Influence of geometric and operating parameters on the performance of the helical capillary tube for the R744 refrigerant

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Abstract. This paper reveals the numerical investigation of the helical capillary tube. The study is carried out for an adiabatic, homogenous capillary tube with the R744 refrigerant. The basic principles of conservation of mass, momentum, and energy are used to develop the mathematical model. The results of the present coiled capillary tube model are verified with previously published test results. The refrigerant properties of R744 are employed from CO2PROP. The influence of various geometric parameters of the capillary tube-like tube diameter, roughness, and coil diameter on the performance of the tube been computed. The consequence of the tube diameter on the tube performance is larger than other geometric parameters. As the tube diameter increase by 18%, the mass flow rate increase by nearly 55%. Similarly, as an increment in the coil diameter took place by 10%, the mass flow rate increase by 5%. The negligible change is observed owing to the change in surface roughness. While the surface roughness increases by 18%, the mass increases by 1%. Moreover, the influence of various operating factors is evaluated. A significant variation in the tube and system performance is seen owing to the change in gas cooler temperature. As the gas cooler temperature rises by 5%, the drop in mass flow rate is about 18%. Comparatively, less effect is recorded due to an evaporator temperature. As the evaporator temperature rises by 15%, the mass flow decreases by nearly 11%. The considerable impact is perceived owing to the change in pressure of the gas cooler. As the gas cooler pressure increases by 4%, the mass flow rate increases by nearly 7%. Similarly, the influence of operating and geometric parameters on the coefficient of performance and cooling capacity of the transcritical R744 system is evaluated. For optimum performance of the R744 cycle, the selection of proper gas cooler temperature and gas cooler pressure is a key factor. This work is useful to design the helical capillary tube with R744 refrigerant.

1. Introduction
In day to day life, the use of the heating ventilating air conditioning and refrigeration system increases continuously. However, owing to environmental problems of the CFCs, HCFCs, and HFCs, there is a
need to use ecological refrigerant. Carbon dioxide (R744) is a perfect refrigerant, which fulfills these requirements. On the other hand, due to the low critical temperature, the R744 cycle works in a transcritical manner. The transcritical cycle is altogether different from the subcritical cycle. Many more modifications need to do in various components of the R744 system. Among various expansion devices, the capillary tube is simple, economical and self-actuating devices, owing to that it is used in many small capacity refrigeration systems. The combination of R744 refrigerant and capillary tubes may be widely useful in many applications. However, in the transcritical R744 cycle, temperature and pressure are not dependent on each other. Consequently, the flow through the capillary tube is complicated. So there is a need to do a detailed study on this complex aspect. Accordingly, that must do a comprehensive work of this complex aspect. The design of the capillary tube may be done by selecting proper tube dimensions, to achieve the desired refrigerant flow rate. Various researches have been done on the capillary tubes with different refrigerants. Jabaraj et al.[2] developed an experimental prototype model. They compared the blends of refrigerants with R22 refrigerant for the straight capillary tube. The mass flow rate through the capillary tube of the blend (R600a and R290) was more than three percentages than the R22 refrigerant. Wang et al. [3] created a theoretical model for the helically coiled capillary tube using metastable flow conditions. Under the similar operating condition, the length was calculated by using homogenous and separated two-phase flow conditions. Agrawal and Bhattacharyya [4] explained the experimental model for the R744 system. The performance of the system was observed in various operating conditions. The system gave a good response at the larger charge of the refrigerant than the lower charge of the refrigerant. Jadhav and Agrawal [5] studied the flow behavior of the straight and spiral capillary tube. The study was accompanied by R744 and R22 refrigerants. Under the same operating situations, the mass flow rate of the spirally coiled capillary was relatively smaller than the straight-sized capillary tube. Kim et al. [6] conducted experiments at various test conditions for the helical capillary tube. For R22, R-407C, and R-410A refrigerant, the mathematical correlation was developed. It was reported that the refrigerant mass flow rate of 410A and R-407C was larger than that of R22. Fiorelli and Silvares [7] studied two-phase flow through a capillary tube employing homogenous and separated situations. The study is carried out for the R22 refrigerant. It was observed that the error level in both cases (homogenous and separated) was the same. Agrawal and Bhattacharya [8] investigated the performance of the straight capillary tube. The work was conducted with R744 refrigerant and employing adiabatic flow conditions. The flow characterization through the capillary tube was reported. Moreover, the system behavior was also reported. Bansal and Wang [9] presented a straight capillary tube model. That study was carried out for R600a, R134a, and R22 refrigerant. They developed a unique diagram for the tube to identify the unchoked and choked flow situation. Various authors [10-14] carried out studies toward the helically coiled capillary tube. The studies were carried out with different friction factors correlations. They observed that Mori and Nakamaya relation was the most practical relation among all. Similarly, Dubba and Kumar [15] discussed the flow characterization of the helical capillary tube. The impact of various operating parameters on the mass flow rate is discussed. The dimensionless correlation is developed which gave good agreement with the measured mass flow rate.

Comparatively, less work was found in the open literature on transcritical R744 with a helical capillary tube. The flow behavior with the helically coiled capillary tube is complicated from that of the straight-sized capillary tube. Similarly, under a transcritical R744 cycle the flow characterization in a capillary tube more intricate. Hence it is a need to do an extensive study on this complex aspect of the coiled capillary tube and R744 refrigerant, this paper reported the same. In this paper, the flow behavior of helical capillary tube and R744 refrigerant is reported. In this work, the influence of various factors on the system performance and flow behavior of the helical capillary tube is calculated. Moreover, this work is beneficial to design the helically coiled tube employing R744 refrigerant.

2. Mathematical Model
The numerical model is created using basic equations of mass, momentum, and energy. To make computation precise, the capillary tube is discretized in miniature parts to catch the variation in refrigerant properties. The addition of lengths, at each element, gave the total length of the tube. In the R744 transcritical system, supercritical and transcritical regions are a 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 R744 refrigerant by employing their properties. Some insignificant features of the study are being ignored with the following assumptions as:

1. The internal surface roughness and tube diameter are uniform throughout the length of the tube.
2. An adiabatic capillary tube, with the flow, is assumed in one direction.
3. Flow in the subcritical region is considered as homogeneous flow and no metastable conditions.
4. Since very less loss at the entrance of the tube, the entrance losses are ignored.

2.1 Supercritical and transcritical flow region (Single-phase model)

For an elemental length, using basic equations of mass and energy

\[ \frac{m}{m_1} = \frac{m_2}{m} \]

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

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

The momentum relation is written as

\[ -dP - \frac{f_{sp}Vgd}{2d} = GdV \]

Using equation 1 above equation is written as

\[ dL = \frac{2d}{f_{sp}} \left( \frac{\nu}{d \nu} - \frac{dp}{vG^2} \right) \]

(3)

The Mori and Nakamaya friction factor correlation may be employed for the helical capillary tube.

\[ f_{sp} = \frac{C_2 (\frac{d}{\nu})^{0.5}}{\left[ Re \left( \frac{d}{\nu} \right)^{\frac{1}{n+1}} \right]^{\frac{1}{n+1}}} \left[ 1 + \frac{C_2}{\left[ Re \left( \frac{d}{\nu} \right)^{\frac{1}{n+1}} \right]^{\frac{1}{n+1}}} \right] \]

(4)[11][13]

The constants \( C_1 \) and \( C_2 \) are computed using the following equations

\[ \ln C_1 = \frac{1}{n+1} \left\{ \frac{1}{4} \left[ \frac{\ln (2n+1)}{(8n-3)} \right] + \frac{\ln (4n-1)}{(8n-3)^2} + \frac{\ln (6n-1)}{(8n-3)^3} \right\} \]

\[ \ln C_2 = \frac{1}{n+1} \left\{ \frac{1}{4} \left[ \frac{\ln (2n+1)}{(8n-3)} \right] + \frac{\ln (4n-1)}{(8n-3)^2} + \frac{\ln (6n-1)}{(8n-3)^3} \right\} \]
In the above equation, the ‘\( n \)’ is taken as 5, since the Reynolds Number is greater than \( 10^5 \). Similarly, the ‘\( \alpha \)’ determined to employ

\[
f_s = \alpha \, Re^{-\frac{1}{n}}
\]

### 2.2 Subcritical flow region (Two-Phase Model)

Likewise, section 2.1. For the subcritical region the basic equations are applied. The mathematical model of the subcritical region is the same as that of supercritical and transcritical except the calculation of the dryness fraction (\( x \)). The basic momentum equation may be used as

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

In the above equation, \( v_m \) is the average specific volume, and \( f_{tpm} \) is the average friction factor. Two-McAdams relation may be used to evaluate the viscosity of the two-phase region.

\[
\mu_{tp} = \frac{(1 - x)\mu_a + x(\mu_l)}{\mu_1\mu_a}
\]

In the helical capillary, the coiling effect is predominant and that can be indicated with the Dean Number. Dean number is indicated from the following equation

\[
D_n = Re \left( \frac{d}{D_C} \right)^{\frac{3}{2}}
\]

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

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

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

As indicated in figure 1, the capillary tube is divided into miniature elements. The basic equations of fluid dynamics are used to calculate the length of the tube. The CO2PROP, property code is applied to these parts to calculate the mass flow rate [16]. As shown in figure 2, the mass flow rate is calculated for a given length. Initially, for given operating and geometric conditions, the mass flow rate is assumed. Consequently, the length has calculated for different region viz. supercritical, transcritical, and subcritical regions. The length is added to obtain the total length of the capillary tube. Afterward, the calculated length is compared with the given length. If there was a considerable difference in the lengths of the capillary tube, then the mass flow rate is changed and repeated the same procedure. At last, if the error level is negligible, then the calculated mass flow rate is desired for the given length.
4. Result and Discussion
The simulation results are obtained for the transcritical R744 cycle using a helical capillary tube. For the simulation study, the operating conditions are $P_{gc}=100$ bar, $T_{gc}=313$ K, and $T_{ev}=273$ K. However, the geometric dimensions of capillary tube dimensions are $d=1.4 \text{ mm}$, $L=1.8 \text{ m}$, $D_{e}=50 \text{ mm}$, $\varepsilon = 0.00576 \times 10^{-3}$ mm. To check the accuracy, the result of the existing model is validated with the earlier presented result. The present model is compared with the experimental results of Wang et al. [3]. Figure 3 indicated that the agreement of the results which is good. Figure 4 represents a variation in the mass flow rate by the change in tube diameter. As capillary tube diameter
rises, an increase in the refrigerant flow rate is observed since resistance to the flow decreases, since 
resistance to the flow decreases. It is observed a 55% rise in the refrigerant flow rate as the tube 
diameter rises by 18%.

Figure 3. Validation of the present model with Wang et al.[3]

Figure 4. Variation in refrigerant mass flow rate with change in the tube diameter.

Figure 5. shows the results for changes in the system COP and cooling capacity with a variation in 
tube diameter. As the diameter increases mass increases, due to the decrease of throttling, cooling 
capacity increases. However, a minimal increase of COP has observed due increase in work done. 
Figure 6 shows the variation in mass with respective coil diameter. While the coil diameter of the 
helically coiled tube increases, the resistance to flow decreases due to that the refrigerant mass flow 
rate rises gradually. The mass flow rate increases nearly by 5%, as the coil diameter increase by 10%.

Figure 5. Changes in the COP and cooling capacity with a change in tube diameter at gas cooler 
temperature 313K, and gas cooler pressure 100bar.

Figure 6. Change in Mass flow rate with the 
Coil diameter of the capillary tube at gas 
cooler temperature 313K, and gas cooler 
pressure 100bar.

Figure 7. shows the results for changes in the system COP and Cooling capacity with a change in 
coil diameter. The mass increases, as the coil diameter increases due to the decrease of the coiling 
effect results in an increase in COP and cooling capacity. However, that increasing trend is remarkable
up to 100mm coil diameter. Ahead of 100mm coil diameter, the mass increases slowly due to coiling effect. Figure 8 indicates a variation in the mass with surface roughness. Due to the increase in friction, the mass flow rate decreases, as the surface roughness increases. A negligible change in mass is observed as the surface roughness increases by 18% the mass increases only by 1%.

Figure 7. Change in COP and Cooling capacity with the Coil diameter at gas cooler temperature 313K, and gas cooler pressure 100bar.

Figure 8. Change in Mass flow rate with surface roughness at gas cooler temperature 313K, and gas cooler pressure 100bar.

Figure 9 shows the result of the change in the system COP and cooling capacity with a change in surface roughness. As the surface roughness increases, the mass flow rate decreases. However cooling capacity and COP decreases. However, that decreasing trend is very less. As the tube geometric parameters are changes, minimal change is observed in the system COP and cooling capacity. Similar to the geometric factor, many operating parameters influence the performance of the capillary tube and system.

Figure 9. Change in the performance of the system with a change in surface roughness at gas cooler temperature 313K, and gas cooler pressure 100bar.

Figure 10. Change in refrigerant flow rate with change in evaporator temperatures at gas cooler temperature 313K, and gas cooler pressure 100bar.
For the present study, the different operating parameters considered as gas cooler pressure, gas cooler temperature, evaporator temperature. The performance of the system is observed with the change in these parameters. Figure 10 shows the change in the mass flow rate with the change in evaporator temperatures. Due to a decrease in the pressure difference, the mass flow rate decreases significantly, as the evaporating temperature increases. As the evaporator temperature increases by 15% the mass flow rate decrease by 11%. Figure 11 presents the variation in cooling capacity and COP with a change in evaporator temperature. Because of the increase in the mass flow rate, the cooling capacity enhances, while the evaporator temperature increases. However, the work of compression decreases, and hence the COP increases. Figure 12 shows a graph of mass flow per unit time with change gas cooler pressure. Because of a rise in the pressure difference, the mass flow per unit time increases, as the gas pressure increases. The mass flow increases by 7%, as the gas cooler pressure increases by 4%.

Figure 11. Change in cooling capacity and COP with a change in evaporator temperatures at gas cooler temperature 313K, and gas cooler pressure 100bar.

![Graph](image1)

Figure 12. Change in mass flow rate with change in gas cooler pressure at gas cooler temperature 313K.

Figure 13 exhibits the change in cooling capacity and COP with gas cooler pressure. Because of the increase in pressure, the mass flow rate increases, and hence the cooling capacity increases. For given geometric conditions, maximum COP observed at 100bar, whiles the minimum at 99bar. Thus to achieve the best performance of the system, it needs to operate at optimum gas cooler temperature. Figure 14 indicates a graph of the mass flow rate with the gas cooler temperature. With an increased gas cooler temperature, a decrease in the mass flow rate is observed. As the gas cooler temperature increases by 5%, the mass flow decreases by 18%. Figure 15 shows the change in cooling capacity and COP with gas cooler temperatures. Because of the increase in gas cooler temperature, the mass flow rate decrease, and hence the cooling capacity decreases. Similarly, COP decreases due ‘s’ shaped isotherm above the critical point and transcritical behavior of the R744 system.
Figure 13. Change in cooling capacity and COP with gas cooler pressure at gas cooler temperature 313K.

Figure 14. Mass flow rate with a respective temperature of the gas cooler at gas cooler pressure 100bar.

Figure 15. Cooling capacity and COP with a change in gas cooler temperature at gas cooler pressure 100bar.

5. Conclusion
This study performed the performance analysis of a transcritical R744 cycle. Moreover, the flow behavior of the helically coiled capillary tube is presented for unchoked flow situations. Similarly, the influence of geometric and operating factors on system performance is observed. The performance of the system is measured by calculating the COP and the cooling capacity. Similarly, the mass flow rate is measured for various tube geometry. The influence of tube diameter has a significant effect on the mass flow rate. It is observed a 55% rise in the mass flow rate with an 18% rise in internal tube diameter.
diameter. Comparatively, less influence is observed with coil diameter, the mass flow rate increase by 5%, as the coil diameter increase by 10%. A lesser variation is seen with a change in surface roughness. The mass increases only by 1%, as the surface roughness increases by 18%.

Moreover, the impact of operating parameters on system behavior is calculated. A notable influence is detected with the gas cooler temperature. The mass flow decreases by 18%, while the gas cooler temperature rises by 5%. Comparatively, less effect is observed due to the change in temperature of the evaporator and pressure of the gas cooler. The mass flow rate drops by 11%, as the evaporator temperature rises by 15%. Similarly, The mass flow rate rises by 7%, as the gas cooler pressure rises by 4%. As the tube geometric parameters are changes, minimal change is observed in the system performance. The system performance largely depends on the gas cooler and evaporator temperature, gas cooler pressure. For optimum performance of the R744 cycle, the selection of proper gas cooler temperature and gas pressure a key factor.

6. References
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