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Extending the capillary tube of a propane air-conditioner to reduce the refrigerant charge

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Abstract

Hydrocarbon refrigerants are employed to replace synthetic refrigerants due totheir low global warming and ozone depletion potentials; however, these refrigerants are flammable; therefore, their mass in the system should be minimized to reduce the associated risks. This study deals with the novelidea of extending the capillary tube of a portable air-conditioner in orderto decrease the amount of refrigerant charge in the system. Extending the length of the capillary tube will shorten thelength of the liquid line in applications in which the distance between theoutlet of the condenser and the inlet of the evaporator cannot be reduced. A script was developed and its accuracywas experimentally assessed.It was then used to estimate the amount of charge for the existing design and the extended capillary tube. The results show a significant reduction (63.9%) in the amount of propane in the capillary tubeand liquid line, and a reasonable decrease (8.3%) in the maximumspeed of refrigerant for the airconditioner used in this work.

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*Keywords:*capillary tube; liquid line; charge.

1. Introduction

Synthetic refrigerants are being phased out, and their production will be banned entirely by 2030 due to their global warming and ozone depletion impacts [1]. Hydrocarbons such as propane are used as replacement of synthetic refrigerants due to their negligible global warming, near zero ozone depletion potentials, and their similar thermodynamic properties. However, the amount of propane in the system should be limited due to its flammability.

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In regard to the use of hydrocarbon refrigerants, several technical committees, which are responsible for standards development, have been trying to include the additional safety measures in recent years. The maximum allowable amount of hydrocarbon refrigerants in a system is specified according to room floor area, installation height and lower flammability limit of the refrigerant. A formula that is well accepted by most standards suggests a maximum of approximately 300 grams hydrocarbon refrigerants for portable factory sealed single package air conditioners [2].Recently, some amendments have been proposed for more restrictive regulations [3].

Most previous research focused on decreasing the amount of charges in components with the maximum charge of the system. The results show that a significant mass reduction can be achieved by (a) using miniature heat exchangers [4], (b) increasing the condensing temperature [5]. The length of the capillary tube is expected to have only a negligible impact on the required charge of the system, as capillary tubes contain only a small fraction of the refrigerant charge [6].For this reason, the research in this area has been very limited.

In a previous work [6], it has been shown that reducing the inside diameter of capillary tubes significantly reduces the refrigerant charge in the tube; however, this technique has a negative impact on the overall life of the capillary tube and the noise level by increasing the maximum velocity of the refrigerant. An alternative approach isto use a shorter and helical capillary tube to lower the maximum velocity. This method is effective but not practical if the distance between the outlet of the condenser and the inlet of the evaporator cannot be reduced. Here, shortening the length of capillary tubes extends the length of the tube between the outlet of the capillary tube and the inlet of the evaporator (liquid line). The liquid line has a larger diameter compared to the capillary tube; consequently, this arrangement may actually increase the overall refrigerant charge of the system. In such cases, it is suggested to both extend the capillary tube and eliminate or shorten the liquid line.

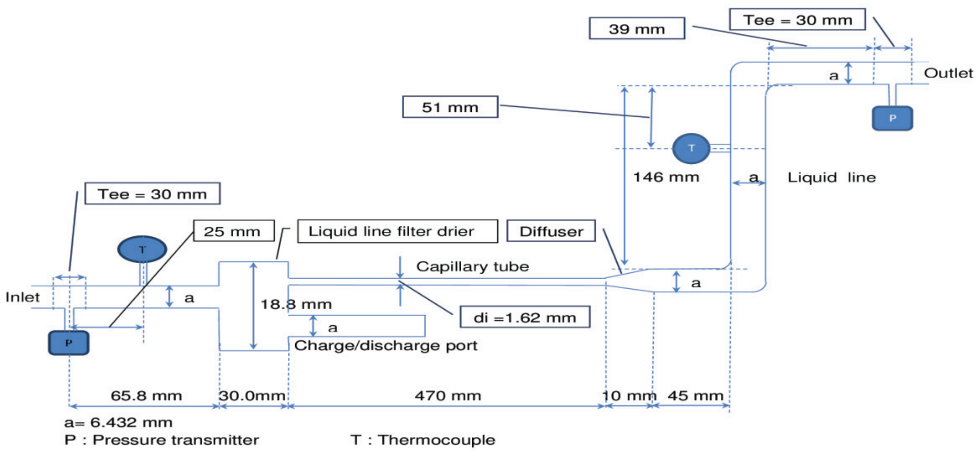

Fig. 1. Schematic of the original capillary tube and instrumentation (not to be scaled)

In the proposed technique, the length of the new capillary tube equals the sum of the length of the original capillary tube and the length of the liquid line. However, in this work, the length of the new capillary tube equals the sum of the original capillary tube and the part of the liquid line which is located between the diffuser and downstream pressure transducer (see Fig.1).This provides an opportunity to compare the computational results with the experimental data. The diameter of the proposed capillary tube is chosen in such a way that the pressure loss remains unchanged in the two cases. This is necessary because increasing the pressure loss can increase the size of the required compressor, which may lead to an increase in the overall refrigerant charge of the system.

2. Methodology

The approach in this study incorporates computational and experimental methods. First, a script was developed to model the capillary tube and liquid line, and then its accuracy was assessed by comparing the computational results with those of experiments. At the next step, the model was used to estimate the mass of propane in the capillary tube and liquid line. It should be noted that the experimental setup in the present study does not provide an explicit measure of the mass of the refrigerant in the system. Therefore, the accuracy of the code was validated based on the measured outlet pressures.

The following assumptions were made for simplicity of the script. The flow in the capillary tubes and liquid line was assumed to be one dimensional, steady, and adiabatic. Thermodynamic data such as enthalpy, density and viscosity of propane were obtained from Reprop 6.02 thermodynamic data base [7] and were set in the script. After defining the geometry of the capillary tube in the code, the inlet flow properties were calculated based on the inlet pressure and temperature. It should be noted that the refrigerant is in a state of saturation except at the entrance of the capillary tube where the refrigerant is in a subcooled state. The calculation in the saturation condition is based on the average properties of the liquid and vapour.

The Colebrook equation is frequently used to determine major head loss coefficient for turbulent flow. In spite of this, previous studies on two phase flows suggest that a more accurate result might be obtained through the use of Churchill formula [8], which is defined as follows,

$$f = 8\left[\left(\frac{8}{Re_D}\right)^{12} + \frac{1}{(A+B)^{1.5}}\right]^{1/12}, \qquad A = \left\{2.457.\,ln\left[\frac{1}{(\frac{7}{Re_D})^{0.9} + 0.27.\frac{e}{D}}\right]\right\}^{16}, \; B = \left(\frac{37530}{Re_D}\right)^{16} \tag{1}$$

Where e denotes roughness (m), D stands for diameter (m), and f and Re_D represents friction factor and Reynolds number, respectively. The properties of refrigerant using McAdams method can be calculated as,

$$\frac{1}{\varphi} = \frac{x}{\varphi_v} + \frac{(1-x)}{\varphi_l} \tag{2}$$

Where ϕ can denote quality (x), viscosity (μ), specific enthalpy (h), thermal conductivity (k), or density (ρ). The subscripts l and v refer to liquid and vapour states, respectively.

In this work, initially both McAdams and Colebrook formulas were used to determine the accuracy, and then the Colebrook formula which shows better results than other method was used for the rest of the study.

The capillary tube was initially divided into equal segments of0.7 mm. The first segment was next to the pressure transducer behind the capillary tube (see Fig. 1). It should be stressed that there is a 2.5 cm gap between the pressure transducer and thermocouple due to technical restrictions. The effect of this gap was ignored in this work, and the measured temperature was assumed to be the inlet temperature of the refrigerant. The final results confirm that the change of temperature in this distance is very small, thus the effect is very minor. The outlet pressure and enthalpy of the segment were initially assumed to be equivalent to those of the inlet properties which were obtained from the measured values. After calculating the average properties, the flow Reynolds number was calculated based on the inside diameter of the tube. All minor head loss coefficients with the exception of the filter were obtained from the reference [9].According to the reference, the minor head loss coefficients were found to be 0.5 for sudden contraction, 0.95 for the sudden expansion at the inlet of the filter, 0.79 for the expansion at the inlet of the diffuser, and0.3 for the elbow. The outlet enthalpy was computed using the first law of thermodynamics. More details of the script can be found in our previous work [6].

The capillary tube and liquid line of the propane air-conditioner modeled in this work are those of a 2.85 kW portable air-conditioner. The length of the nearly straight capillary tube and liquid line are 0.470 m, and 0.295 m, respectively. The capillary tube and liquid line are made of copper with a mean surface roughness of 0.003 mm [9], and have nominal internal diameters of 1.620 mm, and 6.432 mm, respectively.

The measuring instruments include a Coriolis mass flow meter at the inlet of the condenser, two piezoresistive pressure transducers, and two J-type thermocouples at the inlet and outlet of the capillary tube (see Fig. 1). The second thermocouple and pressure transducer are located on the downstream side in the middle of the liquid line at distances of 170 mm and 295 mm from the end of the capillary tube, respectively. As will be seen later, the gap between the thermocouple and pressure transducer does not cause a significant error due to marginal changes of pressure and temperature in the liquid line.

3. Results

The experimental setup in the present study does not provide an explicit measure of the mass of the refrigerant in the system. Therefore, the accuracy of the code was validated based on the measured outlet pressure of the refrigerant at the location of the second pressure transducer in the middle of the liquid line. Table 1 summaries and compares the computational results with the experimental data for five cases. The first and third cases were performed with a charge of 390 gram propane at two ambient temperatures of 35.5˚C and 20.5˚C, respectively, while the other three case studies were conducted with 367.1 gram charge at three ambient temperatures of 20.5 ˚C, 26.5 ˚C, and 35.5 ˚C. It should be noted that the ambient temperature was maintained constant with an error of ±0.5˚C error. For example, in the first case in which the desired temperature was 35.5 ˚C, the temperature varied between 35 ˚C and 36 ˚C, but was on 35.97 ˚C throughout most of the experiment.

Due to lack of sufficient information regarding the head loss coefficients for the filter and charge/discharge ports, the experimental results of the first case were used to estimate their overall minor head loss coefficient by Colebrook's equation. The results show a minor loss of 10.42 provides the best outcomes with zero error for the first case. This value leads to a relative error of -7.96% for the second case, 19.55% for the third case, -23.40% for the fourth case, and 15.01% for the fifth case. The results of the second experiment were used to determine the coefficient of minor loss by McAdams technique. It was found that a value of 2.54 produces zero error in calculation of pressure. However, using this value does not produce any results (cases 1 &3), or presents a relative error greater than that of Colebrook equation (cases 4 & 5). Comparing the accuracy of the two computational approaches, Colebrook method was selected for the rest of this study.

Table 1. Experimental and computational results for different propane charges and ambient temperatures

Case	1	2	3	4	5
Measured ambient temperature (˚C)	35.5±0.5	20.5 ±0.5	20.5 ±0.5	26.5±0.5	35.5±0.5
Measured total charge (gram)	390	367.1	390	367.1	367.1
Measured inlet pressure (MPa)	1.596	1.116	1.273	1.302	1.564
Measured mass flow rate (kg/s)	0.014316	0.011527	0.014156	0.011302	0.013104
Measured inlet temperature (˚C)	35.97	21.15	21.4	26.75	35.84
Measured pressure loss (MPa)	0.968	0.641	0.752	0.748	0.946
Calculated pressure loss (MPa), Colebrook	0.968	0.590	0.899	0.573	0.804
Calculated pressure loss (MPa), McAdams	N/A	0.641	N/A	0.429	0.780
Relative error (%), Colebrook	0.00	-7.96	19.55	-23.40	15.01
Measured outlet pressure (MPa)	0.627	0.475	0.522	0.554	0.618
Calculated outlet pressure (MPa), Colebrook	0.627	0.526	0.374	0.729	0.760
Calculated outlet pressure (MPa), McAdams	N/A	0.475	N/A	0.873	0.784
Relative error (%), McAdams	N/A	0.00	N/A	42.65	17.55
Diameterof extended capillary tube (mm)	1.7	1.7	1.7	1.7	1.7
Charge in extended capillary tube (gram)	2.011	2.135	2.117	2.112	2.027
Charge in existing system (gram)	5.574	6.087	5.851	6.098	5.728
Decrease of propane charge (%)	63.92	64.93	63.81	65.36	64.62

In the second step, the script was run for the existing capillary tube and liquid line at the inlet pressure of 1.596 MPa and inlet temperature of 35.97 ˚C (subcooled condition) (case 1). The flow characteristic and mass of refrigerant in the system were determined and the results are presented in Table 1 and Fig. 2. It should be noted that the presented results are only for the length between the two pressure transducers, and not along the entire length of the capillary tube and liquid line. Consequently, it is expected that the computational results underestimate the reduction of the refrigerant charge (if any) but still makes the point.

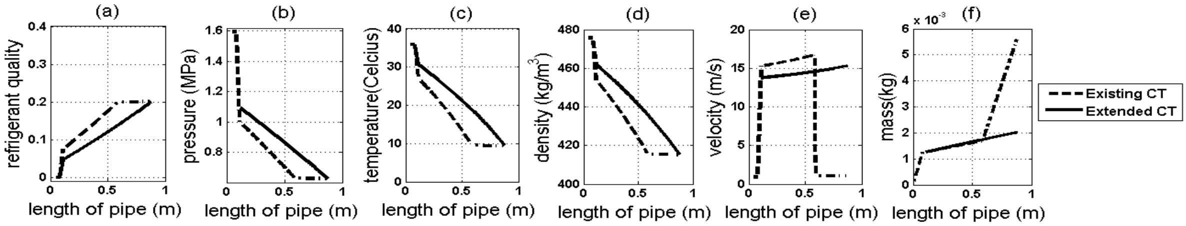

Fig. 2. changes of refrigerant properties along the existing and proposed extended capillary tubes at ambient temperature of 35.5 °C ± 0.5 °C and charge of 390 g, a) quality, b) pressure, c) temperature, d) density, e) velocity, and f) mass.

Figure 2a shows that the refrigerant remains in subcooled state along the pipe before entering the filter of the original capillary tube. In this distance, the changes of pressure, density, temperature, and velocity are negligible (Figs. 2b & 2c & 2d & 2e).At the filter, the minor losses were calculated using equation 4, assuming zero length filter. Nevertheless, Figure 2 includes the length of the filter, assuming a linear variation of properties along the filter.

From the entrance to the end of the filter, the quality increases from zero to 0.076, pressure drops from 1.596 MPa to 1.002 MPa, temperature falls from 35.97 ˚C to 27.04 ˚C, density decreases from 475.73 kg/m^3 to 453.94 kg/m^3, and the velocity increases from 0.926 m/s to 15.30 m/s. The properties continue to change along the capillary tube, and the refrigerant reaches a quality of 0.198, pressure of 633.8 kPa, temperature of 9.81˚C, density of 416.16 kg/m^3, and velocity of 16.69 m/s at the inlet to the diffuser. The change of the quality, pressure, temperature, and density are insignificant along the diffuser and liquid line, but the velocity drops to 1.06 m/s.

In the third step, the extended capillary tube was modelled. The length of the extended capillary tube is 0.765 m which is the sum of the lengths of the original capillary tube and the part of the liquid line located between the two pressure transducers. The diameter of the new capillary tube was adjusted (using trial and error method) until the outlet pressure of the extended capillary tube equalled that of the original configuration. The new diameter is1.70mm.

The results show that the refrigerant experiences similar properties changes to those of the original case along the entrance and filter before entering the capillary tube. As the new capillary tube has a larger diameter, the entrance velocity decreases to 13.75 from 15.30 m/s of the original case. The change of the properties is approximately linear along the new capillary tube, and the properties at the end of the elongated capillary tube are the same as those of the previous case, except for the velocity which is 15.31 m/s, far greater than the original case (1.06 m/s). It should be noted that the velocities along the path where previously the original capillary tube were located are smaller for the extended capillary tube. Conversely, the velocities along the length where the liquid line were situated are much higher than that of the original case. It is interesting to see that the maximum velocity in the case of the extended capillary tube (15.31 m/s) is less than that of the original case (16.69 m/s).

Figure 2f shows the mass of the refrigerant from the entrance of the capillary tube. According to the figure, the rate of the increase of the mass between the upstream pressure transducer and the entrance of the capillary tube, where the refrigerant is in a subcooled state, is sharp and the accumulated mass reaches a value of 1.252 gram in both cases. The rate of increase of the mass along the rest of the original capillary tube is moderate and decreases slightly as the quality increases. At the exit of the original capillary tube, the total mass of refrigerant reaches a value of1.674 gram. For the case of the extended capillary tube at the same distance, the total mass reaches to a value of 1.728 gram which is 3.2% greater than that of the original case. The sudden increase of the mass in the liquid line is due to its larger diameter than that of the extended capillary tube. The total mass accumulated in the original capillary tube and the liquid line is 5.574 grams, whereas the total mass in the extended capillary tube is 2.011 grams, representing a decrease of 63.9%.

In the fourth step, the script was run for the existing and extended capillary tube for remaining cases (cases 2 to 5), and the results are presented in Table 1. The results demonstrate similar characteristics to those in case 1. The overall decrease in the refrigerant charge is 64.93%, 63.81%, 65.36% and 64.62% for the cases 2 to 5, respectively.

4. Discussion

It was found that McAdams method does not always yield a result. Further investigation showed that the calculated pressure drop throughout the tube was high, and the calculated pressure became less than the minimum pressure that was embedded in the script (0.05 MPa).The formula (2) is re arranged and presented as follows,

$$\rho = \frac{\rho_v}{x+(1-x)\frac{\rho_v}{\rho_f}} \tag{3}$$

As the ratio of ρ_v/ρ_f is usually small, the density can be approximated as ρ_v/x. For example, when the quality of refrigerant is 0.5, the density is approximately twice of ρ_g which is lower than that of Colebrook method, in which the density is the average of ρ_v and ρ_f. The low density results in high velocity, which in turn increases the calculated pressure drop.

The results show that in both original and proposed extended capillary tubes, the changes in the temperature, density, and velocity of the refrigerant along the tube are insignificant prior to entering the capillary tube. The refrigerant undergoes significant changes along the capillary tubes, but in the case of the original tube, the rates of the changes decreases inside the liquid line. The main difference between the two tubes is associated with the rate of the changes in the properties of the refrigerant. In the case of the extended tube, the rates at which the properties change are smaller due to its larger diameter. The accumulated mass of the refrigerant in the original capillary tube is slightly less than that of the extended capillary tube along the same length due to its smaller diameter. Nevertheless, due to the smaller diameter of the extended capillary tube compared to the liquid line, the total mass of the refrigerant becomes smaller than that of the original case. At the operational conditions of this study, the total mass of refrigerant was reduced by 63.9%. The actual value of the reduction is expected to be higher if the entire liquid line was modelled. It should be noted that the total mass of the refrigerant in the air-conditioner experiences only a small decrease. In the conditions that this work was carried out, the total mass of the refrigerant decreases from 390 grams to 386.44 grams, representing a reduction of 0.9%.

One interesting point is the decrease of the maximum refrigerant velocity in the extended capillary tube compared to that of the original tube. In the original case, the velocity increases along the capillary tube as the quality of the refrigerant increases. The maximum velocity occurs at the exit of the capillary tube before entering the liquid line which has a larger diameter. The velocity increases continuously along the extended tube which has a constant diameter. Subsequently, the maximum velocity occurs at the end of the capillary tube. In comparison, the maximum velocity of the refrigerant in the extended capillary tube is 8.3% ((16.69-15.31)/16.69) less than that of the original case, which offers an additional advantage over the existing configuration.

5. Conclusion

The computational results support the concept of substituting the capillary tube and liquid line of propane air-conditioners with a larger diameter and longer capillary tube which causes an equivalent head loss. The results show that such a technique is effective in reducing the mass of propane. In addition, another benefit of using the elongated capillary tube and removal of the liquid line isa reasonable decrease in the maximum velocity of refrigerant.

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