Dinamika Teknik Mesin 16(2) (2026) 150-157

Dinamika Teknik Mesin
Jurnal Keilmuan dan Terapan Teknik Mesin
http://dinamika.unram.ac.id/index.php/DTM/index

Numerical investigation of capillary tube diameter and thermostatic expansion valve effects on the performance of an R-134a automotive air conditioning system

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ARTICLE INFO

Article History:
Received 04-06-2026
Accepted 02-09-2026
Available online 01-10-2026

Keywords:
Automotive air conditioning
R-134a
Capillary tube
Thermostatic expansion valve
Numerical simulation

ABSTRACT

Automotive air-conditioning (AC) systems require an expansion device that provides appropriate refrigerant flow while maintaining cooling performance and energy efficiency. This study develops a MATLAB-based steady-state numerical model to investigate the effect of capillary-tube diameter on an R-134a automotive AC system and compare it with a thermostatic expansion valve (TXV) model under identical operating conditions. Capillary diameters of 0.8, 1.0, 1.2, and 1.4 mm were evaluated at evaporator and condenser temperatures of 5 °C and 45 °C, respectively. Increasing the capillary diameter increased refrigerant mass flow rate from 0.00268 to 0.01212 kg/s and cooling capacity from 0.330 to 1.604 kW, while reducing pressure drop from 694.22 to 606.63 kPa. Compressor power increased from 0.088 to 0.320 kW, with COP values ranging from 3.751 to 5.013. The 1.4 mm capillary tube achieved the highest predicted cooling capacity and COP (5.013) within the evaluated range. The TXV model yielded a lower predicted COP of 2.955; however, this difference reflects the simplified numerical representation of the TXV and does not establish general practical superiority of the capillary tube. The results indicate that capillary diameter is a critical design parameter and that the proposed model provides a transparent screening tool for preliminary expansion-device selection. Experimental validation is required before practical performance claims can be established.

1. INTRODUCTION

Automotive air-conditioning (AC) systems are essential for cabin thermal comfort but can contribute significantly to vehicle energy consumption. The expansion device is a critical component because it regulates refrigerant flow into the evaporator and consequently affects evaporating pressure, mass flow rate, cooling capacity, compressor power, and coefficient of performance (COP). Capillary tubes provide a simple, low-cost,

https://doi.org/10.29303/dtm.v16i2.1338

and passive expansion solution, whereas Thermostatic Expansion Valves (TXVs) regulate refrigerant flow according to evaporator superheat and can respond to variations in thermal load (ASHRAE, 2022; Çengel and Boles, 2019).

Previous studies have demonstrated that expansion-device characteristics strongly influence refrigeration-system performance. Hmood et al. (2021) showed that capillary geometry and refrigerant properties affect pressure drop and steady-state mass flow, while Alfa et al. (2023) experimentally confirmed the sensitivity of R-134a refrigeration performance to capillary-tube diameter. Ridhuan et al. (2024) reported that capillary-tube length affects compressor power and COP in an R-22 split-air-conditioning system.

For automotive applications, Alkan and Hoşöz (2010) experimentally compared an R-134a automotive air-conditioning system equipped with different expansion devices, while Alkan et al. (2021) demonstrated the influence of expansion-device selection on cooling and exergy performance. Park (2021) emphasized the role of TXV-based superheat regulation in stable operation. More recently, Firdaus et al. (2026) experimentally investigated the effects of capillary-tube diameter and TXV operation in an R-134a automotive AC system.

A specific research gap therefore remains in the direct and consistent numerical comparison of capillary-tube diameter and TXV performance under identical automotive AC operating conditions. Previous studies have generally examined capillary-flow characteristics, experimental system performance, or TXV control behavior separately. Consequently, the effects of capillary diameter cannot always be isolated from differences in system configuration or operating conditions. A transparent framework linking capillary diameter and hydraulic resistance to refrigerant mass flow, cooling capacity, compressor power, and COP, while allowing comparison with a TXV under matched conditions, is therefore needed for preliminary expansion-device assessment.

To address this gap, the present study develops an integrated MATLAB-based steady-state numerical framework for an R-134a automotive AC system. The main novelty is the direct model-to-model comparison of different capillary-tube diameters and a simplified constant-superheat TXV under the same prescribed evaporator and condenser temperatures and system assumptions. The model couples expansion-device pressure loss and refrigerant flow with evaporator cooling capacity, compressor power, and COP, providing a consistent basis for evaluating the effect of capillary diameter. Rather than claiming general practical superiority of one expansion device, the framework is intended to clarify the influence of expansion-device representation and provide a transparent screening tool before experimental validation.

Accordingly, this study quantifies the effects of capillary-tube diameter on refrigerant flow characteristics and thermodynamic performance and compares the resulting COP with that predicted by the simplified TXV model under identical nominal operating conditions. The analysis focuses on the relationship between hydraulic resistance, refrigerant circulation, cooling capacity, compressor power, and COP, while explicitly recognizing the limitations of the simplified TXV representation and the need for quantitative experimental validation.

2. RESEARCH METHODS

This study employed a MATLAB-based steady-state numerical model to investigate the effect of capillary-tube diameter on the performance of an R-134a automotive air-conditioning (AC) system. The system was represented by a vapor-compression refrigeration cycle consisting of a compressor, condenser, expansion device, and evaporator. Two expansion-device configurations were considered: a capillary tube and a Thermostatic Expansion Valve (TXV). Refrigerant thermodynamic properties were obtained from R-134a saturation-property data (ASHRAE, 2022; Çengel and Boles, 2019). The model was developed as a transparent engineering-level framework for preliminary numerical screening, rather than as a detailed predictive model of an actual automotive AC system.

To provide a consistent basis for parametric comparison, the following assumptions were adopted: (i) steady-state operation; (ii) one-dimensional refrigerant flow through the capillary tube; (iii) negligible heat loss from refrigerant lines and components except for the heat transfer represented in the evaporator and condenser models; (iv) constant compressor isentropic efficiency of 80%; (v) fixed evaporator effectiveness; (vi) prescribed evaporating and condensing temperatures of 5 and 45 °C, respectively; and (vii) thermodynamic equilibrium based on the R-134a property data used in the model.

The prescribed evaporating and condensing temperatures eliminate the need to resolve detailed heat-exchanger geometry and distributed heat-transfer processes. These assumptions reduce model complexity and allow the effect of expansion-device diameter to be isolated. However, the model does not explicitly resolve two-phase flow and phase change inside the capillary tube, detailed heat-transfer behavior in the heat exchangers, compressor map characteristics, refrigerant-charge effects, or dynamic TXV operation. Consequently, the predicted results should be interpreted as preliminary numerical estimates rather than quantitatively validated representations of a production automotive AC system.

The refrigerant mass flow rate through the capillary tube was calculated using a simplified orifice-flow formulation (Çengel and Boles, 2019; Hmood et al., 2021):

https://doi.org/10.29303/dtm.v16i2.1338

$$\dot{m} = C_d A \sqrt{2\rho \Delta P} \tag{1}$$

The cross-sectional area of the capillary tube is expressed as:

$$A = \pi d^2/4 \tag{2}$$

where $\dot{m}$ is the refrigerant mass flow rate (kg/s), C_d *is* the discharge coefficient, is the capillary-tube cross-sectional area (m²), ρ is the refrigerant density (kg/m³), and ΔP is the pressure difference between the condenser and evaporator.

The refrigerant flow characteristics were evaluated using the Reynolds number (Incropera et al., 2017):

$$\mathrm{Re} = \frac{\rho V d}{\mu} \tag{3}$$

where V is the refrigerant velocity (m/s), d is the capillary-tube diameter (m), and μ is the dynamic viscosity of the refrigerant (Pa·s).

For the present simplified hydraulic representation, the Darcy friction factor was estimated using the Blasius correlation. This approximation is intended to characterize the sensitivity of hydraulic resistance to capillary diameter and does not represent the detailed two-phase frictional behavior occurring during refrigerant expansion. More comprehensive capillary models, such as that of Hmood et al. (2021), account explicitly for phase change and two-phase flow along the tube. Therefore, the present capillary model is used primarily for parametric engineering assessment.

The pressure drop along the capillary tube was subsequently determined using the Darcy–Weisbach formulation (Çengel and Boles, 2019; Incropera et al., 2017):

$$f = 0.3164\, Re^{-0.25} \tag{4}$$

where L is the capillary-tube length (m). This formulation provides a consistent basis for evaluating the effect of diameter on hydraulic resistance while maintaining the other nominal operating parameters constant.

The evaporator cooling capacity was calculated from the refrigerant enthalpy difference and the prescribed evaporator effectiveness (ASHRAE, 2022):

$$\Delta P_f = f_D \frac{L}{D} \frac{\rho v^2}{2} \tag{5}$$

where $\dot{Q}evap$ *is the cooling capacity (W),* h_{out} and h_{in} are the refrigerant specific enthalpies at the evaporator outlet and inlet, respectively, and ε_{evap} is the evaporator effectiveness.

The compressor power consumption was calculated from the refrigerant enthalpy increase using a prescribed compressor isentropic efficiency of 80% (Çengel and Boles, 2019).

$$Q_e = \dot{m}(h_1 - h_4)\varepsilon_e \tag{6}$$

where $\dot{W}_{comp}$ is the compressor power (W) and h_1, h_2 is the refrigerant specific enthalpy at the compressor outlet.

The heat rejected by the condenser was determined from the corresponding refrigerant enthalpy difference (ASHRAE, 2022). The coefficient of performance was subsequently calculated as:(Çengel and Boles, 2019).

$$W_c = \dot{m}(h_2 - h_1) \tag{7}$$

The compressor and heat-exchanger representations are intentionally simplified. The compressor is characterized by a fixed isentropic efficiency rather than a compressor performance map, while the evaporator and condenser are represented using prescribed thermal conditions rather than detailed distributed heat-transfer models. Accordingly, the calculated cooling capacity and compressor power represent model-based estimates under prescribed operating conditions.

For comparison with the capillary-tube configuration, the TXV was represented using a constant evaporator-outlet superheat condition. The prescribed superheat was used as the control criterion, and the refrigerant mass flow rate was iteratively adjusted until the calculated outlet state satisfied the specified superheat condition. The resulting mass flow rate was then used to calculate the compressor inlet state, compressor power, condenser heat rejection, and COP using the same thermodynamic framework applied to the capillary-tube case.

The TXV model does not explicitly simulate valve-opening dynamics, sensing-bulb response, valve hysteresis, transient load response, or detailed valve-flow characteristics. Therefore, the calculated TXV performance represents a steady-state constant-superheat approximation, rather than a complete dynamic representation of a production automotive TXV. This limitation is particularly important when interpreting the COP comparison because practical TXV performance depends on dynamic control and load response (Alkan et al., 2021; Park, 2021).

The simulations were performed at a condensing temperature of 45 °C, an evaporating temperature of 5 °C, a compressor isentropic efficiency of 80%, and a capillary-tube length of 2 m. The capillary-tube diameter was varied using four discrete values: 0.8, 1.0, 1.2, and 1.4 mm.

For the TXV comparison, the same nominal evaporating and condensing temperatures, refrigerant, compressor efficiency, and thermodynamic property framework were maintained. The expansion-device representation was the primary model difference, while the TXV outlet condition was controlled using the prescribed constant superheat. This matched-condition setup was adopted to minimize the influence of unrelated operating-condition differences.

The evaluated performance parameters were refrigerant mass flow rate, evaporating pressure, flow velocity, Reynolds number, pressure drop, cooling capacity, condenser heat rejection, compressor power consumption, and COP.

Because experimental measurements were not available under exactly the same component geometry, refrigerant charge, compressor characteristics, heat-exchanger configuration, and boundary conditions used in the present model, quantitative experimental validation was not performed and is not claimed. Instead, model credibility was assessed by examining whether the predicted trends were physically consistent with published numerical and experimental findings.

The predicted influence of capillary-tube diameter was compared qualitatively with trends reported by Hmood et al. (2021), Alfa et al. (2023), Ridhuan et al. (2024), and Firdaus et al. (2026), while automotive AC expansion-device behavior was considered in relation to Alkan et al. (2021) and Park (2021). These comparisons are intended only to establish trend-level physical consistency, not numerical agreement with experimental measurements.

Accordingly, the results presented in this study should be regarded as preliminary numerical predictions. Differences between the model and practical automotive AC systems may arise from two-phase capillary flow, refrigerant charge, compressor characteristics, heat-exchanger geometry, heat-transfer coefficients, pressure losses in connecting lines, and dynamic TXV behavior. Quantitative experimental validation under matched component and operating conditions is therefore required before the predicted performance values can be generalized to practical automotive AC systems.

The overall numerical procedure is summarized in Figure 1. The simulation begins by specifying the refrigerant, evaporating and condensing temperatures, compressor efficiency, capillary-tube length, and capillary diameter. The corresponding R-134a thermodynamic properties are then determined from the property data.

For each capillary diameter, the flow area, refrigerant mass flow rate, velocity, Reynolds number, friction factor, and pressure drop are calculated sequentially. The resulting refrigerant states are then used to determine evaporator cooling capacity, compressor power, condenser heat rejection, and COP.

For the TXV case, the expansion-device calculation is replaced by the constant-superheat control procedure. The refrigerant mass flow rate is iteratively adjusted until the prescribed evaporator-outlet superheat is satisfied, after which the remaining system performance parameters are calculated using the same thermodynamic framework. The outputs from both configurations are then compared to evaluate the influence of expansion-device representation under identical nominal operating conditions.

3. RESULTS AND DISCUSSION

The simulation results show that capillary-tube diameter strongly affects refrigerant flow and the overall performance of the R-134a automotive air-conditioning system. As shown in Figure 1, increasing the diameter from 0.8 to 1.4 mm increased the refrigerant mass flow rate from 0.00268 to 0.01212 kg/s, while reducing the pressure drop from 694.22 to 606.63 kPa. The larger flow area reduces hydraulic resistance, allowing greater refrigerant throughput through the expansion device and into the evaporator. Thus, the reduction in pressure loss directly influences the refrigerant circulation rate and subsequently the evaporator performance. This trend agrees with Hmood et al. (2021), while the underlying pressure-loss behavior is consistent with the internal-flow principles reported by Çengel and Boles (2019) and ASHRAE (2022). Unlike studies focusing mainly on capillary-flow characteristics, the present analysis links this hydraulic response directly to the thermodynamic performance of the complete refrigeration cycle.

Figure 2 shows that increasing the diameter increased refrigerant velocity from 4.52 to 6.67 m/s and Reynolds number from 20,323 to 52,502. The increase in velocity indicates that the increase in mass flow rate was sufficiently large to overcome the increase in flow area. The increasing Reynolds number indicates increasingly inertia-dominated flow; however, because refrigerant expansion may involve changes in thermodynamic state and two-phase behavior, Reynolds number alone cannot fully characterize the heat-transfer process. The resulting flow enhancement should therefore be interpreted together with refrigerant properties, pressure, and vapor quality. These observations are consistent with Hmood et al. (2021) and the convective heat-transfer principles of Incropera et al. (2017).

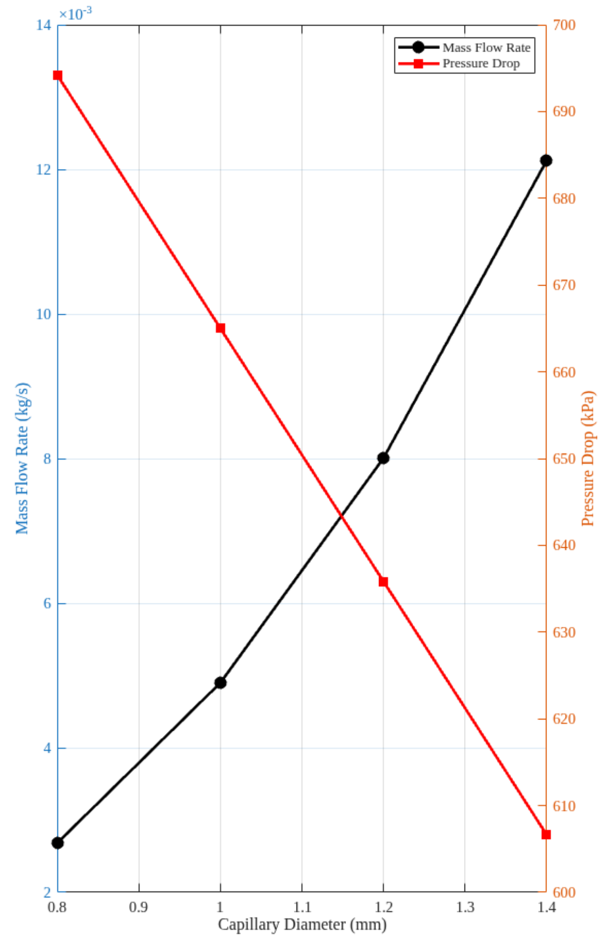

Figure 1. Effect of capillary-tube diameter on refrigerant mass flow rate and pressure drop.

The resulting effect on system performance is presented in Figure 3. Increasing the diameter increased cooling capacity from 0.330 to 1.604 kW and compressor power from 0.088 to 0.320 kW. The cooling capacity can be expressed as indicating that the substantial increase in refrigerant mass flow directly increases the refrigerant heat absorption in the evaporator. Although the higher refrigerant circulation also increases compression work, the increase in cooling capacity is proportionally greater. Consequently, COP increases from 3.751 to 5.013 according to (COP = Qevap/Wcomp. Therefore, the COP improvement results from the combined effects of reduced hydraulic resistance, increased refrigerant circulation, greater evaporator heat absorption, and the relative balance between cooling capacity and compressor work. The observed trend agrees with Alfa et al. (2023), Ridhuan et al. (2024), and Firdaus et al. (2026), although direct quantitative comparison is inappropriate because of differences in system configuration and operating conditions.

Figure 4 compares the capillary-tube and TXV configurations. The capillary-tube model achieved a maximum COP of 5.013, compared with 2.955 for the simplified TXV model. This difference should not be interpreted as evidence that a capillary tube is inherently superior to a TXV. The TXV was represented using a constant-superheat formulation and did not include valve-opening dynamics, sensing-bulb response, hysteresis, or transient superheat regulation. In practical automotive systems, TXVs regulate refrigerant flow according to evaporator superheat and can adapt to changing thermal loads; such behavior has been demonstrated in experimental automotive air-conditioning studies (Alkan et al., 2021). The present result therefore represents a model-to-model comparison under specified steady-state assumptions, rather than a direct comparison of the intrinsic performance of physical expansion devices.

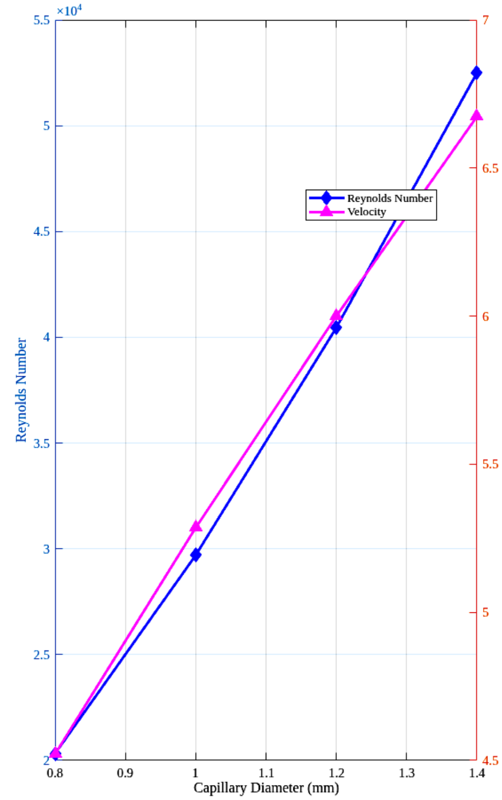

Figure 2. Effect of capillary-tube diameter on refrigerant flow velocity and Reynolds number.

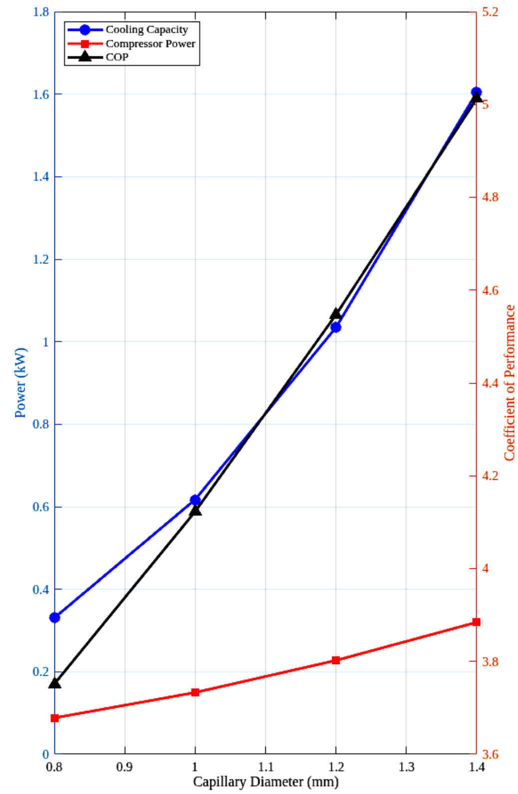

Figure 3. Effect of capillary-tube diameter on cooling capacity, compressor power consumption, and coefficient of performance (COP).

The higher capillary-tube COP results from the model's direct relationship between diameter, hydraulic resistance, and refrigerant mass flow. Within the investigated range, the resulting increase in cooling capacity is greater than the increase in compressor power. Nevertheless, the 1.4 mm diameter should be regarded only as the optimum within the investigated design space, rather than as a universal optimum. A larger diameter could eventually cause excessive refrigerant flow or inadequate pressure reduction depending on compressor, heat-exchanger, refrigerant-charge, and load conditions.

The novelty of the present study lies in the integrated evaluation of expansion-device behavior from refrigerant-flow characteristics to system-level thermodynamic performance. Unlike analyses primarily relating capillary-tube geometry to pressure drop or mass flow rate, the present framework systematically links capillary diameter with pressure drop, mass flow rate, velocity, Reynolds number, cooling capacity, compressor power, and COP under consistent operating conditions. In addition, comparison with a simplified TXV model within the same MATLAB framework demonstrates the sensitivity of predicted system performance to expansion-device modeling assumptions. This provides a practical basis for preliminary expansion-device screening while offering a clearer interpretation of the hydraulic-to-thermodynamic response of the refrigeration system. Nevertheless, the model assumes steady-state operation and simplifies two-phase flow, heat-exchanger characteristics, compressor performance, and TXV dynamics; therefore, the comparison with previous studies represents trend-level rather than quantitative validation. Future work should incorporate matched-condition experimental validation, dynamic TXV modeling, detailed two-phase capillary-flow analysis, calibrated compressor maps, and alternative low-GWP refrigerants.

https://doi.org/10.29303/dtm.v16i2.1338

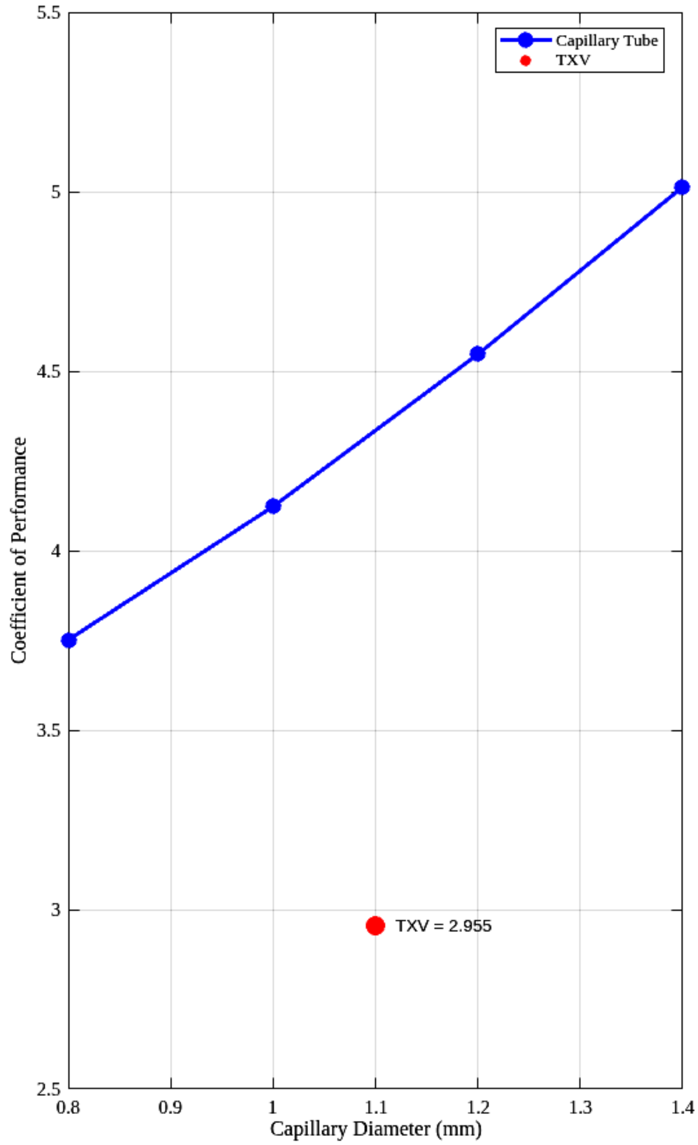

Figure 4. Comparison of the coefficient of performance (COP) between the capillary-tube and thermostatic expansion valve (TXV) systems.

4. CONCLUSION

This study developed a MATLAB-based steady-state numerical model to investigate the effect of capillary-tube diameter on the performance of an R-134a automotive air-conditioning system and to compare it with a simplified constant-superheat Thermostatic Expansion Valve (TXV) model. Increasing the capillary-tube diameter from 0.8 to 1.4 mm increased the refrigerant mass flow rate from 0.00268 to 0.01212 kg/s, reduced the pressure drop from 694.22 to 606.63 kPa, and increased the cooling capacity from 0.330 to 1.604 kW. Although compressor power increased from 0.088 to 0.320 kW, the increase in cooling capacity was proportionally greater, resulting in an increase in predicted COP from 3.751 to 5.013. Thus, within the investigated range and specified operating conditions, the 1.4 mm capillary tube provided the highest predicted COP.

The main contribution of this study is the integrated evaluation of expansion-device behavior by linking capillary-tube diameter with pressure drop, refrigerant mass flow rate, flow characteristics, cooling capacity, compressor power, and COP under consistent operating conditions. The comparison with the simplified TXV model further demonstrates that predicted system performance is sensitive to the mathematical representation of the expansion device. Although the TXV model produced a lower COP of 2.955, this result is strongly influenced by the constant-superheat assumption and the omission of dynamic valve-opening behavior and transient superheat regulation. Therefore, the predicted COP difference should not be interpreted as evidence that capillary tubes are inherently superior to TXVs in practical automotive air-conditioning systems.

The predicted trends are consistent with previous experimental and numerical studies, providing trend-level rather than quantitative validation. The proposed framework can therefore be used for preliminary parametric screening of expansion-device configurations. Nevertheless, the steady-state formulation, simplified TXV dynamics, two-phase flow representation, heat-exchanger characteristics, and compressor performance remain limitations of the present model. Future work should incorporate matched-condition experimental validation, dynamic TXV modeling, detailed two-phase capillary-flow analysis, calibrated compressor performance maps, and alternative low-GWP refrigerants to improve predictive accuracy and practical applicability.

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