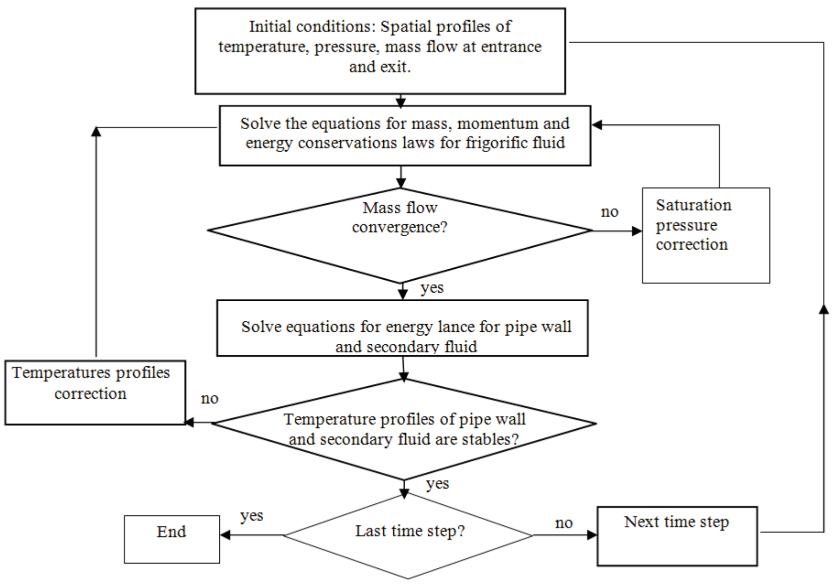

Figure 3. Flowchart of the static evaporator's model

6. METHODOLOGY

Initially it provided a default value for the environmental conditions and the dimensions of the static evaporator. The set of values was called of model base. For environmental conditions it was used the average weather of Belo

Horizonte. For the initial dimensions of the evaporator have been used those of the static evaporator already existing in the Laboratório de Regrigeração e Bomba de Calor of UFMG, showed in Nunes (2007). This consists of a flat copper fin with 1 mm thick which is fixed a tube with 6.35 mm in diameter; fig. 4 illustrates the other dimensions of this heat exchanger, in millimeters.

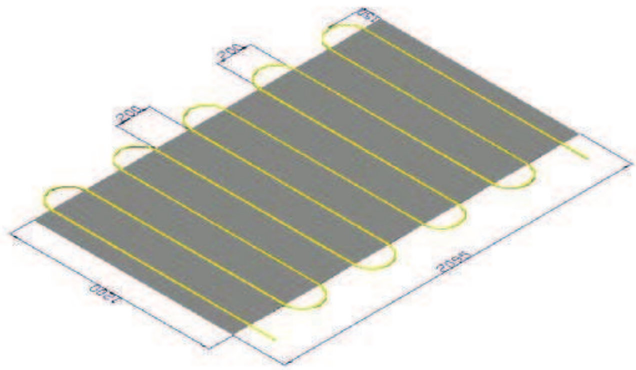

Figure 4. Static evaporator with the dimensions in millimeters.

In the first simulations, where it was used for these the base model, it was tested the grid of the model. First test was a spatial grid, which found that a number of control volumes not less than 565 and no larger than 950 should be adopted. Subsequently a grid temporal test was made that showed that any time step between 2 and 10 seconds could be adopted.

7. ANALYSIS AND RESULTS

The behavior of the evaporation temperature during the start of the heat pump, shown in Fig. 5, can be analyzed as follows. In a first moment (t=0) the fluid inside the evaporator is in an inert state (environmental temperature). After the start, the mass of fluid in the evaporator decreases due to the fact that the output flow is greater than the input flow at the beginning, with the mass reducing is natural that the temperature also reduces, due to decreased pressure. The temperature will continue to decline until the flow output become smaller than the input, this will occur in approximately 20 seconds after the start, this time the evaporation pressure begins to increase. It will increase until the instant that the flow rates become equal so the evaporation temperature stabilizes, which occurs between 150 and 200 seconds, approximately. This behavior is similar to those found by MacArthur (1989), Toub (1981) and Chi (1982).

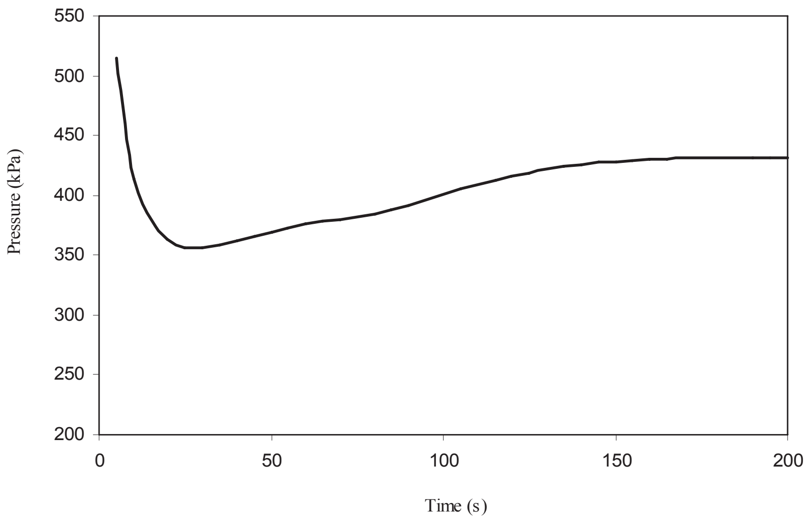

Figure 5. Behavior of evaporating pressure in time

Figure 6 shows the curves of input and output flow. It's possible through this graph to confirm what is observed in Fig 7. The moment of about 20 / 2, where the inflow is now greater than the output and the time between 120 and 200 / 2, where the flows are equal and the evaporation pressure stabilizes. Figure 6 shows the evolution of the mass amount of fluid inside the evaporator.

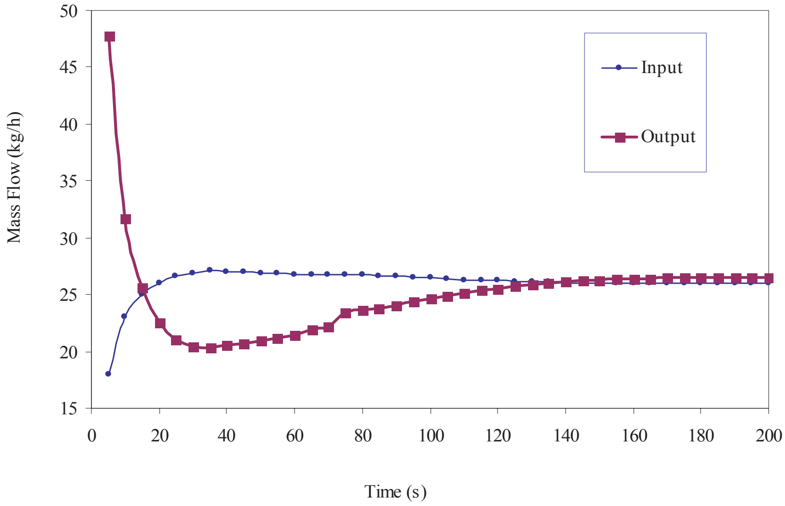

Figure 6. Evolution of the inflow and exit in time

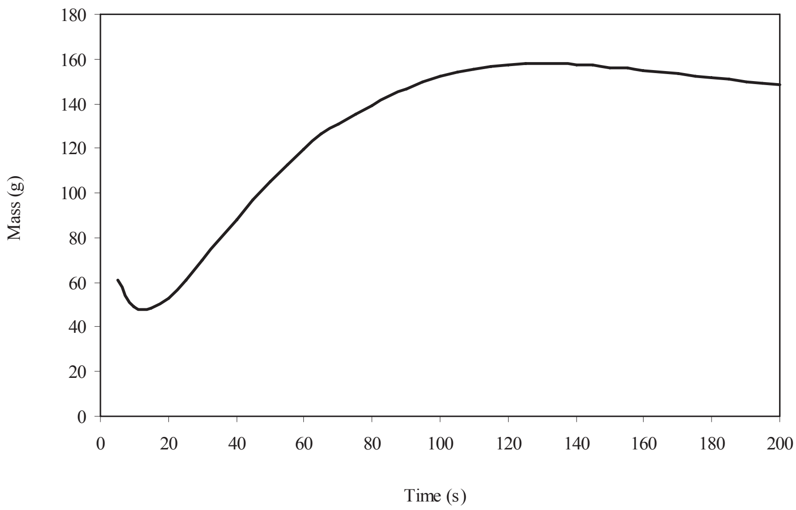

Figure 7. Behavior of the mass of fluid inside the evaporator

7.1. Simulations

The static evaporator will be exposed to ambient air that will face throughout the day and year large variations in temperature. Thus, it is important to know the behavior of the evaporator with the room temperature variations.

During the simulations, was varied room temperature (Tamb) of 15°C to 35°C. The simulations had the purpose to provide a possible trend of the evaporator in response to the gradual increase of room temperature.

Figure 8 shows the time evolution of temperature in different room temperature situations.

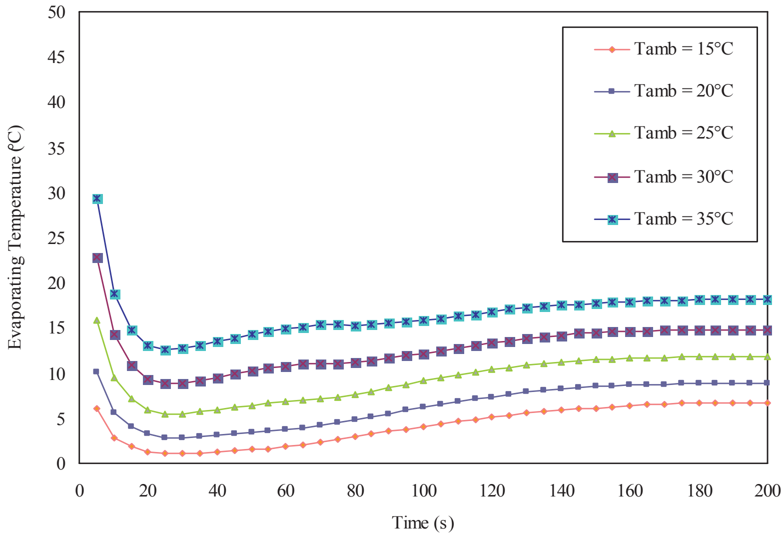

Figure 8. Influence of room temperature on the evaporation temperature

As might be expected, the graph shows an evaporation temperature increase with increasing room temperature. During the passage of fluid through the evaporator, which receives heat from the environment, so an environment with a higher temperature will provide a greater heat transfer resulting in an increase in evaporation temperature. In addition, the fluid already starts from higher temperature.

The fig. 9 and fig. 10 show the response of the input and output flow to a gradual increase of room temperature.

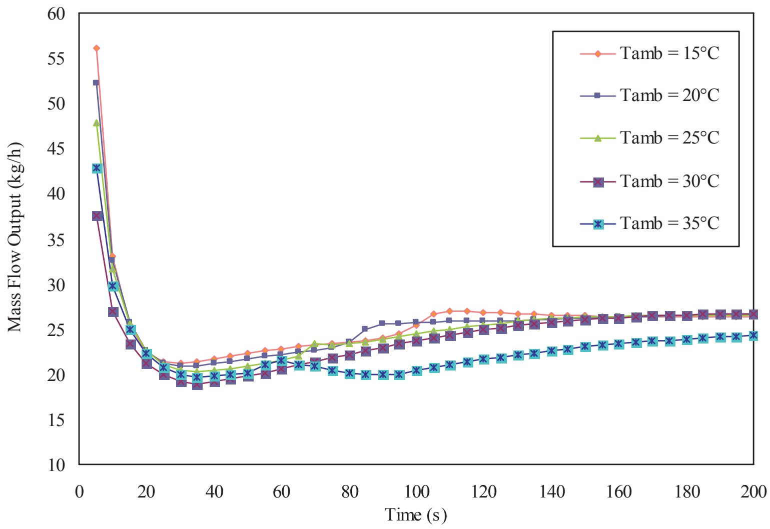

Figure 9. Influence of room temperature on the flow input

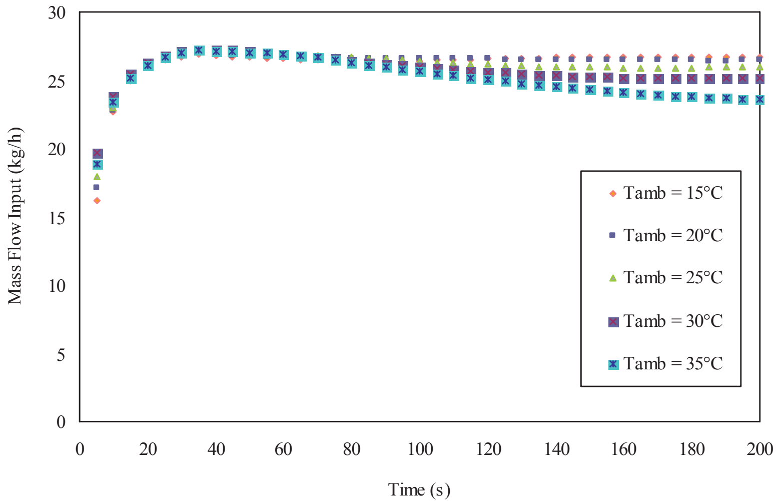

Figure 10. Influence of room temperature on the flow output

Unlike what happened with the evaporation temperature, flow rates don't show significant change with variations of room temperature.

From these simulations, It can be concluded that the room temperature will influence the performance of the heat pump. A variation in the evaporation temperature will cause changes in the work of compression, so it was calculated the COP (Coefficient of Performance) for different room temperatures, as shown in fig. 11.

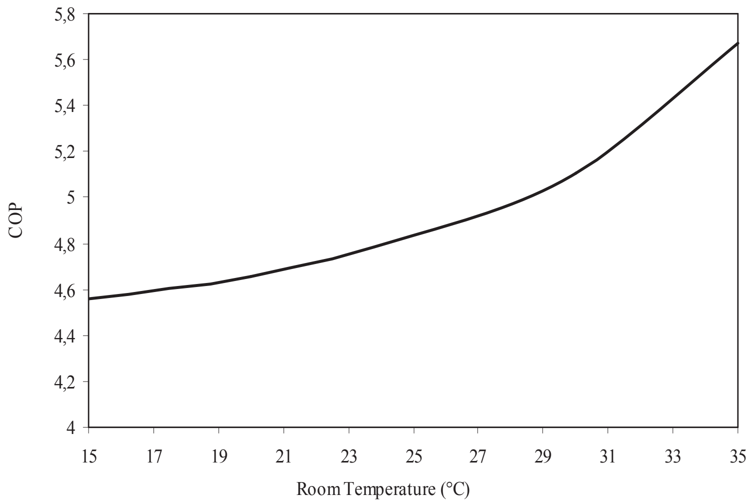

Figure 11. Variation of COP with room temperature

The calculation of the COP was made with the evaporation temperature already stabilized. The graph confirms what was expected. The change in room temperature has a significant role in the performance of heat pump. The curve shows a trend that indicates that an increase in room temperature causes an increase in COP of heat pump, which was already expected since an increase in room temperature leads to a greater external convective coefficient and the refrigerant starts from a higher temperature, reducing the compression work.

7.2. Length of the fin

One goal of this work is to know the detailed operation of the static evaporator and from this to be able to design it. For the initial dimensions of the evaporator have been used those of the static evaporator already installed at the Laboratório de Refrigeração e Bomba de Calor of UFMG, as it has already been mentioned in this paper.

Through simulations on the mathematical model it was changed the dimensions of the fin and tube length keeping the same thermal load. These simulations were intended to creating various types of settings for the evaporator with the same thermal load, so it could be possible to determine the most economically viable option for the static evaporator.

Figure 12 illustrates the relation found for the length of the tube and the length of the fin with the same thermal load.

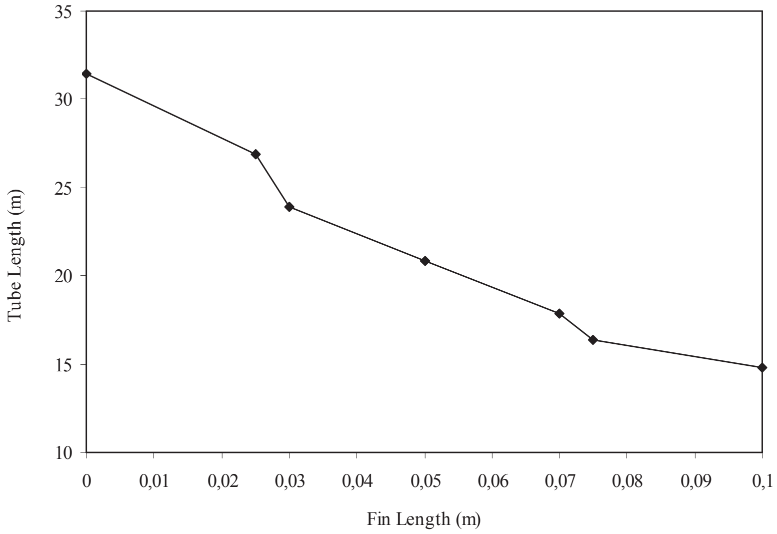

Figure 12. Relation between tube length and the fin length

Using this relation of the above graph it can be possible to make the cost calculation for different configurations of the static evaporator. The copper tube was quoted at U$ 4.00 per meter of pipe. Copper plate has a market price of U$ 23.00 per kilogram. The mass of fluid (R134-a) was U$ 21.00 per kilogram. To quote these products, research was done in several houses in the industry. The values of the cost of manufacturing were not considered in the calculation, because it is judged that the process would be almost the same independent of the length of the tube or plate.

The relationship between price and fin efficiency is shown in fig. 13.

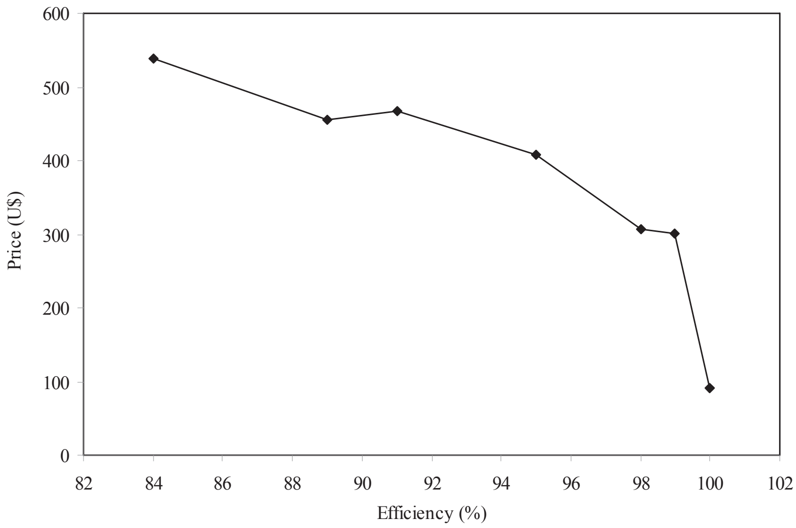

Figure 13. Relation between cost and fin efficiency

It is noticed that the price decreases with increasing fin efficiency. The efficiency of a fin is a measure of the ratio of the real heat transmitted through it and the heat was transmitted if the whole fin were at the same temperature as the base, so the shorter fin guarantees a greater efficiency. The 100% efficiency occurs, of course, when the length of the fin becomes zero, or when there is no fin.

From the results of these simulations it's possible to draw a conclusion. Looking only at the static evaporator operation, due to the high cost of copper plate, it is more economically viable a static evaporator with no fin. There is a variable that has not been discussed, the pressure loss would influence the work of compression, but since this work only deals with the static evaporator this variable was not analyzed. In the future a more detailed model with a complete heat pump would bring a relation between cost and efficiency that would include the pressure loss.

8. CONCLUSIONS

The mathematical model of static evaporator proved to be very stable and a powerful tool for analyzing its behavior in the transient regime. The processing time of each simulation is less than two minutes, a fact that allowed to test a wider range of simulations. Besides the low processing time, the model showed little sensitivity to changes in its input data, the adjustments in the estimation of evaporation pressure almost wasn't necessary during the simulations. Although it has not yet been possible to make the experimental validation it's possible to say that the results presented by the model were consistent.

Despite the results, some interesting conclusions were drawn. About the environmental conditions, it is obvious the great influence of room temperature in static evaporation operation, highlighted in this article. Another interesting conclusion is that it would be more economically practicable to build a static evaporator with no fin, due to the high cost of copper plate. A static evaporator with no fin and a greater tube length could supply the required thermal load with a lower cost, analyzing only the evaporator, since the pressure loss can not be evaluated.

It is hoped that with the information contained here, and some other that can be extracted from the model, since this is available for more tests, to build an air-water heat pump with static evaporator. With this mathematical model could be built a more efficient and cheaper heat pump. After this build, this model could be validated thoroughly.

9. ACKNOWLEDGEMENTS

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11. RESPONSIBILITY NOTICE

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