A Study of the Flow through Capillary Tube Tunned up for the Cooling Circuit with Fluoroinert Refrigerants

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Abstract:

Capillary tube expansion devices are widely used in refrigeration equipments, nevertheless the mechanism of the flow is still not fully described and understood, so the experimental verification of most predictions is still necessary. We have prepared a modified numerical model of the capillary flow and verified it both through the available data from literature and also via our own measurements performed in a real cooling circuit with pure fluoroinert refrigerant.

Keywords: Capillary tube, fluoroinert refrigerant, cooling circuit, two-phase flow, model comparisons

1. INTRODUCTION

Capillary tubes as expansion devices are mostly used in smaller refrigeration and air conditioning devices because they are relatively simple and inexpensive parts. They substitute more expensive and complex thermostatic valves. Nevertheless, one can find another reason for their use in highly specialized cooling circuits where, for instance, the space constrains are dominant or highly magnetic and radiation hard surrounding eliminates the use of more sophisticated throttling devices. The use of capillaries in the complex cooling system for the particle detector became the motivation for this study. In such application one has to face also other problems such as the multiple manifolding due to the high number of individual evaporators. Another problem arises from the fact that the capillary does not connect the condenser and evaporator directly, but it is bound between the pressure regulator and backpressure regulator, usually far away from the compressor-condenser unit and placed in a relatively unapproachable area. Then the proper sizing of the capillary (that should also cope with the slightly changing refrigerant mass flow) becomes an essential problem.

A capillary is a tube of small inner diameter mostly between 0.5 - 1.5 mm and length about 1.5 - 6 m. Applications of new alternative refrigerants yield a request of more accurate and general methods for predicting capillary thermal and fluid-dynamic behavior. A number of both theoretical and experimental studies have been carried out especially during the past 10 years.

Its inner diameter and its length define the pressure distribution over the capillary-tube. The first small pressure drop occurs at the inlet as the sub-cooled liquid coolant enters the capillary Fig.1a. Between points 1 and 2 the pressure decreases linearly until the saturation pressure is reached. The most basic studies assume beginning of the refrigerant two-phase flow with non-linear decrease of pressure just at the point 2. The capillary-tube outlet conditions are defined by point 3, where the flow can reach its critical speed (local speed of sound) or lower speed. An uncontrolled expansion connected with shock wave generation may occur between points 3 and 4 in case of critical conditions at the capillary outlet. Final pressure value at point 4 is equal to discharge pressure set after the capillary.

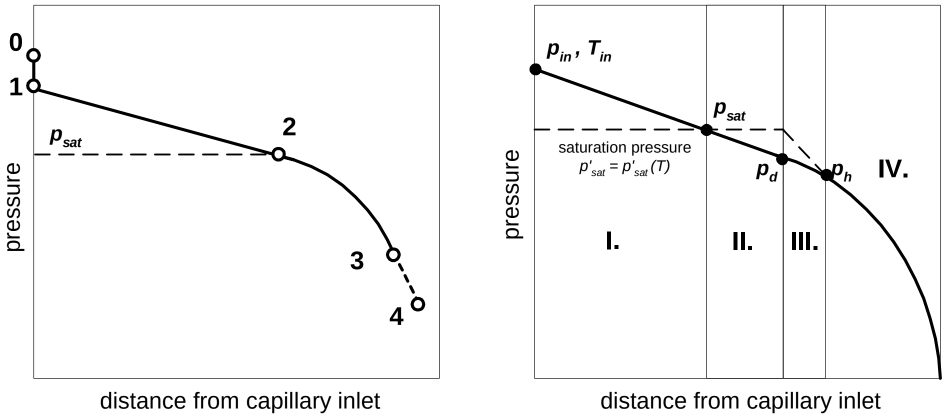

Fig. 1. a) Pressure distribution along the capillary tube;
b) Regions of capillary flow *I. Subcooled single liquid phase, II. Metastable single liquid phase, III. Metastable two-phase region (liquid - vapor), IV. Thermodynamic equilibrium two-phase region (liquid - vapor)*

Mikol [1] and Li et al. [2] reported in their experimental research works that the evaporating flow through capillary consists of four different regions. There are not only two thermodynamic equilibrium phenomena of sub-cooled liquid and two-phase mixture, but also two metastable regions marked in Fig.1b. The saturation condition and the flash point define metastable single liquid region - II. The evaluation of pressure of vaporization at the flash point has been studied by Chen [3] and Lackme [4]. The second region is a zone where both the liquid and two-phase flow remain in metastable region - III.

2. SOME OF PREVIOUS WORKS AND LITERATURE REVIEW

The majority of theoretical research works assume just two thermodynamic equilibrium regions of capillary flow and use the homogeneous two-phase flow with same velocities of both vapor and liquid. For instance Bansal [5] has presented a model of homogeneous two-phase flow through adiabatic capillary tube. In this case of simplified capillary flow pattern, the one-phase region was solved analytically. Numerical solution applied to the two-phase region was based on Lins approximation of frictional pressure gradient. Kritsadathikarn [6] has developed a similar numerical model of the adiabatic capillary flow. Pressure drop in the two-phase region was calculated from the friction factor f_{TP} obtained from the two-phase Reynolds number using the Duklers viscosity of a two-phase mixture. Sami and Tribes [7] presented a numerical study of capillary tube performance working for alternative refrigerants and their binary mixtures. Fairly good agreement between theoretical results of the proposed model and their experimental data was reached. Their homogeneous model did not consider the effect of two-phase component velocity difference. None of the presented models has provided direct information about the slip ratio, the variable defining the ratio between two-phase component velocities. Due to the lack of correlation of the slip ratio the separated models of capillary flow have received little attention in the past in literature.

Wong and Ooi [8] developed a numerical model with separated two-phase flow solved with Miropolskiy [9] slip ratio correlation and Lins approximation of two-phase frictional pressure gradient. Comparison with experimental data and simplified homogeneous model showed that the separated numerical model gave better prediction of the capillary tube performance. Wongwises and Chan [10] studied the effects of various correlations of slip ratio, frictional pressure gradient and friction factor on the prediction computed with separated model of adiabatic capillary flow.

All above mentioned studies assume pure adiabatic capillary flow. Sinpiboon and Wongwises [11] developed numerical study of the refrigerant flow through non-adiabatic capillary. The mathematical model is categorized into three different cases, depending on the position of the capillary in the heat exchanger. A set of differential equations is solved by the explicit method of finite-difference method. Another model providing non-adiabatic capillary flow solution was introduced in a numerical study by Escanes and Pérez-Segarra [12]. The set of four differential equations (mass, momentum, energy and entropy balance) is solved on the base of one-dimensional finite volume method. Heat transfer coefficient is defined from Gnielinski correlations [13] and assumes the known temperature distribution on the capillary inner wall.

A very advanced complex model of the capillary flow was presented recently by García-Valladares and Pérez-Segarra [14]. Their model represents solution of the capillary flow with all four partial regions (two thermodynamic equilibrium and two metastable) in an adiabatic and non-adiabatic capillary tube.

The numerical models are somewhat time-consuming and they require programming skills, so this approach to the problem is not always easy for engineers. So the analytical approximate models are also favored in everyday practice, even if they are less precise and do not describe capillary behavior in details.

A novel approximate analytical approach was developed by Yilmaz and Ünal [15]. Their work simplifies the model of an adiabatic capillary tube. Zhang and Ding [16] presented approximate analytic solution of capillary tube valuable for theoretical analysis and engineering calculation. Two solutions of adiabatic capillary behavior were developed. The first one is the explicit function for the capillary tube length prediction. The second one is represented by the explicit function estimating the refrigerant mass flow rate.

A lot of various experimental works were presented in literature besides theoretical studies of capillary tube performance over the years. Just to mention some of them – Mikol [1] in his very detailed study described the values of friction factor in both one-phase and two-phase flow based on his own experimental data for refrigerants R12 and R22. Koizumi and Yokoyama [17] measured temperature and pressure development over glass capillary. The existence of metastable region in capillary flow was proved from their measurements and from visual observations as well. The theoretical pressure drop prediction was compared with their experimental data.

Li and Lin [2] investigated experimentally the metastable phenomenon of R12 through two different capillary tubes. The effects of the inner capillary diameter, the backpressure, the mass flow-rate and the inlet sub-cooled temperature of the refrigerant on the delay of vaporization of the refrigerant were discussed. The model for two-phase flow in capillary tubes, considering the thermodynamic non-equilibrium phenomenon during vaporization, was developed in their other study [18]. A comparison, being in a good agreement with their experimental data has shown that model is practical and could be used for size prediction of the capillary tube as an expansion device.

Let us conclude with an experimental study of the capillary flow presented by Melo and Ferreira [19]. The two-phase flow of three different refrigerants R12, R134a and HC-600a was investigated for eight capillaries with varied combinations of length, diameters and tube roughness. A conventional, dimensionless analysis was utilized to derive correlations to predict the mass flow rates through capillary tube.

3. OBJECTIVES, METHODOLOGY AND PREPARATION STUDIES

All previously mentioned studies - numerical, analytical and experimental require some preparation activities or studies concerning namely adequate thermodynamic and transport properties of the refrigerant used in the models and also some information about capillary parameters. Mainly the correct evaluation of the inner diameter and inner surface roughness is quite important.

Thermophysical data of our target refrigerants, i.e. R218, octafluoropropane (referred to as C_3F_8 in the text) and the R610, i.e. perfluorobutane (referred to as C_4F_{10} in the text), are not so frequent in the literature compared to the standard group of refrigerants. We have used the REFPROP database [20] with provisional data files to generate required properties and correlated them with our previously published measurements realized with fluoroinerts and their mixtures [21], [22]. The retrieved data were than fitted mostly via polynomial or exponential functions in the regions relevant to our studies.

The inside diameters of our capillaries (made of stainless steel, copper and copper nickel) were measured usually by two methods. The first was the weighting method - the capillary tubes were filled with fluoroinert liquids (perfluorohexane - C_6F_{14} or perfluorooctane - C_8F_{18}) having the density 1.7 times higher then water at normal laboratory temperatures. A precise balance determining the mass of the filled liquid in the capillary served then for the ID evaluation. The second method used for verification of our ID measurement, applied to a number of capillary sample cuts from the same batch, was an optical one.

To verify or estimate inner surface capillary roughness and consequently to use the most suitable relation for the friction factor led us to a detailed investigation of the liquid phase flow through the capillaries. The influence of the inner surface capillary roughness can be seen from Fig. 2.

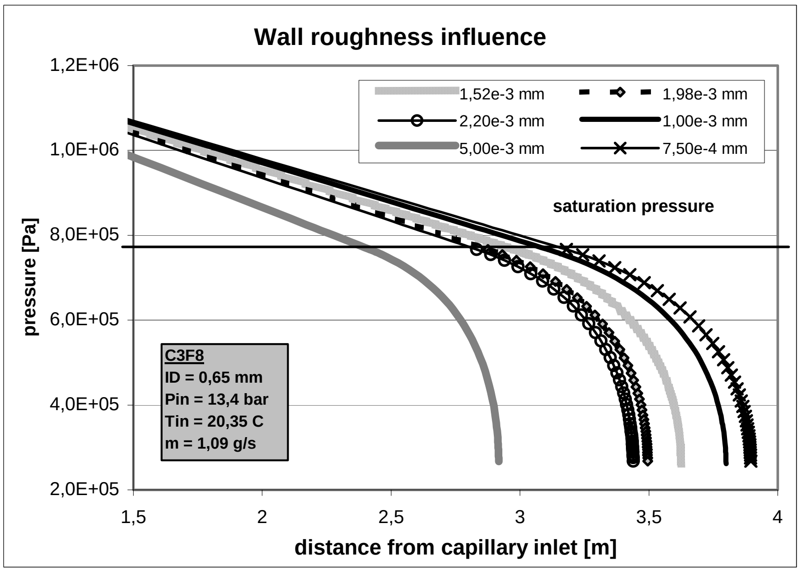

Fig. 2. Inner wall surface roughness impact on the capillary length and pressure profile

For the liquid phase flow studies, at laboratory temperatures, we have used the setup introduced in the Fig. 3. The C_6F_{14} and C_8F_{18} fluoroinert liquids were utilized since they exhibit similar properties (densities, viscosities, surface tension, etc.) at laboratory temperatures as the C_3F_8 and the C_4F_{10} in real cooling circuits under the adequate technological temperatures.

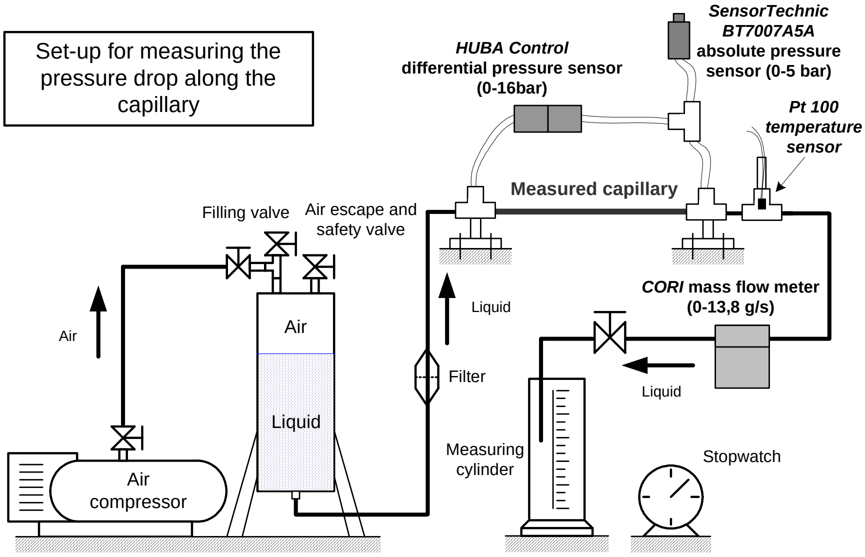

Fig. 3. Experimental set up for capillary behavior in liquid phase flow (C_6F_{14}, C_8F_{18})

Characteristics of the several different capillaries were evaluated by measuring the pressure drop using a precise differential HUBA pressure sensor together with a reference absolute Sensor Technics pressure transducer. The mass flow was monitored with the precise Corriolis type mass flow meter CORIFLOW and it was also initially checked through the direct weighting method. Some 18 capillary samples were investigated (ranging in the ID from 0.6 to 1.1 mm and in lengths from 0.7 to 3 m). The four different friction factor approximation (Churchill, Haaland, Colebrook and classical equation) usually used in capillary flow models were compared with our measured results. Evaluation resulted in a conclusion to use the Colebrook friction relation for our studies with copper capillary and the inner surface roughness was selected to be around 2 x 10^{-6} m. Data comparisons are demonstrated in Fig. 4.

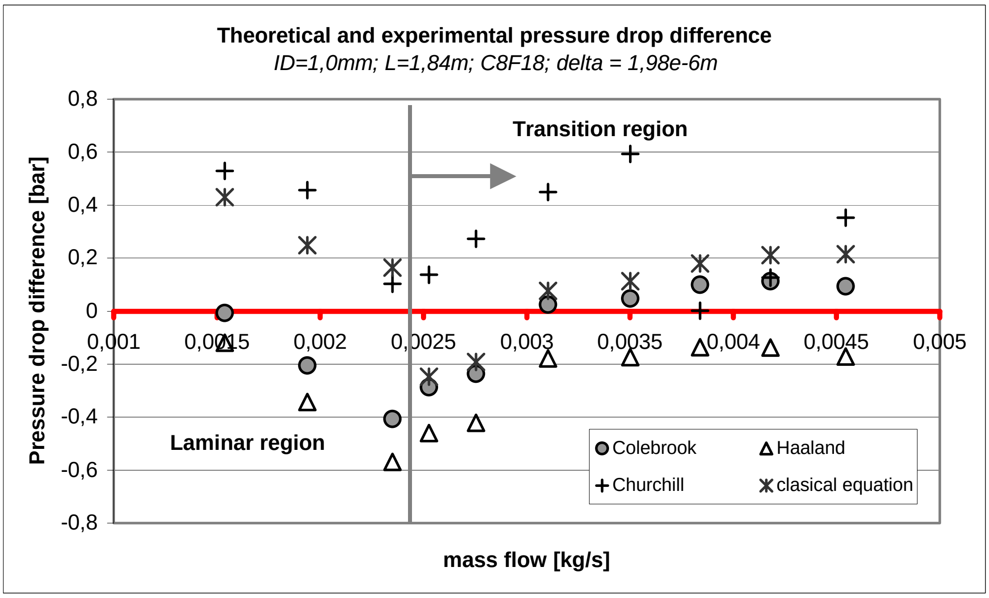

Fig. 4. Evaluation of the different friction factors against the measured data

4. MODIFIED MODEL OF THE CAPILLARY FLOW

The aim of this study was to develop a steady state numerical simulation of the capillary flow behavior. The current study includes some modifications of previous models presented in the literature (mainly [10], [12] and [14]). The model assumes two equilibrium regions of capillary flow. Obtained solution provides the developments of all main flow characteristics (pressure, temperature, vapor quality, etc.) along a capillary tube.

4. 1 Mathematical formulation and numerical solution

Governing equations of the flow (continuity, momentum, energy and entropy) in differential form were discretized using one-dimensional finite volume method in a similar manner as in [12].

Continuity equation: $\frac{\partial \rho}{\partial t}+\frac{\partial(\rho.v)}{\partial z}=0$

Energy equation: $\frac{\partial(\rho.E)}{\partial t}+\frac{\partial(\rho.v.E)}{\partial z}+\frac{\partial}{\partial z}(p.v-\tau_{zz}.v+q)=\rho.v.K_z$

Momentum equation: $\frac{\partial(\rho.v)}{\partial t}+\frac{\partial(\rho.v.v)}{\partial z}=-\frac{\partial p}{\partial z}+\frac{\partial \tau_{zz}}{\partial z}+\rho.K_z$

Entropy creation: $\frac{\partial(\rho.s)}{\partial t}+\frac{\partial(\rho.v.s)}{\partial z}\geq-\frac{1}{T}\frac{\partial q}{\partial z}$

The capillary flow may be defined by the following parameters:

- Refrigerant mass flow rate
- Temperature and pressure at the capillary inlet
- Inner, outer diameter and wall roughness of the capillary tube
- Discharge diameter, i.e. approximately hydraulic diameter of evaporator
- Temperature of the capillary inner wall or average ambient temperature around non-insulated capillary
- Thermophysical properties of a considered refrigerant

The following simplifications were assumed in our model:

- Capillary flow is solved as one-dimensional steady state problem
- Two equilibrium capillary flow regions are considered
- Inner, outer diameter and wall roughness are not varying along capillary tube length
- Gravity is not put in account, capillary is solved for the horizontal position

Taking into account these simplifications, the governing equations for ***one-phase region*** will have the following form after the discretization:

Continuity equation (*the same for both capillary flow regions*):

$$\dot{m}_{i+1}-\dot{m}_i=0 \tag{1}$$

Momentum equation:

$$\left[\dot{m}.v\right]_{i+1}-\left[\dot{m}.v\right]_i=\left[p_i-p_{i+1}\right].A-\tilde{\tau}_w.P.\Delta z \tag{2}$$

Energy equation for specific energy ($e=h+v^2/2=E+p/\rho$):

$$\left[\dot{m}.e\right]_{i+1}-\left[\dot{m}.e\right]_i=-\tilde{q}_w.P.\Delta z \tag{3}$$

In the process of discretization of momentum, respectively energy, adequate equations in formulas were considered $\partial\tau_{zz}.A=-\tilde{\tau}_w.P.dz$, respectively $\partial q.A=-\tilde{q}_w.P.dz$.

The enthalpy variations have been evaluated by neglecting their dependence on pressure variations: that is $dh \cong c_p.dT$. The discretized energy equation solved for the control volume outlet temperature gives

$$T_{i+1} = \frac{\overline{\dot{q}}_w.P.\Delta z + T_i.\overline{\dot{m}}.c_p - \overline{\dot{m}}.\left(\frac{v_{i+1}^2 - v_i^2}{2}\right)}{\overline{\dot{m}}.c_p} \tag{4}$$

The discretized momentum equation solved for the control volume outlet pressure may be expressed as

$$p_{i+1} = p_i - \frac{\Delta z}{A}\left[\frac{\dot{m}_{i+1}.v_{i+1} - \dot{m}_i.v_i}{\Delta z} + \frac{\overline{f}}{4}\frac{\overline{\dot{m}}^2}{2.\rho.A^2}.P\right] \tag{5}$$

The shear stress at the wall $\tilde{\tau}_w$ in equation (2) is given as

$$\tilde{\tau}_w = \frac{\overline{f}}{4}.\frac{\overline{\dot{m}^2}}{2.\rho.A^2} \tag{6}$$

Discretized governing equations of the ***two-phase region*** have the following form.
Momentum equation:

$$\left[\dot{m}^l.v^l\right]_{i+1} - \left[\dot{m}^l.v^l\right]_i + \left[\dot{m}^g.v^g\right]_{i+1} - \left[\dot{m}^g.v^g\right]_i = \left[p_i - p_{i+1}\right].A - \tilde{\tau}_w.P.\Delta z \tag{7}$$

Energy equation:

$$\overline{\dot{m}}\left(e_{i+1}^l - e_i^l\right) + \dot{m}_{i+1}^g\left(e_{i+1}^g - e_{i+1}^l\right) - \dot{m}_i^g\left(e_i^g - e_i^l\right) = -\tilde{q}_w.P.\Delta z \tag{8}$$

Entropy creation:

$$\overline{\dot{m}}\left(s_{i+1}^l - s_i^l\right) + \dot{m}_{i+1}^g\left(s_{i+1}^g - s_{i+1}^l\right) - \dot{m}_i^g\left(s_i^g - s_i^l\right) - \frac{1}{T_w}.\tilde{q}_w.P.\Delta z \geq 0 \tag{9}$$

The control volume outlet vapor quality is expressed using the two-phase region discretized energy equation (8)

$$x_{i+1}.\dot{m}_{i+1}\left[h_{i+1}^g - h_i^g + \frac{{v_{i+1}^g}^2 - {v_{i+1}^l}^2}{2}\right] = x_i.\dot{m}_i\left[h_i^g - h_i^l + \frac{{v_i^g}^2 - {v_i^l}^2}{2}\right] - \\ - \overline{\dot{m}}\left[c_P^l\left(T_{i+1} - T_i\right) + \frac{{v_{i+1}^l}^2 - {v_i^l}^2}{2}\right] + \overline{\dot{q}}_w.P.\Delta z \tag{10}$$

The discretized momentum equation solved for control volume outlet pressure gives

$$p_{i+1} = p_i - \frac{\Delta z}{A}\left[\frac{\overline{f}}{4}\frac{\overline{\dot{m}}^2}{2.\rho.A^2}.P + \frac{\dot{m}_{i+1}\left\{x_{i+1}.v_{i+1}^g + \left(1 - x_{i+1}\right).v_{i+1}^l\right\} - \dot{m}_i\left\{x_i.v_i^g + \left(1 - x_i\right).v_i^l\right\}}{\Delta z}\right] \tag{11}$$

The control volume outlet temperature is evaluated from saturation condition $T_{i+1} = T_{sat}\left(p_{i+1}\right)$. Critical length is defined as the distance from the capillary tube inlet to the control volume where critical conditions are reached (i.e. when the entropy creation equation (9) is not verified). In this case the capillary flow reaches its critical conditions and the flow velocity is equal to local speed of sound.

4.2 Modifications of the model

Our approach via simulation allows solution of both homogeneous and separated flow through adiabatic or non-adiabatic capillary tube. Various optional correlations of empirical coefficients were implemented into the model algorithm:

- *Two-phase viscosity*: Mc Adams, Lin, Owen, Dukler
- *Friction factor*: Haaland, Churchill, Colebrook
- *Slip ratio*: Chisholm, Zivi, Premoli
- *Frictional pressure gradient (two-phase multiplier)*: Friedel, Lin

The effects of relevant parameters on the prediction of capillary flow model were investigated. The underlined correlations were chosen as default settings of our model.

In a non-adiabatic solution of the capillary flow, two kinds of heat transfer coefficient can be evaluated. The first one assumes that the capillary is placed in a heat exchanger, the inner wall temperature is known and just the forced convection inside the capillary has to be solved. The second one is the case with non-insulated capillary. The natural convection between ambient and capillary outer surface, conduction through its wall and forced convection of flowing refrigerant inside the capillary have to be solved in this case.

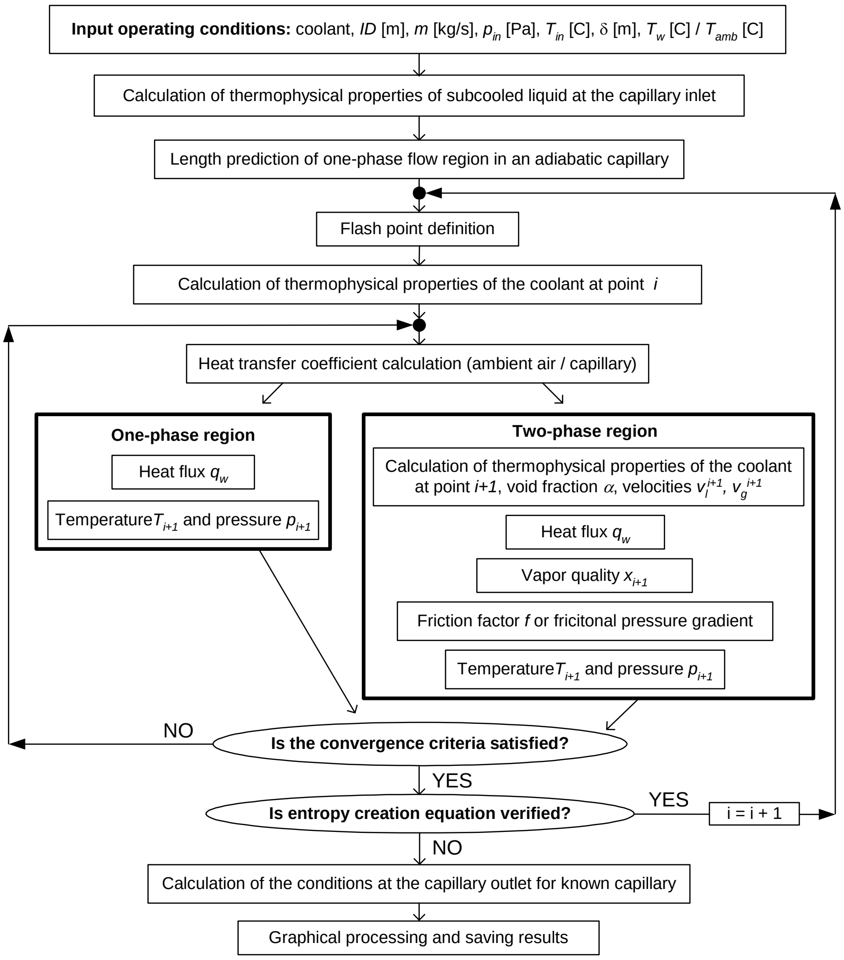

Fig. 5. Flow chart of the capillary model solved by Matlab program

The model allows finding two different solutions at the outlet region for a capillary with defined basic parameters. The variables at the discharge (temperature, pressure, density, vapor quality...) are calculated both for the case with the capillary outlet placed in one-phase region and for the case with the outlet situated in two-phase region. The conditions at the discharge are calculated from governing equations using the known value of the discharge diameter.

The model was tailored to provide capillary flow performance for the fluoroinert refrigerants, namely C_3F_8 and C_4F_{10}. Theoretical results of our numerical simulation were verified with our own experimental data.

The presented model based on one-dimensional finite volume method discretization is structured in such manner that it is free to further modifications and improvements compared to most other numerical models mentioned in the literature review. This means that two metastable regions could be easily implemented in the model. A simple modification of the code can adopt the evaluation of the critical (maximum) mass flow rate through the capillary tube of given length.

5. VERIFICATION OF THE MODEL

The initial evaluation of the presented model was performed for typical refrigerants (R12, R22, R134a) and compared to available experimental data from literature. Table I summarizes a map of various parameters in the process of validation.

Table I. Summary of the combined parameters in two-phase flow

Experiment	experimental data	
Psat = Psat(T)	saturation pressure calculated from the measured temperature	
Notation	**Results of our numerical model** **Two-phase multiplier** Φ_{LO}	**Slip ratio** S
Num. model: Lin; Premoli	Lin	Premoli
Num. model: Lin; Chisholm	Lin	Chisholm
Num. model: Lin; Zivi	Lin	Zivi
Num. model: Friedel; Premoli	Friedel	Premoli
Num. model: Friedel; Chisholm	Friedel	Chisholm
Num. model: Friedel; Zivi	Friedel	Zivi
Num. model: homogeneous	homogeneous model *	S = 1

** in case of homogeneous model* Φ_{LO} *is not calculated, friction factor is evaluated from* Re_{TP}

Some results typical for the refrigerant R12 are introduced in Fig. 6 and Fig. 7.

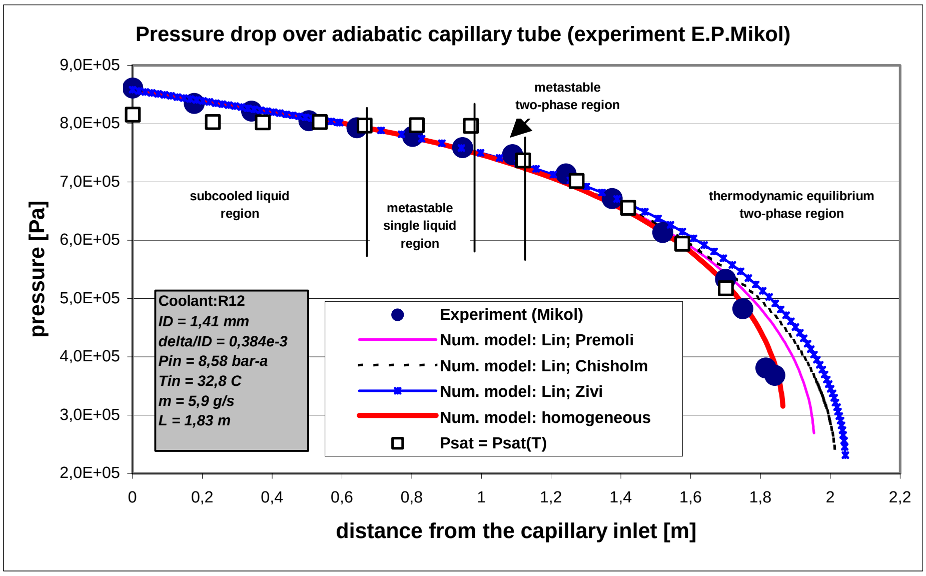

Fig. 6. Verification of the model and comparison with Mikol experimental data

One can see that the numerical model with simplified homogeneous two-phase flow region ($v^g = v^l$) reproduces experimental data reasonably well. Results obtained from the model of the separated two-phase flow show some variations. The discrepancies are caused by the choice of the different approximation of the two-phase multiplier and slip ratio – see Table I. It can be concluded that the Friedel approximation of the two-phase multiplier delivers usually a higher pressure drop and consequently the predicted length of the capillary is shorter. Concerning the optimal choice of the parameters for the separated two-phase flow we would recommend the combination of the Lin two-phase multiplier and Premoli slip ratio.

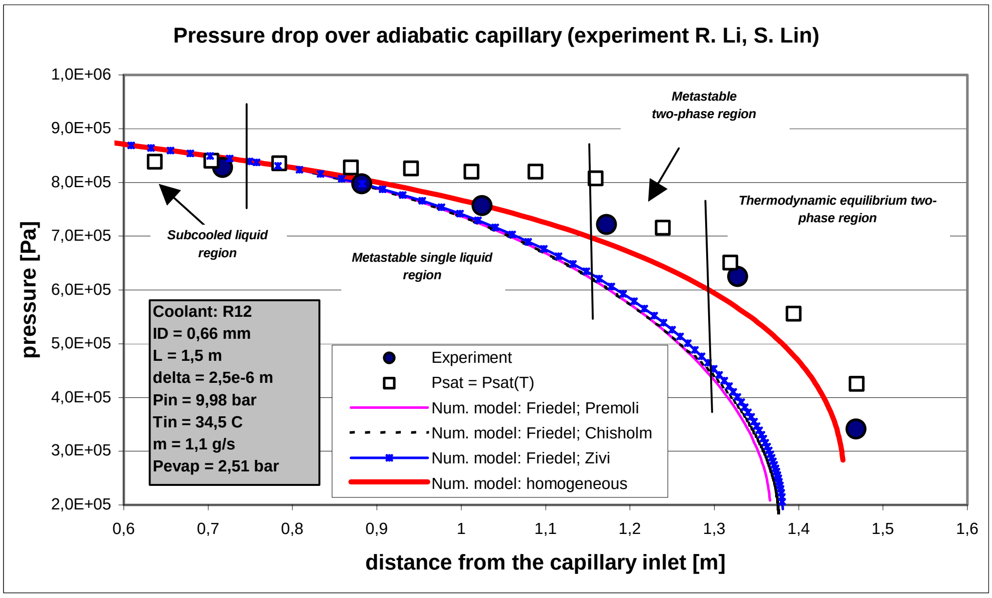

Fig. 7. Verification of the model and comparison with Li and Lin experimental data

6. EXPERIMENTAL VERIFICATION OF THE MODEL

Experimental measurements with two in-house made copper capillaries were performed inside a real circuit working with the fluoroinert C_3F_8 in completely dry oil-free mode. Fig. 8 shows the schematics of the circuit.

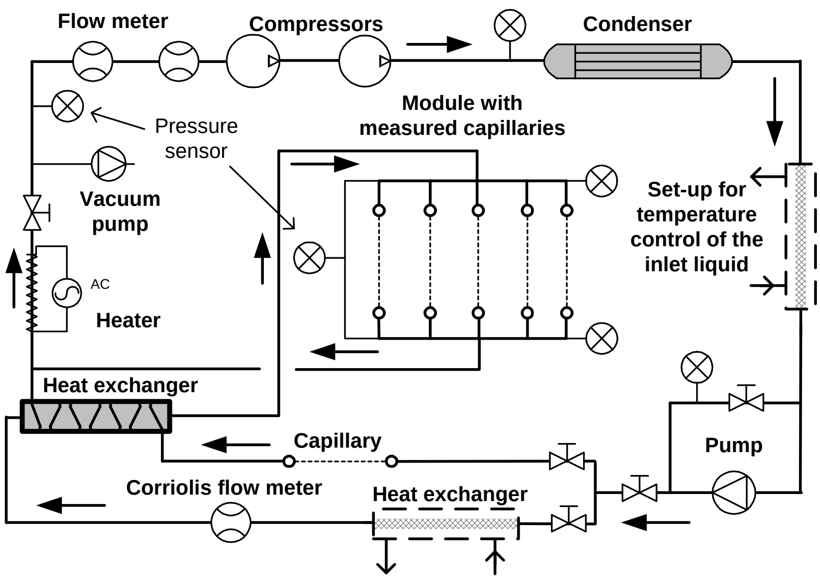

Fig. 8. Real cooling circuit using the fluoroinert refrigerant C_3F_8

The circuit with two stage compression is versatile, capable of changes of parameters such as the inlet pressure via pressure regulator, evaporation pressure via back pressure regulator, degree of sub-cooling realized through liquid vapor heat exchanger. Mass flow is recorded via a Danfoss corriolis mass flow meter on the liquid side and it is also checked on the vapor side. The pump serves for fast increase of the supply pressure if necessary.

The capillary implemented into the circuit for our measurements was divided into uneven length sections where the temperatures were measured with calibrated mini Pt1000 and NTC sensors inserted at the level of the inner surface of the capillary. Pressure taps were made with small diameter drills just opposite to the temperature placement. Pressure drops are monitored across the section and three extra absolute pressure sensors were also installed for the reference. The lengths of the section were made larger in the expected liquid phase flow length and become shorted in the expected two-phase flow length as it is shown in the Fig. 9. All setup was insulated with Armaflex material. The verified ID of the capillary was 1.03 mm and two lengths (L_1 ~ 3.25 m and L_2 ~ 5.08 m) were measured as the pilot experimental project.

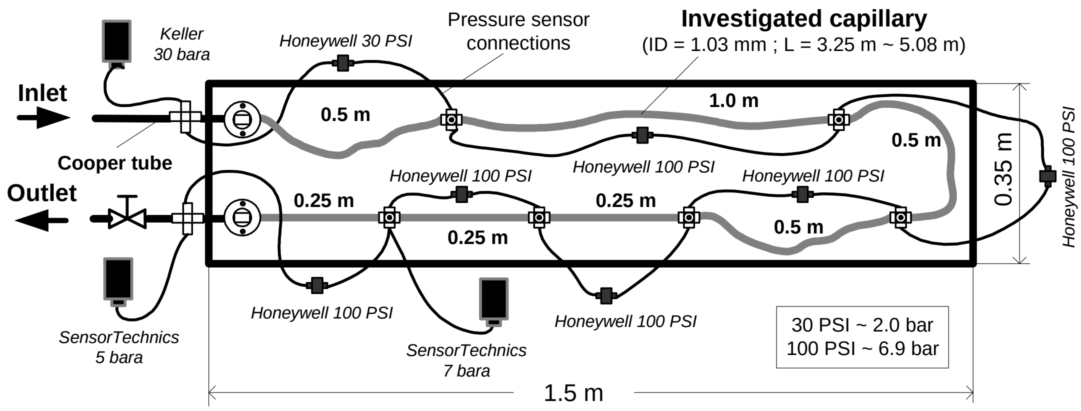

Fig. 9. Schematics of the model capillary (ID ~ 1.03 mm and L_1 ~ 3.25 m, L_2 ~ 5.08 m)

Experimentally obtained pressure profile over the capillary installed in the circuit working with refrigerant C_3F_8 was compared to the pressure profile predicted by the numerical model. Results with running parameters during the experiment are introduced in Fig. 10 for the capillary having the length L_1 ~ 3.25 m and in Fig. 11 for the capillary with L_2 ~ 5.08 m.

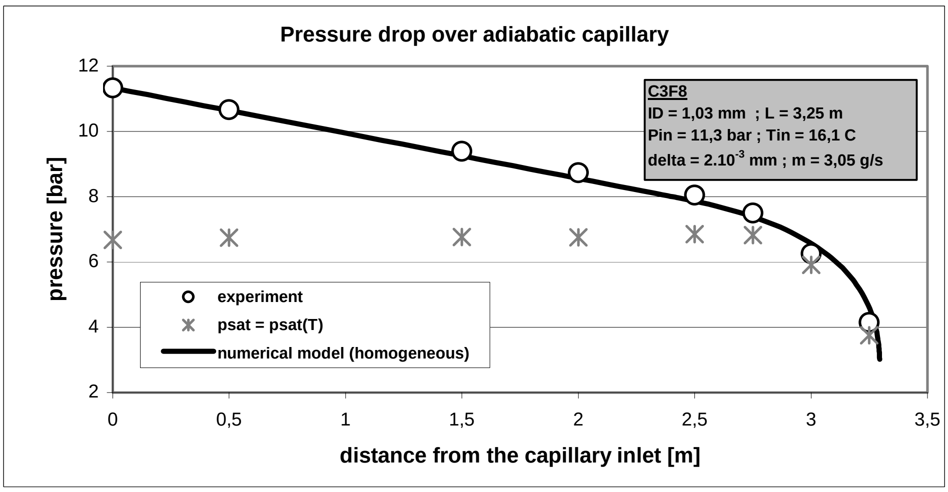

Fig. 10. Capillary behavior within the circuit / comparison between experimental and predicted data

There is reasonably good agreement between experimental and numerical results of the pressure profile plotted as the solid line. Points corresponding to the saturation pressure were calculated according to the monitored temperature along the capillary. Nevertheless there are small discrepancies in expected starting point of vaporization at the end of liquid flow region between the experimental data and data obtained from the simulation model. Denser sensor placement in the transition region could probably help to clarify it.

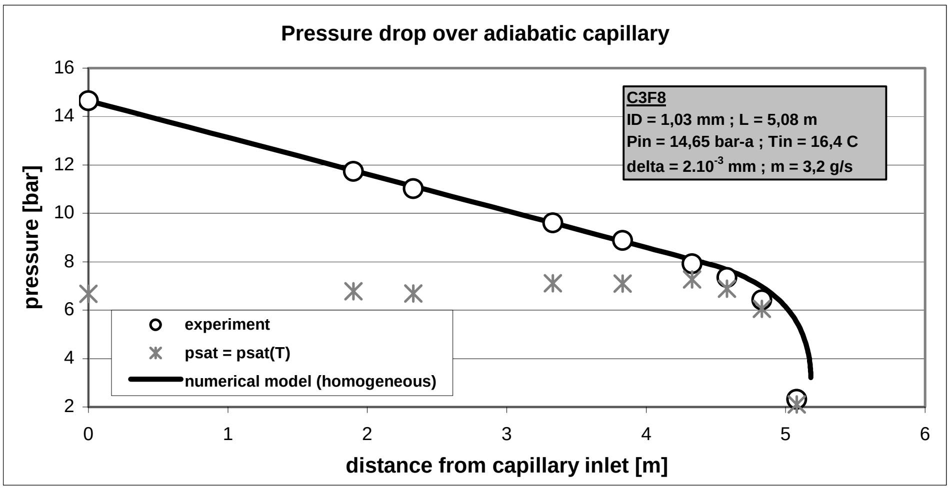

Fig. 11. Capillary behavior within the circuit / comparison between experimental and predicted data

The numerical model was used to study the influence of the three important parameters for capillary behavior:

a) Mass flow of the refrigerant.

The increased mass flow decreases both the single phase and two-phase length of the capillary and increases the critical pressure at the capillary outlet.

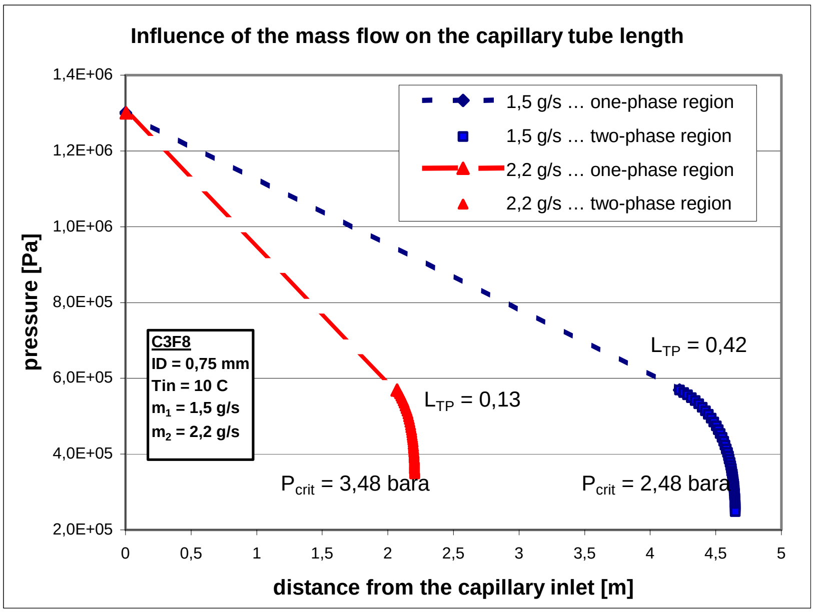

Fig. 12. Influence of the changed parameters for the capillary flow

b) *Inner diameter of the capillary.*

It is the primary design parameter of the capillary that influence significantly the capillary flow pattern. Its incorporation with capillary length (both for the liquid phase and two-phase flow) is demonstrated in Fig. 13. There is also indication that critical pressure is lower for bigger ID when capillary can eventually become “choked”. Plotted curves of four different ID represent points where model predicts critical flow condition.

c) *Refrigerant inlet capillary temperature.*

The higher sub-cooling of the refrigerant, the lower inlet temperature is achieved and consequently the saturation pressure that indicates the transition region between the one and two-phase flow is lowered. Then the ratio between adequate capillary flow regions is changing and the overall capillary length is increasing.

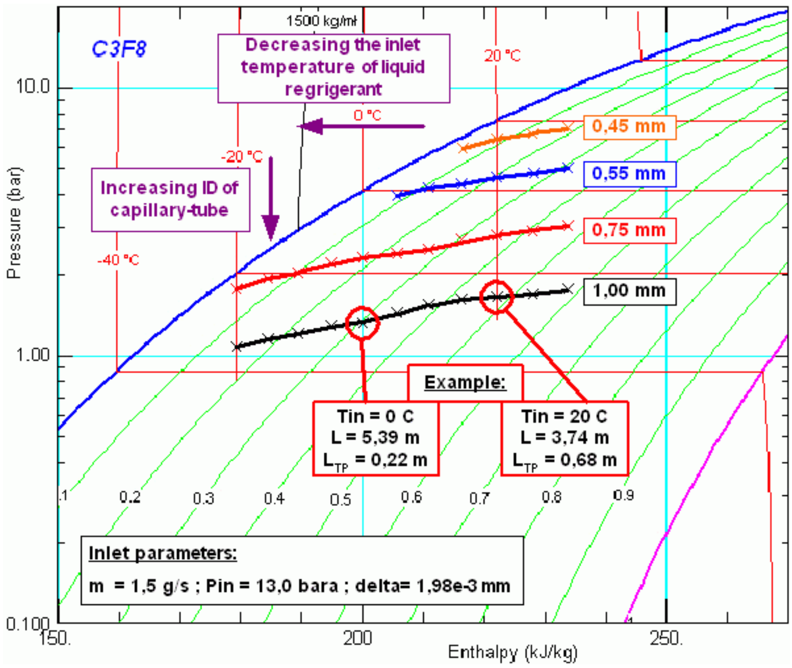

Fig. 13. Influence of the changed parameters for the capillary flow

7. CONCLUSION

A numerical model was prepared for analyzing the behavior of a capillary tube as a throttling device in cooling circuits working with fluoroinert refrigerants. The reliability of the model was verified both with available data from literature, mostly for the traditional refrigerants and then comparisons were performed also with our own experimental data obtained from measurements within the real cooling circuit using C_3F_8.

The need to verify or estimate inner surface capillary roughness and friction factor led us to detailed investigation of the liquid phase flow through the capillaries via experiment in single liquid flow region.

The agreement between experimental and simulation results, mostly for the adiabatic capillary tube with separated two-phase flow indicates that the model can be used to predict complex flow behavior in capillary tubes. It can also help to select appropriate dimension of capillaries in refrigeration cycles even with infrequently used coolants – in special applications.

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Nomenclature:

z [m] … axial coordinate
d [m] … inner diameter
δ (*delta*) [m] … capillary wall roughness
P [m] … capillary inner perimeter
A [m^2] … capillary inner cross section
Δz [m] … control volume length
L [m] … capillary tube length
t [s] … time
$\dot{m}$ [kg. s^{-1}] … mass flow rate
ρ [kg.m^{-3}] … density
p [Pa] … pressure
T [K] … temperature
v [m.s^{-1}] … velocity
f [-] … friction factor
τ [Pa] … shear stress
E [J.kg^{-1}] … energy (sum of internal energy and kinetic energy)
h [J.kg^{-1}] … enthalpy
e [J.kg^{-1}] … specific energy ($e = h + v^2/2 = E + p/\rho$)
s [J.kg^{-1}.K^{-1}] … entropy
x [-] … vapor quality
c_p [J.kg^{-1}.K^{-1}] … specific heat
q [W.m^{-2}] … heat flux
K_z [m.s^{-2}] … body forces (gravity)
α [-] … void fraction
S [-] … slip ratio
Φ_{LO} [-] … two-phase multiplier
Re [-] … Reynolds number

Subscripts

i … control volume inlet (point i of the computational grid)
$i+1$ … control volume outlet (point $i+1$ of the computational grid)
TP … two-phase
w … condition at the capillary inner wall
sat … saturation properties
$crit$ … critical conditions
in … capillary inlet
amb … ambient
$evap$ … evaporation

Superscripts

g … vapor phase
l … liquid phase
– … arithmetical average over a control volume
~ … integral average over a control volume

REFERENCES

[1] E. P. Mikol: Adiabatic single and two-phase flow in small bore tubes, ASHRAE Journal, (1963), pp. 75-86

[2] R. Y. Li, S. Lin, Z.Y. Chen, Z.H. Chen: Metastable flow of R12 through capillary tubes, International Journal of Refrigeration 13, (1990), pp. 181-186

[3] Z. H. Chen, R. Y. Li, S. Lin, Z. Y. Chen: A correlation for metastable flow of refrigerant 12 through capillary tubes, ASHRAE Transactions 96 (1), (1990), pp. 550-554

[4] C. Lackme: Incompleteness of the flashing of a supersaturated liquid and sonic ejection of the produced phases, International Journal of Multiphase Flow 5, (1979), pp. 131-141

[5] P.K. Bansal, A.S. Rupasinghe: A homogeneous model for adiabatic capillary tubes, Applied Thermal Engineering 18, (1998), pp. 207-219

[6] P. Kritsadathikarn, T. Songnetichaovalit, N. Lokathada: Pressure Distribution of Refrigerant Flow in an Adiabatic Capillary Tube, Research Article, Science Asia 28, (2002), pp. 71-76

[7] S.M. Sami, C. Tribes: Numerical prediction of capillary tube behavior with pure and binary alternative refrigerants, Applied Thermal Engineering 18 (6), (1998), pp. 491-502

[8] T. N. Wong, K. T. Ooi: Adiabatic capillary tube expansion devices: A comparison of the Homogeneous flow and the Separated flow models, Applied Thermal Engineering 16 (7), (1996), pp. 625-634

[9] Z. L. Miropolskiy, R. I. Shneyerova, A. I. Karamysheva: Vapor void fraction in steam fluid mixture flowing in heated and unheated channels, International Heat Transfer Conference, Vol. 5, Paris (1970), paper B 4.7.

[10] S. Wongwises, P. Chan: Two-phase separated flow model of refrigerants flowing through capillary tubes, International Comm. Heat Transfer 27 (3), (2000), pp. 343-356

[11] J. Sinpiboon, S. Wongwises: Numerical investigation of refrigerant flow through non-adiabatic capillary tubes, Applied Thermal Engineering 22, (2002), pp. 2015-2032

[12] F. Escanes, CD. Perez-Segarra, A. Oliva: Numerical simulation of capillary-tube expansion devices, International Journal of Refrigeration 18, (1995), pp. 113-122

[13] V. Gnielinski: New equations for heat and mass transfer in turbulent pipe and channel flow, International Chemical Engineering 16, (1976), pp. 359-368

[14] O. García-Valladares, C.D. Pérez-Segarra, A. Oliva: Numerical simulation of capillary tube expansion devices behavior with pure and mixed refrigerants considering meta-stable region, Pergamon, Applied Thermal Engineering 22, (2002), pp. 173-182

[15] T. Yilmaz, S. Ünal.: General equation for the design of capillary tubes, ASME Journal of Fluids Engineering 118(2), (1996), pp. 150-154

[16] C. Zhang, G. Ding: Approximate analytic solution of adiabatic capillary tube, Elsevier, International Journal of Refrigeration 27, (2004), pp. 17-24

[17] H. Koizumi, K. Yokoyama: Characteristics of refrigerant flow in a capillary tube, ASHRAE Transaction 86, (1980), pp. 19-27

[18] R. Y. Li, S. Lin, Z.H. Chen: Numerical modeling of thermodynamic non-equilibrium flow of refrigerant through capillary tubes, ASHRAE Transactions 96 (1), (1990), pp. 542-549

[19] C. Melo, R.T.S. Ferreira, C. Boabaid Neto, J.M. Concalves: An experimental analysis of adiabatic capillary tubes, Pergamon, Applied Thermal Engineering 19, (1999), pp. 669-684

[20] REFPROP (2000), Thermodynamic properties of refrigerants and refrigerant mixtures, version 6.0, Gaithersburg, M. D. National Institute of Standards and Technology

[21] V. Vacek, G. Hallewell and S. Lindsay: Velocity of sound measurements in gaseous per-fluorocarbons and their mixtures, Fluid Phase Equilibria 185 (2001), 305-314

[22] V. Vacek , G. Hallewell , S. Ilie and S. Lindsay, Perfluorocarbons and their use in Cooling Systems for Semiconductor Particle Detectors, Fluid Phase Equilibria 174 (2000), 191-201