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 rush hour, has stimulated the study of water heating systems that substitute the electric shower. 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 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 improve this heat pump would be the substitution of its conventional evaporator for a static evaporator. 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. Some simulations had been carried through, that had allowed testing geometric parameters of the system, 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.

Keywords: Heat Pump, Static Evaporator, Mathematical Model

NOMENCLATURE

A	heat transfer area, m^2
D	tube diameter, m
F	friction factor
f_{LM}	Lockhart-Martinelli correction factor
G	mass velocity, $kgm^{-2}s^{-1}$
h	specific enthalpy, $kJ\ kg^{-1}$
H	heat transfer coefficient, $Wm^{-2}K^{-1}$
N	rotational speed, rpm
$\dot{m}$	mass flow rate, $kg\ s^{-1}$

Greek symbol

α	void fraction
η	efficiency
ρ	specific mass, kgm^{-3}
ω	humidity

Subscripst

1	input of the volume control
2	output of the volume control
a	air
comp	compressor
cond	condensation
exp	expansion

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 significant 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since water heating by electrical resistance method that is used almost exclusively in Brazil, is considered an outdated technology.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 presented 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.

In recent years, it is possible to observe an increase in interest in heat pump assisted by radiation due to the fact that the systems converting primary energy into thermal energy in a much more efficient than the traditional heat pumps resulting in COP (Coefficient of Performance) higher.

Nowadays, studies have shown that the combination of the evaporator collector and forming a single unit using a coolant improves the performance of the whole system as compared to traditional collectors using air or water as working fluid. In these systems, the refrigerant from the condenser is sent to the collector-evaporator unit where it then evaporates by absorbing solar radiation, and these so called systems of heat pumps for solar radiation with direct expansion. The uses of solar evaporator heat pumps are very convenient since addition of solar radiation. Other energy can be collected as, for example, the sensible heat and latent heat from the atmosphere coming from the condensation effect.

Gobbé (1983) developed a thorough study for the determination of the main modes of heat transfer (heat inputs) in a static evaporator coupled to a heat pump operating with R12. In order to provide for heat exchange with the evaporator means, various models have been proposed in order to conduct the geometrical optimization of the system.

Alloula (1986) presented a model relating to the operation of a static evaporator coupled to a heat pump. It works with R12 for heating water. In this study, we evaluated the importance of the influence of each meteorological parameter (thermal contributions) in the energy balance of the evaporator in order to have a proper sizing of the same.

Pereira Neto (1988) presented a theoretical and experimental study of a heat pump for heating water, where the expansion of R12 is made in a solar collector. The author developed a mathematical model of the evaporator, which was confronted with experimental results obtained from tests with the prototype. By simulating the operation of the heat pump was analyzed the influence of important parameters such as overheating of the refrigerant, the compressor speed and efficiency than solar radiation incident on the system performance.

Kuang et al. (2003) conducted an analytical and experimental heat pump evaporator plan with 2 m² of area for solar water heating. A simulation model has been developed for calculating the thermal performance of the system. The monthly average presented COP values of the order 4-6 with the collector having an efficiency of 40 to 60%. The results obtained from the model were used to optimize the design and determine the control strategy to be adopted. The influence of various parameters such as solar radiation, ambient temperature, the area of the collector, the tank volume and speed of the compressor in the pump performance was investigated. The results indicated that system performance is strongly influenced by the variation of solar radiation, solar collector area and the compressor speed.

Chata et al. (2005) conducted a research on the thermal performance of a heat pump with solar radiation by operating with direct expansion refrigerants with two different configurations for the collector: with and without coverage. The fluids used were 12-R,-R 22, R-134a, R-404A and R-407C. The results show that R-12 has the higher COP, followed by the R-22 and R-134a. For the mixtures, R-410A was more effective than R-407C and 404A-R, but compared to the results were lower than R134a.

Li et al. (2007) performed an exergy analysis of a heat pump with direct expansion solar radiation operating with R-22, power of 750 W through experiments conducted in the winter. According to Li et al. Elements with higher exergy loss in the compressor and the evaporator were, followed by collecting condenser and the expansion valve. According to the authors, to ensure a perfect match between the compressor capacity and the capacity of the evaporator collector, when operating under the same thermal load and environmental variables, must use a compressor with variable speed and the electronic expansion valve.

Chow et al. (2010) developed a numerical simulation model for a heat pump using direct expansion R-134a as the working fluid to investigate the capability of the pump in heating water for domestic use in the city of Hong Kong. From the simulation results obtained from the weather data, the authors obtained a COP of 6.46, which is well above a traditional heat pump.

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 the simulation 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, and others), materials for the pipes and plates and different weather conditions.

These tests had indicated that the model represents a good tool to project static evaporators.

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.

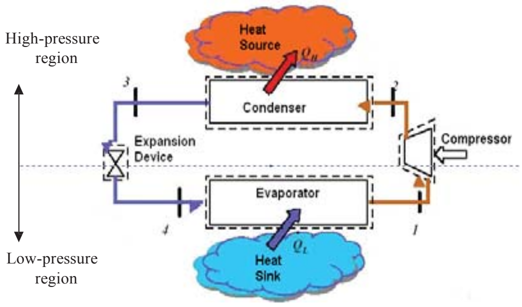

Figure 1. Schematic drawing of a cooling and heating system by vapor compression.

Four components can be highlighted in this cycle. Compressor, condenser, expansion device, and evaporator. The latter is the main theme of this paper, as it has 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.

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.

MODEL OF THE DEVICE OF EXPANSION

The evaporator model requires the value of the mass flow of refrigerant in its entry ($\dot{m}_{exp}$). 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.

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

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

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.

MODEL OF COMPRESSION

Another crucial variable for the model of the evaporator is the flow of refrigerant in its output ($\dot{m}_{comp}$). 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, pressure loss in aspiration and discharge valves are negligible, mass flow is constant during compression, refrigerant is pure, uncontaminated by the compressor oil.

$$\dot{m}_{comp} = NV\eta_v v_{comp1}^{-1} \qquad (2)$$

Maia (2005) brings the equation shown above to calculate the mass flow of compressor. Where N is the rotation, V is the volume of the compressor, v_{comp1} is the specific volume of the refrigerant at the compressor inlet, and η_v is the volumetric efficiency of the compressor.

$$\begin{aligned} \eta_v = {} & 0,757 - 1,244\times10^{-7} P_{cd} + \\ & 7,791\times10^{-7} P_{ev} - 5,047\times10^{-15} P_{cd}{}^2 - \\ & 1,185\times10^{-12} P_{ev}{}^2 + 2,089\times10^{-13} P_{cd} P_{ev} \end{aligned} \qquad (3)$$

Volumetric efficiency was calculated based on Eq. (3) that was shown in Maia (2007).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.

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 the room temperature.

In static evaporator 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.

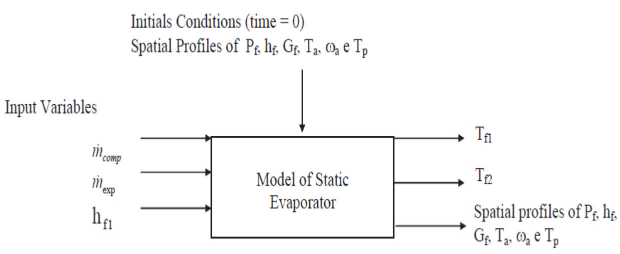

Figure 2. Block diagram of the model of the static evaporator.

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.

$$\frac{\partial \rho_f}{\partial t} + \frac{\partial G_f}{\partial z} = 0 \qquad (4)$$

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

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

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

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, ν_l the specific volume of liquid, ν_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.

Figure 3 shows a detailed flowchart of the static evaporator model.

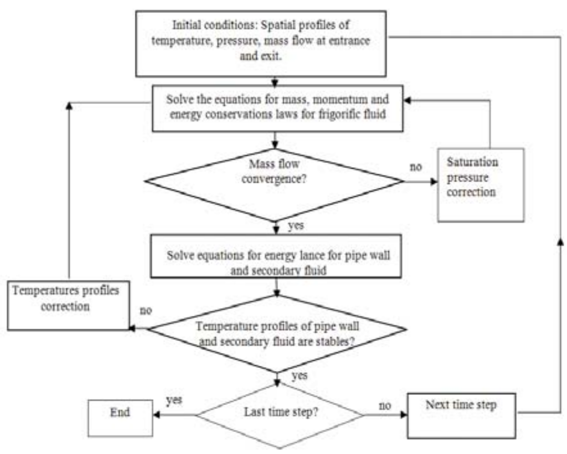

Figure 3. Flowchart of the static evaporator's model.

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 Lab of Heat Pump and Refrigeration in UFMG, showed in Nunes (2007). This consists of a flat copper fin with 1mm (0.0393 in) thick which is fixed a tube with 6.35 mm (0.25 in) in diameter.

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.

ANALYSIS AND RESULTS

The behavior of the evaporation temperature during the start of the heat pump, shown in Fig. 4, 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), Touber (1981) and Chi (1982).

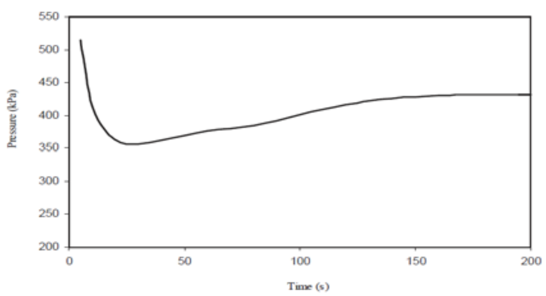

Figure 4. Behavior of evaporating pressure in time.

Figure 5 shows the curves of input and output flow. It's possible through this graph to confirm what is observed in Fig. 6, evolution of the mass amount.

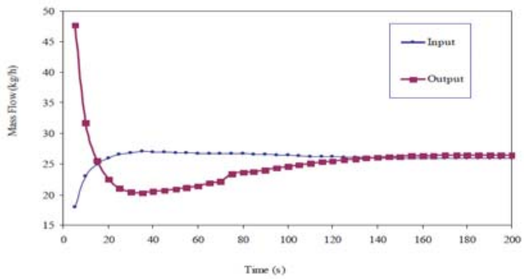

Figure 5. Evolution of the inflow and exit in time.

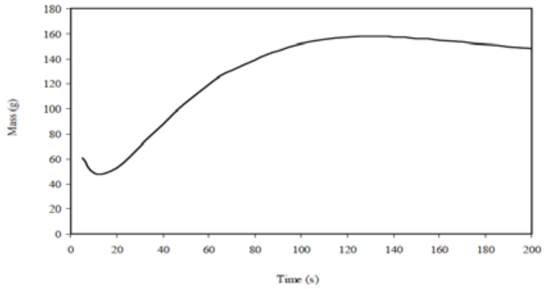

Figure 6. Behavior of the fluid mass inside the evaporator.

The moment of about 20 s, where the inflow is now greater than the output and the time between 120 s and 200 s, where the flows are equal and the evaporation pressure stabilizes.

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. The simulations had the purpose to provide a possible trend of the evaporator in response to the gradual increase of room temperature, Fig. 7.

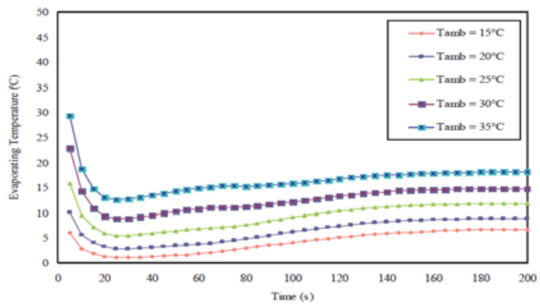

Figure 7. Influence of room temperature on the evaporation temperature.

As might be expected, the graph shows an evaporation temperature increase with increasing room temperature.

During the passage of fluid through the evaporator, this 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.

Figure 8 and Fig. 9 show the response of the input and output flow to a gradual increase of room temperature.

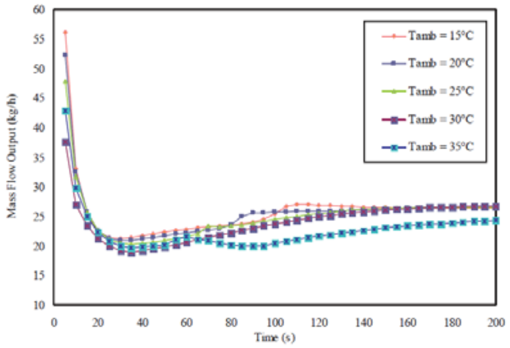

Figure 8. Influence of room temperature on flow input.

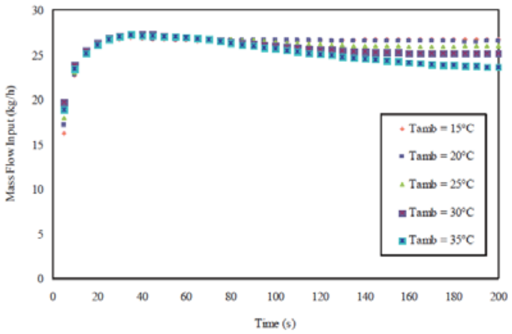

Figure 9. Influence of room temperature on flow output.

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.10.

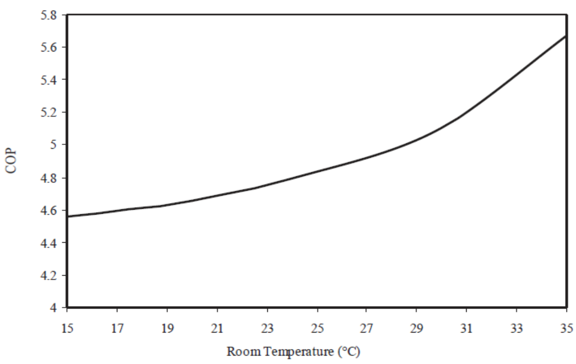

Figure 10. Variation of COP with room temperature.

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.

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 Lab of Heat Pump and Refrigeration in 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 11 illustrates the relation found for the length of the tube and the length of the fin with the same thermal load.

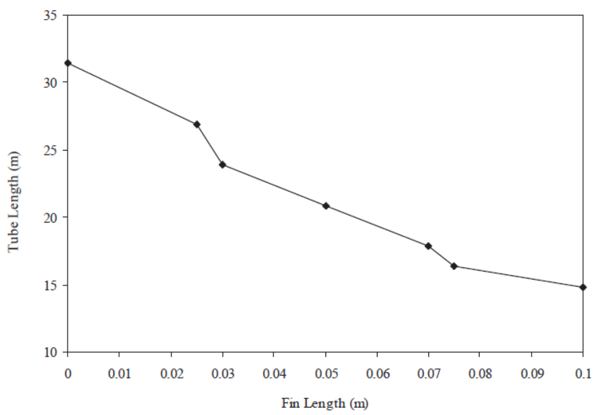

Figure 11. Relation between tube length and the fin length.

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. 12.

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.

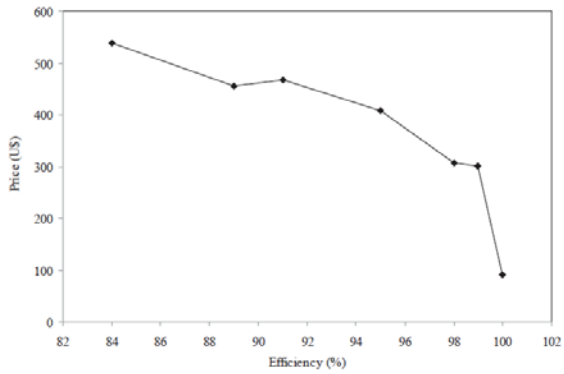

Figure 12. Relation between cost and fin efficiency.

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.

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 testing 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 cannot 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.

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