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Homogeneous versus separated two phase flow models: Adiabatic capillary tube flow in a transcritical CO_2 heat pump

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Abstract

A comparative study of flow characteristics of an adiabatic capillary tube in a transcritical CO_2 heat pump system have been investigated employing separated and homogeneous two phase flow models. Separated flow model is employed considering the annular flow pattern. The models are based on fundamental equations of mass, momentum and energy which are solved simultaneously. Two friction factor empirical correlations (Churchill, Lin et al.) and McAdams viscosity model are used. Chisholm correlation is used to calculate slip ratio while void fraction is calculated based on Premoli correlation. Sub-critical and super-critical thermodynamic and transport properties of CO_2 are calculated employing a precision in-house property code.

The results indicate that both two phase flow models predict reasonably well. Discrepancy between the separated flow model and homogeneous flow model has a maximum about 8 to 11%. Void fraction is influenced by vapour quality and follows the same trend as vapour quality. Liquid velocity and vapour velocity difference is relatively lower compared to R22.

Keywords: Capillary tube; Transcritical CO_2; Adiabatic; Separated flow model; Homogeneous flow model

1. Introduction

A recent release of the assessment report [1] of the Intergovernmental Panel on Climate Change (IPCC) has led to environmental groups calling on governments around the world to accelerate the phase-out of HCFCs in the wake of findings that action under the ozone layer treaty could do more to combat global warming than the Kyoto Protocol. While a renewed interest has already been observed in environmentally benign natural refrigerants, such findings are likely to accelerate a switch to these alternates. Carbon dioxide is a preferred choice over other natural refrigerants with its excellent thermophysical and heat transfer properties. In addition, its low price, easy availability, non-toxicity, non-flammability, low pressure ratio and high volumetric capacity make it a promising alternative [2]. The transcritical nature of carbon dioxide heat pump systems offer extensive possibilities in simultaneous heating and cooling applications due to the large temperature glide present in the gas cooler [3].

Capillary tube is one of the simplest kinds of expansion device employed in small capacity vapour compression systems. Nevertheless it is an essential and critical component whose flow characteristics directly affect system performance. Flow inside the capillary tube is complex in nature and numerous combinations of bore and length can be provided to obtain the desired flow restriction. Tube geometry (diameter and length) at a given operating condition is primary in the design of a capillary tube. In retrofitting existing systems for new refrigerants, therefore, it is vital and critical to select a capillary tube which is compatible with the system components. In a transcritical CO_2 refrigeration cycle, employing a capillary tube is quite different from subcritical systems; here pressure and temperature are two independent parameters unlike the conventional sub-critical cycle. The flow factor of the expansion valve determines the gas cooler pressure and hence it becomes interesting to investigate the flow characteristics in the capillary tube for CO_2 systems, where the flow is transcritical in nature.

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Nomenclature

A	cross-sectional area	m^2
a	acceleration	$m\,s^{-2}$
D	capillary tube diameter	m
f	friction factor (–)	
F	frictional force	N
G	mass flux	$kg\,m^{-2}\,s^{-1}$
h	specific enthalpy	$J\,kg^{-1}$
L	capillary tube length	m
$\dot{m}$	refrigerant mass flow rate	$kg\,s^{-1}$
p	pressure	Pa
q	heat flux	$W\,m^{-2}$
Re	Reynolds number (–)	
S	slip ratio (–)	
u	velocity	$m\,s^{-1}$
v	specific volume	$m^3\,kg^{-1}$
x	vapour quality (–)	

Greek symbols

α	void fraction (–)	
ε	internal surface roughness	mm
μ	dynamic viscosity	Pa s
ρ	density	$kg\,m^{-3}$
τ	shear stress	$N\,m^{-2}$
ϕ	two phase frictional multiplier (–)	

Subscripts

1–4	state points on capillary tube
c	capillary tube
g	saturated vapour
i	segment
k	phase (liquid/vapour)
l	saturated liquid
lg	liquid–vapour phase
lw	liquid–wall interface
sp	single phase
tp	two phase

Capillary tubes have been studied extensively by several researchers over the years. In majority of the studies, flow characterisation of adiabatic capillary tube is based on the homogeneous two phase flow model where both liquid and gas phase are assumed to flow with equal velocities. Most of these studies are with halocarbon and hydrocarbon refrigerants. Bansal and Rupasinghe [4] developed a homogeneous two phase flow model to study the performance of adiabatic capillary tubes for R134a. Jung et al. [5] modelled the pressure drop through a capillary tube to predict its size in residential air conditioners for R22 and its alternatives, R134a, R407C and R410A. Effects of sudden contraction at capillary tube inlet, degree of subcooling, friction factors and various viscosity models were also reviewed. Sami et al. [6] proposed a numerical model to predict the capillary behaviour for R22 alternatives such as R410A, R410B and R407C under different flow regimes. Trisaksri and Wongwises [7] developed a new correlation based on simulation studies for refrigerants R12, R22, R134a, R407C and R410A considering homogeneous flow to predict the tube size. Choi et al. [8] presented a generalised correlation for calculating refrigerant mass flow rate in an adiabatic capillary tube based on experimental data for R22, R290, and R407C. Kritsadathikarn et al. [9] presented a numerical study on the local pressure distribution for refrigerants R12, R134a, R409A and R409B. Wongwises et al. [10] compared the flow characteristics of many pairs of refrigerants flowing through adiabatic capillary tubes employing a homogeneous two phase flow model. Theoretical studies on flow characteristics of an adiabatic capillary tube employing hydrocarbons and their blends as refrigerants have been reported as well [11,12].

Metastable flow is a phenomenon where thermodynamic equilibrium does not prevail in a system. In a capillary tube flow, there is existence of metastable region since the flashpoint does not occur at the location where the pressure is equal to the saturation pressure but is delayed. Several authors [13–18] investigated the flow behaviour of an adiabatic capillary tube including the metastable region. Recently Garcia-Valladares [19] carried out transient one dimensional analysis of a capillary tube suction line heat exchanger considering the metastable region. Lately, artificial neural network technique was employed to predict the mass flow rate of refrigerant through an adiabatic capillary tube based on the homogeneous flow model [20,21].

Lin et al. [22] investigated, experimentally and theoretically, local friction pressure drop of two-phase flow during vaporisation of R12. Gu et al. [23] analysed and modelled an adiabatic capillary tube for the azeotropic R407C. Wongwises and Pirompak [24] also proposed an adiabatic capillary tube model to predict the flow characteristics of refrigerant mixtures. Melo et al. [25] investigated experimentally the effects of condensing pressure, size of adiabatic capillary tube, subcooling and choice of refrigerant, namely, R12, R134a and R600A on mass flow rate.

Accuracy of the homogeneous model depends on the physical properties (density and viscosity) of the refrigerants and the tube diameter. During the two phase flow in a capillary tube, slip may exist between the phases due to velocity differences of liquid and vapour phases where vapour phase tends to flow at higher velocity than the liquid phase; separated two phase flow model is more appropriate in these situations. Liang and Wong [26] presented a two phase drift flux model to simulate the flow of R134a refrigerant in the capillary tube. Flow characteristics such as pressure distribution, void fraction, dryness fraction, phase velocity were presented. Wong and Ooi [27] presented a comparative study of the capillary tube flow characterisation employing homogeneous and separated two phase flow model for R12. Wongwises and Chan [28] reported effects of various friction pressure gradient and slip ratio correlations on the prediction of a separated flow model for simulating adiabatic capillary tubes employing R12 and R22 refrigerants. Wongwises and Suchatawut [29] presented the refrigerant flow characteristics in an adiabatic capillary tube including metastable region employing separated flow model with simplified annular flow pattern for refrigerants R12 and R22.

It is observed that majority of such studies have concentrated on the HFCs, hydrocarbon refrigerants and their mixtures. Relatively, much less information is currently available in the open literature on two-phase flow characteristics of CO_2 in a transcritical capillary tube. It must be noted that medium to large CO_2 systems would require expansion valves so the optimum gas cooler pressure can be set by controlling the valve. However, smaller systems could still employ a capillary tube for simplicity and inexpensiveness sacrificing pressure control to attain optimum discharge pressure. Recently Agrawal and Bhattacharyya [30,31] investigated the flow characteristics of an adiabatic and non-adiabatic capillary tube in a CO_2 transcritical heat pump cycle employing homogeneous two phase model, including tube geometry effects on system performance. Garcia-Valladares [32] numerically simulated the short tube orifice flow of transcritical carbon dioxide based on a one dimensional finite volume formulation under transient condition. Choking phenomenon of the flow through short tube orifice was also investigated. Chen and Gu [33] developed a non-adiabatic homogeneous capillary tube model for the transcritical CO_2 cycle. They proposed a new transcritical refrigeration cycle combining the characteristics of heat transfer and expansion into one capillary tube by assembling the capillary tube in an accumulator or suction line. Parametric study for cooling pressure, evaporating temperature, ambient heat transfer coefficient and capillary size was presented. Lately Madsen et al. [34] investigated an adiabatic capillary tube in a transcritical CO_2 refrigeration system employing homogeneous flow model considering isenthalpic expansion. Flow characterisation of an adiabatic capillary tube in a transcritical CO_2 cycle employing a separated flow two phase model has not been reported yet.

The objective of this paper is to present a comparative study of an adiabatic capillary tube flow in a transcritical heat pump cycle using both homogeneous and separated two phase flow models. Longitudinal variation of void fraction is presented to obtain a better insight into the flow phenomenon.

2. Mathematical model

The capillary tube can be divided into three distinct flow regions, namely, supercritical flow region 1–2, transcritical flow region 2–3 and the subcritical flow region 3–4 as shown in Fig. 1. State '2' lies on the critical temperature line (Fig. 2). Therefore, in region 2–3, the fluid is considered as a subcooled liquid. In the supercritical and transcritical single phase region, temperature does not remain constant, unlike subcritical refrigeration cycles, due to the unique shape of isotherms in the CO_2 cycle. In the subcooled region as the flow progresses, the pressure continues to decrease due to presence of friction; temperature also decreases due to the unique shape of the isotherms, unlike the subcritical systems where temperature remains constant

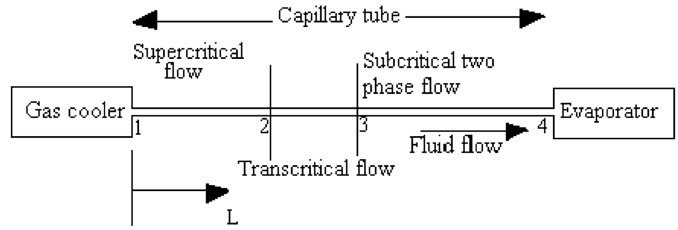

Fig. 1. Adiabatic capillary tube showing different flow regions.

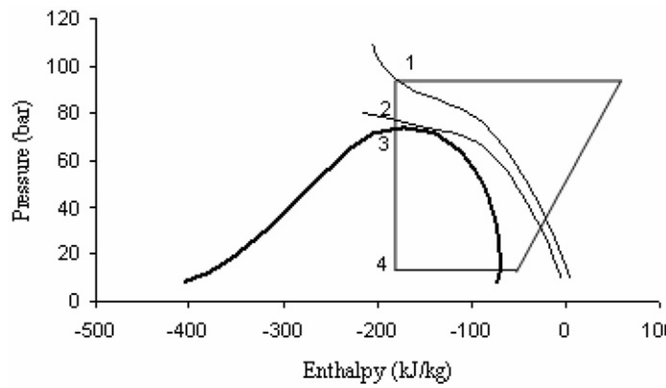

Fig. 2. Representation of corresponding capillary tube flow regions on P-h diagram of carbon dioxide transcritical cycle.

with pressure in the subcooled region. Consequently, probability of occurrence of metastable liquid region is less.

Total tube length is expressed as:

$$L = L_{sup} + L_{subliq} + L_{tp}$$

The expansion process from point 1 to 4 is shown in Fig. 2 on the pressure-enthalpy cycle plot.

The capillary tube flow model is developed employing the following assumptions:

- Straight horizontal tube with constant inner diameter and roughness.
- One-dimensional steady flow through the tube.
- Thermodynamic equilibrium prevails in the system (i.e. no metastable phenomenon is present).
- Refrigerant is free of oil.
- Flow through the tube is fully developed turbulent flow.
- Entrance losses are negligible. Since there is a large reduction of pressure from inlet to exit of the capillary in transcritical systems, entrance loss is negligible as a fraction of the total and this was verified by using the available correlations.

The model is based on fundamental equations of conservation of mass, momentum and energy and it incorporates variation in property values. This latter feature is essential for simulation of transcritical CO_2 system as property variation is extremely large in the neighbourhood of the critical point. The entire flow domain involves single phase as well as two phase fluid and hence is discretised into a number of longitudinal elements to enable the sharp changes in CO_2 property to be captured in the analysis (Fig. 3).

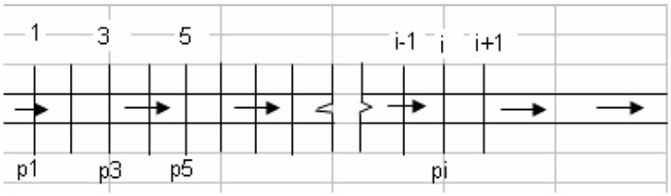

Fig. 3. Longitudinal discretisation for the capillary tube.

2.1. Single-phase flow region

The single-phase flow region includes supercritical and transcritical zone where refrigerant is in gas and liquid state, respectively. The conservation of mass for steady flow in an element of fluid yields:

$$\mathrm{d}\left(\frac{A_c u}{v}\right)=0 \tag{1}$$

For steady adiabatic flow with no external work and neglecting the elevation difference, the energy conservation equation reduces to:

$$\mathrm{d}h+\frac{G^2}{2}\mathrm{d}v^2=0 \tag{2}$$

From the conservation of momentum equation, the difference in forces applied to the element of fluid due to drag and pressure difference on opposite ends of the element should be equal to that needed to accelerate the fluid yielding:

$$\frac{\mathrm{d}p}{\mathrm{d}L}=-G^2\left(f_{\mathrm{sp}}\frac{v}{2D}+\frac{\mathrm{d}v}{\mathrm{d}L}\right) \tag{3}$$

Single-phase friction factor, f_{sp}, is calculated from Churchill correlation [35]:

$$f_{\mathrm{sp}}=8\left[\left(\frac{8}{Re_{\mathrm{sp}}}\right)^{12}+\left(A^{16}+B^{16}\right)^{-3/2}\right]^{1/12} \tag{4}$$

where

$$A=2.457\ln\frac{1}{(7/Re_{\mathrm{sp}})^{0.9}+0.27\varepsilon/D},$$

$$B=\frac{37530}{Re_{\mathrm{sp}}},\qquad Re_{\mathrm{sp}}=\frac{GD}{\mu}$$

2.2. Two phase flow region

In the two phase flow region, the liquid flashes into vapour due to reduction in pressure as the flow progresses in the adiabatic capillary tube which further intensifies due to acceleration of the vapour. The two phase pressure drop is a combined result of the tube wall friction and the fluid acceleration, as shown below:

$$\frac{\mathrm{d}p}{\mathrm{d}L}=\left(\frac{\mathrm{d}p}{\mathrm{d}L}F\right)+\left(\frac{\mathrm{d}p}{\mathrm{d}L}a\right) \tag{5}$$

It may be noted that in a transcritical CO_2 cycle, the pressure drop in single phase is also sum of the friction and fluid acceleration as the temperature does not remain constant in single phase. However, velocity of the fluid is not as high as in case of two phase flow. This unique feature of transcritical CO_2 flow adds to the complexity in simulating capillary tube flow.

2.3. Homogeneous flow model

Principles of mass, energy, and momentum conservation are employed to a discretized element of the capillary tube. The conservation of mass and energy for steady flow in an element of fluid follows the single-phase regime model given by Eqs. (1) and (2).

Two phase enthalpy and specific volume are expressed as:

$$h_i=h_{l,i}+x_i h_{\mathrm{lg},i},\qquad v_i=v_{l,i}+x_i v_{\mathrm{lg},i} \tag{6}$$

Further, the energy equation can be written as:

$$\left(\frac{G^2}{2}v_{\mathrm{lg},i}^2\right)x_{i+1}^2+\left(h_{\mathrm{lg},i+1}+G^2 v_{l,i+1}v_{\mathrm{lg},i+1}\right)x_{i+1}$$
$$+\left\{h_{l,i+1}-h_i+\frac{G^2}{2}\left(v_{l,i+1}^2-v_i^2\right)\right\}=0 \tag{7}$$

This quadratic equation can be solved to calculatex_{i+1}.

Conservation of momentum can be written in the same manner following the single-phase region model, as expressed in Eq. (3), and the differential length expression for the capillary tube is obtained similarly:

$$\mathrm{d}L=\frac{2D}{f_{\mathrm{tp}}}\left\{\frac{\mathrm{d}\rho}{\rho}-\frac{\rho}{G^2}\mathrm{d}p\right\} \tag{8}$$

Lin correlation [22] is used to calculate the two phase friction factor, given as:

$$f_{\mathrm{tp}}=\phi_{\mathrm{tp}}f_{\mathrm{sp}}\left(\frac{v_{\mathrm{sp}}}{v_{\mathrm{tp}}}\right) \tag{9}$$

where

$$\phi_{\mathrm{tp}}=\left[\frac{(8/Re_{\mathrm{tp}})^{12}+(A_{\mathrm{tp}}^{16}+B_{\mathrm{tp}}^{16})^{-3/2}}{(8/Re_{\mathrm{sp}})^{12}+(A_{\mathrm{sp}}^{16}+B_{\mathrm{sp}}^{16})^{-3/2}}\right]^{1/12}$$
$$\times\left[1+x\left(\frac{v_g}{v_l}-1\right)\right] \tag{10}$$

having

$$A=2.457\ln\frac{1}{(7/Re_{\mathrm{tp}})^{0.9}+0.27\varepsilon/D}$$

$$B=\frac{37530}{Re_{\mathrm{tp}}},\qquad Re_{\mathrm{tp}}=\frac{GD}{\mu_{\mathrm{tp}}}$$

McAdams [36] correlation, shown below, is employed to estimate two phase viscosity:

$$\frac{1}{\mu_{\mathrm{tp}}}=\frac{(1-x)}{\mu_l}+\frac{x}{\mu_g} \tag{11}$$

2.4. Separated flow model

The separated flow model considers the phases to be artificially segregated into two streams; one of liquid and one of vapour. Slip occurs between the two phases since vapour phase tends to flow with higher velocity. From a consideration of the various flow patterns it is observed that this model is most valid for annular flow pattern in which only liquid film with uniform film thickness wets the tube wall [37]. Fig. 4 depicts the annular flow pattern in which the thin liquid film is in contact with

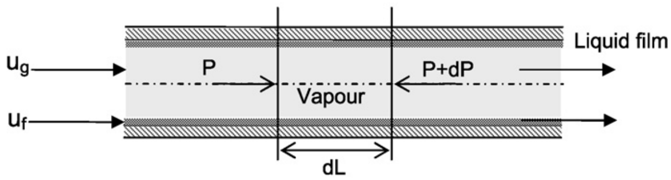

Fig. 4. Schematic diagram of annular flow pattern with control volume.

the tube wall while the vapour moves in the core zone of the tube. The slip occurs at the interface of the two phases while the frictional force exists between the liquid film and tube wall.

The conservation of mass for steady state two phase gas/liquid flow in a constant area channel is expressed by:

$$\frac{\mathrm{d}}{\mathrm{d}L}(A_g\rho_g u_g)+\frac{\mathrm{d}}{\mathrm{d}L}(A_l\rho_l u_l)=0 \tag{12}$$

Sum of the forces (pressure forces and shear forces at wall and interface) is equal to rate of change in momentum for each phase. Hence, for a steady state two-phase gas/liquid flow in a constant area channel, the momentum equation yields:

$$A_l\,\mathrm{d}p-\tau_{\mathrm{lw}}(\pi D\,\mathrm{d}L)+\tau_{\mathrm{lg}}(\pi D\,\mathrm{d}L)+u_l\frac{\mathrm{d}\dot m_l}{\mathrm{d}L}\delta L=\dot m_l\,\mathrm{d}u_l \tag{13}$$

$$-A_g\,\mathrm{d}p+\tau_{gl}(\pi D\,\mathrm{d}L)+u_g\frac{\mathrm{d}\dot m_g}{\mathrm{d}L}\delta L=\dot m_g\,\mathrm{d}u_g \tag{14}$$

At the interface of the liquid and gas, the momentum conservation yields:

$$\tau_{\mathrm{lg}}(\pi D\,\mathrm{d}L)+u_l\frac{\mathrm{d}\dot m_l}{\mathrm{d}L}\delta L=\tau_{gl}(\pi D\,\mathrm{d}L)+u_g\frac{\mathrm{d}\dot m_g}{\mathrm{d}L}\delta L \tag{15}$$

Combining Eqs. (13)–(15), the momentum equation can be written as:

$$-(A_l+A_g)\frac{\mathrm{d}p}{\mathrm{d}L}-\tau_{\mathrm{lw}}(\pi D)=\frac{\mathrm{d}}{\mathrm{d}L}(\dot m_l u_l+\dot m_g u_g) \tag{16}$$

In two phase flow, the void fraction α and mass flux G are defined as: $\alpha=A_g/A_c$ and $G=\dot m/A_c$. Hence the phase velocity and mass flow rate are given by:

$$u_g=\frac{Gx}{\rho_g\alpha},\qquad u_l=\frac{G(1-x)}{\rho_l(1-\alpha)},$$
$$\dot m_g=GA_c x,\qquad \dot m_l=GA_c(1-x)$$

After rearranging and substituting, Eq. (16) yields:

$$-\left(\frac{\mathrm{d}p}{\mathrm{d}L}\right)=\frac{\tau_{\mathrm{lw}}(\pi D)}{A_c}+G^2\frac{\mathrm{d}}{\mathrm{d}L}\left[\frac{x^2v_g}{\alpha}+\frac{(1-x)^2v_l}{(1-\alpha)}\right] \tag{17}$$

where τ_{lw} is the wall shear stress, defined as: $\tau_{\mathrm{lw}}=f_{\mathrm{lw}}\rho_l u_l^2/8$.

Accordingly, Eq. (17) can be written as:

$$-\left(\frac{\mathrm{d}p}{\mathrm{d}L}\right)=\frac{f_{\mathrm{lw}}\rho_l u_l^2}{2D}+G^2\frac{\mathrm{d}}{\mathrm{d}L}\left[\frac{x^2v_g}{\alpha}+\frac{(1-x)^2v_l}{(1-\alpha)}\right] \tag{18}$$

where the first term on RHS is the frictional pressure drop and the second term is the acceleration pressure drop.

Chisholm correlation [38] is employed to calculate slip ratio between the two phases:

$$S=\left[1-x\left(1-\frac{\rho_l}{\rho_g}\right)\right]^{1/2}$$

The void fraction is calculated based on the Premoli [39] correlation:

$$\alpha=\frac{xv_g}{(1-x)v_lS+xv_g}$$

The void fraction is a function of x and p expressed as:

$$\alpha=f\big[x(L),p(L)\big] \tag{19}$$

Hence, acceleration pressure drop in Eq. (18) can be expressed as:

$$\begin{aligned}\frac{\mathrm{d}}{\mathrm{d}L}&\left[\frac{x^2v_g}{\alpha}+\frac{(1-x)^2v_l}{(1-\alpha)}\right]\\&=\frac{\mathrm{d}x}{\mathrm{d}L}\left[\left\{\frac{2xv_g}{\alpha}-\frac{2(1-x)v_l}{(1-\alpha)}\right\}\right.\\&\quad\left.+\left(\frac{\partial\alpha}{\partial x}\right)_p\left\{\frac{(1-x)^2v_l}{(1-\alpha)^2}-\frac{x^2v_g}{\alpha^2}\right\}\right]\\&\quad+\frac{\mathrm{d}p}{\mathrm{d}L}\left[\frac{x^2}{\alpha}\frac{\mathrm{d}v_g}{\mathrm{d}p}+\left(\frac{\partial\alpha}{\partial p}\right)_x\left\{\frac{(1-x)^2v_l}{(1-\alpha)^2}-\frac{x^2v_g}{\alpha^2}\right\}\right]\end{aligned} \tag{20}$$

where

$$\left(\frac{\partial\alpha}{\partial x}\right)_p=\frac{C}{D} \tag{21}$$

$$\begin{aligned}C&=\left[(1-x)v_l\left\{1-x\left(1-\frac{\rho_l}{\rho_g}\right)\right\}^{1/2}+xv_g\right]v_g\\&\quad+xv_g\left[v_l\left\{1-x\left(1-\frac{\rho_l}{\rho_g}\right)\right\}^{1/2}\right.\\&\quad\left.+\left\{\frac{(1-x)v_l(1-\rho_l/\rho_g)}{2\sqrt{1-x(1-\rho_l/\rho_g)}}\right\}-v_g\right]\end{aligned}$$

$$D=\left[(1-x)v_l\left\{1-x\left(1-\frac{\rho_l}{\rho_g}\right)\right\}^{1/2}+xv_g\right]^2\ \text{and}$$

$$\left(\frac{\partial\alpha}{\partial p}\right)_x=\frac{\alpha_{i+1,x}-\alpha_{i,x}}{\Delta p} \tag{22}$$

Energy equation in separated two phase flow can be obtained by equating the rate of increase of total energy for a phase k, interfacial energy transfer which includes the energy transfer by virtue of the mass transfer together with work done on phase k.

Neglecting elevation difference and no external work and heat transfer, energy equation can be written for steady state adiabatic separated flow in a constant area channel, given as:

$$\begin{aligned}&\mathrm{d}\left[\dot m_g\left(h_g+\frac{u_g^2}{2}\right)\right]\\&\quad=q_{gl}\pi D\,\mathrm{d}L+u_g\tau_{gl}\pi D\,\mathrm{d}L+\frac{\mathrm{d}\dot m_g}{\mathrm{d}L}\left(h_g+\frac{u_g^2}{2}\right)\delta L\end{aligned} \tag{23}$$

$$\begin{aligned}&\mathrm{d}\left[\dot m_l\left(h_l+\frac{u_l^2}{2}\right)\right]\\&\quad=q_{\mathrm{lg}}\pi D\,\mathrm{d}L+q_{\mathrm{lw}}\pi D\,\mathrm{d}L+u_l\tau_{\mathrm{lg}}\pi D\,\mathrm{d}L\\&\qquad+\frac{\mathrm{d}\dot m_l}{\mathrm{d}L}\left(h_l+\frac{u_l^2}{2}\right)\delta L\end{aligned} \tag{24}$$

Energy balance at the interface of liquid/gas yields

$$q_{gl}\pi D\,\mathrm{d}L + u_g\tau_{gl}\pi D\,\mathrm{d}L + \frac{\mathrm{d}\dot{m}_g}{\mathrm{d}L}\left(h_g + \frac{u_g^2}{2}\right)\delta L$$
$$= q_{\mathrm{lg}}\pi D\,\mathrm{d}L + u_l\tau_{\mathrm{lg}}\pi D\,\mathrm{d}L + \frac{\mathrm{d}\dot{m}_l}{\mathrm{d}L}\left(h_l + \frac{u_l^2}{2}\right)\delta L \tag{25}$$

Neglecting heat transfer between the capillary tube wall and the liquid film q_{lw}, the energy equation can be written as:

$$\frac{\mathrm{d}}{\mathrm{d}L}(\dot{m}_g h_g + \dot{m}_l h_l) + \frac{\mathrm{d}}{\mathrm{d}L}\left(\dot{m}_g\frac{u_g^2}{2} + \dot{m}_l\frac{u_l^2}{2}\right) = 0 \tag{26}$$

which can be further simplified as:

$$x_i h_{g,i} + (1-x_i)h_{l,i} + \frac{x_i u_{g,i}^2}{2} + (1-x_i)\frac{u_{l,i}^2}{2}$$
$$= x_{i+1}h_{g,i+1} + (1-x_{i+1})h_{l,i+1} + \frac{x_{i+1}u_{g,i+1}^2}{2}$$
$$+ (1-x_{i+1})\frac{u_{l,i+1}^2}{2} \tag{27}$$

Eq. (27) is solved to calculate x_{i+1}.

3. Results and discussion

A comparative study of flow characteristics of an adiabatic capillary tube, based on both homogeneous and separated two phase model is reported here for a gas cooler pressure and temperature of 100 bar and 313 K, respectively. The evaporator temperature is taken as 288 K while the capillary tube diameter and internal surface roughness are taken as 1.0 and 0.0015 mm, respectively. Refrigerant mass flow rate is fixed at 0.01 kg/s. A simulation code is employed to solve the discretized governing equations numerically using finite difference approximation. Employing the new equation of state for CO_2 and transport property correlations available in the literature, a separate property code CO2PROP, employing a technique based on derivatives of Helmholtz free energy function using efficient iterative procedures, has been developed to calculate sub-critical and super-critical thermodynamic and transport properties of CO_2 [3]. Thermophysical properties of R22 are calculated through REFPROP (ver. 6.01).

To the best of the authors' knowledge, no results from theoretical studies based on the separated flow model of adiabatic capillary tube flow with CO_2 refrigerant in the transcritical cycle is available in the open literature; results from the present simulation model (separated two phase flow and for R22) are compared with the published results from Mikol [40] for the specified capillary tube and specified conditions for refrigerant R22 as shown in Fig. 5. In the validation exercise, metastable liquid region is included based on Chen et al. [15] correlation. It is observed that the agreement is excellent except near exit of the capillary tube which could be attributed to the fact that choked flow conditions are not considered in the model. The present model exhibits a discrepancy of about 5%. Since the included metastable liquid region correlation is originally for R12, employing this correlation with R22 may also lead to some degree of error.

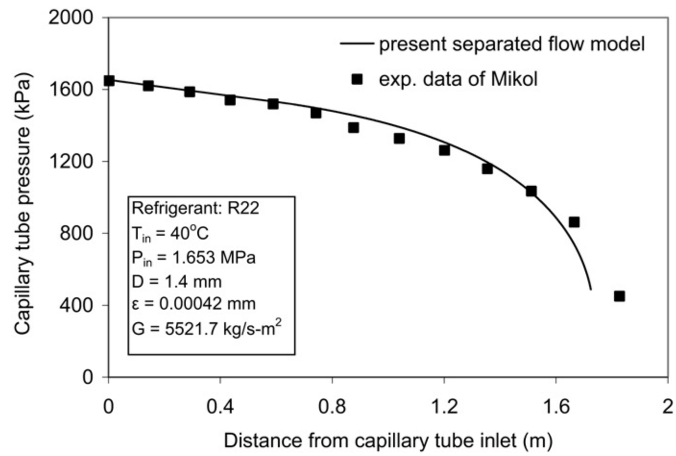

Fig. 5. Validation of the present model with the experimental results of Mikol [40].

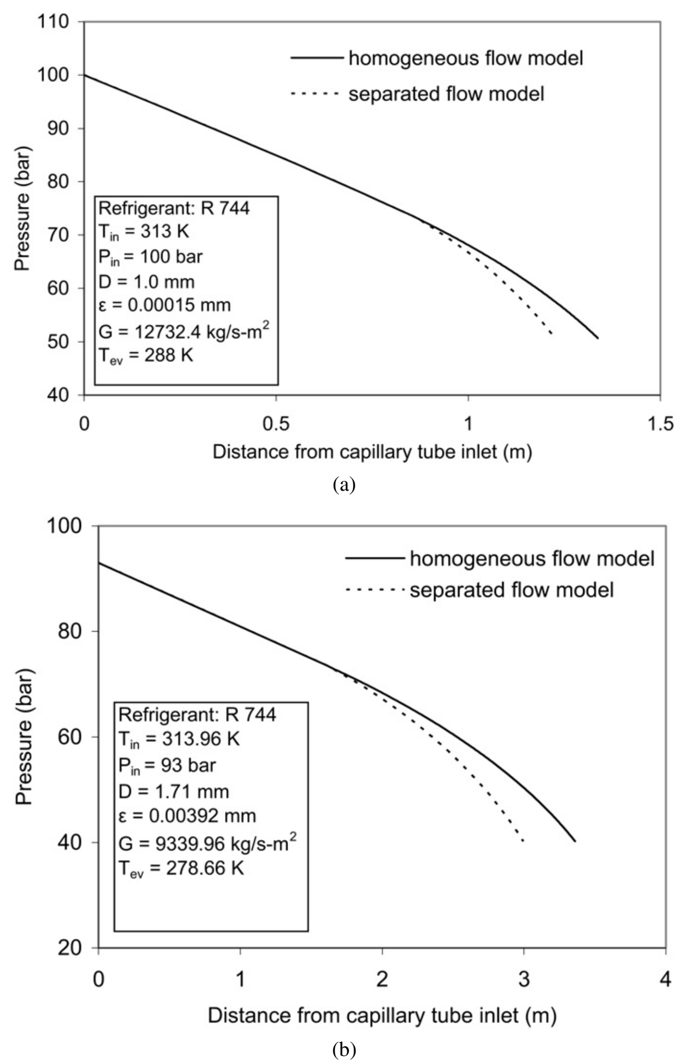

Fig. 6. Comparison of pressure variation along the capillary tube.

Figs. 6(a) and 6(b) show the pressure variation along the capillary tube considering homogeneous and separated flow model, respectively. The results reveal that both the two phase flow models yield reasonably good results. Discrepancy between the

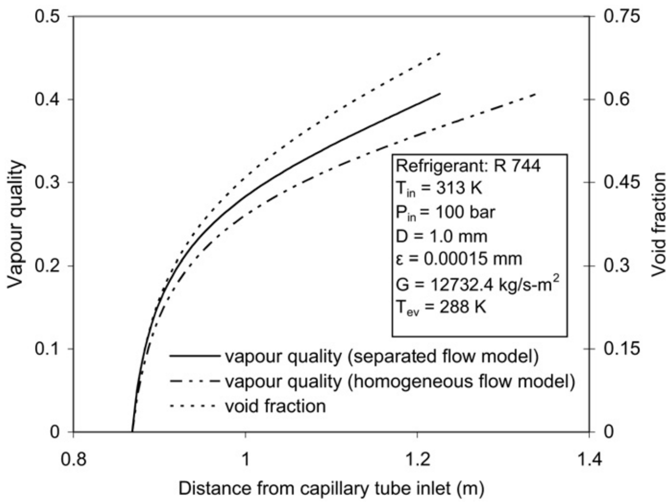

Fig. 7. Void fraction and vapour quality variation along the capillary tube.

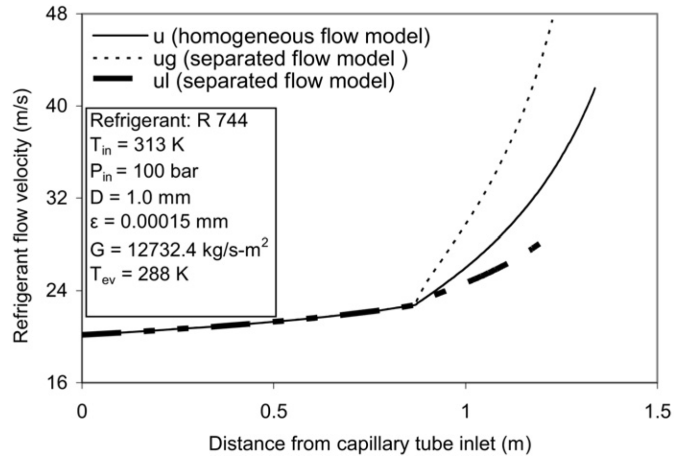

Fig. 8. Refrigerant flow velocity in a capillary tube for transcritical CO_2 cycle.

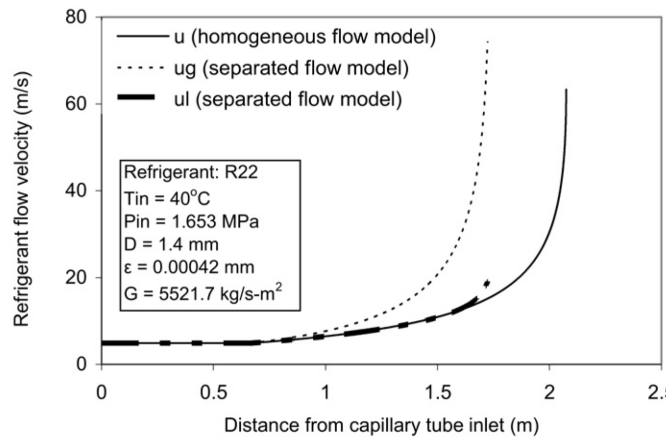

Fig. 9. Refrigerant flow velocity of R22 in a capillary tube for subcritical cycle.

separated flow model and homogeneous flow model is about 8 to 11%. This may be due to the relatively higher flow friction at higher vapour quality with separated flow (Fig. 7). As the capillary length increases, the pressure decreases fairly linearly up to the saturation point (i.e. for the single phase region). On inception of vaporisation, pressure drop increases rapidly and non-linearly.

Variation of vapour quality and void fraction along the tube length is exhibited in Fig. 7. Vapour quality and void fraction trends are almost similar i.e. at the onset of vaporisation, quality and void fraction both increase rapidly since the increments of length needed to drop the saturation pressure become progressively smaller. Up to nearly 20% of the two phase length, quality and void fraction are almost similar. This may be attributed to the fact that initially liquid percentage is less, compared to latter part and moreover CO_2 vapour density is relatively higher. It implies that void fraction is more influenced by vapour quality due to higher vapour density and also the correspondingly lower density ratio of liquid and vapour phases of CO_2 which is not there in case of conventional refrigerants [30]. Moreover, in a transcritical CO_2 cycle, initially vapour quality is high due to inception of vaporisation near the critical point where constant dryness fraction lines are in close proximity unlike in subcritical cycles where the inception of vaporisation is away from the critical point. Absolute value of vapour quality is relatively higher in case of the separated flow model, particularly towards the exit of the capillary tube (Fig. 7). Further, vapour quality increases at a faster rate during the initial part.

Figs. 8 and 9 exhibit variation in refrigerant velocity along the tube length for CO_2 and R22 refrigerant, respectively. The variation is quite modest up to the saturation point (i.e. in single phase zone) for CO_2 while velocity is constant in single phase (i.e. subcooled region) for R22. This can be attributed to the fact that for CO_2, due to the unique shape of isotherms in single phase region, temperature is not constant unlike the conventional refrigerant. Refrigerant velocity varies sharply in the two phase region due to presence of vapour phase, for both the refrigerants CO_2 as well as R22. However, owing to high vapour density of CO_2, acceleration of the fluid in two phase region is moderate compared to that for conventional refrigerants. It can be observed that in two phase region, R22 vapour velocity increases rapidly and hence liquid and vapour velocity difference is significantly large for R22; in case of CO_2, the liquid vapour density difference is relatively smaller along with a higher vapour quality. Homogeneous two phase velocity of refrigerant is much smaller then that of the vapour velocity for R22 while it is almost at mean value of vapour and liquid velocities in case of CO_2 (Figs. 8–9). Further, viscosity of CO_2 liquid is lower than traditional refrigerants; however, the vapour viscosity is comparable.

4. Conclusions

A comparative study of flow characteristics of an adiabatic capillary tube with CO_2 in a transcritical cycle employing homogeneous and separated two-phase flow model is presented here. Simulation is based on the annular flow pattern with separated two-phase flow model. In a CO_2 transcritical cycle, simulation of the capillary tube flow is different from other refrigerants. Vapour fraction is higher as the inception of vaporisation is near the critical point.

Variation of pressure along the capillary tube length is predicted to be almost similar by both the two phase flow models.

Although there is a small disagreement near the exit of the capillary, both the two phase flow models produce reasonably good prediction. Discrepancy between the separated flow model and homogeneous flow model is about 8% for the chosen conditions. Due to relatively more homogeneous two-phase flow of CO_2, void fraction follows the same trend as vapour quality. Vapour quality is relatively higher due to inception of vaporisation near the critical point. Liquid velocity and vapour velocity difference is relatively lower due to higher vapour density and low density ratio (liquid to vapour). Results with both the flow models are almost similar for an adiabatic capillary tube flow in the CO_2 transcritical heat pump cycle. However, homogeneous model is always preferred for its simplicity to simulate the adiabatic capillary tube flow with CO_2. Nevertheless these predictions must all be verified with extensive test data work on which is currently underway.

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