NUMERICAL AND EXPERIMENTAL STUDY OF A HEAT PUMP FOR RESIDENCIAL APPLICATION

Maia, Gustavo Fonseca Freitas
Mechanical Engeneering Department, UFMG, Av. Antônio Carlos, 6627, CEP 31270-910, Belo Horizonte-Minas Gerais-Brasil
gufofma@ig.com.br

Machado, Luiz
Mechanical Engeneering Department, UFMG, Av. Antônio Carlos, 6627, CEP 31270-910, Belo Horizonte-Minas Gerais-Brasil
luizm@demec.ufmg.br

Ziviani, Márcio
Mechanical Engeneering Department, UFMG, Av. Antônio Carlos, 6627, CEP 31270-910, Belo Horizonte-Minas Gerais-Brasil
ziviani@fco.eng.ufmg.br

Koury, Ricardo Nicolau Nassar
Mechanical Engeneering Department, UFMG, Av. Antônio Carlos, 6627, CEP 31270-910, Belo Horizonte-Minas Gerais-Brasil
koury@demec.ufmg.br

Abstract. *The purpose of this paper is the development of a numerical model for dynamic conditions able to determine the spacial temperatures and velocities profiles for water stored into heat pump reservatory for water heating at homes. This model will be developed according to finite volume technique by CFX-5 software, in which mass, energy and momentum balance equations are applied and solved in each control volume. This software has a great advantage for the other ones, because it has a grafic window, where the user draws the domain, set boundary conditions and convergence criteria that make more easily work with complex geometry. The experimental device used for tests contains temperature and pressure meters conected to a data aquisition system and na analogic flow meter. An energy balance in steady state conditions at heat pump condenser was made and the heat flux obtained experimentally was used like entry variable in developed model. Two situations were analysed: at first time the heat pump working with capillary tube, and later thermostatic valve like expansion device. The model was validated by comparation of its results with experimental results obtained in test under dynamic conditions made at heat pump. The maximum error obtained with model for two cases was 3K between water temperature acquired experimentally and with model. The performance of the heat pump working wtih the thermostatic valve was a little better than it works wtih capillary tube.*

Keywords: *heat pump, finite volume, thermal storage.*

1. Introduction

After Industrial Revolution in England, the humanity started to care about more with energy production, been that in these time had beginning the industrial production process through machines and equipment that wanted steam to operate and, therefore, energy generation.

How the electricity is the most expended energy form in the world, its rational use implicates directly in maintenance of non-renewable sources. This form, the research for more efficient equipment is growing in domestic and industrial use.

Almost total of energy applied for bath in Brazil is generated by electric resistance. It's important notice that this kind of heating causes big overcharge for electricity generation in times of highest use, besides to be little efficient, since the electrical resistance is an equipment that presents low efficient as Second Law of Thermodynamics. In order of solution this problem; some equipment had been used in industry and residences, like solar collect and heat pumps, both to water heating. The heat pumps, don't necessity of a seasonal thermal charge like solar collect, besides its can be used at any time. In cold regions its use like residential heater also avoids the electricity loss.

The paper had for objective the development of a mathematics model for determination of temperature profile of water contained in tank of a heat pump for heating water for residential use. This model will be developed applying the volume finite techniques through software named CFX-5, and its validation will be obtained with comparison of experimental results obtained in heat pump earlier mentioned. Besides, its desire to evaluate the behavior of heat pump utilized in tests, because this is the first work to be developed in these one, and finally comparate the work of the heat pump operating with capillary tube and with expansion valve like expansion device.

2. Principle of function of a steam compression frigorific machine

Its four main components of a steam compression machine, the condenser, evaporator, expansion device and compressor as showed in Fig. 1. 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 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 reduce 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.

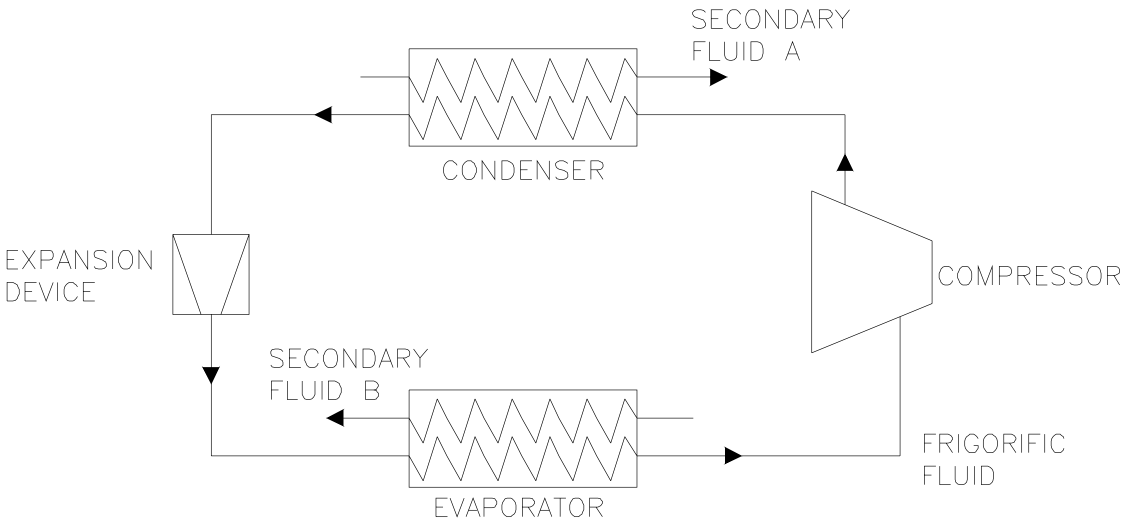


Figure 1 - Main components of a refrigeration system.

3. Sensible heat storage

A thermal storage system is basically constituted by three main items: the storage material, the heat transfer equipment and the thermal tank. In relation of storage material must be analyzed the necessary thermal level, the thermal proprieties and storage capacity. The heat transfer equipment has like function the transport of energy for the work fluid. This transfer can occur using heat exchangers, called indirect method, or using the work fluid oneself, called direct method. The thermal reservoir that contains the work fluid must be isolated or be constructed by isolante material in order to the thermal losses for the environment has been minimized.

Although specific thermal capacity of the liquid sensible heat storage has been less than latent heat storage, this kind of storage presents many advantages that justifies its apply, we can detach the easier operation and system control, short costs of equipment and the possibility of operates directly with work fluid.

Ismail and Carroci (1985) presented a convective model to stratified liquid water storage. The results were comparate with an unidimensional analytic solution. A little later, at 1986, they presented a bidimensional model to stratified tanks including wall thermal losses and performed experimental comparation for its validation.

Recently, Eames and Norton (1998) developed a tridimensional finite volume model in dynamic conditions to verify thermal performance of sensible heat storage tanks with different geometry. The model was validated by a comparation with temperature measure in a test series in which velocity and entrance temperatures and initial stratification profile were changed.

In recent work, Leal (1999) made a numerical and experimental study of stratified liquid sensible heat storage tanks. She analyzed the stratification degradation in natural cooling, and the numerical and experimental results were compared with good approximation at curves.

4. Developed model

4.1. Government equations

To establishment of the government equations for the problem, in order to develop a mathematic model that provides the water temperature in each point of thermal reservoir, it was used the three-dimensional Cartesian coordinate system in dynamic conditions. The tank has 200L of storage capacity, but it utilized only 170L in tests due to constructive problems. Figure 2, in which dimensions in meters, presents the tank with the principal dimensions.

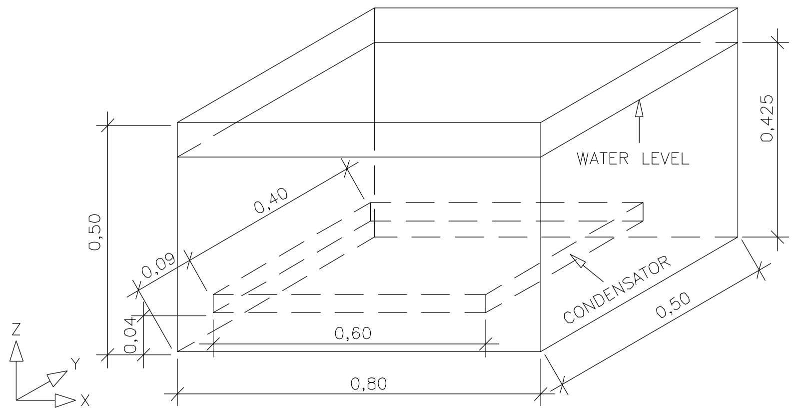


Figure 2 – Water tank in study

There isn't water flux entering or leaving the tank, but how exists a source yielding heat to water in reservoir, the natural convection flow were generated. In the general energy equation, the viscous dissipation term was despised, because there isn't thermal energy generated by mechanical energy conversion, so like the thermal expansion term, due to a little variation of thermal expansion coefficient (β) at typical temperature zone of water heating for residential use.

4.2. Source terms

The energy generation term, S_E, was obtained experimentally measuring the refrigerant fluid flow in the condenser, immersed in tank, and the refrigerant fluid temperatures at inlet and outlet of the condenser. How the water heating in tank goes in a very slow way, the process can be treated like a steady state series. To simplification, an uniform heat flux distribution was admitted for all of condenser surface. This doesn't happen really, so in the phase change zone the convective coefficient is very bigger than monophase zone. The machine used in tests allows to choice what the expansion device will be employed, capillary tube or thermostatic valve. In each case, was defined one heat generation equation. It's important to emphasize that from the experimental points obtained was made a least square approximation, resulting in heat transfer tax equation.

This source term appears in model when the analyzed control volume find oneself in the region taken by the condenser, how shows the Fig. 3, where the point P represents the coordinate system origin. Therefore, the zone where heat generation exists is defined by the following limits:

X: $0 \le x \le 0.60$ m; Y: $0.09 \le y \le 0.49$ m; Z: $0.04 \le z \le 0.08$m.

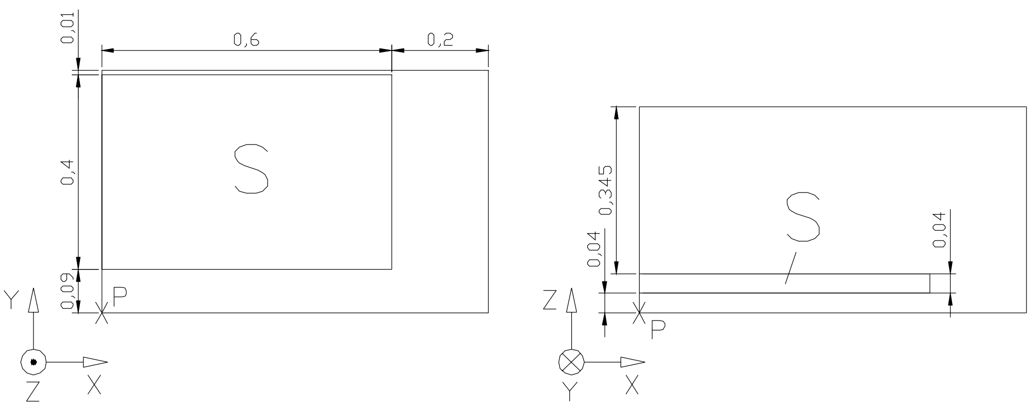


Figure 3 – Zone with energy source term (S_E) in model

The natural convection flow appears due to specific mass difference existent between hot water next to condenser and cold water in the top of the tank. The hot water tends to go up and the cold water goes down to occupy the position left by hot fluid, then this is heated maintaining the displacement.

For the study problem, the specific mass variation during the test is relatively small, that allows the use of Boussinesq model. This model utilizes a standard specific mass fixed, but applies a local gravitational body force what is linear function of fluid's thermal expansion coefficient (β) and of the difference between local temperature and reference temperature for which specific mass was adopted. This body force is applied like source term in momentum equation in the following mode according to Burmeister:

$$S_M = (\rho - \rho')g \quad (1)$$

Where the specific mass approximated for Boussinesq model is given by

$$\rho' = \rho\beta(T - T_{ref}) \quad (2)$$

Where ρ fluid's standard specific mass (fixed) and Tref is the reference temperature for which specific mass was adopted.

These approximation supplies good results for situations in that Mach number is short (M<0.1) and where temperatures difference are relatively small (β(T-Tref) <0.1). Both conditions are satisfied in studied problem.

4.3. Solve equations method

The differential equations were discretized according to finite volume techniques, creating linear equations that will solved by CFX-5 software, for numerical simulation of transport phenomena. This one divide in three parts: Build, Solver e Post. The Build matches preprocessor, where the user defines the problem will be solved, solution domain, kind of fluid, boundary conditions, adopted mesh, initial conditions and convergence criteria. It's a large advantage of CFX in relation to others programs, because from a graphics interface, the solution domain, application boundary conditions places were defined, that becomes more easier introduce complex geometry. The Solver is where the discretized equations are properly solved, and the Post produces many graphic forms since the obtained results.

5. Materials and methods

5.1. Experimental device

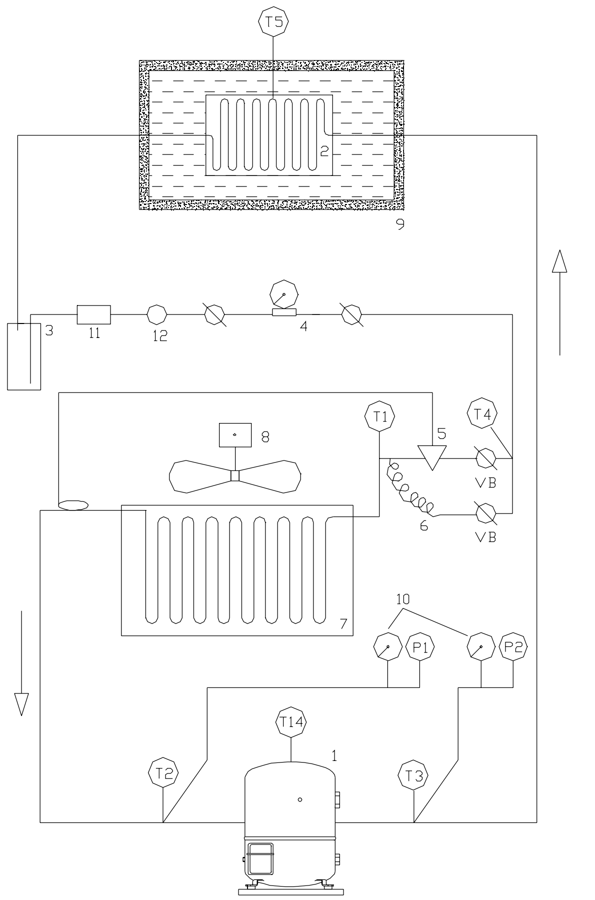
The experimental device used consists in a steam compression heating system working with R134a as primary fluid and, air as secondary fluid in evaporator and water as secondary fluid in condenser. The storage water tank of heat pump has 170L of utility volume, fixed in this value during all the tests madden.

The system is compose by a hermetic compressor for R134a, 115W and 60 Hz, produced by Tecunseh; an air evaporator madden by copper tubes with 8mm external diameter and 5.4m length; a water condenser madden by copper tubes with 8mm external diameter and 16.5m length; and two expansions devices in parallel, one is capillary tube and the other one is a thermostatic valve, that they work alternately through two block valves previous choice.

The tests equipment has 14 thermocouples (model K) installed, two manometers and two capacitive pressure meters for high and low pressure determination, and a volumetric flow meter. The thermocouples and pressure meters signal are registered in an acquisition data system. The heat pump was constructed by Friominas Company in 2003. In order to have a better equipment understood, and temperature, pressure and flow meter points view, the Fig. 4 (schematic drawing) and Fig. 5 are following presented.

5.2. Experimental procedure

The experimental procedure that will be describe in these moment was used for two madden tests, one with thermostatic valve and the other one with capillary tube like expansion device, that will be utilized in model validation. In experiment preparations, fill the heat pump tank with 170L of water, and the thermocouples and pressure meters were connected to data acquisition system. The machine was turned on and maintained in work during three hours sequence. The temperatures and pressures values were saved in a data file with 30 seconds sample tax. At once the flow values was read manually in ten to ten minutes intervals, because the flow meter install at equipment is analogic, it's not possible connect it to data acquisition system. The heating process happen slow mode, and for that reason, it was treated like a short steady state series. The thermodynamics cycle points and the water tank temperatures were obtained for each steady state, through average of read values in before five minutes to next five minutes of read flow points intervals.



Legend	
1	Compressor
2	Condenser
3	Liquid bottle
4	Flow meter
5	Thermostatic valve
6	Capillary tube
7	Evaporator
8	Fan
9	Water tank
10	Manometer
11	Dry filter
12	Viewer
T_n	Thermocouples
P_n	Pressure meters
V_B	Block valves

Figure 4 – Equipment's schematic drawing

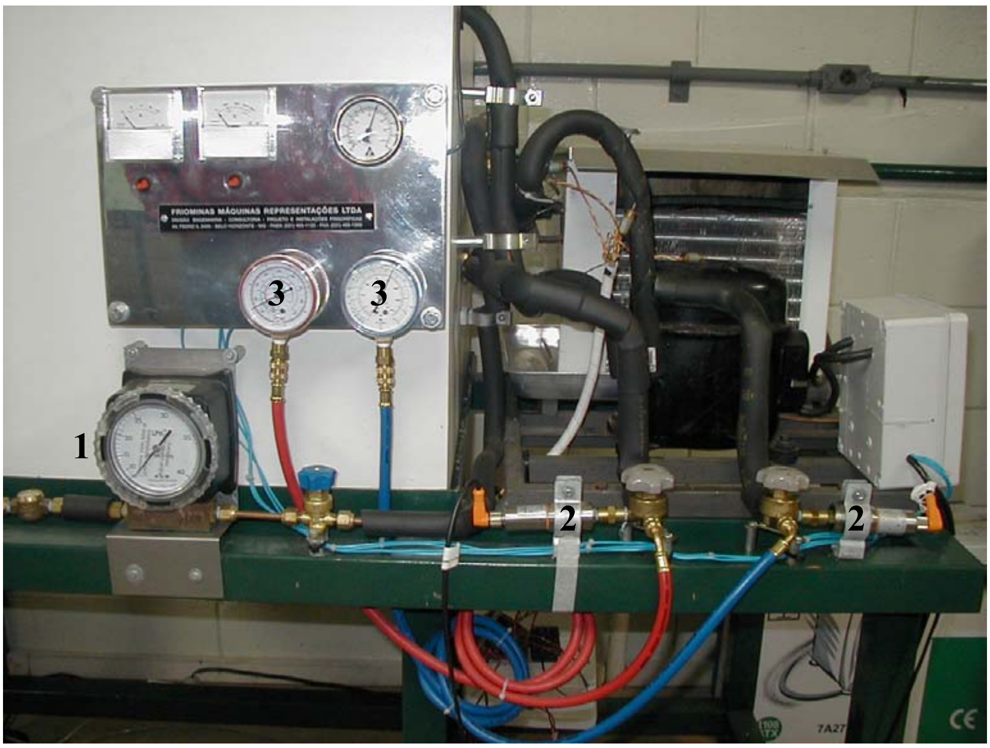


Figure 5 – View of installed flow meter (1), pressure meters (2) and manometers (3)

In both tests of dynamic conditions, we note the degradation of heat pump performance coefficient. Figure 6 shows such fact so to heat pump working with capillary tube as much as thermostatic valve like expansion device. This occurs because all along the test, the condensation pressure rises, since this directly depends water temperature was heated, e consequently increases the compressor work. In the two cases, the mean performance coefficient obtained in tests is 3.3. The average water final temperature in tank is 2K bigger when the heat pump works with thermostatic valve as expansion device, but the thermostatic valve price is very larger than capillary tube price. A more detail evaluation must be maddened to verify the two devices economics viability; however in this preliminary analysis I would use the capillary tube in the residential heat pump final design.

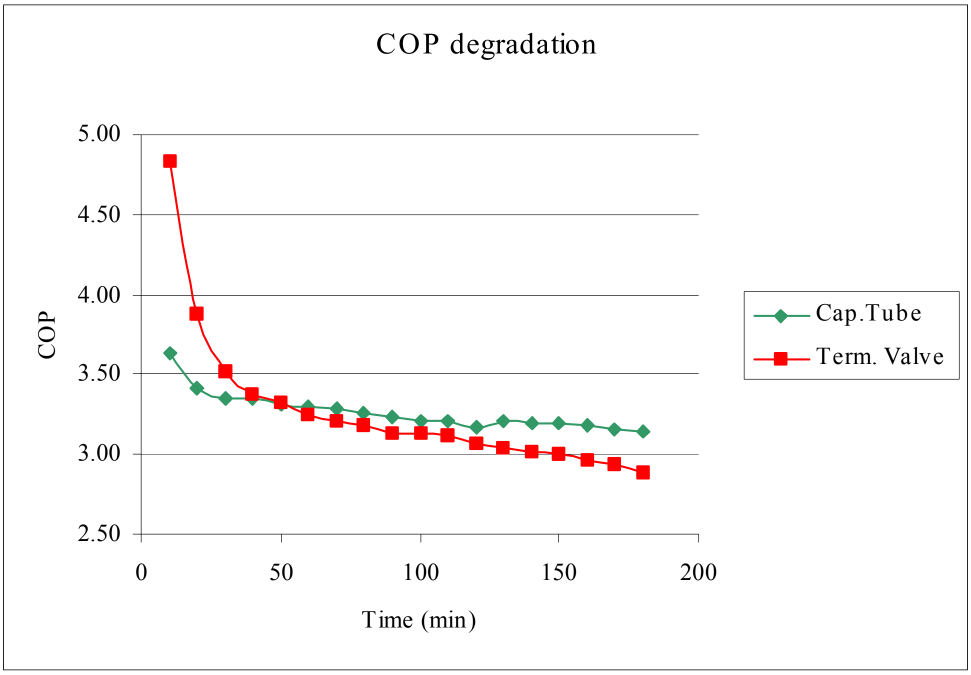


Figure 6 – Heat pump COP (performance coefficient) degradation

6. Results and analysis

Our mathematic model needs to be validated so that its results can be used like support in future projects. For that, a comparation between model results and experimental results was done. However, before this comparation is made, a time and space mesh test was done, in order to identify ideal mesh to problem solution. A space mesh obtained as result is composed by 8cm edge tetrahedrons, and the time mesh adopted is that one Δt is equal to 2 minutes.

6.1. Experimental results comparation

At these part of paper will be compared the model temperatures values and experimental temperatures values obtained through thermocouples read. They are installed at middle of each lateral face inside the tank at two height levels (10cm e 26cm). Table 1 shows this comparation for heat pump operating with each one of expansion devices.

Finally, when we observe the temperatures values on table 1, where T_{EXP} is experimental temperature and T_{MOD} is model temperature, we perceive that two first hours test points the error between the value acquired by model and the experimental value stays inside the 1 K range. At third hour, the model temperatures grow up in bigger ratio that experimental results, increases the error, resulting a maximum error of 3.3 K when the heat pump works with capillary tube and 2.9 K when the thermostatic valve is used as expansion device. This fact can occur due there is air in top of water tank. The air contained in reservoir is heated during the test, and due internal pressure tank raise, it tends to escape carrying part of heat that should be utilized to heat the water. This heat loss doesn’t consider in model. The error obtained all along the test was lower in thermostatic valve case as so as expected due the better adjustment to had as result to heat transfer tax equation.

Through this analysis, we conclude that model developed to determine water temperature profile in heat pump tank was validated and can be used in other cases that involve natural convection since right adaptations were madden.

Table 1 - Comparison between model results and experimental results

Time (min)	Capillary Tube						Thermostatic Valve					
	H = 10 cm			H = 26 cm			H = 10 cm			H = 26 cm		
	T_{EXP} (K)	T_{MOD} (K)	Dif (K)	T_{EXP} (K)	T_{MOD} (K)	Dif (K)	T_{EXP} (K)	T_{MOD} (K)	Dif (K)	T_{EXP} (K)	T_{MOD} (K)	Dif (K)
0	298.1	298.0	-0.1	298.1	298.0	-0.1	298.1	298.0	-0.1	297.9	298.0	+0.1
10	300.7	298.9	-1.8	299.8	298.9	-0.9	299.9	299.1	-0.8	300.0	299.0	-1.0
20	301.5	300.1	-1.4	301.0	300.0	-1.0	301.3	300.3	-1.0	301.1	300.3	-0.8
30	302.5	301.2	-1.3	302.1	301.1	-1.0	302.4	301.6	-0.8	302.4	301.6	-0.8
40	303.4	302.3	-1.1	303.1	302.3	-0.8	303.6	302.9	-0.7	303.7	302.9	-0.8
50	304.3	303.4	-0.9	303.7	303.4	-0.3	304.9	304.2	-0.7	304.8	304.2	-0.6
60	305.8	304.6	-0.8	305.0	304.6	-0.4	306.0	305.5	-0.5	305.9	305.5	-0.4
70	306.3	305.8	-0.5	305.9	305.7	-0.2	307.2	306.8	-0.4	307.2	306.8	-0.4
80	307.2	307.0	-0.2	306.8	307.0	+0.2	308.4	308.2	-0.2	308.2	308.2	0
90	308.2	308.2	0	308.0	308.2	+0.2	309.5	309.5	0	309.3	309.5	+0.2
100	309.5	309.4	-0.1	308.9	309.4	+0.5	310.6	310.8	+0.2	310.4	310.8	+0.4
110	310.3	310.6	+0.3	309.9	310.6	+0.7	312.0	312.2	+0.2	311.8	312.2	+0.4
120	311.1	311.9	+0.8	310.9	311.9	+1.0	313.2	313.5	+0.3	312.7	313.5	+0.8
130	312.3	313.2	+0.9	312.0	313.2	+1.2	313.9	314.9	+1.0	313.7	314.9	+1.2
140	313.3	314.5	+1.2	312.8	314.5	+1.7	315.1	316.2	+1.1	314.9	316.2	+1.3
150	314.3	315.9	+1.6	314.0	315.9	+1.9	316.2	317.5	+1.3	316.0	317.6	+1.6
160	315.3	317.2	+1.9	315.1	317.2	+2.1	317.3	318.9	+1.6	317.3	318.9	+1.6
170	316.4	318.6	+2.2	316.0	318.6	+2.6	318.5	320.3	+1.8	318.1	320.3	+2.2
180	317.2	320.0	+2.8	316.7	320.0	+3.3	319.1	321.6	+2.5	318.7	321.6	+2.9

7. Conclusions

This paper accomplished all of its general and specific objectives. The heat pump tank mathematic model according to finite volume techniques was developed and its results presented good agreement when compared to experimental results. The maximum error obtained between model temperatures and experimental temperatures was 3.3 K and 2.9 K, if the heat pump works with capillary tube or with thermostatic valve as expansion device respectively. Besides, was made an experimental equipment function evaluation, as since this work is the first perform in it. We realized that condenser inside the tank is over dimension, providing a very high sub cooling at expansion device inlet, that decreases machine COP. It must be changed by a more adequate one. Finally, a more specific work objective was comparate the heat pump results operating with capillary tube or thermostatic valve as expansion device, for future project of a compact, low price heat pump for residential use construction. We observed the heat pump presented very closer results for both of expansion devices. In order to reduce manufacture costs, and emphasizing that a more detail analysis will be done, in a preliminary analysis, I affirm that heat pump must be constructed with capillary tube as expansion device.

8. References

Burmeister, L. C., "Convective Heat Transfer", 2ª ed., Wyley-Interscience Publication, 1993, 617p.

Eames, P. C., Norton, B., "The Effect of tank geometry on thermally stratified sensible heat storage subject low Reynolds numbers flow", Int. J. Heat Mass Transfer, v.41, n.14, 1998, p.2131-2142

Ismail, K. A. R., Carroci, L. R., "A convective model for stratified storage tank", In: 7th Miami International Conference on Alternative Energy Sources, 12, 1985, Miami, p. 275-282.

Ismail, K. A. R., Carroci, L. R., “A two dimensional model for liquid stratified storage tanks”, In: World Congress III of Chemical Engeneering, Tokio, 9, Japan, 1986, p. 681-683.

Leal, J. F. B., “Estudo Numérico e Experimental de Fenômeno de Estratificação Térmica em Armazenamento de Calor e Frio”, Campinas: Universidade Estadual de Campinas, 1999, 78p. (Tese, Doutorado em Engenharia Mecânica)