DYNAMIC MODEL OF A HEAT PUMP FOR RESIDENTIAL WATER HEATING

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Abstract. *One of the goals of this work is to study the dynamic behavior of small heat pump for residential water heating. For this, a vapor compression heat pump was designed, modeled and built. The heat pump is constituted basically by a small alternative compressor, a capillary tube, an air evaporator and a condenser coil immersed in a 200 liter water reservoir. Mathematical models for the evaporator and the condenser were developed from mass, momentum, and energy conservation laws, while the mathematical model for the capillary tube was based mainly on momentum conservation. These models use some relationships from bibliography to estimate the heat transfer coefficients, pressure drop and void fraction. The compressor model was developed based on experimental data from manufacturer and also of the energy conservation law. The coupling among the four component models allowed the determination of the spatial and temporal profiles of temperatures, pressures and mass flows, as well as the refrigerant mass distribution in the heat exchangers during the water heating process. The model was validated thought from experimental results obtained in a heat pump test bench operating with R134a. Measurements of temperatures, pressures, mass flows, electric current and voltage allowed the temporal evaluation of the temperatures and heat pump performance during the water heating process. Additional simulations were accomplished in order to analyze the system operation under various operating conditions.*

Keywords*: Heat Pump; Mathematical Model; Dynamic Operation*

1. INTRODUCTION

The water heating using electrical resistence is responsible aproximately by 26% of residencial expenditure electrical energy in Brazil nowadays, according BEN (2003). Therefore a new way to obtain heat water spending less electrical energy would take away a significative reduction on national lawsuit, reducing the risks of an energetic collapse, like occurred in 2002.

With the objective to implement a new water heating system, its propose a method that already much diffused in Europe. This is based in use of a free energy present in ambient air for water heating using an equipament called heat pump. With this system we get a significative reduction of electrical energy expenditure when it compared with electrical shower.

In many years ago, the researchs in refrigeration and heating areas through vapour compress cycle was made by simple experiments and calculus. In sixty decade, the model technique started to be used. With the advance info in the seventies, the numerical models utilization consisted in an important tool to make studies about vapour compress machines behaviour. In modeling of a system costituted by many components 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 otimization 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 ozon layer across refrigerant behavior simulations. To this, a simple data bench shift in model is made.

The objective of this work is to present the development of a numerical model to simulate the behaviour of a heat pump to water haeting for residential use. Still in this work the model will be validated by a comparation of its results with experimental results obtained in a prototype developed by UFMG´s Refrigeration Group. The UFMG´s Refrigeration Group wishes to build a low cost heat pump to water heating for residential use, with the objective to introduce an efficient alternative to use electrical resistence for residential water heating.

2. FRIGORIFIC MACHINE FUNCTION PRINCIPLE

There are four main components of a vapour compress machine (frigorific systems or heat pumps), the condenser, the evaporator, the expansion device and the compressor, Fig.1. In this machine, a refrigerant fluid is submited a thermodynamics cycle composed by physical phenomena of condensation, expansion, evaporation and compression. In the condenser, a refrigerant fluid condenses rejecting the energy absorbed during its passage by the evaporator and compressor of machine. This energy is received by a secondary fluid that involves the condenser, named secondary fluid A. After left the condenser, the refrigerant fluid will cross the expansion device and next the evaporator. The objective of the expansion device is reducing the high pressure of condenser to the low pressure required in evaporator. In this component, the frigorific fluid evaporates receiving energy from fluid B. After the evaporator, the frigorific fluid goes to compressor that raises its pressure and it discharges back to the condenser closing the cycle.

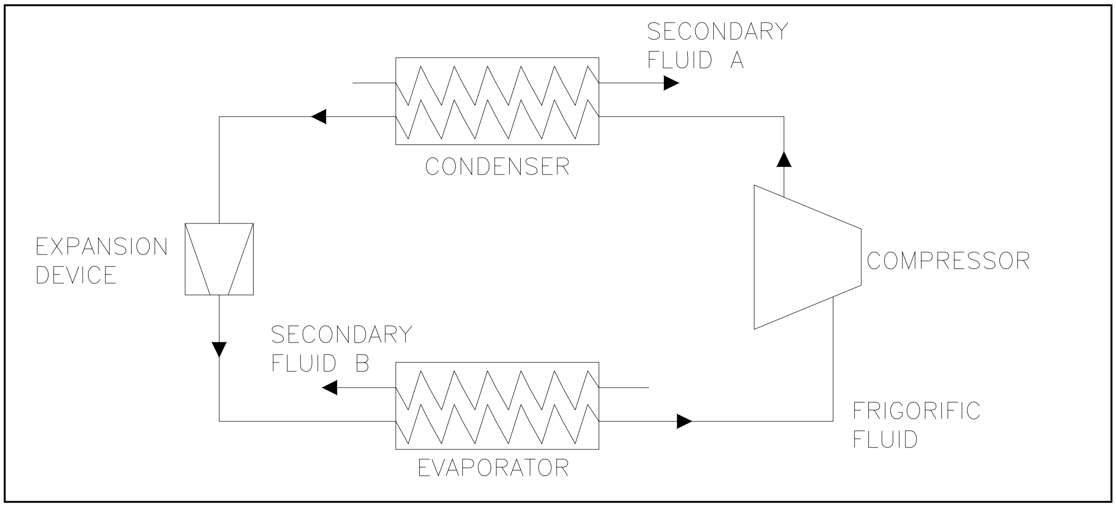

Figure 1. Main components of a refrigeration system.

The machine modeled in this paper is composed by a hermetic compressor, a capillary tube, a finned multitube evaporator where the secondary fluid is the air and a condenser in coil form who secondary fluid is the water that we wish to heat.

3. HEAT EXCHANGERS MODEL

The heat pump model divides in two main blocks that are the heat exchangers models, evaporator and condenser. In beginning, the heat exchanger to be divided in many control volumes. In each control volume, was applied the mass, momentum and energy conservation laws and since who are originated the enthalpy, mass flow and pressure equations for the frigorific fluid that are solved to obtain the outlet control volume values for these variables. To solve these equations it's necessary to know these values at inlet control volume. The enthalpy and mass flow are known for the first volume control, because they are entry variables to heat exchangers model, both imposed by the preceding component model, the compressor in the condenser case and the capillary tube in evaporator case. Therefore, the inlet pressure in the first control volume isn't known, as long as it's an outlet variable heat exchanger model. To solve this problem, we arbitrate any value to saturation pressure that will allow the refrigerant fluid equation system resolution. The pressure, enthalpy, and mass flow values calculated at outlet control volume will be used to feed the next inlet control volume. So, the variables' spatial profile relating to refrigerant fluid might obtain. During these profiles determination, the calculated value for enthalpy must be compared constantly with saturated liquid and saturated vapour enthalpy values with the objective to become determined the transition points between one-phase and two-phase flow regions. In one-phase flow regions, we used the Dittus-Boelter correlation to calculate heat transfer coefficient. In two-phase flow regions, we used Shah correlation, Shah (1979), to estimate heat transfer coefficient for condensation, and at evaporation case we used the Dengler and Addoms correlation. The pressure loss in two-phase flow regions was determined by Lockhart and Martinelli (1949) correlation and the void fraction model according to Hugmark (1962). To the last control volume, the calculated mass flow represents the outlet exchanger mass flow. This mass flow must be compared with the next component model's mass flow, the compressor for evaporator case and the capillary tube for condenser case.

The difference between these values is made and this value will be compared to a predetermined error. If it's bigger than error, so the saturation pressure must be changed and the calculus are repeated. The Newton Raphson algorithm makes this correction. The calculus must be repeated until then an acceptable convergence was obtained between two mass flow values. Next, an energy balance at pipe wall and secondary fluid was made and the obtained equations are solved until then these temperatures profiles convergence been reached. This procedure was repeated until then the last time step been reached. The capillary tube model is similar to heat exchangers models, however the equations of mass and energy conservation laws are null, due to we consider the component as adiabatic and the mass flow is constant along its length. The compressor model determines the provided mass flow by component and it was developed based on experimental data from manufacturer and also from the energy conservation law. The Fig. 2 presents the flow chart of heat exchanger model function.

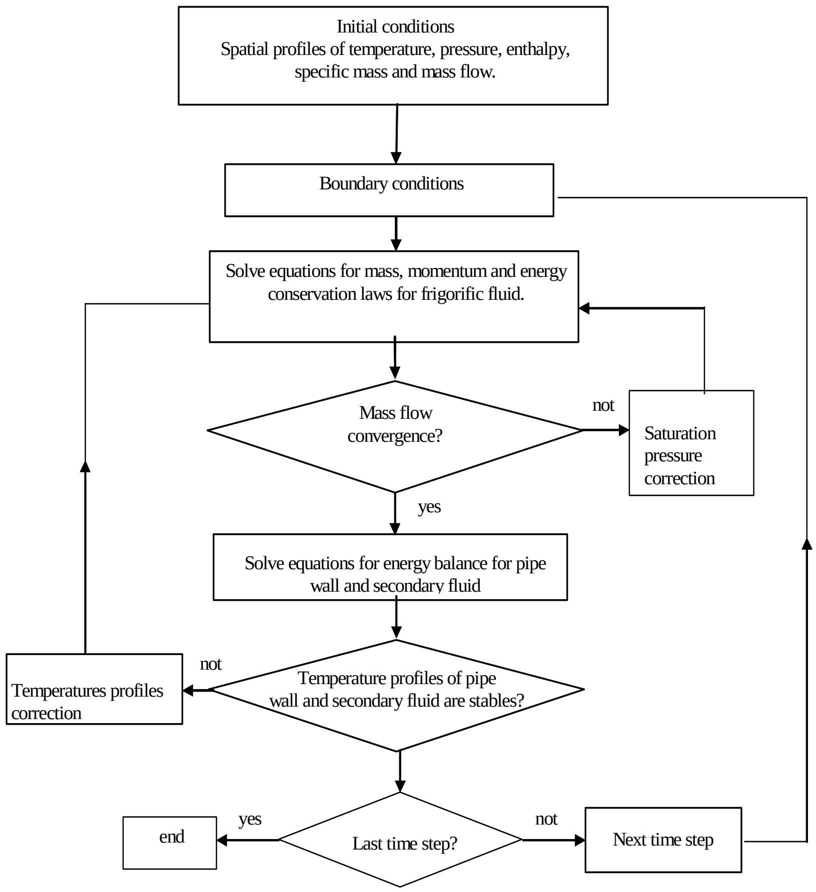

Figure 2. Flow chart of heat exchanger model function

4. RESULTS AND DISCUSSION

At this moment, the results obtained with FORTRAN language developed model will be presented for each main component for heat pump for water heating to residential application. For complete machine model development is necessary that each model works perfectly when it's individually analysed, because, for example, the results of compressor model are used like entry data at condenser model and so sucessively.

4.1. Compressor model results

Due the heat pump function the evaporation and condensation pressure values, so like the difference between the compressor admission temperature and evaporation temperature, suffer variations along machine operation zone, since those are closely connected to water reservoir temperature and the air cross flow at evaporator. It's necessary that this model be able to absorbe those changes, and it provides the compressor dicharge mass flow and enthalpy values for differents entry conditions, that must occur during the complete machine simulation.

4.2. Capillary tube model results

Before show the obtained results with capillary tube model is necessary to define the standard configuration used to capillary tube during the simulation, so, the entry data supplied in order that the model works. These data are the following: condensation temperature, 40°C ; evaporation temperature, -5°C ; the difference between the capillary tube admission temperature and condensation temperature, 7°C ; internal capillary tube diameter, 1,2mm; capillary tube length, 0,8m. Typical pressure and temperature distribution all along capillary tube, obtained by the model working with standard configuration, are presented on Fig. 3. The fluid temperature along the capillary tube stays constant until the flow reach the point C, meaning the beginning of two-phase flow region and since falling on accelerated way. At once the pressures falls of a linear way in one-phase flow region and since point C, the pressure fall give itself rough way. The title goes out zero at one phase flow region, reaching more than 25% at capillary tube outlet.

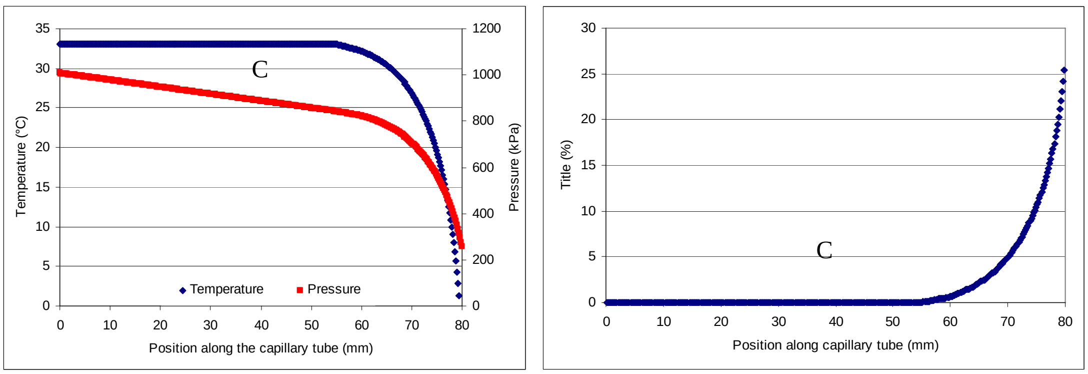

Figure 3. Temperature, pressure and title along the capillary tube

4.3. Evaporator model results

The Fig. 4 presents the frigorific fluid mass flow temporal evolution impose by compressor and capillary tube, and mass fluid evolution present in evaporator since the equipment start. In this figure the evaporation temperature, the difference between the compressor admission temperature and evaporation temperature, and the mean air temperature after it crosses the heat exchanger temporal evolution are represented. We can see that let’s get started the compressor aspire big fluid quantities while that the expansion device still works below the steady station flow condition. In that way, the refrigerant fluid been retired of the evaporator, that results in an abrupt reduction at low pressure that can be verified through the saturation temperature drop during this time interval. Since to reach a maximum, the compressor mass flow begins to fall while that the expansion device still going up. How the compressor flow still bigger, fluid mass continues leaving the evaporator and the fluid temperature remains falling. At 30 s, the mass flows equalize and during a time interval the capillary tube flow is bigger than the compressor flow. In that way, the evaporator starts to receive a mass gain, that determines a pressure raise and, consequently, at R134-a saturation temperature. Finally, around the 500 s, these mass flows equalize and the evaporator reaches the steady state.

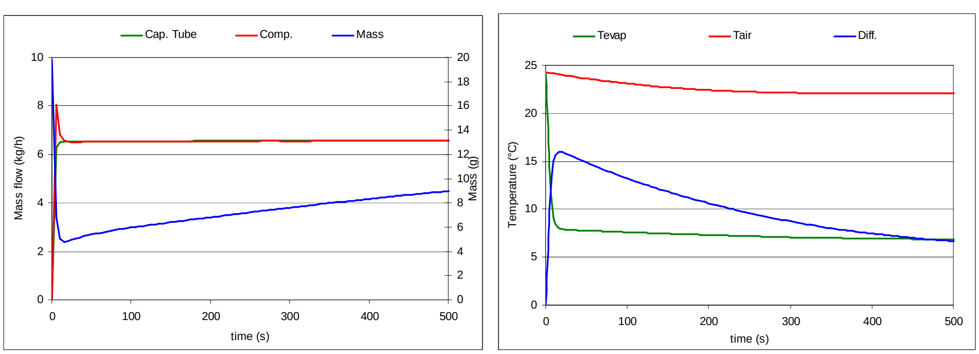

Figure 4. Mass flow, mass fluid and temperatures temporal evolution after an evaporator start

The Fig. 5 shows the R134-a, air and pipe wall temperature along the evaporator length. The almost constant fluid temperature at region included between the evaporator inlet and almost 3,5 meters, indicates that is the evaporation zone. This temperature fall in the two-phase flow region occurs due the pressure drop considered during the frigorific fluid flow inside the evaporator. The landing observed in the pipe wall and air temperature curves are results of fluid circuit in evaporator and the air cross flow. So, in the first 21cm of evaporator, the tube is receiving a previously cooled air by a control volume corresponding to a before or a next tube, that blended with heat transfer through the fins, results in a still more cooled air at control volume outlet.

Figure 5. Fluid, pipe wall and air temperatures along evaporator length

4.4. Condenser model results

The Fig. 6 presents the frigorific fluid mass flow temporal evolution impose by compressor and capillary tube, and mass fluid evolution present in condenser since the equipment start. In this figure the condensation temperature and the heat pump reservoir's water mean temperature temporal evolution are represented. We can see that let's get started the compressor aspire big fluid quantities while that the expansion device still works below the steady station flow condition. In that way, the refrigerant fluid has been inject in the condenser, which results in an abrupt increase at high pressure that can be verified through the saturation temperature raise during this time interval. Since to reach a maximum, the compressor mass flow begins to fall while that the expansion device still going up. How the compressor flow still bigger, fluid mass continues arriving the condenser and the fluid temperature remains growing up. At 20 s, the mass flows equalize and during a time interval the capillary tube flow is bigger than the compressor flow. In that way, the condenser starts to suffer a mass decrease, that determines a pressure drop and, consequently, at R134-a saturation temperature. Finally, around the 100 s, these mass flows equalize and the condenser starts to work in shorts steady states, because the water heating causes a condensation pressure raise and consequently a new operation point for the condenser.

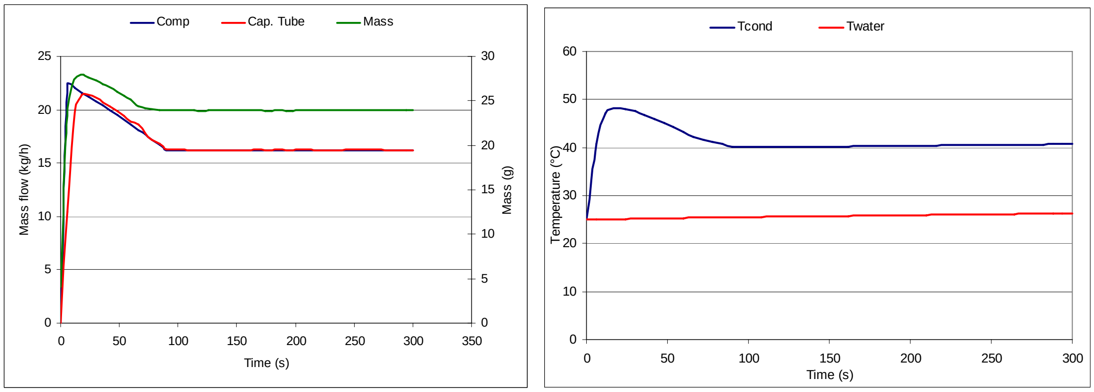

Figure 6. Mass flow, mass fluid and temperatures temporal evolution after a condenser start

The Fig. 7 shows the R134-a, water and pipe wall temperature along the condenser length. At condenser inlet, the most elevated temperature was found. This falls due the flow drop pressure, until that at a determinate point the saturation point was reached. The phase change occurs in a greatest portion of condenser length, and when the fluid was totally at liquid phase; its temperature turns to fall. The pipe wall temperature behavior was same the expected, because in the two-phase flow region, the heat transfer coefficient was bigger at the beginning and after goes lower gradually, that makes the pipe wall temperature approaches more the fluid temperature at this region start.

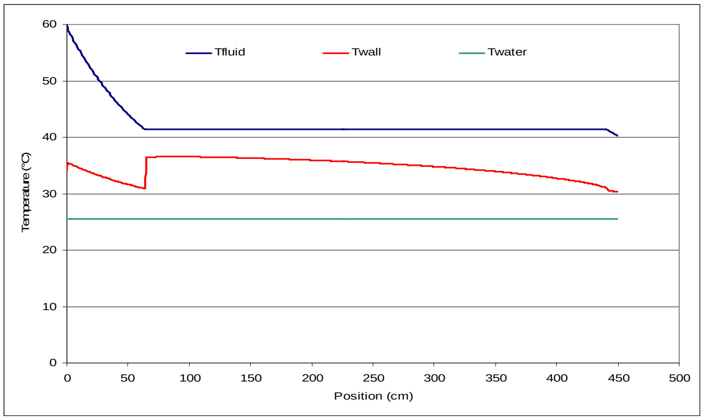

Figure 7. Fluid, pipe wall and water temperatures along condenser length

This mathematical model, without the validation, can be found on Koury, Maia, Castro e Machado (2007).

5. VALIDATION OF THE MODEL

The results that will be shown were found through the validation of the complete model of the heat pump on Maia (2007). There are two observations that must be presented at this moment about the developed model to simulate the functioning of the machine. The first observations is that the pipes, which are among the components, had not been considered. The model assumes that the results at the exit of the last volume of control in a component are the data of entrance for the subsequent component, and thus successively.

The second observations is that, due to the great difficulty to esteem the initial distribution of the refrigerant fluid in the exchangers of heat in the equipment, the values of necessary mass in each component in the machine's band of functioning had been calculated and assumed this value as initial condition of the model. After presenting these observations, the validation of the results will be showed.

In the Fig. 8 there are the condensation's profiles of temperature (Tcd), the evaporation's profiles of temperature (Tev) and the water's profiles of temperature (Ta), gotten through the simulation with the developed model (mod) and in the experimental tests (exp). The trend of the curves is similar, however the temperature of the water in the model grows in a more accented form, following the growth of the temperature of condensation, while in the experimental test, during the first hour, an oscillation in the temperatures of condensation and evaporation occurs, and then there is the trend showed in the simulation.

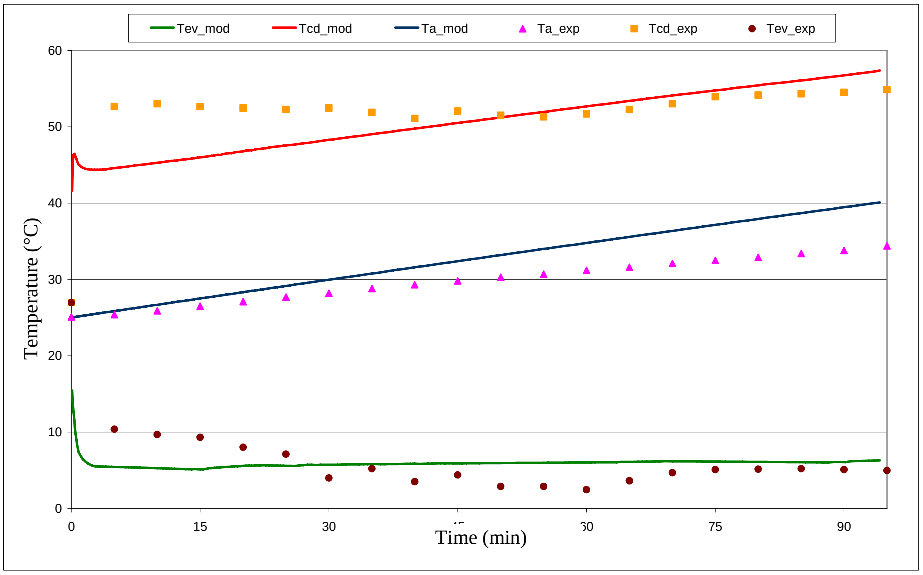

Figure 8. Evolution of the temperatures in the process of water heating

The Fig. 9 shows the mass distribution in the heat exchangers during the simulation being carried out. The initial value of the used for the mass is next to that existing in the system during all the functioning band of the machine. There is a variation of the total mass in the system that occurs because of the addition of the errors stored in the convergences carried through during the functioning of the model, nevertheless this oscillation is less than 4% of the mass initial value.

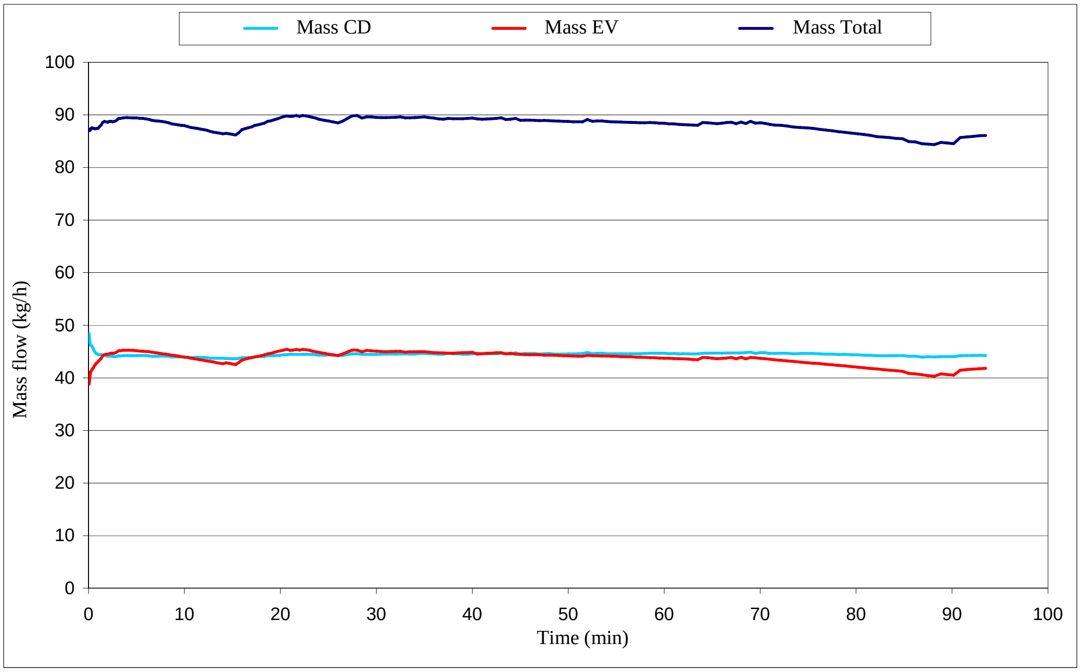

Figure 9. Mass Distribution in the exchangers of heat during the simulation

There is a possibility of reduction of this error through convergence tolerance adjustment in the pressures and in the degree of superheating, although this alteration becomes the program slower and unstable. Then, it must be evaluated the computational time and the reply of the model before taking this decision

The Fig. 10 shows the thermodynamic cycle developed by the machine for one determined instant of time, either for the experimental results or the points gotten for the model.

In the experimental cycle case, it was considered that there isn't a head loss in the exchangers of heat due to measurement impossibility of this phenomenon.

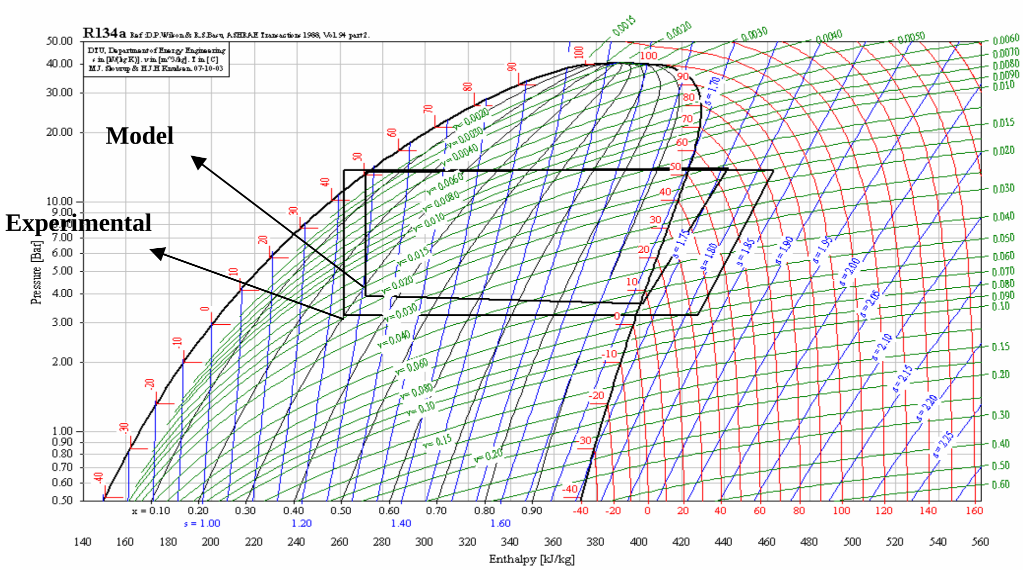

Figure 10. Thermodynamic cycle of experimental and model

Through this figure, it can be concluded that the work of compression considered for the model during the simulation is very less than the work of real compression, and then it results in a higher COP and in a lesser time for water heating. Others factors that contribute for the bigger water heating in the model are:

- In the physical model there is a parcel of lubricating oil that circulates together with the refrigerant fluid in the condenser, harming the heat transference for the water that it wishes to heat.

- The degrees of subcooling in the condenser and superheating in the evaporator are considerably lower in the model, because it doesn't consider the lines of liquid and vapor to the entrance of these components. A better model, that takes these lines into account consideration, it's in development. This program doesn't have stability necessary to execute a complete simulation, however some preliminary results show that due to the addition of mass in the system caused for this change, the thermodynamic cycle also answers with the increase of the subcooling and of the superheating.

- The coefficient of heat transference for the side of the water was calculated using a correlation for natural convection on a long cylinder. This correlation does not represent the occurred physical phenomenon very well, resulting in a global coefficient higher than the real. This fact implies in a reduction of the delta of temperature in each instant of time, a slower heating of the water. It's hoped to research into literature a correlation for the calculation of the coefficient of heat transference for natural convection on bank of pipes, and it must be used in the more sophisticated model than instead the one in development.

The Fig. 12 and 13 present a map of the evolution of the refrigerant fluid temperature throughout the length of the condenser and the evaporator, respectively. The model also is capable to show the maps of pressure, enthalpy, specific volume, mass flow, quality, void fraction and coefficient of heat transference the refrigerant fluid, and also the maps of the temperature of the wall and secondary fluid.

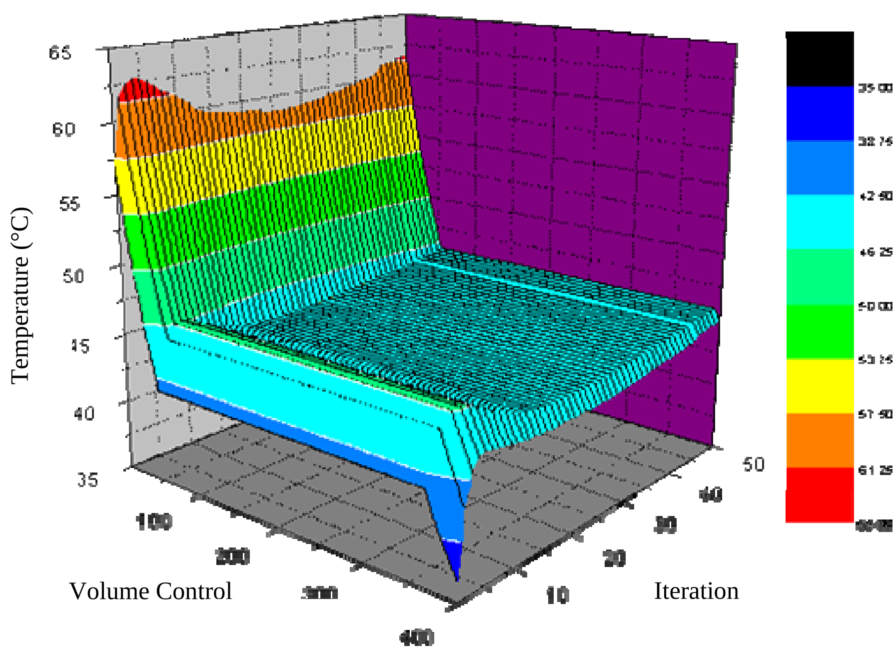

Figure 12. Evolution of the refrigerant fluid temperature throughout the length of the condenser

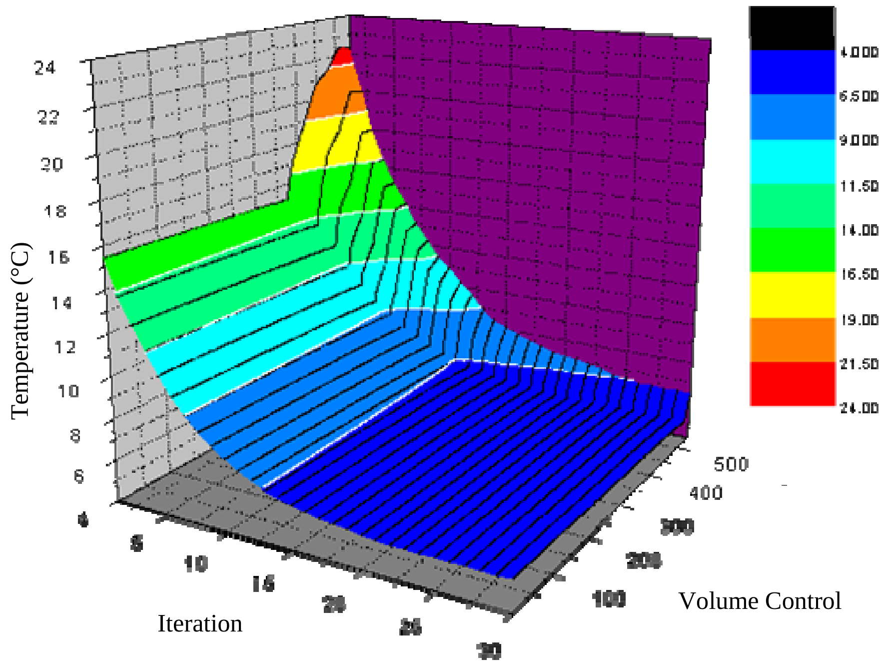

Figure 13. Evolution of the refrigerant fluid temperature throughout the length of the evaporator

The graphs represent only the first instants of simulation, because there are a lot of numbers of data. When one considers great intervals of time, it is not possible a clear visualization of the results. For each instant of time converged for the model (called iteration in the axle graphs), there are 400 and 540 values of temperature in the cases of the condenser and evaporator, respectively (a value for each volume of control).

In the Fig. 12, it's perceived that most of the heat exchanger is really the region of condensation (horizontal part of the graph) as gotten in the project. The temperature of condensation increases in a slow form after an initial peak and the subcooling degree decreases the measure while the time of functioning of the equipment grows. In virtue of the heating of the water of the reservoir, the condenser always works in transient regimen.

The Fig. 13 shows the profile of fall of temperature in the evaporator. The temperature of evaporation falls a lot up to the condition of permanent regimen in the heat exchanger and the superheating degree decreases as the machine follows it's regimen of work.

After the results gotten with the complete model of the heat pump to be presented and correlated with the measured ones on the physical model. It was verified some reasons for shunting lines occurred and possible corrections that can be used in the model to improve the correlation.

6. CONCLUSION

The main objective of this work was to develop a mathematical model to simulate the behavior of a heat pump for residential water heating. This model was developed and it was validated through the comparison between the model results and the experimental results.

Some deviations between the results of the mathematical model and the physical model can be observed. One factor that caused these deviations is the lubricating oil presence together with the refrigerant fluid in the condenser, and this presence harms the heat transference for the water that it wishes to heat, other factor is the lines of liquid and vapor to the entrance and exit of the heat exchangers can't be considered influencing in the degrees of sub-cooling and superheating of the mathematical model and the last factor is that the heat transference coefficient for the side of the water used in the calculations is not representing the physical model correctly, so it's necessary to reduce this value for the heat transference for the water be slower as occurred in the experimental tests.

To decrease the deviations observed between the experimental results and the model results, it is hoped to develop a model more sophisticated, in which some possible causing factors of these deviations will be smoothed or even though eliminated.

7. REFERENCES

BEN, Balanço Energético Nacional, 2003, publication of Ministério das Minas e Energia.
Hughmark, G. A., 1962, "Hold up in gas-liquid flow". Chemical Engineering Progress, Vol. 58, Nº 4, p. 62-65.

Koury, R. N. N., Maia, G. F., Castro, L. F. N. and Machado, L., 2007, "Numerical model and experimental study of a low cost heat pump for residential water heating". International Congress of Refrigeration - ICR, Beijing, China, v.1

Lockhart, R. W. and Martinelli, R. C., 1949, "Proposed correlation of data for isothermal two-phase two-component flow in pipes". Chemical Engineering Progress, Vol. 45, Nº 1, pp. 39-48.

Shah, M. M., 1979, "A general correlation for heat transfer during film condensation inside pipes", International Journal of Refrigeration, p. 446-457.

Maia, G. F. F., 2007, "Modelagem Matemática e Estudo Experimental de uma Bomba de Calor Ar-Água de Baixo Custo para Uso Residencial", PhD thesis, Departamento de Engenharia Mecânica; Universidade Federal de Minas Gerais; Belo Horizonte, Brazil, pp.119

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