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Performance evaluation of a non-adiabatic capillary tube in a transcritical CO_2 heat pump cycle

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Abstract

A non-adiabatic capillary tube in a transcritical CO_2 heat pump cycle has been simulated to investigate the effect of parameters such as gas cooler and evaporator temperature, capillary tube diameter and heat exchanger length on various performance indicators. The homogeneous flow model is employed to simulate two-phase flow in the non-adiabatic capillary tube. Fundamental equations of mass, energy and momentum are solved simultaneously through an iterative process. Single and two-phase heat transfer coefficients are calculated by employing appropriate empirical correlations. Subcritical and supercritical thermodynamic and transport properties of CO_2 are calculated employing an in-house precision property code.

Lowering evaporator temperature is found to be more effective for heat transfer from the capillary tube compared to the gas cooler temperature. Heat transfer rate variation with respect to gas cooler temperature in case of CO_2 is distinctly different compared to conventional refrigerants due to its transcritical nature and is influenced by initial quality, mass flow rate of the refrigerant and the prevailing temperature difference at the gas cooler. Increase in gas cooler temperature causes the heat transfer rate to first increase and then to decrease. Lowering evaporator and gas cooler temperature increases the cooling capacity. Throttling effect decreases rapidly as internal tube diameter becomes larger leading to higher mass flow rate of the refrigerant. Shorter inlet adiabatic capillary length with larger heat exchanger length is better for heat transfer. This study is an attempt to allay the scepticism prevailing in the parlance of CO_2 based transcritical systems overemphasising the need for a throttle valve to control the optimum discharge pressure.
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Keywords: Capillary tube; CO_2; Non-adiabatic; Capillary tube-suction line heat exchanger

1. Introduction

Capillary tubes are a type of refrigerant flow control device and are widely accepted as an expansion device in small vapour compression refrigerating and air conditioning systems due to its simplicity, low initial cost and low starting torque of the compressor. Flow inside the capillary tube is a flashing process where it undergoes phase change and is complex in nature. It is a common practice to solder the capillary tube to the outer surface of the suction line (Fig. 1) to improve the system performance, and to avoid liquid entry into the compressor by superheating the suction vapour. Such a configuration of capillary tube and suction line form a counter-flow heat exchanger commonly known as capillary tube-suction line heat exchanger (CL-SLHX). Under such conditions, the capillary tube is considered non-adiabatic and has the unusual characteristic of flashing, while simultaneously being cooled by heat transfer to a wall where large frictional effects enhance flashing, while heat transfer to the suction line retards flashing. These complexities of the non-adiabatic two phase flow make it challenging to analyse.

Natural refrigerants have become the preferred choice to replace conventional refrigerants in view of their benign nature and CO_2 appears to be leading the pack. In addition to its environmental benefits, CO_2 has attractive thermo-physical and safety characteristics compared to the currently used refrigerants.

A capillary tube employing CO_2 as the refrigerant is altogether a different phenomenon because of the transcritical

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Nomenclature

A	inside cross-sectional area of capillary tube	m^2
c_p	specific heat	$kJ\,kg^{-1}\,K^{-1}$
D	capillary tube diameter	mm
f	friction factor	
G	mass flux	$kg\,m^{-2}\,s^{-1}$
h	specific enthalpy	$kJ\,kg^{-1}$
k	thermal conductivity	$kW\,m^{-1}\,K^{-1}$
L	capillary tube length	m
L_c	capillary tube fin length	m
L_{hx}	capillary tube-suction line heat exchanger length	m
L_{in}	upstream adiabatic capillary length	m
L_{out}	downstream adiabatic capillary length	m
L_s	suction line fin length	m
$\dot{m}$	mass flow rate	$kg\,s^{-1}$
m	fin constant	
Nu	Nusselt number	
p	pressure	bar
Pr	Prandtl number	
q	heat transfer rate	kW
Re	Reynolds number	
T	temperature	K
t	fin thickness	mm
UA	overall heat transfer conductance	$kW\,K^{-1}$
V	velocity	$m\,s^{-1}$
v	specific volume	$m^3\,kg^{-1}$
x	dryness fraction	
X_{tt}	Lockhart–Martinelli factor	

Subscripts

1–4	capillary tube state points
c	capillary
ev	evaporator
gc	gas cooler
hx	heat exchanger
i	internal
l	saturated liquid
o	external
s	suction line
sp	single phase
tp	two phase

Greek symbols

ρ_l	density of saturated liquid	$kg\,m^{-3}$
ρ_v	density of saturated vapour	$kg\,m^{-3}$
μ_l	dynamic viscosity of saturated liquid	Pa s
μ_v	dynamic viscosity of saturated vapour	Pa s
η	fin efficiency	
ε	internal surface roughness	mm
α	heat transfer coefficient	$W\,m^{-2}\,K^{-1}$

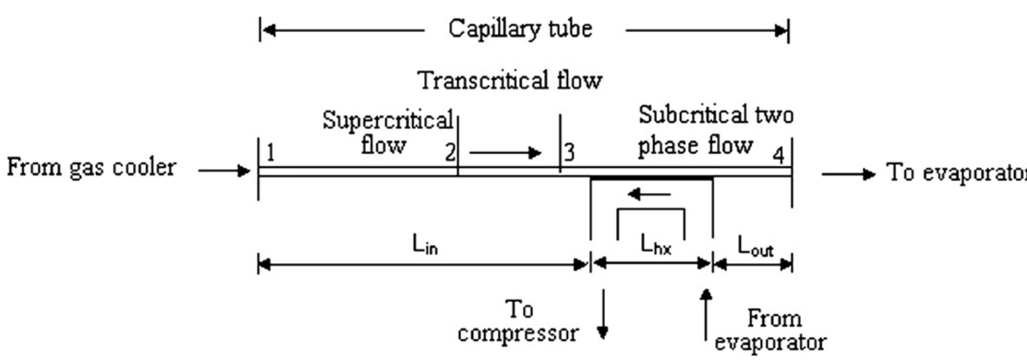

Fig. 1. Schematic diagram of the capillary tube-suction line heat exchanger.

nature of the process. The transcritical heat pump cycle has two independent parameters in pressure and temperature, while conventional sub-critical cycles have only one in temperature. Consequently, unlike the subcritical cycle, pressure can be independently controlled in a transcritical heat pump cycle to optimise the system performance. Selection of a capillary tube must look to optimise the performance of a transcritical heat pump system. It must be noted that medium to large CO_2 systems would essentially require expansion valves so that the optimum gas cooler pressure can be set by controlling the valve. There exists a good amount of scepticism regarding use of capillary tubes in CO_2 transcritical systems with respect to this pressure control issue. However, smaller systems could still employ a capillary tube for simplicity and inexpensiveness sacrificing pressure control to attain optimum discharge pressure. It would be interesting to note whether an optimisation can be carried out in sizing the capillary tube so that the optimised gas cooler pressure is also attained in the process.

Being an influential component of a refrigeration system, capillary tubes have been studied widely over many years. Numerous researchers have studied the design and flow behaviour of adiabatic capillary tubes with halocarbon and hydrocarbon refrigerants [1–7]; however, such studies for a transcritical CO_2 system is scant. Madsen et al. [8] investigated an adiabatic capillary tube in a transcritical CO_2 refrigeration system. They concluded that a system operates closer to optimum conditions with capillary tube than a pre-fixed high pressure. Recently

Agrawal and Bhattacharyya [9] presented the flow characteristics of an adiabatic capillary tube flow in the CO_2 transcritical heat pump cycle. Occurrence of choking, cause and effects, were also discussed in their analysis. Comparatively, flow behaviour in non-adiabatic tubes received much less attention in the literature.

Various aspects of the non-adiabatic capillary tube with halocarbon and hydrocarbon refrigerants have been presented by several researchers [10–15]. Pate and Tree [10] developed a two-phase flow model employing a linear quality profile. They concluded that the flow behaviour with linear quality assumption is better than those obtained from a non-linear profile. Peixoto and Bullard [11] theoretically studied the lateral and concentric tube CL-SLHX. Sinpiboon and Wongwises [12] theoretically investigated the flow characteristics in a non-adiabatic capillary tubes for three different heat exchanging regions. Numerical results reveal that the performance of non-adiabatic capillary tubes depends on the position of the starting point of the heat exchanging process. Xu and Bansal [13] developed a homogeneous model to simulate the refrigerant flow behaviour in a non-adiabatic capillary tube. It is concluded that the flow behaviour depends on relative influence of heat transfer and frictional effects. Bansal and Xu [14] presented a parametric study of refrigerant flow behaviour in a non-adiabatic capillary tube. Effects of various geometric and thermodynamic parameters on the refrigerant flow characteristics were reported. Effects and possibilities of re-condensation within the heat exchanger were also presented. Recently Yang and Bansal [15] presented performance of a CL-SLHX with HFC-134a and HC-600a refrigerants with respect to various thermodynamic and geometric parameters.

Although bulk of the capillary tube studies were engaged to subcritical refrigeration cycles with HFCs and hydrocarbon refrigerants, very little effort has been spared to understand the flow characteristics of adiabatic and non-adiabatic capillary tube in a transcritical CO_2 heat pump cycle. Recently Chen and Gu [16] presented parametric studies of a non-adiabatic capillary tube for the transcritical CO_2 cycle. A new transcritical refrigeration cycle was proposed, combining the characteristics of heat transfer and expansion into one capillary tube by assembling the capillary tube in an accumulator or suction line. Two phase flow through short tube orifice in a transcritical CO_2 system was investigated by Chen et al. [17]. Based on test results, a correlation was developed to predict mass flow rate. Recently Garcia [18] numerically simulated the short tube orifice flow of transcritical carbon dioxide based on an one-dimensional finite volume formulation under transient condition. Choking phenomenon of the flow through short tube orifice was also investigated.

It is interesting and vital to study the flow behaviour of a non-adiabatic capillary tube in a CO_2 transcritical heat pump cycle under different geometric and thermodynamic boundary conditions to optimise the system. This study was targeted to obtain the basic understanding of the non-adiabatic capillary tube flow characteristics for transcritical CO_2 heat pump cycle under various operating conditions.

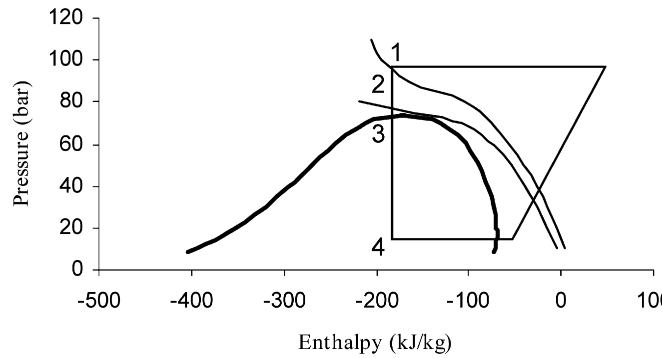

Fig. 2. Location of state points 1–4 in the transcritical cycle on p–h plane.

2. Mathematical modelling

Schematic representation of the non-adiabatic capillary tube discussed in the present study is shown in Fig. 1. The capillary flow in a transcritical CO_2 cycle passes through 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 Figs. 1 and 2. Unlike the conventional refrigerants where temperature lines are almost vertical in the subcooled region on the p–h plot, temperature lines for CO_2 are of ‘S’ shape in the supercritical and transcritical region (Fig. 2). Consequently, the modelling procedure for capillary tubes in CO_2 transcritical heat pump cycles is distinctly different from that undertaken for cycles based on conventional refrigerants. Suction line is attached to the capillary tube in the subcritical two-phase region. Consequently, initial length of the capillary L_{in} and post heat exchanging length L_{out} are adiabatic (Fig. 1). Governing equations and solution methodology are suitably selected and correlated as per the capillary flow conditions.

2.1. Assumptions

The present model is based on the following simplifying assumptions:

(i) Straight tube with constant inner diameter and roughness.
(ii) Homogeneous and one-dimensional steady flow through the tube.
(iii) Lateral contact of capillary tube-suction line heat exchanger.
(iv) Negligible thermal resistance at soldering.
(v) Negligible heat transfer to the surrounding.
(vi) Thermodynamic equilibrium (i.e. no meta-stable phenomenon).
(vii) Oil free refrigerant.
(viii) Fully developed turbulent flow.
(ix) Negligible entrance losses. Since there is large reduction of pressure from inlet to exit of the capillary, entrance losses are negligible and this was verified by using the available correlations.
(x) No suction superheat in the evaporator.

The homogeneous flow model is employed to simulate two-phase flow in the non-adiabatic capillary tube. Homogeneous

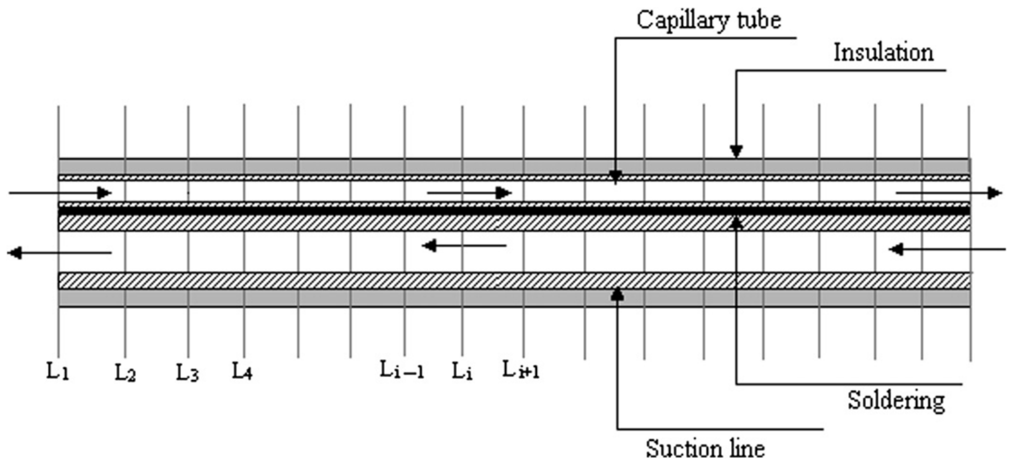

Fig. 3. Discretization in the capillary tube-suction line heat exchanger.

flow model is justified based on the fact that the low density of liquid CO_2 and the low ratio of liquid to vapour density contribute to make the two phase flow homogeneous. The model is formulated employing fundamental equations of conservation of mass, energy and momentum and it incorporates variation in property values since it is extremely large in the neighbourhood of the critical point. The entire flow domain, adiabatic as well non-adiabatic, is discretised into a number of longitudinal elements (Fig. 3) to enable the sharp changes in CO_2 property to be included appropriately in the analysis.

2.2. *Conservation of mass*

The conservation of mass for steady flow in an element of fluid for the capillary tube and the suction line is given by:

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

2.3. *Conservation of momentum*

Neglecting the force due to elevation, the momentum conservation equation of the capillary tube can be written as:

$$\frac{\mathrm{d}p}{\mathrm{d}L}=-f_{\mathrm{sp}}\frac{G^2 v}{2D}-G^2\frac{\mathrm{d}v}{\mathrm{d}L} \tag{2}$$

Churchill [19] and Lin [20] correlations are employed to calculate the single-phase and two-phase flow friction factor, f_{sp}& f_{tp} respectively. McAdams [21] model is used to calculate two phase viscosity, μ_{tp}.

2.4. *Conservation of energy*

2.4.1. *Adiabatic region*

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

$$\mathrm{d}h+\frac{G^2}{2}\,\mathrm{d}v^2=0 \tag{3}$$

2.4.2. *Capillary tube-suction heat exchanger*

Energy equation for the capillary tube fluid flow, with no external work and negligible elevation difference, is expressed as:

$$\mathrm{d}h+\frac{G^2}{2}\,\mathrm{d}v^2+\frac{\mathrm{d}q}{\dot{m}}=0 \tag{4}$$

For single phase flow (vapour phase) in the suction line, energy equation is given by:

$$\mathrm{d}q=\dot{m}c_{\mathrm{p}}\,\mathrm{d}T_{\mathrm{s}} \tag{5}$$

For each longitudinal element of the capillary tube suction line heat exchanger, dq is given by:

$$\mathrm{d}q=UA(T_{\mathrm{c}}-T_{\mathrm{s}}) \tag{6}$$

In Eq. (6), a simple arithmetic temperature difference is justified by considering a small incremental length of 1 mm in the numerical solution.

The capillary tube-suction line combination forms a twin tube heat exchanger (Fig. 1). Accordingly, overall heat transfer coefficient is calculated treating capillary tube and suction line as extended surfaces [22]. Therefore, overall heat transfer conductance is given by:

$$\frac{1}{UA}=\frac{1}{\eta_{\mathrm{c}}\alpha_{\mathrm{c}}A_{\mathrm{c}}}+\frac{1}{\eta_{\mathrm{s}}\alpha_{\mathrm{s}}A_{\mathrm{s}}} \tag{7}$$

where

$$A_{\mathrm{c}}=\pi D_{\mathrm{i,c}}\,\mathrm{d}L,\qquad A_{\mathrm{s}}=\pi D_{\mathrm{i,s}}\,\mathrm{d}L$$

$$\eta_{\mathrm{c}}=\frac{\tanh m_{\mathrm{c}}L_{\mathrm{c}}}{m_{\mathrm{c}}L_{\mathrm{c}}},\qquad \eta_{\mathrm{s}}=\frac{\tanh m_{\mathrm{s}}L_{\mathrm{s}}}{m_{\mathrm{s}}L_{\mathrm{s}}}$$

$$m_{\mathrm{c}}=\sqrt{\frac{\alpha_{\mathrm{c}}}{t_{\mathrm{c}}k_{\mathrm{c}}}},\qquad m_{\mathrm{s}}=\sqrt{\frac{\alpha_{\mathrm{s}}}{t_{\mathrm{s}}k_{\mathrm{s}}}}$$

$$L_{\mathrm{c}}=\frac{\pi D_{\mathrm{i,c}}}{2},\qquad L_{\mathrm{s}}=\frac{\pi D_{\mathrm{i,s}}}{2}$$

$$t_{\mathrm{c}}=\frac{(D_{\mathrm{o,c}}-D_{\mathrm{i,c}})}{2},\qquad t_{\mathrm{s}}=\frac{(D_{\mathrm{o,s}}-D_{\mathrm{i,s}})}{2}$$

Convective heat transfer coefficient for single phase flow is calculated employing Gnielinski [23] correlation given by:

$$Nu=\frac{(f/8)(Re-1000)Pr}{1.07+12.7(f/8)^{1/2}(Pr^{2/3}-1)} \tag{8}$$

Table 1
Parameters used in the analysis

Parameter	Value
Gas cooler pressure (p)	100 bar
Gas cooler exit temperature (T_{gc})	313 K
Evaporator temperature (T_{ev})	288 K
Capillary internal diameter ($D_{i,c}$)	1 mm
Capillary external diameter ($D_{o,c}$)	3.17 mm
Capillary tube internal surface roughness (ε)	0.0015 mm
Capillary tube length (L)	1.3 m
Suction line internal diameter ($D_{i,s}$)	9.35 mm
Suction line external diameter ($D_{o,s}$)	12 mm
Thermal conductivity of suction line and capillary (k)	15 W m^{-1} K^{-1}
Length of heat exchanger (L_{hx})	0.3 m
Inlet adiabatic length (L_{in})	0.9 m
Outer adiabatic length (L_{out})	0.1 m

Two phase heat transfer rate is estimated through Wattelet–Carlo [24] correlation and is given by:

$$\alpha_r = F.\alpha_l, \quad \alpha_l = 0.023\frac{k_l}{D_{i,c}}Re_l^{0.8}Pr_l^{0.4}$$

(Dittus–Boelter correlation) (9)

$F = 1 + 1.925X_{tt}^{-0.83}$ where X_{tt} is the Lockhart–Martinelli parameter given by:

$$X_{tt} = \left(\frac{1-x}{x}\right)^{0.9}\left(\frac{\rho_v}{\rho_l}\right)^{0.5}\left(\frac{\mu_l}{\mu_v}\right)^{0.1} \tag{10}$$

The baseline parameters used in the study are presented in Table 1.

3. Numerical solution

The homogeneous model is employed to simulate two-phase flow in the capillary tube-suction line heat exchanger to carry out the thermodynamic analysis of the heat pump cycle. For the given total length, capillary tube is simulated for mass flow rate. In each calculation, inlet adiabatic length is fixed at 0.9 m. Entire length of the heat exchanger is discretised into number of small elements, each 1 mm long. For each control volume, discretised momentum and energy equations are employed to estimate enthalpy and pressure at the outlet of the control volume. These governing equations are coupled equations, and hence are solved simultaneously by an iterative technique. Using the new equation of state for CO_2 and transport property correlations available in the literature, a separate 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 carbon dioxide [25]. Suction line calculation is based on the thermodynamic and transport properties at the outlet temperature of the control volume, guessed initially. Mass flow rate and outlet temperature of the suction line are guessed initially. The solution process continues till the length and temperature match the capillary length and evaporator temperature within tolerances. Finite difference approximation is employed to solve the discretised governing equations. Pressure is taken as a marching parameter for adiabatic section, while length is selected as the marching parameter for non-adiabatic section. Parameters are chosen in such a way that there is no choking at the exit of the capillary tube. Onset of negative entropy change is used as a criterion to detect the choked flow condition in the capillary tube flow [9].

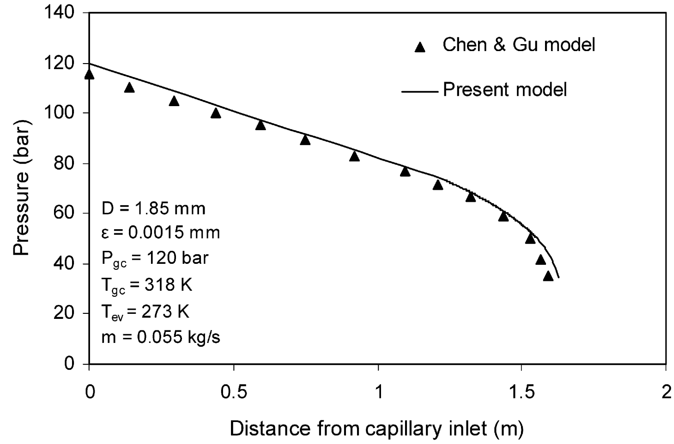

Fig. 4. Validation of the present model with the results of Chen and Gu [16].

4. Results and discussion

Numerical results are presented to study the effect of various parameters on the performance of a CT-SLHX. As mentioned earlier, the suction line is attached to a capillary tube in the subcritical two phase region. Unless otherwise mentioned, heat exchanger inlet and outer adiabatic lengths are fixed. Variation of exit gas cooler and evaporator temperature, capillary tube diameter, and heat exchanger length are suitably incorporated in the study.

Results from the present simulation model are compared with the published results from Chen and Gu [16] for the specified capillary tube. It is observed (Fig. 4) that the agreement is excellent and additionally, there is a close match in mass flow rate as well.

Fig. 5 exhibits the longitudinal variation of temperature and quality in the capillary tube and the temperature variation in the suction line for a typical case when $T_{gc} = 313$ K, $T_{ev} = 288$ K, $P_{gc} = 100$ bar and $D_{i,c} = 1$ mm. It is observed that the absolute value of local quality in the non-adiabatic region is lower than that of an adiabatic capillary tube. However, the drop is not very appreciable due to a lower heat transfer rate for this typical case.

4.1. Effect of evaporating temperature

Heat transfer rate profile with respect to evaporator temperature in the range of 276–288 K is shown in Fig. 6. As the evaporator temperature decreases, the total pressure drop across the capillary tube increases. Consequently, mass flow rate increases (Fig. 8). At lower evaporating temperature, available temperature difference between capillary tube and suction line fluid is large. The cumulative effect of increased mass flow rate and temperature difference causes the heat transfer rate to increase by about 57% depicted in Fig. 6. The higher mass flow

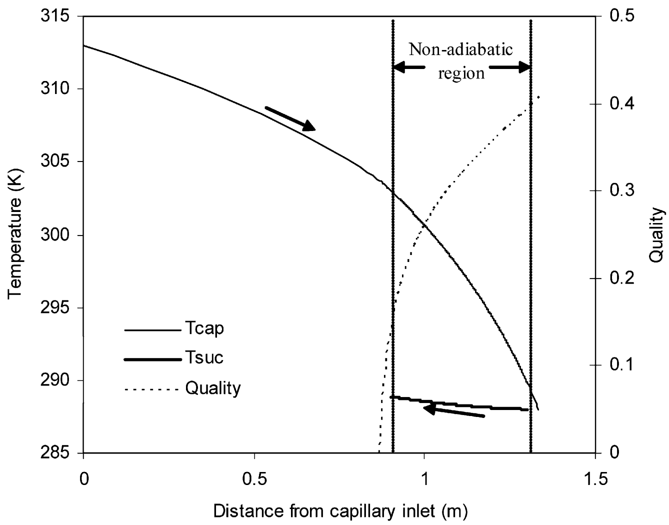

Fig. 5. Longitudinal variation of temperature and quality in capillary tube.

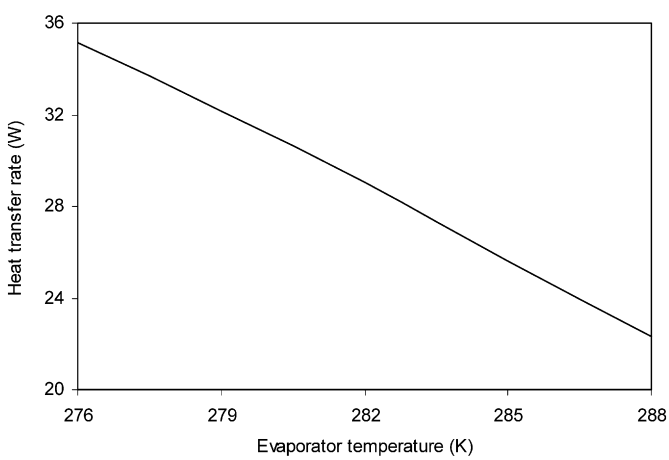

Fig. 6. Profile of heat transfer rate with respect to evaporator temperature.

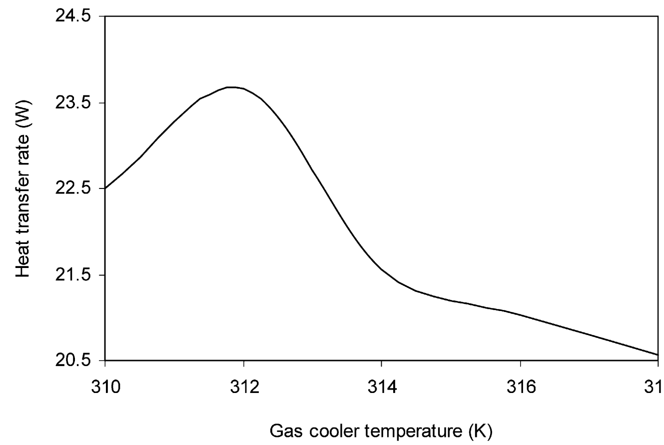

Fig. 7. Profile of heat transfer rate with varying gas cooler temperature.

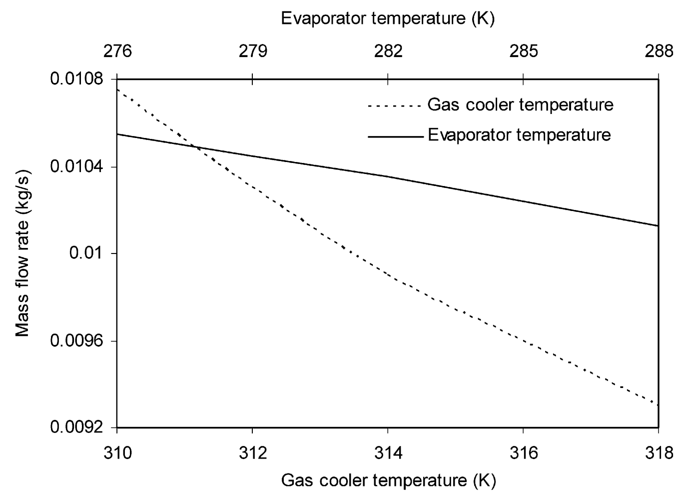

Fig. 8. Refrigerant mass flow rate with respect to gas cooler and evaporator temperature.

rate at lower evaporator temperatures causes an early flashing within the inlet adiabatic region resulting in higher quality at the inlet of the heat exchanger which leads to a larger pressure drop. However, the increased heat transfer at lower evaporator temperature suppresses the quality within the heat exchanger region. Overall quality does not change significantly unlike the conventional refrigerants due to higher vapour density and transcritical nature of CO_2.

4.2. Effect of gas cooler temperature

Variation of heat transfer rate with gas cooler temperature is shown in Fig. 7. Being a transcritical cycle, the pressure and temperature are independent parameters which make the variation of gas cooler temperature in a capillary tube flow distinct from the conventional refrigerant case. Consequently, despite changes in gas cooler temperature, pressure limits at the capillary tube remains the same. The heat transfer rate variation depends upon the relative influence of temperature difference across the capillary tube and suction line, mass flow rate of refrigerant and, the flash point location. It can be observed that as the gas cooler temperature increases, heat transfer rate first increases then decreases with the peak value at $T_{gc} = 312$ K. This can be attributed to the fact that with increase in gas cooler temperature, mass flow rate decreases due to shorter inlet single phase flow region, leading to early flashing of the refrigerant (Fig. 8). Hence quality is relatively higher at the initial stage of flashing of the refrigerant in the capillary which causes higher pressure drop for a given capillary length. In addition, at higher gas cooler temperatures, in the CT-SLHX section, heat transfer takes place relatively at a lower pressure of capillary tube; this results in a lower temperature difference between the capillary tube fluid and the suction line fluid. This can be explained by the fact that at higher gas cooler temperatures, refrigerant quality is relatively higher at the time of flashing as mentioned earlier leading to higher pressure drop in the given section of the capillary tube. The aggregate effect is that beyond a certain gas cooler temperature, heat transfer rate decreases despite increase in gas cooler temperature.

Variation of cooling capacity with gas cooler and evaporator temperature is shown in Fig. 9. It can be observed that a lower evaporator temperature brings about a higher cooling capacity, unlike traditional refrigerants. This unique feature is due to the peculiar thermodynamic properties of CO_2. Lowering

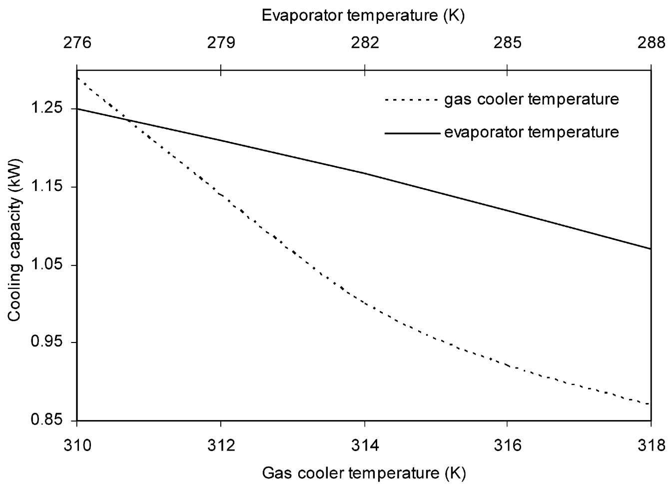

Fig. 9. Variation of cooling capacity with gas cooler and evaporator temperature.

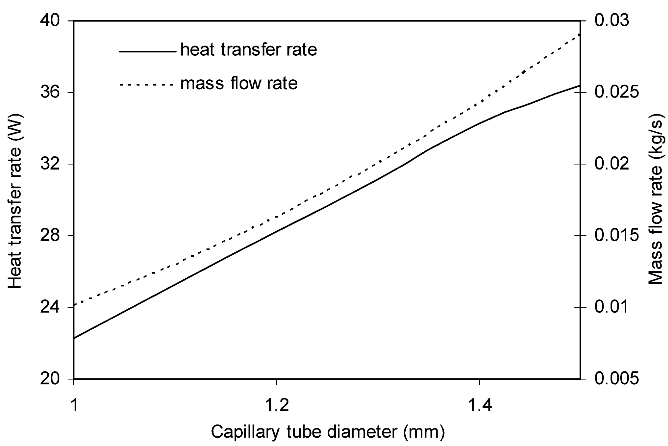

Fig. 10. Heat transfer rate profile and mass flow rate with varying capillary tube diameters.

gas cooler temperature also enhances the cooling capacity due to higher mass flow rate of refrigerant. At a higher mass flow rate of refrigerant, throttling effects are more prominent in the capillary tube flow which leads to increase in specific cooling capacity at a lower gas cooler temperature.

4.3. Effect of the capillary tube inlet diameter

The heat transfer profile and the mass flow rate of refrigerant at various capillary tube diameters are shown in Fig. 10. Capillary tube diameter was varied from 1.0 mm to 1.5 mm. Mass flow rate increases rapidly with the tube diameter due to rapid decrease in throttling effect. The increased mass flow through larger tubes causes increase in heat transfer rate by about 68% for a tube diameter increase of 50%.

4.4. Effect of capillary tube length

Profiles of heat transfer rate at two capillary tube suction line heat exchanger lengths of $L_{hx} = 0.3$ and 0.2 m are shown in Fig. 11. Inlet adiabatic capillary length L_{in} was varied from 0.85 to 0.95 m. It can be observed that as the inlet adiabatic capillary length increases, heat transfer rate decreases due to decrease in the inlet temperature of the refrigerant in the capillary tube heat exchange region. With fixed inlet adiabatic capillary length L_{in}, the higher heat exchanger length L_{hx} yields, as expected, higher heat transfer. Variation in heat transfer with inlet adiabatic capillary length is more prominent at higher heat exchanger length. Shorter inlet adiabatic capillary length with larger heat exchanger length is better for heat transfer.

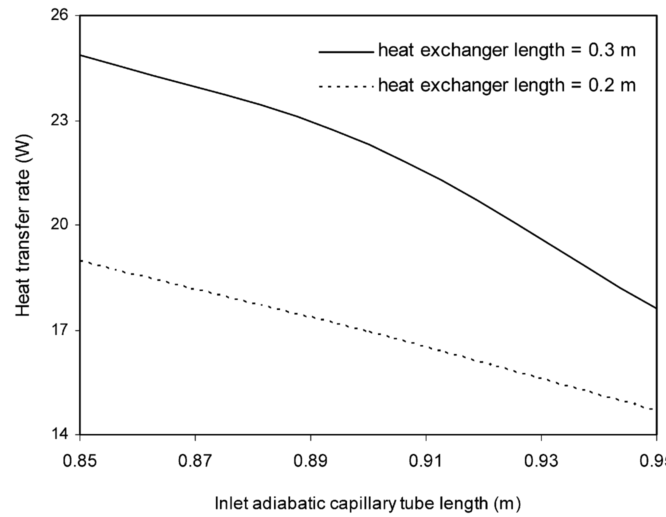

Fig. 11. Profiles of heat transfer rate for varying inlet adiabatic length and heat exchanger lengths.

5. Conclusions

A non-adiabatic capillary tube is modelled in a transcritical CO_2 heat pump cycle to investigate the effects of various geometric and operating conditions on the performance of the capillary tube. Lowering the evaporator temperature is more effective for heat transfer from capillary compared to the gas cooler temperature. Heat transfer rate profile with respect to gas cooler temperature is inconsistent with traditional refrigerant due to transcritical nature of CO_2 and is influenced by initial quality, refrigerant mass flow rate and the temperature difference at the gas cooler. Throttling effect decreases rapidly as internal tube diameter becomes larger leading to higher mass flow rate of the refrigerant. Lowering evaporator and gas cooler temperature increases in cooling capacity. However, variation is sharper with gas cooler temperature. Increase in cooling capacity with evaporator temperature is due to the peculiar thermodynamic properties of CO_2 and is incoherent with the behaviour of conventional refrigerants. It is recommended that the capillary tube with suction line is attached in the early part of the two phase region i.e. a shorter inlet adiabatic length and a larger heat exchanger length should be used for effective heat transfer. Basic validation of the model against test data reported in the literature yielded satisfactory agreement. This study was undertaken to allay the prevailing scepticism regarding the unsuitability of capillary tubes for CO_2 based transcritical systems to attain the optimum discharge pressure.

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