THEORETICAL AND PRACTICAL STUDY OF VOID FRACTION IN A TWO-PHASE FLOW OF R134A

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Abstract: *The main objective of this work was the experimental determination of the R134-a refrigerant fluid void fraction in a stratified two-phase flow for different values of mass flow and temperature. The void fraction was determined by direct reading of the liquid column of refrigerant flow through a test section consisting of a glass tube. This section was at the evaporator entrance of a refrigeration bank test, where the compressor rotational speed, evaporation and condensation temperatures and the superheat and sub-cooling degrees of the refrigerant were adjusted in several values. Correlations removed from the specialized literature were used to calculate the values of void fraction in different operation points of the experimental bank. A comparative study between the experimental and theoretical values of void fraction was conducted, and the conclusion was that the correlation of Hughmark is the most accurate to estimate the void fraction, presenting an error about 5% compared to the experimental values. In the final part in this paper, a simple model of heat transfer and the Hughmark correlation were used to value the influence that the refrigerant mass in an evaporator exerts in the superheat degree, which is a fundamental parameter to a good operation of the refrigeration system.*

Key Words: *void fraction, two-phase flow, refrigerant inventory.*

1. INTRODUCTION

An important tool that helps in the domain of the two-phase flow of the fluid is the void fraction determination. The determination of this parameter is essential to a complete understanding of the fluid hydrodynamic associated with the multiphase flow. Several techniques to measure the void fraction have been developed and refined, in order to measure them with greater speed and precision. The theoretical and practical study of void fraction in a two-phase flow showed in this paper had the following objectives: presentation of a synthetic bibliographical revision, which will guide the theoretical determination proposals in this work; experimental determination of the void fraction at the same conditions of the theoretical calculations; consistency analysis of the theoretical models by comparison with the experimental values; development of a methodology to find the liquid mass ratio in two-phase flow.

2. BIBLIOGRAPHICAL REVIEW

Consider the schematic two-phase flow represented in Figure 1.

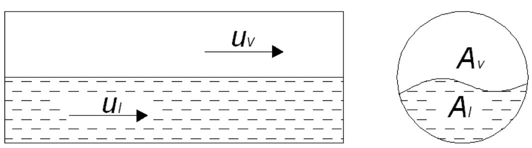

Figure 1 – Schematic flow

Where A_v is the area occupied by vapor, A_l is the area occupied by liquid, and A is the total area of the tube (A_v + A_l). The u_l is the average speed of liquid phase and u_v is the average speed of vapor phase.

The void fraction α is defined as the ratio between the area occupied by vapor and the total area of the tube:

$$\alpha = \frac{A_v}{A_v + A_l} = \frac{A_v}{A} \tag{1}$$

The void fraction can be expressed in terms of the quality x, the density of the two phases, ρ_v and ρ_l, and the slip γ.

$$\alpha = \frac{1}{1 + \left(\frac{1-x}{x}\right) \cdot \frac{\rho_v}{\rho_l} \cdot \gamma} \tag{2}$$

The slip ratio will be defined as the ratio between the speeds of the vapor phase and liquid phase:

$$\gamma = \frac{u_v}{u_l} \tag{3}$$

Next, the main theoretical models to calculate the void fraction in stratified flows are presented. In the homogeneous model, it is assumed that the phases have equal speeds, so $\gamma = 1$. Thus, the void fraction is expressed by:

$$\alpha = \frac{1}{1 + \left(\frac{1-x}{x}\right) \cdot \frac{\rho_v}{\rho_l}} \tag{4}$$

To show with bigger accuracy the physical reality of two-phase flow, it is necessary that the difference between the speeds on the liquid-vapor interface, called slip, either considered. Thom (1964), based on experimental results, proposed an expression to calculate the slip in function of the density and dynamic viscosities μ_v and μ_l of phases, where the values of γ were present in the table form. Butterworth (1975) approached the values of γ by:

$$\gamma = \left(\frac{\rho_v}{\rho_l}\right)^{0,89} \left(\frac{\mu_l}{\mu_v}\right)^{0,18} \tag{5}$$

Zivi (1964) proposed a model based on the principle of minimum entropy production, conditions of null friction on the wall and without liquid movement:

$$\gamma = \left(\frac{\rho_v}{\rho_l}\right)^{\frac{1}{3}} \tag{6}$$

Some correlations are based on Martinelli parameter X, which can be obtained by:

$$X = \left(\frac{1-x}{x}\right)^{0,9} \left(\frac{\rho_v}{\rho_l}\right)^{0,5} \left(\frac{\mu_l}{\mu_v}\right)^{0,1} \tag{7}$$

Lockhart and Martinelli (1949) represented the evolution of void fraction in function of the parameter X to many experimental results. The curves obtained were approximate by equations developed by Wallis and refined by Domanski and Didion to the domain of $X > 10$. These approaches were cited by Rice (1987):

$$\alpha = \left(1 + X^{0,8}\right)^{-0,378} \quad \text{to} \quad X \le 10 \qquad \text{or} \qquad \alpha = 0,823 - 0,157\ln(X) \quad \text{to} \quad X > 10 \tag{8}$$

Other correlation based on the Martinelli parameter was established by Baroczy (1965) in the table forms. The void fraction was approximate by an equation developed by Butterworth (1975), where X appears modified:

$$\alpha = \left[1 + \left(\frac{1-x}{x}\right)^{0,74} \left(\frac{\rho_v}{\rho_l}\right)^{0,65} \left(\frac{\mu_l}{\mu_v}\right)^{0,13}\right]^{-1} \tag{9}$$

Some authors verified that the decrease of mass flow implied an increase in the void fraction. Hughmark (1962) established a correlation to calculate the void fraction that takes into account the effect of the mass flow. The void fraction is calculated from a correction factor, K_H, which was introduced in the model homogeneous equation:

$$\alpha = \frac{K_H}{1+\left(\frac{1-x}{x}\right)\frac{\rho_v}{\rho_l}} = K_H \alpha_{\text{hom}} \tag{10}$$

α_{hom} is the void fraction calculated by Equation 4. The factor K_H is function of a parameter Z, showed in the Table 1.

Table 1
Values of parameter K_H and Z of Hughmark correlation

Z	1,3	1,5	2,0	3,0	4,0	5,0	6,0	7,0	10,0	15,0	20,0	40,0	70,0	130,0
K_H	0,185	0,225	0,325	0,490	0,605	0,675	0,720	0,767	0,780	0,808	0,830	0,880	0,930	0,980

The parameter Z is given by:

$$Z = \left[\frac{d_{\text{int}} G}{\mu_l + \alpha(\mu_v - \mu_l)}\right]^{\frac{1}{6}} \left\{\frac{1}{g \cdot d_{\text{int}}}\left[\frac{G \cdot x}{\rho_v \cdot \alpha_{\text{hom}} \cdot (1-\alpha_{\text{hom}})}\right]^2\right\}^{\frac{1}{8}} \tag{11}$$

Where d_{int} is the internal diameter of the tube, g is the gravity acceleration and G is the ratio between the mass flow and the total area of the tube. The other parameters were defined previously.

Prémoli (1982) developed a correlation to determine the void fraction in a great domain of experimental conditions. The void fraction is calculated using the Equation 2, where the slip is a function of density of each phase, the dynamic viscosity of the liquid phase, the surface tension, σ, between the two phases, the quality, the tube diameter and speed of liquid mass:

$$\gamma = 1 + E_1 \sqrt{\frac{y}{1 + y \cdot E_2} - y \cdot E_2} \tag{12}$$

The dimension numbers that appear in this expression are calculated by:

$$E_1 = 1{,}578 \cdot \text{Re}_l^{-0,19}\left(\frac{\rho_l}{\rho_v}\right)^{0,22} \quad ; \quad E_2 = 0{,}0273 \cdot We_l \cdot \text{Re}_l^{-0,51}\left(\frac{\rho_l}{\rho_v}\right)^{-0,08} \quad \text{and} \quad y = \frac{x}{1-x}\left(\frac{\rho_l}{\rho_v}\right) \tag{13}$$

The number of Reynolds and Weber are defined by:

$$\text{Re}_l = \frac{G \cdot d_{\text{int}}}{\mu_l} \quad \text{and} \quad We_l = \frac{G^2 \cdot d_{\text{int}}}{\sigma \cdot \rho_l \cdot g} \tag{14}$$

σ is the superficial tension between the vapor and liquid phases.

3. MATERIALS AND METHODS

To obtain experimental data that served of basis in this study was used the experimental bank tests, represented in Figure 2. The bank is a system of cooling and heating by vapor compression, with primary fluid the R134a and the secondary fluid pure water in the evaporator and condenser. The system consists of an alternative compressor, a condenser, a sub-cooling, an evaporator, two expansion valves and measuring systems and data acquisition. Before the entering of evaporator, it was installed a test section that consists of a translucent tube, where it was possible to record images of the refrigerant flowing through the section. A scale was located in front of the tubing, so that the instrument allowed the direct reading of height of the liquid column beyond serving as reference to the dimensional analysis. The best images had been taken to Auto CAD 2000 Software, where a dimensional analysis of the column height of the cooling liquid was possible. The images were registered with a digital camera, mark OLYMPUS, FE100 model. They were captured in the resolution 4.0 mega pixels, maximum equipment resolution. In this work, the compilation of calculations was done by software ESS - Engineer Equation Solver - from where, in addition to the mathematical processes, thermodynamic properties and transport properties were also extracted. The graphic generation was done to the software Excel.

To the evaluation of consistency between mathematical models and experimental results, sixteen measurements were conducted. A rigorous test plan was elaborated before to do the experimental procedures. This plan was supported by the ideas proposed by Figliola (2000). The test bank was operated in steady-state with superheat evaporator of 7^{o}C. This adjustment was done by a thermostatic valve. Four evaporation temperatures were used: -5, 0, 5 and 10^{o}C. The adjustment and control of these temperatures were obtained by the water temperature control at the evaporator entrance. It was done with help of an electrical resistance and a PID controller. Each one of the evaporation temperatures was combined with two condensation temperatures: 50 and 45^{o}C. The adjustment of the condensation temperature was obtained by controlling the flow and temperature water at the condenser entrance, which was, in fact, a mixture of hot water produced in the condenser and cold water. The compression speeds were adjusted to 470 rpm and 600 rpm, obtained by the frequency inverter.

Legend:
1 e 7. Water tanks
2 e 12. Recirculation pumps
3 e 4. Electric heat (sub-cooling / evaporator)
5. Sub-cooling
6. Evaporator
8. Condenser
9. Expansion valves
10. Alternative compressor and Electric motor
11. Heat water tank (condenser)
13. Cool water tank (condenser)
14. Entry of water from supplying net
15. Glass Tube
V. Measurer of mass flow
Pa e Pb. Measurer of pressure
S1 e S2. Measurer of temperature (PT-100)
T1e T5. Measurer of R134a temperature
E1 e E2. Measurers of water temperature evaporator
C1 e C2. Measurers of water temperature condenser

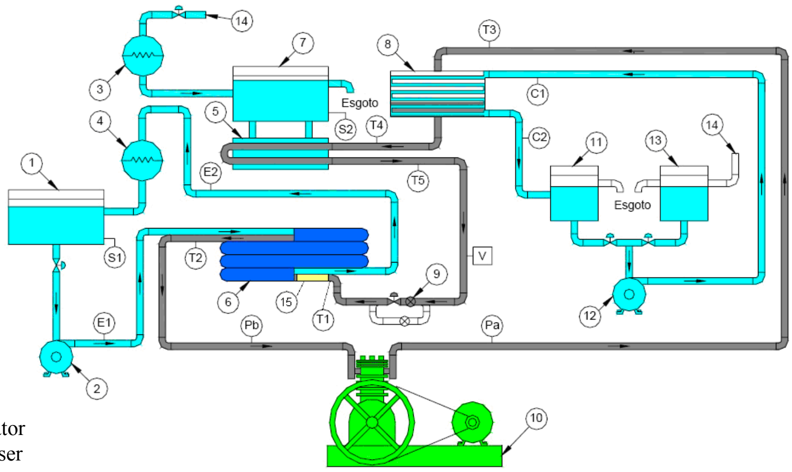

Figure 2 – Schematical representation of the experimental bank

In relation to the mass flow, an expected performance is its increase with the increase of evaporation temperature. It can be justified because the theoretical flow in the compressor, $\dot{m}$, is defined by:

$$\dot{m} = n \cdot V \cdot \rho_{asp} \cdot \eta_v \tag{15}$$

Where n, V, ρ_{asp} and η_v are, respectively, the compressor speed, volumetric capacity, the refrigerant density in the compressor aspiration and volumetric efficiency. The density ρ_{asp} increases with the increase of the evaporation pressure, whereas the efficiency η_v increases with the decrease of the compression rate, defined as the relation between condensation and evaporation pressures. Thus, the increase of the temperature evaporation results in an increase of product $\rho_{asp} . \eta_v$, so the refrigerant mass flow also increases, which it is essential for analysis in this work.

4. UNCERTAINTY ANALYSIS OF MEASUREMENT

When a measurement result is presented, it is essential to give some quantitative indication of the result quality. According to Gonçalves (1999), two base parameters must be estimated in any measurement operation: the correction C and expanded uncertainty U. The measurement result Rm can be obtained in terms of C and U:

$$Rm = MI + C \pm (\Delta I_{\max} + U_{95}) \tag{16}$$

Where MI is the average of n available indications, C is the correction, ΔI_{max} is the absolute value of maximum difference between the indications and its average value, U_{95} is the expanded uncertainty of the measurement process calculated for a confidence level of 95%. The evaluation of U in direct measurements occurs when there is only one entrance value involved and the indication is a consequence of the application of the measurement system in the measured. In this case, C is numerically equal to the symmetrical value of the tendency (Td). The tendency corresponds to the arithmetic average of the finite number of measurements of the same measured, deducted from the true value of measured, (VV). As VV is generally unknown, it can be used the VVC, True Value Conventional. Besides, to this case, Gonçalves (1999) defines that:

$$\Delta I_{\max} = |I_i - MI|_{\max} \tag{17}$$

Where MI is the average of n available indications and I_i is the value of each indication.

$$U_{95\%} = Re \quad \text{and} \quad Re = t \cdot s \tag{18}$$

Where: t is the Student's coefficient and s is the standard experimental deviation of the sample.

The standard experimental deviation is defined as:

$$s(q) = \sqrt{\frac{\sum_{k=1}^{n}(q_k - \overline{q})^2}{n-1}} \tag{19}$$

Where q_k is the independent value obtained to the variable q; and $\overline{q}$ represents the average of k comments from q.

The methodology showed above was used to do the measurements analysis in this experiment. Just to illustrate the calculations, it is presented in the Table 2 the experimentally measured values with their uncertainties, only to the case of condensation temperature of the 50°C and rotation of the 600rpm. Similar values were obtained to the condensation temperature of 50°C and rotation of 470rpm, condensation temperature of 45°C and rotations of 470 and 600rpm.

Table 2

Base values *(MI)* and uncertainties of measured ($\Delta I_{max}+U_{95}$) to condensation temp. of 50°C and rotation of 600 rpm

	Evaporation Temperatures							
	-5°C		0°C		5°C		10°C	
	Base Value	Uncer-tainty	Base Value	Uncer-tainty	Base Value	Uncer-tainty	Base Value	Uncer-tainty
Temperature in the exit of compressor (C)	62,73	0,66	62,59	0,70	62,62	0,55	61,75	0,82
Temperature before of expansion valve (C)	32,81	0,95	32,29	1,43	30,84	0,25	29,81	1,29
Temperature after of expansion valve (C)	-4,47	0,64	-3,26	0,67	4,47	0,48	5,35	1,20
Temperature in the entrance of compressor (C)	6,64	1,40	7,18	0,91	11,48	0,52	12,36	1,63
Flow (kg/h)	30,90	1,34	36,10	1,35	41,80	1,34	49,20	1,36
Height of liquid (mm)	2,53	0,05	2,71	0,04	3,10	0,08	3,25	0,08

5. RESULTS AND DISCUSSIONS

The evolutions of the R134a mass flow as a function of evaporation temperature is shown in Figures 3 and 4. As expected, this mass flow increases with the increase of the evaporation temperature.

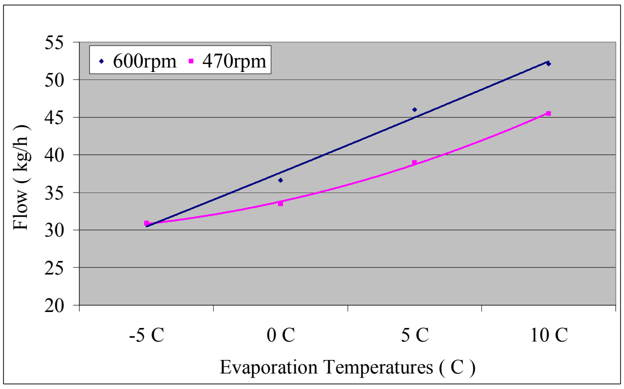

Figure 3 - Variation of flow as function of evaporation temperature. Condensation temperature of 45°C.

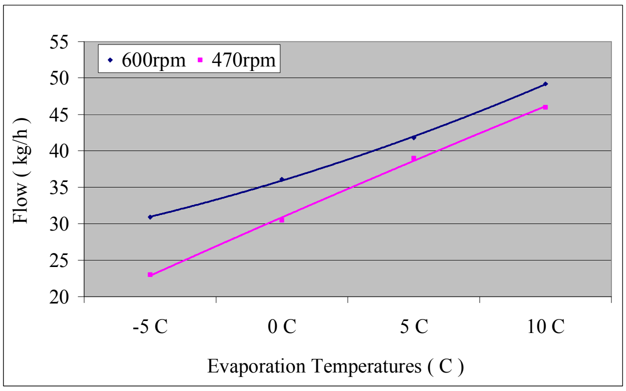

Figure 4 - Variation of flow as function of evaporation temperature. Condensation temperature of 50°C.

Two examples of the photographic registers used to determinate the void fraction are showed on the Figures 5 and 6. As it can be seen, these columns can be measured by direct reading. The indications on the figures represent the height of the columns in millimeter. Naturally, the void fraction can be obtained by simple calculations of this height.

In the Table 3 it is presented the values of the void fraction calculated theoretically and experimentally for a condensation temperature of 45ºC, a compressor speed of 470 rpm and evaporation temperatures of -5ºC, 0ºC, 5ºC and 10ºC. In the table 4 it is presented the variations of theoretical results in relation to experimental corresponding to these tests. In the Graphics 7, 8, 9 e 10 are presented the void fraction in function of the evaporation temperature for the condensation temperatures of 45ºC and 50ºC and for compressor speeds of 470 rpm and 600 rpm.

Among the theoretical models studied, the proposed model of Hughmark showed the lowest variation in relation to the experimental data. The variations of this model were less than 5%. Analyzing these graphics, it is also possible to observe that to the high values of the evaporation temperature, the models described (except the Homogeneous and Butterwoth model) presented next values of void fraction. This occurs because in higher temperatures the liquid and vapor densities are near. Consequently, the velocity of the vapor is near to liquid, the slip value tends to 1, and its influence on the void fraction loses significance. It can be concluded that in the condenser the other correlations also will provide good results.

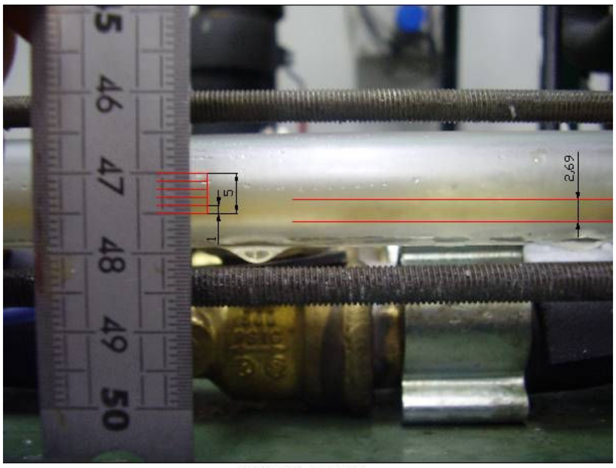

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Figure 5 - Evaporation temperature of -5°C and condensation of 45°C. Rotation of 470rpm

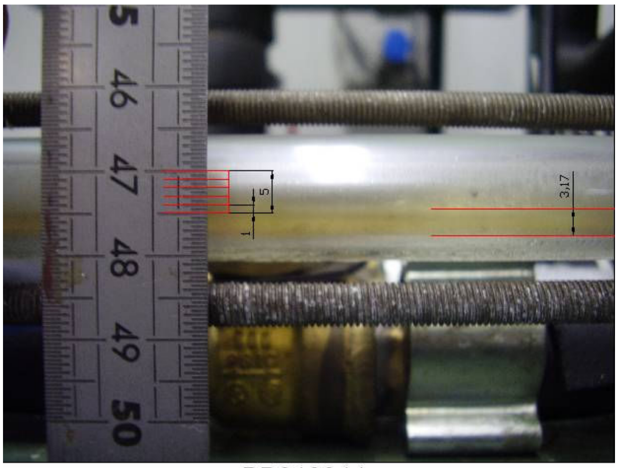

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Figure 6 - Evaporation temperature of 10°C and condensation of 45°C. Rotation of 470rpm

Table 3
Values of void fraction. Condensation temperature of 45oC and of 470rpm.

	Evaporation Temp.			
Model	-5ºC	0ºC	5ºC	10ºC
Homogeneo	0,966	0,963	0,936	0,909
Butterworth	0,999	0,999	0,997	0,996
Zivi	0,859	0,853	0,777	0,713
Lockhart e Martinelli	0,855	0,853	0,812	0,781
Baroczy/Butterworth	0,837	0,830	0,767	0,718
Hughmark	0,767	0,763	0,726	0,701
Prémoli	0,830	0,829	0,784	0,750
Experimental	0,781	0,769	0,755	0,721

Table 4
Results variation of the void fraction. Condensation temperature of 45°C and 470rpm.

	Evaporation Temp.			
Modelo	-5ºC	0ºC	5ºC	10ºC
Homogeneo	-23.7%	-25.3%	-24.0%	-26.2%
Butterworth	-27.9%	-30.0%	-32.2%	-38.2%
Zivi	-9.9%	-10.9%	-2.9%	1.0%
Lockhart e Martinelli	-9.5%	-11.0%	-7.6%	-8.3%
Baroczy/Butterworth	-7.2%	-8.1%	-1.6%	0.4%
Hughmark	1.7%	0.7%	3.8%	2.7%
Prémoli	-6.3%	-7.8%	-3.9%	-4.1%
Experimental	Ref.	Ref.	Ref.	Ref.

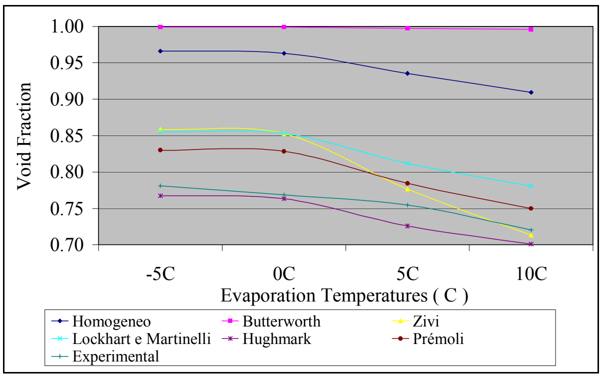

Figure 7 – Void fraction in function of the evaporation Temp. Condensation Temp. of 45oC and 470rpm.

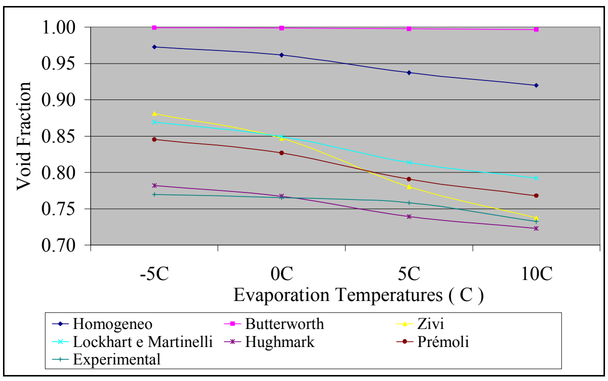

Figure 8 – Void fraction in function of the evaporation Temp. Condensation Temp. of 45oC and 600rpm.

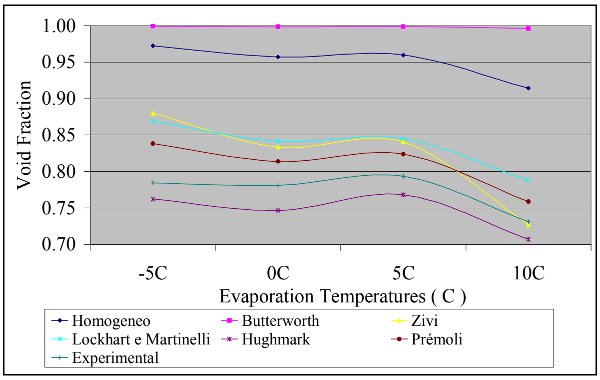

Figure 9 – Void fraction in function of the evapo-ration Temp. Condensation Temp. of 50^{o}C and 470rpm.

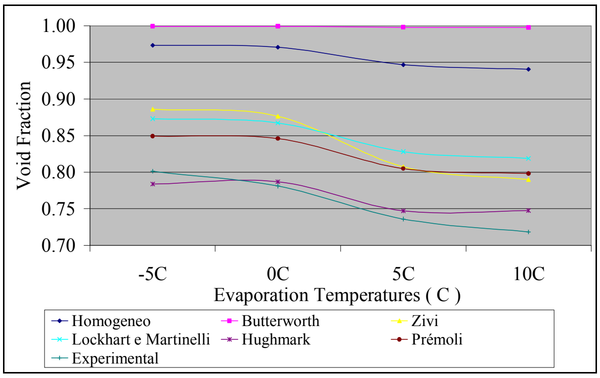

Figure 10 – Void fraction in function of the evapo-ration Temp. Condensation Temp. of 50^{o}C and 600rpm.

6. APPLICATION OF HUGHMARK CORRELATION

A simulation was developed to demonstrate the application of the Hughmark correlation where it was examined the influence that the refrigerant mass in the evaporator exerts on the superheat degree of this heat changer. To do this, an air cooling with tube diameter of 8 mm and total length of 10 m was considered. A mass flow of R22 equal to 30 kg/h flows through the evaporator with an evaporation temperature of 0^{o}C. The quality in the entrance of the heat exchanger is 20%. A linear variation of the quality along the section of exchange was assumed. The values of the superheat degree were simulated with unit variations of 5 to 20^{o}C. The average temperature of the external air was considered equal to 25°C and an external convective coefficient equal to 200 W/m² °C was used in the calculations.

In an evaporator, there are basically two heat exchange zones: an evaporation zone of length Z_{evap} and a superheat zone of length Z_{sa}. The length superheat zone can be calculated by:

$$Z_{sa} = A_{sa} / \pi d \qquad \text{and} \qquad \dot{Q}_{sa} = U A_{sa} \Delta T_{ml} \therefore A_{sa} = \dot{Q}_{sa} / \left(\Delta T_{ml} U \right) \tag{20}$$

Where d is the tube diameter, $\dot{Q}_{sa}$ is the heat flow between the refrigerant and air, ΔT_{ml} is the difference of the logarithmic average temperature between the refrigerant in the superheat zone and external air, and U is the global coefficient of exchange. This coefficient is defined as:

$$U = \frac{1}{1/h_f + 1/h_a} \tag{21}$$

Where h_f is the coefficient of convective heat exchange of the fluid side, calculated by the Dittus Boelter (1930) equation, and h_a is the coefficient of convective heat exchange of the air side, fixed in this simulation.

After calculating the superheat zone, the evaporation zone can be obtained by subtraction of the total length Z_{tot}:

$$Z_{evapo} = Z_{tot} - Z_{sa} \tag{22}$$

In the superheat zone, the refrigerant mass m_{sa} is function of the vapor density ρ_v and the volume of this region:

$$m_{sa} = \rho_v \, \pi d Z_{sa} \tag{23}$$

In the two-phase region, the refrigerant mass m_{evap} can be calculated using the Hughmak correlation, and the conditions described previously. Thus, this mass is given by:

$$m_{evap} = \left[\rho_l + \alpha \left(\rho_v - \rho_l \right) \right] V \tag{24}$$

Equation 24 shows that, in two-phase flow, the fluid mass is function of the void fraction. In turn, the void fraction is function of the quality. Knowing that the quality varies linearly throughout the test section, the evaporator must have its length divided in several sections and the calculation of the fluid mass must be processed in each section separately. After the individual calculations, it is necessary combine all the values to come to the total mass in the two-phase region. In this article, the two-phase region was divided in 200 elements. For the processing of these calculations, it was used FORTRAN programming languages.

The total refrigerant mass in the evaporator can be obtained by the sum of the mass in the two-phase region and the mass in the one phase region. The results are presented in Figure 11.

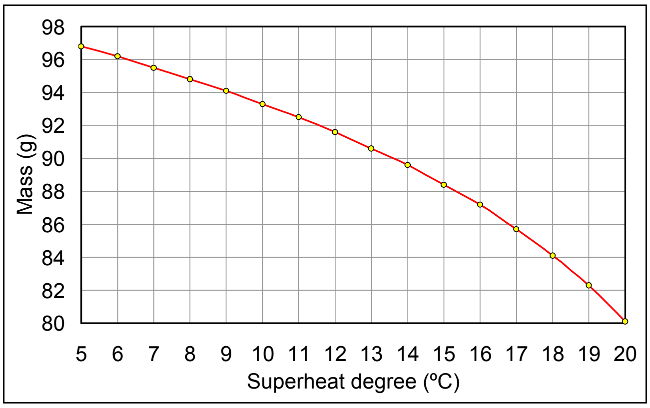

Figure 11 – Variation of superheat degree in function of the frigorific fluid mass

It is possible to see, in Figure 11, that small changes in the refrigerant mass reflect on significant variations of the superheat degree. It is possible to see that an error in 10% of the mass triples the superheat degree. The load of refrigerant and, consequently, the superheat degree control must be very well controlled to prevent performance problems system and possible failures of the components, as previously discussed. This result illustrates how important it is that the correct mass is put in evaporators and condensers refrigeration systems, which can be done by the help of an appropriate void fraction correlation.

7. CONCLUSIONS

A comparative study between the experimental and theoretical values of void fraction was conducted, and the conclusion was that the correlations of Hughmark are the most accurate correlation to estimate the value of the void fraction, it showed a variation less than 5% compared to the experimental values. With this correlation, and using a simple model of the heat transfer to estimate the length of the two-phase flow zone, the refrigerant inventory in the evaporator and condenser refrigeration system can be determined for different operation points of this system. Thus, the determination of the void fraction by Hughmark model, and its application to obtain the refrigerant inventory is a powerful tool for the correct balance of fluid in the refrigerator system. With this procedure it is possible minimize errors about the refrigerant load, since a very close initial parameter to the necessary can be foreseen.

8. REFERENCES

BAROCZY, C.J. A systemic correlation for two phase pressure drop. Chem. Eng. Progr. Symp. Ser, Vol. 62, p. 232 – 249, 1966.

BUTTERWORTH, D. A comparison of some void-fraction relationships for co-current gas-liquid flow. Int. J. Multiphase Flow, 1975, Vol. 1, p. 845-850.

DITTUS, F. W.; BOELTER, L. M. K. Heat transfer in automobile radiators of the tubular type. University of California Publications in Engineering, Berkeley, v. 2, n. 8, p. 443-461, 1930.

FIGLIOLA, R. S.; BEASLEY D. E. Theory and Design for Mechanical Measurements, 3rd ed., John Wiley & Sons, New York, NY, 2000.

GONÇALVES, A.A.JR. Metrologia – Parte I, 1a edição. Universidade Federal de Santa Catarina, Florianópolis, 1999.

HUGHMARK, G.A. Hold up in gas-liquid flow. Chemical Engineering Progress, 1962, Vol. 58, N° 4, p. 62-65.

LOCKHART, R.W., MARTINELLI, R.C. Proposed correlation of data for isothermal two-phase two-component flow in pipes. Chemical Engineering Progress, 1949, Vol. 45, N° 1, p. 39-48.

RICE, C.K. The effect of void fraction correlation and heat flux assumption on refrigerant charge inventory predictions. ASHRAE Transactions, 1987, Vol. 93, Part 1, p. 341-367.

THOM, J.R.S. Prediction of pressure drop during forced circulation boiling of water. Int. J. Heat Mass Transfer 7, 1964, p. 709–724.

ZIVI, S.M.. Estimation of steady state steam void fraction by means of the principle of minimum entropy production. Trans. ASME, J. Heat Transfer 86, 1964, p. 247 - 252.

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