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Numerical study on the straight, helical and spiral capillary tube for the CO_2 refrigerant

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Abstract

A numerical study has been carried out for straight, spiral and helical capillary tubes and their performance has been compared with CO_2 refrigerant. The numerical models are developed based on the fundamental conservation principles of mass, momentum, and energy. Within this, outer loop, the ordinary differential equations are solved from the inlet to the exit of the capillary tube. The study has been carried out to calculate the mass flow rate by bisection method where the mass is iteratively calculated at the specified capillary length or vice versa. In-house coding programming employs the finite difference approach for numerical solutions. The characterization of the capillary tube has been done by calculating the length for the given mass or by calculating mass for the given length. The comparison of the straight capillary with helical capillary tube (50 mm coil diameter) and spiral capillary tube (50 mm pitch) has been reported. For a change in tube diameter, surface roughness, and length, the percentage reduction in mass flow rate in capillary tubes (straight, helical, and spiral) is calculated. The percentage reduction in mass in a helical capillary tube compared to the straight capillary tube is about 7–9 %. The percentage reduction in mass in a spiral tube compared to the straight capillary tube is nearly 23–26 %. Additionally, the percentage reduction in mass in a spiral tube compared to the helical capillary tube is almost 17–19 %. Additionally, the percentage reduction in length in a spiral tube compared to straight capillary tube ranges from 37 % to 43 %. Similarly, the percentage reduction in length in a spiral tube compared to helical capillary tube is ranging from 25 % to 32 %.

Keywords

straight tube, helical tube, spiral tube, capillary tube, adiabatic, mass flow rate

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УДК 621.564

Численное исследование прямой, винтовой и спиральной капиллярных трубок для хладагента CO_2

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Аннотация

Выполнено численное исследование прямых, спиральных и винтовых капиллярных трубок с хладагентом CO_2 и проведено сравнение их характеристик. Численные модели разработаны на основе фундаментальных принципов сохранения массы, импульса и энергии. Решены обыкновенные дифференциальные уравнения во

внешнем контуре от входа до выхода капиллярной трубки. Проведены расчеты массового расхода капиллярной трубки и хладагента методом деления отрезка пополам, где масса итеративно рассчитана при заданной длине капилляра или наоборот. Использовано собственное программирование численным методом конечных разностей. Определение характеристик капиллярной трубки выполнено путем расчета ее длины для заданной массы или расчета массы для заданной длины. Выполнено сравнение прямой капиллярной трубки с винтовой (внутренний диаметр 50 мм) и спиральной (шаг 50 мм). Для изменения диаметра трубки, шероховатости поверхности и длины рассчитан процент снижения массового расхода в прямых, винтовых и спиральных капиллярных трубках. В результате сравнения с прямой капиллярной трубкой получено процентное уменьшение массы трубок: для винтовой трубки — 7–9 %, для спиральной — 23–26 %. Для спиральной трубки также получено процентное уменьшение показателей: массы по сравнению с винтовой трубкой — 17–19 %; длины по сравнению с прямой трубкой — 37–43 % и винтовой — 25–32 %.

Ключевые слова
прямая трубка, винтовая трубка, спиральная трубка, капиллярная трубка, адиабатическая, массовый расход

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Introduction

The use of heat pumps, refrigeration systems increased by almost 10 % in terms of total and interest income. Among these systems, a large area is covered by small capacity applications, and a capillary tube is used as an expansion device. The capillary tube is a relatively low-cost and simple narrow tube with different cross-sections: straight, helical, and spiral. The coiled capillary tube is preferable compared to the straight capillary tube. Along with the economic system, an environmentally friendly system is the need of the day. CO_2 refrigerant is a favorite among all-natural refrigerants. The sustainable solution for small capacity HVAC applications is the transcritical CO_2 system and capillary tube [1–2]. Numerous numerical studies of a straight capillary tube with various refrigerants were carried out. Stoecker and Jones [3] designed the numerical model of the straight capillary tube. The numerical model of the straight capillary tube was designed using basic equations of conservation: mass, energy, momentum. Jabaraj et al. [4] mentioned the influencing parameter in the capillary tube. The study was carried out for R22 and M20 refrigerants. Among all the geometric parameters, the tube diameter and tube length were more influential. Rasti and Jeong [5] formed a correlation for calculating the mass flow rate of R-12, R-22, R-134a, R-152a, R-404A, R-407C, R-410A, R-507A, and R-600a for the straight capillary tube. The flow characterization of the adiabatic capillary tubes was carried out. The developed correlations had been in comparatively good agreement with different application range. Choi et al. [6] framed an experimental model for straight capillary tube with R-290, R-22, R-407C refrigerant. The effect of numerous geometric parameters and operating parameters was calculated, and an empirical correlation was developed. Chingulpitak and Wongwises [7] revealed the numerical study for the R-22, R-407C, and R-410A on the helically coiled capillary tube employing the adiabatic and homogenous two-phase condition. After completing rigorous research, they suggest that the Mori and Nakamaya friction factor equation was the best among various correlations. Jadhav and Agrawal [8–10] carried out a numerical study for the helical and spiral capillary tube with CO_2 and R22 refrigerant. They suggest the coil diameter influenced the helical capillary tube and pitch in the spiral capillary tube. Additionally, the gas cooler temperature in the CO_2 system is a crucial parameter to attain the optimum pressure. Mittal et al. [11] developed a numerical model for spiral capillary tubes for R22, R407C, and R410a refrigerants. The model was developed by considering adiabatic and metastable flow conditions. Flow behavior was reported at various operating and geometric conditions. Khan et al. [12] carried out an experimental study on an adiabatic spiral capillary tube with an R-134a refrigerant. The influence of geometrical parameters on the flow behavior of the capillary tube and operating parameters such as subcooling was investigated. A reduction of 5–15 % in the mass flow rate of the spiral capillary tube compared to the straight capillary tube has been reported. An empirical correlation was also developed to predict the mass flow rate of the refrigerant.

In the past, the literature reveals the studies on straight, helical, and spiral capillary tubes individually. However, capillary tubes comparative studies have not been done at the mature level for CO_2 refrigerant. Additionally, one must compare the straight, helically, and spirally coiled capillary tubes for a compact design at one glance. A compact and environmentally sustainable HVAC system is the need of today′s world, and that can be addressed with various tube configurations of capillary tube and CO_2 refrigerants. The objective of the present study is to exhaustively compare the straight, helical, and spiral capillary tubes with carbon dioxide refrigerant, which helps to design the straight and coiled capillary tube. The comparison was carried out for more practical tube dimensions and operating conditions of the transcritical CO_2 system.

Mathematical Modeling

Mathematical modeling of the capillary tube is carried out for the straight, helical and spiral capillary tube. The capillary tube is divided into small discretized elements as shown in Fig. 1 for straight, helical and spiral capillary tube. In CO_2 system, two different flow regions are observed in the capillary tube, viz. single phase and two-phase region. For simplicity, the mathematical modeling of capillary tube is divided into single phase and two-phase region.

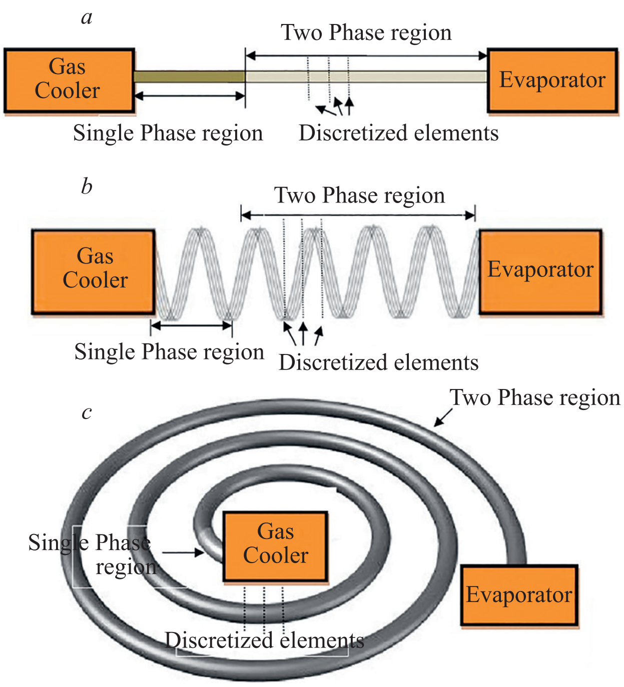


Fig. 1. Straight (*a*), helical (*b*) and spiral (*c*) capillary tube with discretized elements

Single Phase region

After the refrigerant cooling in the gas cooler, the single phase gaseous refrigerant pass through the capillary tube. To calculate flow characteristics of single phase region, basic equations of conservation of mass, momentum and energy are applied for straight, helical, and spiral capillary tube. The conservation of mass is applied for the elemental area, then it may be written as

$$m = m_1 = m_2$$

$$\frac{AV}{v} = \frac{A_1V_1}{v_1} = \frac{A_2V_2}{v_2}.$$

Here m, m_1, m_2 are the mass of CO_2 refrigerant flowing through of the elemental area of the tube, kg/s; A is the cross-sectional area tube, m^2; V is the velocity of the fluid, m/s; and v is the specific volume, m^3/kg. The flow in the capillary tube is steady and unidirectional. Applying Steady Flow Energy Equation (SFEE) to elemental area we get

$$h_1 + \frac{V_1^2}{2} + z_1 + Q = h_2 + \frac{V_2^2}{2} + z_2 + W.$$

The flow of the refrigerant is passing through the discretized session 1 and 2. Here h is the specific enthalpy, kJ/kg; V is the velocity of the fluid, m/s; z is the head across tube, m; Q is the heat transfer; and W is work done, kW. Assuming that capillary tube is adiabatic ($Q = 0$) with no any work done ($W = 0$) and the tube is horizontal ($z_1 = z_2$), above equation can be written as

$$h_1 + \frac{V_1^2}{2} = h_2 + \frac{V_2^2}{2}.$$

Using above equations

$$dh + \frac{dv^2G^2}{2} = 0.$$

Here G is the mass flux, kg/(m^2·s). Additionally, conservation of momentum is applied to the elemental area. The summation of the forces on the elemental area is equal to the acceleration of the refrigerant. The net momentum is given as

$$\Sigma f = ma$$

$$-dP - \frac{f_{sp}VGdL}{2d} = GdV$$

$$dL = \frac{2d}{f_{sp}}\left(\frac{v}{dv} - \frac{dp}{vG^2}\right).$$

Here f is force, N; P is the pressure, N/m; a is the acceleration of the fluid through tube, m/s^2; L and dL are the total length and elemental length, respectively, m; f_{sp} is the friction factor in single capillary tube. In addition to the basic equations, friction factor equations are calculated by using following friction factor correlation for straight, helical and spiral capillary tube as follows.

1) For straight tube, Churchill [13], friction factor correlations may be expressed as

$$f_{st} = 8\left[\left(\frac{8}{\mathrm{Re}}\right)^{12} + (A^{16} + B^{16})^{-\frac{3}{2}}\right]^{\frac{1}{12}}.$$

Here f_{st}, is friction factor in the straight capillary tube, Re is the Reynolds number, A and B may be calculated as

$$A = 2.457\ \ln\left(\frac{1}{\left(\frac{7}{\mathrm{Re}}\right)^{0.9} + \frac{0.27\varepsilon}{d}}\right),\ B = \frac{37\ 530}{\mathrm{Re}},\ \mathrm{Re} = \frac{Gd}{\mu},$$

$$dL = \frac{2d}{f_{sp}}\left(\frac{v}{dv} - \frac{dp}{vG^2}\right),$$

where ε is the surface roughness, mm; and μ is the dynamic viscosity, Pa·s.

2) For helical capillary tube, the elemental length is calculated using Mori and Nakamaya [14] friction factor relation

$$f_{hsp} = \frac{C_1\left(\frac{d}{D_c}\right)^{0.5}}{\left[\mathrm{Re}\left(\frac{d}{D_c}\right)^{\frac{n}{2}}\right]^{\frac{1}{n+1}}}\left[1 + \frac{C_2}{\left[\mathrm{Re}\left(\frac{d}{D_c}\right)^{\frac{n}{2}}\right]^{\frac{1}{n+1}}}\right].$$

Here d is tube diameter; D_c is coil diameter of the helical capillary tube; f_{hsp} is the friction factor of the helical capillary tube. The value of n is considered as 5 because the Reynolds Number (Re) is larger than 10^5.

$$\ln C_1 = \frac{1}{n+1} \times$$

$$\times\left\{\begin{matrix}\frac{1}{4}[-3\ln(2n+1) + (16n-7)\ln(2n-1) - (8n-3) \times \\ \times [\ln n + \ln(4n-1)] + (6n-1)] + n\ln\alpha + 9\ln 2\end{matrix}\right\},$$

where α is the linearity constant, and it is calculated by employing the general friction factor equation

$$\ln C_2 = \frac{1}{n+1} \times$$

$$\times \left\{ \begin{array}{c} \frac{1}{4}[3\ln(2n+1) - (15n+4)\ln n + (19n-4) \times \\ \times \ln(2n-1) - (7n-4)\ln(4n-1)] + n\ln\alpha - \\ - n\ln(6n-1) - 9n\ln 2 \end{array} \right\}$$

$$dL = \frac{2d}{f_{hsp}}\left(\frac{v}{dv} - \frac{dp}{vG^2}\right).$$

3) Similarly, for spiral capillary tube elemental length is calculated using Ju et al. [15] friction factor relation

$$f_{csp} = f_{st}(1 + 0.11\mathrm{Re}^{0.23}(d/D_c))^{0.14},$$

f_{csp} is the friction factor of the spiral capillary tube and f_{st} is the friction factor in straight capillary tube

$$f_{st} = 0.1(1.46\varepsilon/d + 100/\mathrm{Re})^{0.25}$$

$$dL = \frac{2d}{f_{csp}}\left(\frac{v}{dv} - \frac{dp}{vG^2}\right)\left(\frac{\cos\left(\frac{d\theta}{2}\right)}{\cos\left(\frac{d\theta}{4}\right)}\right).$$

Here, ε is the surface roughness, mm; and θ is angular displacement, rad.

Two-Phase region

While passing the refrigerant through subcritical region, two-phase region is existed. In CO_2 refrigerant the density ratio is low, so the two phase flow is considered as a homogenous flow. The modeling of the capillary tube is similar to the single-phase region except the dryness fraction (x) in the subcritical region. Using the dryness fraction, the enthalpy, entropy and specific entropy of the refrigerants are calculated as

$$h = h_f + xh_{fg},$$
$$v = v_f + xv_{fg},$$
$$s = s_f + xs_{fg}.$$

Here h, h_f, h_g, are specific enthalpy, specific enthalpy at saturated liquid, and specific enthalpy at saturated gas, respectively, and latent heat of evaporation is given as $h_{fg} = h_g - h_f$. Similarly, v, v_f, v_g, are specific volume, specific volume at saturated liquid, and specific volume at saturated gas, respectively, and difference of specific volume of saturated gas and saturated liquid is termed as $v_{fg} = v_g - v_f$. Similarly, s, s_f, s_g, are specific entropy, specific entropy at saturated liquid, and specific entropy at saturated gas, respectively, and $s_{fg} = s_g - s_f$. The mean values of the friction factors (similar to section "Single phase region") for straight, helical, and spiral capillary tube and specific volume are used to calculate the length of the respective capillary tube.

Solution Technique

The characterization of the capillary tube has been done by calculating the length for the given mass or calculating mass for the given length. The CO_2 refrigerant properties are given for this calculation. The calculation of mass is termed as simulation of the capillary tube and determination of length is called as designing of the capillary tube.

In the designing of the capillary tube, initially, for a given tube geometry and mass flow rate, evaporator temperature (T_{ev}) gas cooler temperature (T_{gc}) and pressure (P_{gc}) used in calculating elemental length. Adding the elemental length gives the total tube length. Likewise, total length is calculated for every flow regions, viz. supercritical, transcritical and subcritical region (as shown in Fig. 2). The pressure is considered as a marching parameter to indentify the flow regions in the capillary tube. While the pressure is above the critical value (71 bar), the flow region is super-critical region. If the pressure is in between critical pressure and saturation pressure, the flow region is the transcritical region. At last, the flow region is subcritical region if it is between the saturation pressure and the evaporator pressure.

In the simulation study, as shown in Fig. 3, mass of the refrigerant is calculated for the given length. Initially mass is assumed and calculated for the total length of the tube. Comparing the calculated length (c) with the desired length (d) and if deviation is found in it, then the mass is to be changed, and again calculated length of tube. The cycle is repeated until the desired length gets equal to the calculated length and at that time assumed mass is the exact mass of the tube. The better approximate value may be found by bisection method. For numerical solutions, FORTRAN programming is used with in-house coding using the finite difference approach.

Result and Discussion

The straight, helical and spiral capillary tube has been compared. In the comparison considered, T_{gc} and

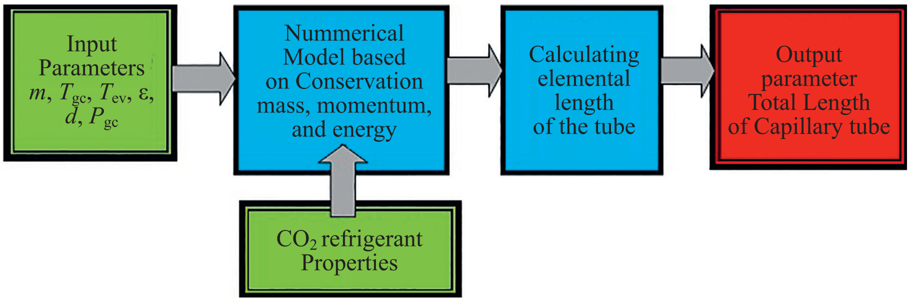


Fig. 2. Block diagram of designing of the capillary tube

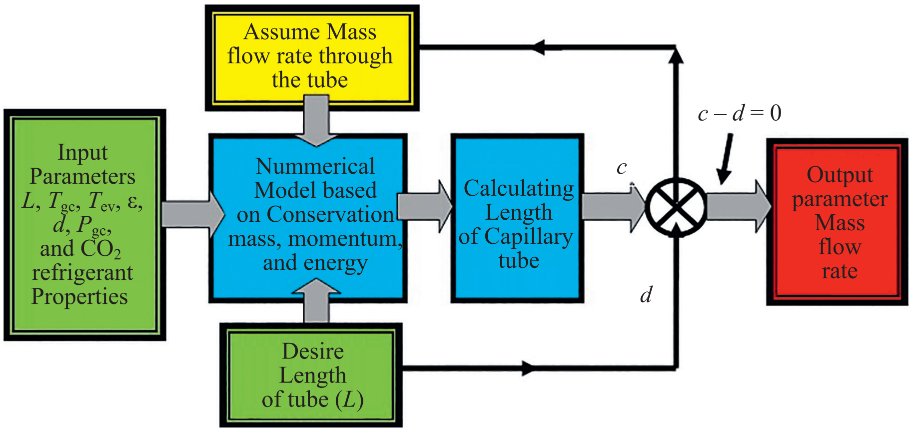


Fig. 3. Block diagram of simulation of the capillary tube

P_{gc} are gas cooler temperature and gas-cooler pressure, respectively. Similarly, T_{ev} and P_{ev} are evaporator temperature and evaporator pressure, respectively. Fig. 4 shows validation of present model result with the test result of Agrawal and Battacharyya for straight capillary tube. The present model results agree with the experimental results.

Fig. 5 indicates the validation of present model result with Wang et al. for helical capillary tube. The results agree with the test results of Wang el al.

Fig. 6 indicates the validation of present model result with Jadhav and Agrawal for spiral capillary tube. The results agree with the test results of Jadhav and Agrawal.

The comparison of straight capillary tube with helical tube (50 mm coil diameter) and spiral capillary tube (50 mm pitch) has been reported. For change in tube diameter, surface roughness and length of tube, the percentage reductions in mass flow rate in capillary tubes (straight, helical, and spiral) are calculated. The change in mass flow rate for straight, helical and spiral capillary tube, when changing the tube diameter from 1 mm to 1.8 mm, is indicated. The geometric and operating conditions are: tube length 1.4 m, evaporating temperature 270 K, Surface roughness 0.0005 m, gas cooler temperature 310 K, coil diameter 50 mm (helical capillary tube only), and pitch 50 mm (for spiral capillary tube only). As the tube diameter increases, the mass flow rate increases significantly in straight, helical and spiral capillary tube due to flow resistance minimizing.

Fig. 7, *a* indicates the percentage reduction in mass for straight helical and spiral capillary tube when changing the tube diameter from 1 mm to 1.8 mm in five steps with 0.2 mm intervals, i.e. 1 mm, 1.2 mm, 1.4 mm, 1.6 mm, 1.8 mm. Percentage reduction in mass in a helical tube compared to straight capillary tube is from 7.07 % to 8.73 %. Percentage reduction in mass in a spiral tube compared to straight capillary tube is from 23.52 % to 25.64 %. Percentage reduction in mass in a spiral tube

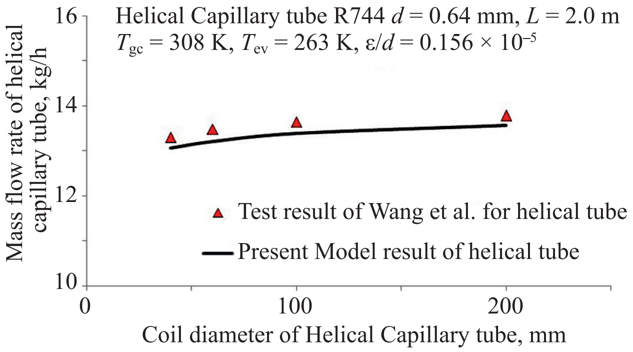


Fig. 5. Validation of the present model results of helical capillary tube with Wang et al. for helical capillary tube

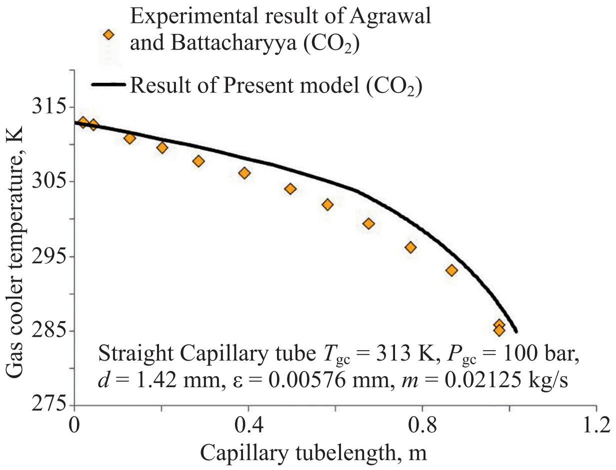


Fig. 4. Validation of present model result with the test result of Agrawal and Battacharyya for straight capillary tube

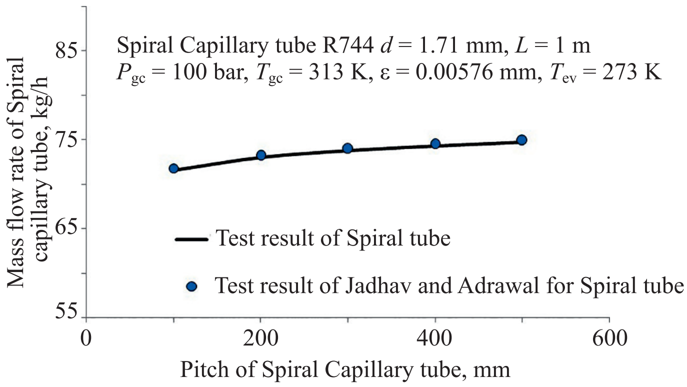


Fig. 6. Validation of present model result with Jadhav and Agrawal for spiral capillary tube

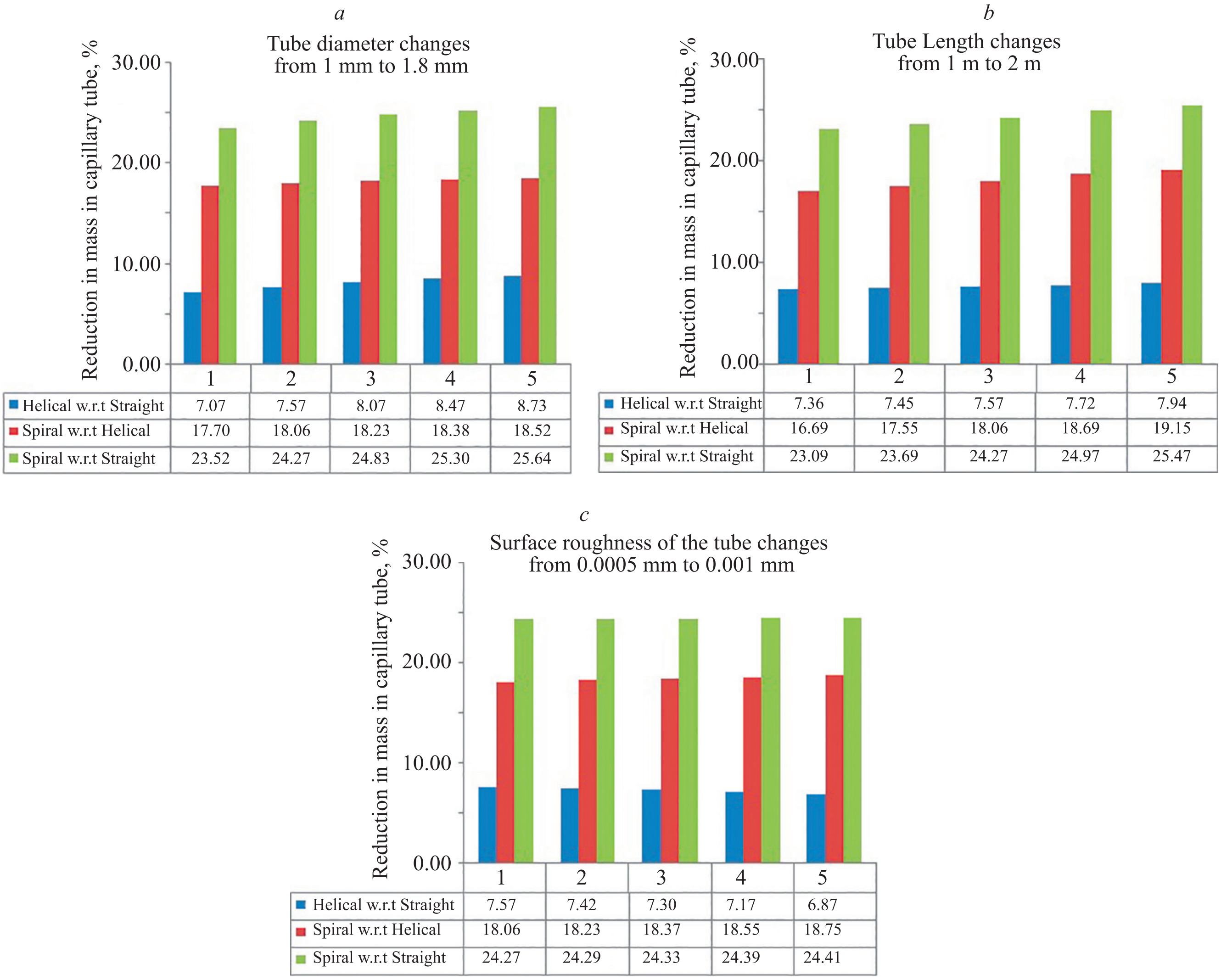


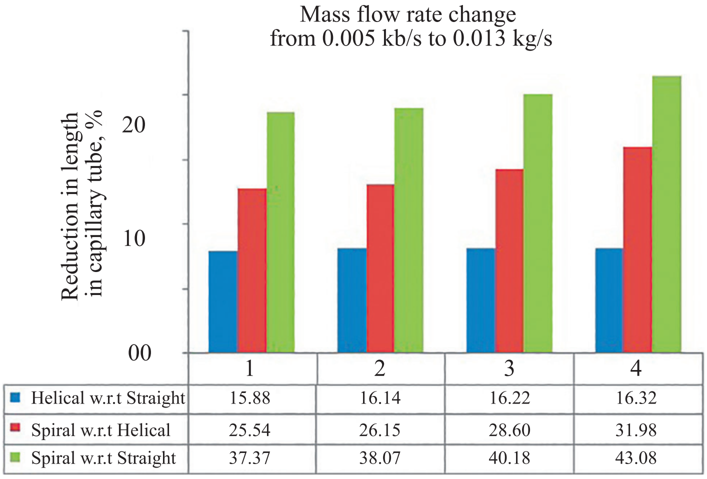
Fig. 7. Comparison of the a mass flow rate in straight helical and spiral capillary tube with change in tube diameter (*a*), tube length (*b*), surface roughness (*c*)

compared to helical capillary tube is from 17.70 % to 18.52 %.

The change in mass flow rate for straight, helical and spiral capillary tube, when changing the tube length from 1 m to 2 m, is presented. The geometric and operating conditions are: tube diameter 1.2 m, evaporating temperature 270 K, surface roughness 0.0005 m, gas cooler temperature 310 K, coil diameter 50 mm, and pitch 50 mm. As the tube length increases, the mass flow rate decreases in straight, helical and spiral capillary tube. The increase in length increases the friction in the tube that results in decrease in mass flow rate. Fig. 7, *b* indicates the percentage reduction in mass for straight helical and spiral capillary tube when changing the tube length from 1 m to 2 m in five steps with 0.25 m intervals, i.e. 1 m, 1.25 m, 1.5 m, 1.75 m, 2 m. Percentage reduction in mass in a helical tube compared to straight capillary tube is from 7.36 % to 7.94 %. Percentage reduction in mass in a spiral tube compared to straight capillary tube is from 23.09 % to 25.47 %. Percentage reduction in mass in a spiral tube compared to helical capillary tube is from 16.99 % to 19.15 %.

The change in mass flow rate for straight, helical and spiral capillary tube, when changing the surface roughness from 0.0005 mm to 0.001 mm, is reported. The geometric and operating conditions are: tube diameter 1.2 m, evaporating temperature 270 K, tube length 1.4 m, gas cooler temperature 310 K, coil diameter 50 mm, and pitch 50 mm. As the surface roughness increases, the resistance to flow increases that results in decrease in mass flow rate for straight, helical and spiral capillary tube. Fig. 7, *c* indicates the percentage reduction in mass for straight helical and spiral capillary tube when changing the surface roughness from 0.0005 mm to 0.001 mm in five steps with 0.000125 mm intervals, i.e., 0.0005 mm, 0.000625 mm, 0.00075 mm, 0.000875 mm, 0.001 mm. Percentage reduction in mass in a helical tube compared to straight capillary tube is from 7.57 % to 6.87 %. Percentage reduction in mass in a spiral tube compared to straight capillary tube is in the range 24.27–24.41 %. Percentage reduction in mass in a spiral tube compared to helical capillary tube is in the range 18.06–18.75 %.

Fig. 8 indicates the percentage reduction in length for straight helical and spiral capillary tube for change in the



	1	2	3	4
🟦 Helical w.r.t Straight	15.88	16.14	16.22	16.32
🟥 Spiral w.r.t Helical	25.54	26.15	28.60	31.98
🟩 Spiral w.r.t Straight	37.37	38.07	40.18	43.08

Fig. 8. Percentage reductions in length for straight helical and spiral capillary tube for change in the mass flow rate

mass flow rate from 0.005 kg/s to 0.013 kg/s. Percentage reduction in length in a helical tube compared to straight capillary tube is from 15.88 % to 16.32 %. Percentage reduction in length in a spiral tube compared to helical capillary tube is from 25.54 % to 31.98 %. Percentage reduction in length in a spiral tube compared to straight capillary tube is from 37.37 % to 43.08 %.

Conclusion

Numerical study of straight, spiral and helical capillary tube and comparison of their characteristics has been carried out. The models are developed based on the basic principles of conservation of mass, momentum, and energy. Within this outer loop, the ordinary differential equations are solved from the inlet to the exit of the capillary tube. The study has been done to find the mass flow rate by bisection method, where the mass is iteratively calculated at the specified capillary length or vice versa. For numerical solutions, FORTRAN programming is used with in-house coding using the finite difference approach. The comparison of straight capillary with helical tube (50 mm coil diameter) and spiral capillary tube (50 mm pitch) has been reported. For changing the tube diameter, surface roughness and length of tube, the percentage reduction in mass flow rate in capillary tubes (straight, helical, and spiral) is calculated. The percentage reduction in mass in a helical tube compared to straight capillary tube is about 7–9 %. The percentage reduction in mass in a spiral tube compared to straight capillary tube is near about 23–26 %. The percentage reduction in mass in a spiral tube compared to helical capillary tube is nearly 17–19 %. The percentage reduction in length is calculated, as the mass flow rate changes from 0.005 kg/s to 0.013 kg/s. The percentage reduction in length in a helical tube compared to straight capillary tube is nearly 16 %. At the same time, percentage reduction in length in a spiral tube compared to straight capillary tube ranges from 37 % to 43 %. Similarly, percentage reduction in length in a spiral tube compared to helical capillary tube is ranging from 25 % to 32 %.

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