Influence of geometric and operating parameters on the flow behavior of the helical capillary tube

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Abstract. This paper reveals numerical study on an adiabatic helical capillary tube employing homogenous and unchoked flow conditions for a CO₂ refrigerant. The numerical model is based on the basic principles of conservation of mass, momentum, and energy. A result of the present model is validated with previously published test results. Thermodynamic and transport properties of CO₂ refrigerant are obtained from property code CO2PROP which is developed by employing an iterative procedure of derivatives of Helmholtz free energy function. The influence of various geometric parameters like tube diameter, roughness, and coil diameter on the mass flow rate of the capillary tube has been evaluated. The mass flow of the helical capillary tube largely depends on the internal diameter. While the very minor change in mass is observed with change surface roughness. Influence of various operating factors like gas cooler pressure, temperature and evaporator temperature evaluated. A significant change in mass is observed with the change in gas cooler temperature, comparatively less effect is observed with evaporator temperature. For optimum performance of the transcritical CO₂ cycle, selection of proper gas cooler temperature, gas cooler pressure is a key factor. This study may be useful for the design of helical capillary tube for CO₂.

1. Introduction
The requirement of the air conditioning and refrigeration system in the global market is continuously increasing, however, due to the environmental issues like ozone depletion potential (ODP) and global warming potential (GWP), etc., there is need to use new technologies for environmentally friendly refrigerants. Owing to ODP and GWP, the use of CFCs and HFCs have been prohibited. A CO₂ is the natural refrigerant having good environmental friendly characteristics, like a zero ODP and less GWP[1]. In many more refrigeration applications, 80% of applications use a vapour compression refrigeration cycle (VCR). In the expansion process of the VCR cycle, the pressure difference across the condenser and evaporator are maintained. There are many expansion devices like orifice, capillary tube, thermostatic expansion valve are employed. Among these devices capillary tube is simple, economical and self-actuating devices, owing to that it is used in many small capacity refrigeration systems. The design of the capillary tube may be done by selecting proper tube dimensions for the desired mass flow rate. The desired mass flow through the capillary tube may be achieved by selecting various combinations of tube length and internal diameter, and surface roughness. The cycle with CO₂ refrigerants is the transcritical cycle that is half cycle above the critical point and half-cycle below the critical point (As shown 1). In transcritical cycle pressure and temperature are independently controlled, which makes the flow inside a capillary tube more complex in nature. So there is a need to do a detailed study on this complex aspect.

Much more research has been done on the capillary tubes. Jabaraj et al [2] carried out an experimental study on the adiabatic straight capillary tube for R22, R600a, and R290 refrigerants.
They observed that the refrigerant mass flow rates of the R600a and R290 blend were about 3.82% greater than the R22 refrigerant. Agrawal and Bhattacharyya [3] built the experimental set up for the prototype of CO₂ heating and cooling system and carried out the experimental study. The change of system COP was observed for the gas cooler and evaporator water flow rate, gas cooler and evaporator water temperature and refrigerant charge. It is reported that the performance of the system is relatively good at the overcharged condition than at the undercharged condition. Zhou and Zhang [4] carried theoretical study and that is validated with experimental work on the helical capillary tube employing R22 refrigerant and results were compared with a straight capillary tube. It was observed that as the coil diameter increases, the refrigerant mass flow rate increases significantly. Comparative study of the straight capillary tube and helical tube having a coil diameter 40mm was carried out. At same operating condition, nearly 10% reduction in a mass flow rate relative to the straight capillary tube. Jadhav and Agrawal [5] compared straight and spiral adiabatic capillary tubes for the CO₂ and R22 refrigerant. For similar working conditions, the refrigerant mass flow rate and length of the tube are considerably larger in CO₂ refrigerant. A reduction in mass flow rate in CO₂ and R22 refrigerants for the spiral capillary tube is about 22% and 15% compared to the straight capillary tube. Kim et al. [6] conducted test work on the helical capillary tube for R-410A, R-407C, and R-22. A mathematical relation was developed for calculating the mass flow rate. It was reported that the mass flow rate of R22 had been about 23% and 4% less than those of R-410A and R-407C respectively. Fiorelli and Silvares [7] developed a numerical model for a straight capillary tube for two-phase homogeneous and separated conditions for R22 refrigerant. The comparison between homogeneous and separated models was conducted, and it was observed nearly the same error level. Agrawal and Bhattacharya [8] studied a numerical model on straight capillary tube employing adiabatic flow conditions for CO₂ refrigerant. A comparative study was carried out for two-phase friction factor correlations. Similarly, four viscosity models were compared for better flow characteristics. Jadhav et al. [9-10] carried out a numerical study for the adiabatic helical capillary tube employing CO₂ refrigerant. They compare various friction correlations and suggested the best friction factor correlation. The mass flow rate increases as the helical coil diameter of the helical tube increases, the coiling effect plays a vital role in the designing of the helical tube. Bansal and Wang [11] conducted a numerical study on an adiabatic straight capillary tube. The model developed for R22 refrigerant, a graphical simulation diagram is formed for choked flow conditions for R134a and R600a in adiabatic capillary tubes which was the useful design of the capillary tube.

Comparatively less work was found in the open literature on transcritical CO₂ with a helical capillary tube. The influence of geometric and operating factors on the capillary tube and system performance is less addressed. In this paper detailed study is carried out on the effect of various factors on system performance. This study is useful to the design of the helical capillary tube with CO₂ refrigerant.

2. Mathematical Model
The model moves around the fundamental principles of mass, momentum, and energy. The total capillary tube is discretized into small elements to capture the change in thermodynamic property and minimize the numerical error. The total capillary length is calculated by adding all discretized elemental lengths of various flow regions. In CO₂ transcritical system, the supercritical and transcritical phase is single phase while the subcritical phase the flow region is the two-phase region (as shown in Figure 1). For simplicity, the mathematical model of a single-phase and two-phase is developed for the CO₂ refrigerant by employing their properties.
To make the model simple, certain less important phenomenon in a preview of the present study are being neglected with the following assumptions as:

1. The cross-sectional area is constant
2. Internal surface roughness is the same throughout the tube.
3. The flow is one-dimensional flow and no heat transfer through the capillary tube.
4. Flow is homogeneous and no metastable occurrence in the capillary tube flow.
5. Entrance losses are negligible.

![Figure 1](image1.png)

**Figure 1** Transcritical CO2 cycle with various flow regions in the capillary tube

![Figure 2](image2.png)

**Figure 2** Schematic diagram of an adiabatic coiled capillary tube showing various flow regions and discretized two-phase flow region

### 2.1 Single-phase flow region

Applying conservation of mass and energy for an element

\[
\frac{AV}{v} = \text{Constant}
\]

\[
G = \frac{m}{A} = \frac{V}{v}
\]

\[
dh + \frac{dv^2G^2}{2} = 0
\]

A momentum conservation equation for the elemental length is given as

\[
-dP - \frac{f_p V G dL}{2d} = G dV
\]

Using equation 1 above equation is written as

\[
dl = \frac{2d}{f_{sp}} \left( \frac{v}{dV} - \frac{dp}{vG^2} \right)
\]

Model is developed using a single-phase and two-phase model with Mori and Nakamaya friction factor model.
\[
f_{sp} = \frac{C_1 \left( \frac{d}{D_c} \right)^{0.5}}{Re \left( \frac{d}{D_c} \right)^n} \left[ 1 + \frac{C_2}{Re \left( \frac{d}{D_c} \right)^{n+1}} \right]
\]

(4)

Where \( C_1 \) and \( C_2 \) are the constant coefficient in the equation which can be calculated as

\[
\ln C_1 = \frac{1}{n+1} \left\{ \frac{1}{4} \left[ -3 \ln(2n+1) + (16n-7) \ln(2n-1) - (8n-3)(\ln n + \ln(4n-1)) + 6n \ln(2n) \right] \right. \\
\ln C_2 = \frac{1}{n+1} \left\{ \frac{1}{4} \left[ 3 \ln(2n+1) - (15n+4) \ln n + (19n-4) \ln(2n-1) - (7n-4) \ln(4n-1) - n \ln(6n-1) - 9n \ln 2 \right] \right. \\
\]

Where the value of \( n \) is considered as 5 (for \( Re \geq 10^5 \)), and \( \alpha \) is relative coefficient calculated using the general friction factor formula indicated as: \( f_s = \alpha Re^{-\frac{1}{n}} \) where \( n \) is the exponent component in equation

2.2 Two-Phase Models

Similar to the single-phase, the fundamental principle of fluid mechanics and thermodynamics is applied for the two-phase straight and spiral capillary tube. The modeling of the two-phase region is analogous to the single-phase, except the computation of the quality of the refrigerant. The conservation of the momentum equation is written similarly to that of the Eq. No. (3). The equation for the differential length is obtained as

\[
dL = \frac{2d}{f_{tpm}} \left( \frac{v_m}{d} - \frac{dP}{v_m G^2} \right)
\]

(5)

Where, \( f_{tpm} \) and \( v_m \) are mean friction factor and mean specific volume of the two phase flow region, respectively. Two phase viscosity have been employed from McAdams model correlation

\[
\frac{1}{\mu_{tp}} = \frac{1-x}{\mu_l} + \frac{x}{\mu_g}
\]

Where \( \mu_g \), \( \mu_l \), and \( \mu_{tp} \) are the dynamic viscosities of saturated vapour, saturated liquid and two-phase, respectively. The secondary flow, also noted as Dean Effect, control the transfer of heat, momentum and mass in coiled capillary tubes. Dean had suggested a dimensionless number (Dean Number) as the ratio of the viscous force enacting on a fluid flowing in a curved pipe to the centrifugal force.

\[
De = Re \left( \frac{d}{D_c} \right)^{\frac{1}{2}}
\]

System performance is measured with system COP and cooling capacity. The cooling capacity is calculated as

\[
Q_c = m (h_A - h_D)
\]

Similarly in the heating capacity is calculated as

\[
Q_H = m (h_B - h_C)
\]

Work done by the compressor is written as

\[
W = m(h_B - h_A)
\]

The system COP is calculated as

\[
COP = \frac{Q_c + Q_H}{W}
\]

3. Solution Technique

The equations are coupled equations and solved by an iterative numerical method to calculate the capillary tube length. The inception of vaporization is captured by taking into account the temperature
and enthalpy simultaneously, in the subcooled region. Thermodynamic and transport properties of CO₂ refrigerant are obtained from property code CO2PROP which is developed by employing an iterative procedure of derivatives of Helmholtz free energy function[14].

Assume \( m_{\text{max}} \) and \( m_{\text{min}} \)

\[
dP = (P_{gc} - P_{\text{crt}})/100, \quad L_{1-2} = 0
\]

\[
P_2 = P_{gc} - dP, \quad T_2 = T_{gc}
\]

Calculate Supercritical Refrigerant Properties by CO2PROP at pressure \( P_2 \) and temperature \( T_2 \)

\[
dL = \frac{2d}{f_{\text{prop}}} \left( \frac{v}{d\nu} - \frac{dP}{\nu v^2} \right)
L_{1-2} = L_{1-2} + dL
\]

\[
P_2 \leq P_{\text{crt}} \quad \text{No}
\]

\[
P_2 = P_{\text{crt}} T_2 = T_{\text{crt}}, \quad L_{2-3} = 0
\]

Calculate critical Refrigerant Properties by CO2PROP at pressure \( P_2 \) and corresponding \( T_2 \)

\[
dL = \frac{2d}{f_{\text{prop}}} \left( \frac{v}{d\nu} - \frac{dP}{\nu v^2} \right)
L_{2-3} = L_{2-3} + dL
\]

\[
P_2 \leq P_{\text{sat}} \quad \text{No}
\]

\[
P_2 = P_{\text{sat}} T_2 = T_{\text{sat}}, \quad L_{3-4} = 0
\]

Calculate Saturated Refrigerant Properties by Properties at pressure \( P_2 \) and corresponding \( T_2 \)

\[
dL = \frac{2d}{f_{\text{prop}}} \left( \frac{v_{m}}{d\nu} - \frac{dP}{\nu v^2} \right)
L_{3-4} = L_{3-4} + dL
\]

\[
P_2 = P_2 - dP
\]

\[
|L - L_{\text{crt}}| < 0, \quad \frac{|L - L_{\text{crt}}|}{2} \leq 0.001
\]

\[
m_{\text{max}} = m
\]

\[
m_{\text{min}} = m
\]

\[
\text{Mass 'm' for Length } L_{\text{crt}}
\]

End

Figure 3 Simulation of capillary tube
In designing of the capillary tube, the mass of the tube is considered and the total length of the capillary tube is calculated. By considering mass momentum and energy equation elemental length is calculated. Addition of these elemental lengths at particular phase region (single-phase or Two-phase) is the total length of the phase region. The total capillary length is the addition of the lengths at single-phase and phase regions. In simulation studies, the mass flow rate is guessed initially for given capillary tube length. Taking mass flow into consideration calculating the total length of the capillary tube. Compare the calculated length with the desired length. If it has the deviation then change the mass flow rate and repeated the same procedure until the desired length is equal to the calculated length. At that time we achieve the desired mass flow for a given length. (Figure 3)

4. Result and Discussion

Flow-through helical is numerically simulated for transcritical CO₂ refrigeration cycles. For CO₂, the analysis is carried under unchoked flow conditions at gas cooler pressure 100bar, gas cooler temperature 313K and evaporative temperature 273K. The internal surface roughness of the capillary tube is taken as 0.00576mm. Capillary length is taken as 1.8m for simulation studies. The authenticity of the model is checked by comparing the experimental results available in the open literature. The present model is validated with experimental results of Wang et al. [3]. The agreement of the results is within the acceptable limit as shown in Figures 4.

The performance of the capillary tube depends on the various geometric and operating parameters. The geometric factors are tube diameter, length, and surface roughness, and the operating factors are gas cooler pressure, gas cooler temperature, evaporator temperature. In this study, the influence of this parameter on the performance of the capillary tube is determined, by measuring mass flow through the capillary tube. Similarly, the system performances are calculated by knowing the COP and cooling capacity of the system. Figure No. 5 represents a change in tube diameter with a mass flow rate. As capillary tube diameter increases a large increase in the mass flow rate observed, since resistance to the flow decreases. It is observed nearly 55% rise in mass flow rate with an 18% increase in the tube diameter. Since the selection of the tube diameter is the crucial thing in the design of the capillary tube.

![Figure 4](validation.png) **Figure 4** Validation of the helical capillary tube with Wang et al.

![Figure 5](mass_flow.png) **Figure 5** Change in Mass flow rate with the tube diameter

Figure No.6 shows the results for changes in the system COP and Cooling capacity with a change in tube diameter. As the tube diameter increases mass increases, due to the decrease of throttling, results in an increase of cooling capacity. However, a marginal increase of COP has observed due increase in work done. Figure 7 exhibits, the change in mass with the respective coil diameter of the tube. As the coil diameter of the helical tube increases the resistance due to secondary flow decreases due to that the mass flow increases gradually. As the coil diameter increase by 10% the mass flow rate increases nearly by 5% only.
Figure 6 Changes in the COP and Cooling capacity with a change in tube diameter

Figure 7 Change in Mass flow rate with the Coil diameter of the capillary tube

Figure No.8 shows the results for changes in the system COP and Cooling capacity with a change in coil diameter. As the coil diameter increases mass increases, due to the decrease of coiling effect results into increase in COP and cooling capacity. However, that increasing trend is gradual due to less variation of mass flow with coil diameter. Figure 9 exhibited Change in Mass flow rate with surface roughness. As the surface roughness increases mass flow decreases, due to the increase in resistance flow. A very minor change in mass is observed, as the surface roughness increases by 18% the mass increases only by 1%

Figure 8 Change in COP and Cooling capacity with the Coil diameter

Figure 9 Change in Mass flow rate with surface roughness

Figure No.10 shows the results for changes in the system COP and Cooling capacity with a change in surface roughness. As the surface roughness increases mass decreases, results into increase in COP and cooling capacity. However, that decreasing trend is minimal.

Similar to the geometric factor many operating parameters influence the performance of the capillary tube and system. These operating parameters are evaporator temperatures, gas-cooler pressure, and gas-cooler temperature. In figure 11 change in mass flow rate is observed for change in evaporator temperatures. As the evaporating temperature increases mass flow rate decreases significantly, due to a decrease in the pressure difference. As the evaporator temperature increases by 15% the mass flow decrease nearly 11%. Fig.12 shows the change in cooling capacity and COP with a change in evaporator temperatures. As the evaporator temperature increases the cooling capacity increases due to the transcritical cycle of CO₂.
Figure 10 change in the system COP and Cooling capacity with a change in surface roughness.

However, as the evaporator temperature increases the work of compression decreases resulting in increase in COP. Figure 13 shows the change in the mass flow rate with a change in gas cooler pressure. As the gas pressure increases the mass flow rate increases, due to an increase in the pressure difference. As the gas cooler pressure increases by 4%, the mass flow increases by nearly 7%.

Figure 12 change in cooling capacity and COP with a change in evaporator temperatures

Figure 14 exhibited the change in cooling capacity and COP with gas cooler pressure. As the gas cooler pressure increases the mass flow increases hence the cooling capacity increases. For a given geometric condition maximum COP observed at 100bar, while the minimum at 99bar. Thus to achieve the best performance of the system, it needs to operate at optimum gas cooler pressure. Figure 15 shows the change in the mass flow rate with a gas cooler temperature. As the gas cooler temperature increases the mass flow rate decreases significantly. As the gas cooler temperature increases by 5% the mass flow decrease nearly 18%. Fig.16 shows the change in cooling capacity and COP with a change in gas cooler temperatures. As the gas cooler temperature increases the cooling capacity decreases due to a decrease in mass flow rate.
Similarly, COP decreases with increase in gas cooler temperature, due’s’ shaped isotherm above the critical point and transcritical behavior of the CO\textsubscript{2} system (as shown in figure 1).

5. Conclusion

This paper presented a theoretical study on a homogenous, adiabatic helical capillary tube using CO\textsubscript{2} as a refrigerant. The flow characterization is done for unchoked flow conditions. The influence of geometric and operating parameters on the mass flow rate of the capillary tube has been evaluated. Similarly, the effect of these parameters on system COP and cooling capacity has been evaluated. The mass flow of the helical capillary tube largely depends on the internal diameter. It is observed nearly 55% rises in mass flow rate with an 18% increase in the tube diameter. Comparatively less influence is observed with coil diameter, as the coil diameter increase by 10% the mass flow increase only by 5% only.

A very minor change in mass is observed, as the surface roughness increases by 18% the mass increases only by 1%. Effect of operating parameters like gas cooler pressure, temperature and evaporator temperature is evaluated. A significant change in mass is observed with the change in gas cooler temperature, as the gas cooler temperature increases by 5% the mass flow decrease nearly 18%. Comparatively less effect is observed with evaporator temperature and gas cooler pressure, As the evaporator temperature increases by 15% the mass flow decrease nearly 11%, similarly, as the gas cooler temperature increases by 4% the mass flow decrease nearly 7%. Cooling capacity and COP of the system is largely depending on the tube diameter, evaporator temperature, gas cooler temperature, and pressure. While very less variation is observed with surface roughness, coil diameter. For optimum performance of the transcritical CO\textsubscript{2} system, selection of proper gas cooler temperature and gas pressure a key factor. This study may be useful for the design of helical capillary tube for CO\textsubscript{2}. 

\begin{figure}[h]
\centering
\includegraphics[width=\textwidth]{figure14.png}
\caption{Change in cooling capacity and COP with gas cooler pressure}
\end{figure}

\begin{figure}[h]
\centering
\includegraphics[width=\textwidth]{figure15.png}
\caption{Shows the change in the mass flow rate with a gas cooler temperature.}
\end{figure}

\begin{figure}[h]
\centering
\includegraphics[width=\textwidth]{figure16.png}
\caption{Change in cooling capacity and COP with a gas cooler temperature}
\end{figure}
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