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# Mathematical modeling and optimum design for capillary tubes in R-410A Air Conditioner

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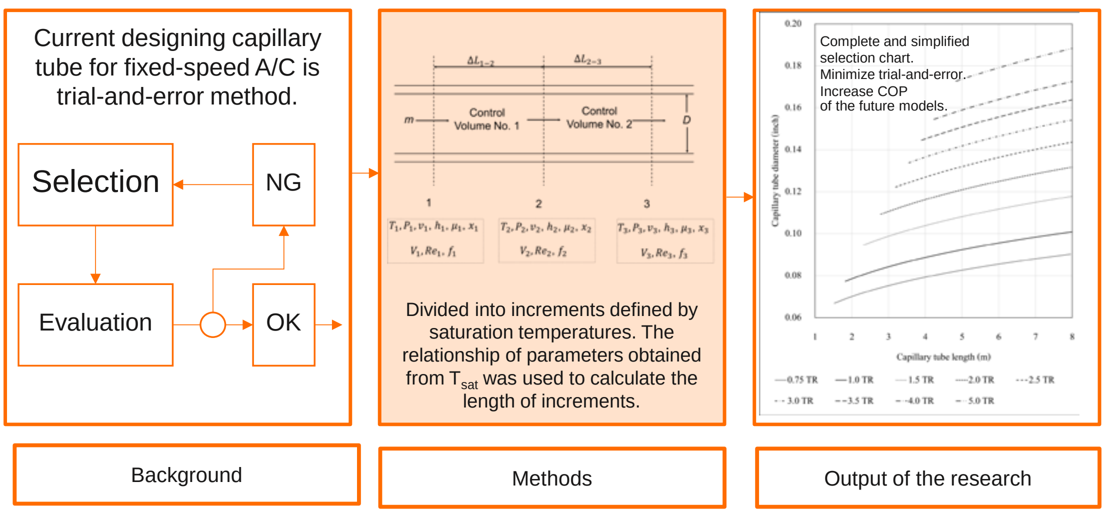
## ABSTRACT




The capillary tube is crucial for vapor compression refrigeration systems to control refrigerant pressure and flow. The selection of capillary tubes is based mainly on the experience of the technicians on the trial-and-error method. Improper and incorrect selections will affect the system's energy efficiency. This research developed a mathematical model of the design of capillary tubes for R-410A refrigeration systems. The model was verified by experiment. The variations in diameter and length of the capillary tube and the effects of these variations were analyzed by using the verified model. The rate of change of refrigerant properties is low during the initial flow and higher when closer to the outlet. The research analyzed the math model with the air conditioner in the 0.75 – 5 TR range. The research outputs the simplified capillary tube selection chart for various cooling capacities which can be conveniently used by the designer. A smaller diameter needs a shorter length but must be larger than the choked flow limit. A larger diameter needs a longer length and resulting in unnecessarily high costs.

*Keywords: vapor compression refrigeration, capillary tube, mathematical modeling, optimum design*

## INTRODUCTION

Air conditioning systems manufactured in Thailand have an export volume of 6,090 million US dollars.[1] Thailand has a total production capacity of all brands of air conditioners at 15 million units per year. 10% are sold domestically and 90% are exported. Thailand's production base is the world's second-largest production base for export. Most of the air conditioners used are air-cooled split types. The air conditioning system consists of 4 main parts: compressor, condenser, evaporator, and metering device.[2,3] The metering device is used to reduce pressure from the high-pressure side (Condensing condition) to the low-pressure side (Evaporating condition). Most of the metering devices are capillary tubes. R-410A refrigerant was selected for this study as a replacement for R-22 refrigerant.[4] R-22 is an HCFC refrigerant that was often used in air-conditioning equipment. The R-22 refrigerants must be discontinued for ODP (Ozone Depletion Potential) reasons. The selection of the capillary tube for the new R-410A becomes the responsibility of the technician and engineer. The knowledge of technicians and engineers in selecting capillary tubes is therefore important. However, the knowledge of sizing the capillary tubes for R-410A is currently limited. The choice of equipment is based mainly on the experience of the operator and the use of trial-and-error. Improper and incorrect selections will result in system malfunctions and also affect the system's energy efficiency.

In this work, a mathematical model of the design of capillary tubes for R-410A refrigeration systems was developed. The model

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evolved from the idea that the refrigerant flowing through the capillary tube changes its properties along the length of the tube. Capillary tube works on the principle that the fluid moves through narrow channels, the pressure drop and kinetic energy increases resulting in lower output pressure. The capillary tube has the advantage of helping to balance the pressure. When the compressor stops, the refrigerant on the high-pressure side will flow through the capillary tube to the low-pressure side. Such a flow causes the system to self-balance the pressure after the system is stopped. Even though capillary tubes cannot operate in a wide range like thermostatic expansion valves, capillary tubes are more economical than other metering devices. Capillary tubes, therefore, are still popular for general fixed-speed refrigerators and air conditioners. The capillary tube is theoretically not compatible with the inverter-type, since it is required different geometry at different operating speeds to optimize the system efficiency[5].

A capillary tube is fixed hardware and a fixed design. Once a device is selected, no customizations can be made. The device size selection must be based on average operating conditions. If the capillary tube selection does not correspond to the actual operating conditions, it will cause problems with interconnected systems[6]. The consequence greatly affects the performance and efficiency of the system. Capillary tubes are sized by the refrigerant mass flow rate[7]. The behavior of refrigerant throttling control creates a differential pressure between the high-pressure side and the low-pressure side. The relationship between capillary tube length and pressure drop is nonlinear[8,9]. This is due to the complexity of friction and the changing refrigerant properties along the capillary tube length. The flow is a two-phase flow and requires sophisticated analysis.[10,11]

The ASHRAE (American Society of Heating, Refrigerating and Air-Conditioning Engineers) standard recommends the selection of capillary tubes[12–14]. ASHRAE developed refrigerant-specific rating charts to predict refrigerant flow rates through adiabatic capillary tubes. These two charts are the mass flow rate chart and the geometric correction factor chart. The mass flow rate chart plots the capillary tube flow rate as a function of inlet condition and inlet pressure for a reference capillary tube geometry of 0.034-inch inner diameter and 130-inch length. The procedure starts with determining the capillary tube inlet pressure condition in conjunction with the inlet condition. The result is the mass flow rate for the reference capillary tube. The next procedure continues with determining capillary tube size (inner diameter and length). The result is the flow factor for mass flow correction. The two quantities necessary to determine the refrigerant flow rate through the capillary tube are the flow rate through the reference capillary tube and the flow factor, which is a geometric correction factor. These two quantities are multiplied together to calculate the flow rate.

In practical design, the designer obtained the refrigerant flow rate from the cooling load analysis first. The flow parameters of the refrigerant were analyzed for the required pressure drop and temperature. The analysis output can be obtained as the inner diameter and length of the capillary tube. Using the ASHRAE standard in the selection of capillary tubes it is necessary to proceed in a trial-and-error manner until the proper size is obtained. Take the results to further laboratory tests to adjust the size until the desired cooling efficiency is achieved. There are several limitations to using charts. The analysis applies only to the refrigerant types available in the chart. The inner diameter and length of the capillary tube are limited only to the range of values provided. Other than the data in the chart, it cannot be selected or analyzed. This research has made the design of the capillary tube practically functional. The analysis is complicated at first but saves time in trial and error in testing for optimization. The method can be applied to all refrigerants without limitation on the inner diameter and length of the capillary tube.

## MATERIALS AND METHODS

### Mathematical Modeling

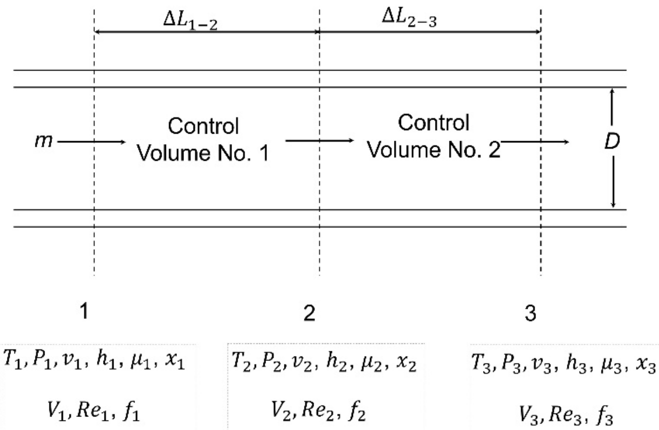


**Figure 1**. The schematic diagram for control volume in the analysis

The capillary tube is divided by finite control volumes. Each of them starts at a given saturation temperature and ends at a lower saturation temperature[15,16]. The first control volume starts at the air conditioner's designed saturation condensing temperature, and the last control volume ends at the designed saturation evaporating temperature. From the properties obtained by a saturation temperature, we can obtain various parameters that can output the length of each control volume as described below.

The conservation of energy law applies to the first control volume. The flow is a steady flow process and there is a change in enthalpy with a change in kinetic energy.

$$1000h_1 + \frac{V_1^2}{2} = 1000h_2 + \frac{V_2^2}{2} \quad (1)$$

The law of mass conservation of mass shows that the flow rate of the refrigerant mass depends on the flow velocity, the cross-sectional area of the pipe, and the specific volume.

$$m = \frac{V_1 A}{v_1} = \frac{V_2 A}{v_2} \quad (2)$$

Since the tube's cross-sectional area is always constant, the ratio between the refrigerant mass flow rate and the tube's cross-sectional area is also constant through every control volume.

By adjusting the equation, it can be seen that the ratio between the refrigerant velocity and the specific volume at any point in the capillary tube system is always constant. This feature will be useful in the further numerical analysis of the model.

$$\frac{m}{A} = \frac{V_1}{v_1} = \frac{V_2}{v_2} \quad (3)$$

At the control volume inlet (point 1), the flow velocity $V_1$ can be analyzed from,

$$V_1 = v_1 \frac{m}{A} \quad (4)$$

At the entrance of control volume (point 1), from the velocity of refrigerant, $V_1$, dynamic viscosity, specific volume and Reynold number can be analyzed as,

$$Re = \frac{VD}{\mu v} \quad (5)$$

The friction factor can be traditionally analyzed by Colebrook's equation. Which is described as[17],

$$\frac{1}{\sqrt{f}} = -2\log\left(\frac{e}{3.7D} + \frac{2.51}{Re\sqrt{f}}\right) \quad (6)$$

Colebrook's equation is widely accepted as the benchmark equation to solve Darcy's friction factor. However, its form can only be solved by iteration, which consumes considerable time and resources and the initial guess is also required. Several researchers provide alternative equations[15,18–27], Blasius's equation was among the most popular to solve Darcy's friction factor, which is described as[16],

$$f = \frac{0.33}{Re^{0.25}} \quad (7)$$

However, recent research shows that Blasius's equation's validity range is limited, due to its lack of consideration for the surface roughness of the tubes and its relative error on the larger Reynold's number, especially greater than 100 000, is large compared to the solution obtained from the Colebrook's equation. This research applies Swamee-Jain's equation, which is proved to have an acceptable relative error to Colebrook's equation, can be solved directly and its form is relatively simple.[28,29]

$$f = 0.25\left[log\left(\frac{e}{3.7D} + \frac{5.74}{Re^{0.9}}\right)\right]^{-2} \quad (8)$$

This mathematical model can be applied to refrigeration and air conditioning systems. It can work with a wide range of refrigerants and different operating conditions. In this research, case studies from air conditioning system work will be used. The analysis was initiated by determining the cooling capacity and evaporator and condenser operating conditions. Based on this information, it is possible to analyze how much refrigerant flow rate the cooling system needs. The first analysis assumes the beginning of the refrigerant flows into the capillary tube. Refrigerant is saturated liquid at condensing temperature and pressure conditions. Saturated liquid conditions of specific volume, specific enthalpy, and viscosity are used as the starting point of the calculation. Parameters at point 1 of the control volume were calculated from the cooling capacity, fluid flow rate, capillary tube diameter, and initial saturation state. The parameters that are calculated and applied for point 1 of the control volume include refrigerant flow speed, Reynold number and friction factor.

The output speed of control volume 1 is shown as follows.

$$V_2 = v_2 \frac{m}{A} \quad (9)$$

Substitute $V_2$ in the conservation of energy equation,

$$1000\, h_1 + \frac{V_1^2}{2} = 1000\, h_2 + \frac{v_2^2}{2}\left(\frac{m}{A}\right)^2 \quad (10)$$

As the refrigerant flows through the capillary tube, the pressure and temperature continuously decrease with the flow distance. Two-phase flow behavior will begin to develop continuously. The refrigerant changes state from liquid to vapor along the flow line. The proportion of refrigerant vapor will increase gradually and affect the specific enthalpy, h, and specific volume, v. Substitute h, v in the conservation of energy equation, we obtain,

$$\left\{1000(h_{f2}) + 1000(h_{g2} - h_{f2})x + \frac{(v_{f2} + (v_{g2} - v_{f2})x)^2}{2}\left(\frac{m}{A}\right)^2\right\} = \left\{1000\, h_1 + \frac{v_1^2}{2}\right\} \quad (11)$$

The equation is in the form of a quadratic equation. The proportion of refrigerant vapor can be analyzed from,

$$x = \frac{-b \pm \sqrt{b^2 - 4ac}}{2a} \quad (12)$$

where,

$$a = (v_{g2} - v_{f2})^2\left(\frac{m}{A}\right)^2\left(\frac{1}{2}\right) \quad (13)$$

$$b = 1000(h_{g2} - h_{f2}) + \left(v_{f2} + (v_{g2} - v_{f2})\right)\left(\frac{m}{A}\right)^2 \quad (14)$$

$$c = 1000(h_{f2} - h_1) + \left(\frac{m}{A}\right)^2\left(\frac{1}{2}\right)v_{f2}^2 - \left(\frac{V_1^2}{2}\right) \quad (15)$$

When analyzing the proportion of refrigerant vapor is done, analysis for specific enthalpy, specific volume, and the viscosity of the solution for point 2 (outlet of control volume 1) will continue using the saturation properties obtained from equations developed in previous researches[30–33].

Analysis of refrigerant properties of point 2 (outlet of control volume 1). The input of the calculation is derived from the ratio between the refrigerant mass flow rate and the cross-sectional area of the tube, the capillary tube diameter, and the properties of the refrigerant inlet (point 1) and the outlet (point 2) of the control volume 1. The parameters from point 1 include velocity and specific enthalpy. The parameters from point 2 are temperature, pressure, specific volume, specific enthalpy, and viscosity.

The conservation of momentum equation applies to control volume 1. The equation explains that the difference in pressure drop at point 1 and point 2 (differential pressure) minus drag force times the cross-sectional area of the capillary tube is equal to the resultant force acting on control volume. The total force acting on the control volume is equal to the difference in the momentum acting on the control volume.

$$\left((p_1 - p_2) - f\frac{\Delta L}{D}\frac{V^2}{2v}\right)A = m(V_2 - V_1) \quad (16)$$

As a result of the previous analysis, the ratio between the refrigerant velocity and the specific volume at any point in the capillary tube system is always constant. Substituting this ratio in the drag force equation, we obtain

$$f\frac{\Delta L}{D}\frac{V^2}{2v} = f\frac{\Delta L}{D}\frac{V}{2}\left(\frac{V}{v}\right) = f\frac{\Delta L}{D}\frac{V}{2}\left(\frac{m}{A}\right) \quad (17)$$

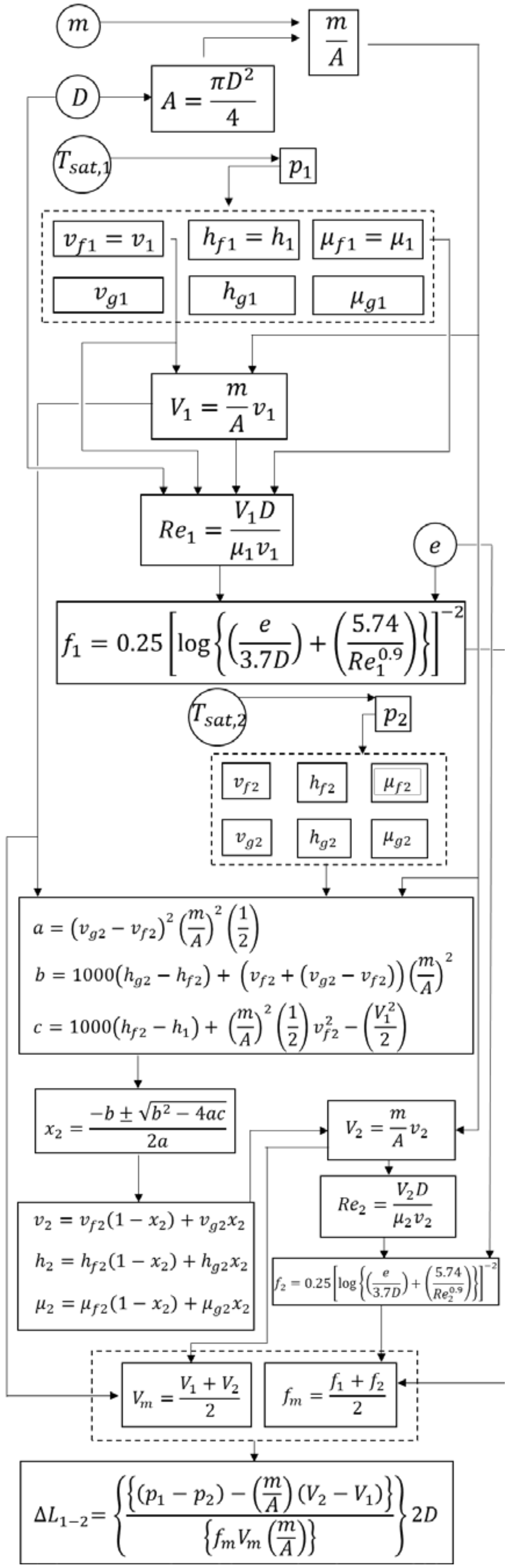


**Figure 2**. The analysis of capillary tube length of control volume 1, the primary parameters are in circles.

The analytical velocity for this resistance pressure equation must be the average velocity from the control volume inlet (point 1) to the control volume outlet (point 2).

$$V_m = \frac{V_1+V_2}{2} \tag{18}$$

The coefficient of friction used in this analysis for the resistance pressure equation is the average coefficient of friction from the control volume inlet (point 1) to the control volume outlet (point 2).

$$f_m = \frac{f_1+f_2}{2} \tag{19}$$

Substitute the mean velocity, $V_m$, and the mean coefficient of friction, $f_m$, in the conservation of momentum equation.

$$\left\{(p_1 - p_2) - f_m \frac{\Delta L}{D}\frac{V_m}{2}\left(\frac{m}{A}\right)\right\} A = m(V_2 - V_1) \tag{20}$$

$$(p_1 - p_2) - f_m \frac{\Delta L}{D}\frac{V_m}{2}\left(\frac{m}{A}\right) = \left(\frac{m}{A}\right)(V_2 - V_1) \tag{21}$$

$$\Delta L_{1-2} = \left\{\frac{\left\{(p_1-p_2)-\left(\frac{m}{A}\right)(V_2-V_1)\right\}}{\left\{f_m V_m\left(\frac{m}{A}\right)\right\}}\right\} 2D \tag{22}$$

The analysis of tube length between points 1-2 (control volume 1) is an analysis of the relationship between the inlet (point 1) and outlet (point 2) of control volume 1. The input of the calculation is derived from the ratio between the refrigerant mass flow rate, the tube cross-sectional area, the capillary tube diameter, the properties of the refrigerant inlet (point 1) and the outlet (point 2) of the control volume 1. The parameters from point 1 include pressure, velocity, and coefficient of friction. The parameters from point 2 are pressure, specific volume, and viscosity. The analysis pattern of the first control volume can be illustrated with a schematic diagram as shown in **figure 2.**

For control volume 2 and after, outlet parameters of control volume 1 (or the previous control volume) become inlet parameters for control volume 2. Analysis of refrigerant properties of point 3 (output of control volume) can be performed. The input of the calculation is derived from the difference between the refrigerant mass flow rate, the pipe cross-sectional area, the capillary tube diameter and the properties of the refrigerant inlet (point 2) and outlet (point 3) of the control volume. The pattern of pipe distance analysis between points 2-3 (control volume 2) is shown in a schematic diagram in **figure 3** and **figure 4**.

The analysis of tube length between points 2-3 (control volume 2) is an analysis of the relationship between the inlet (point 2) and outlet (point 3) of control volume 2. The input of the calculation is derived from the ratio between the refrigerant mass flow rate, the tube cross-sectional area, the capillary tube diameter, the properties of the refrigerant inlet (point 2) and the outlet (point 3) of the control volume 2. The parameters from point 2 include pressure, velocity, and coefficient of friction. The parameters from point 3 are pressure, specific volume, and viscosity.

The process continues until the desired evaporating temperature is reached. Smaller control volumes consume greater calculation resources. While larger control volumes cause a more residual error.

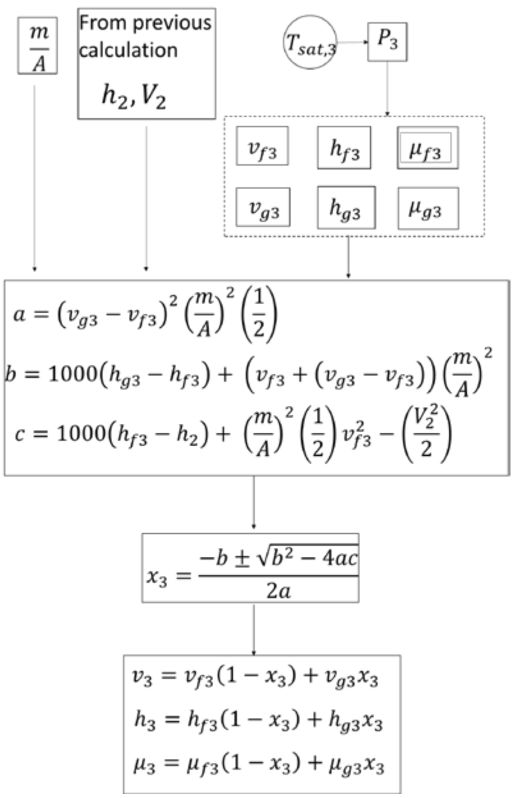


**Figure 3.** Analysis for point 3 (control volume 2 outlet)

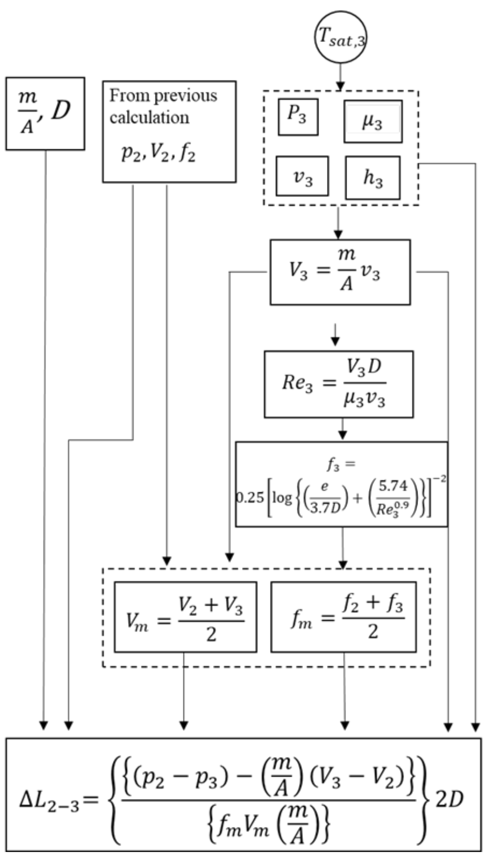


**Figure 4.** Analysis for control volume 2

**Validation of mathematical by an experiment model and case study**

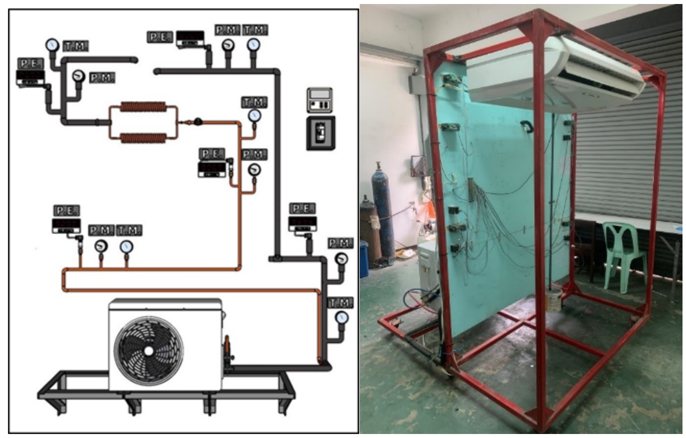

**Figure 5** Experimental setup and instrumentation

A test rig for the air-cooled split type as **figure 5** was built to validate the mathematical model. Split type air-cooled air conditioner with size 1 TR (Ton of Refrigeration) uses R-410A refrigerant. Condensing unit and fan coil unit are mounted with a panel in between. The panel allows air circulation on the FCU (fan coil unit) side without affecting the cooling of the CDU (condensing unit). The installation of pipes and fittings is following the standards of the air conditioner manufacturer. There are 5 temperature and pressure sensors in each pipe section. Two capillary tubes of 0.055-inches diameter, each 4.5-meters long, were installed for use in the experiment. The amount of refrigerant is charged into the system according to the standards of the air conditioner manufacturer. The system works by setting the evaporating temperature by the thermostat to the lowest setting to keep the compressor at full load condition continuously. The cooling capacity analysis was performed by measuring the air side of the FCU and determining the airflow through the evaporator coil during steady-state conditions. Inlet air and outlet air parameters (dry bulb temperature, relative humidity, volume flow rate) FCU were measured.

## RESULTS AND DISCUSSIONS

**Verification experiment results and case study discussion**

**Table 1** Measured air side parameters at FCU

| Air properties | unit | Entering FCU | Leaving FCU |
|---|---|---|---|
| Dry bulb temperature | °C | 32.60 | 26.82 |
| Wet bulb temperature | °C | 26.99 | 25.36 |
| Relative humidity | % | 65.00 | 89.10 |
| Humidity ratio | g.vapor/kg.dry air | 20.37 | 20.00 |
| Air density | kg/m3 | 1.14 | 1.16 |
| Air Enthalpy | kJ/kg | 84.94 | 77.97 |

**Table 1** shows that the air velocity leaving from FCU or supply air is 700 fpm. The net area of the supply air grill is 0.60 ft$^2$. The airflow rate from FCU equals 420 CFM or 195 LPS. These

parameters along with psychrometric analysis can be used to analyze the cooling capacity of the fan coil unit. It was found that the sensible heat is 1.411 kW and the latent heat is 0.224 kW. The sensible heat ratio or the ratio between sensible heat and total heat is 0.863. The cooling capacity from the air side of FCU is 1.634 kW or 0.464 Ton of refrigeration. The heat transfer is passed from the air side to the refrigerant side. Inlet refrigerant and outlet refrigerant parameters (pressure, temperature) at the evaporator coil was measured. The measurement results of the refrigerant system are as **table 2**.

**Table 2** Measured refrigerant side parameters for FCU

| Refrigerant properties | unit | Value |
|---|---|---|
| **Point 1: Outlet from the evaporator** | | |
| Pressure | kPa abs | 877 |
| Temperature | °C | 3.0 |
| Refrigerant Enthalpy | kJ/kg | 421.70 |
| **Point 2: Outlet from the compressor** | | |
| Pressure | kPa abs | 2,300 |
| Temperature | °C | 53.1 |
| Refrigerant Enthalpy | kJ/kg | 447.80 |
| **Point 3: Outlet from the condenser** | | |
| Pressure | kPa abs | 877 |
| Temperature | °C | 38.0 |
| Refrigerant Enthalpy | kJ/kg | 262.00 |
| **Point 4: Outlet from the capillary tube** | | |
| Pressure | kPa abs | 877 |
| Temperature | °C | 3.0 |

Condenser operating condition was 38°C due to outdoor air testing. The operation of the compressor and condenser is therefore dependent on the weather conditions on the test day. Evaporator operating condition was 3°C due to system balancing between CDU and FCU. However, the system operated at 100% continuous full load condition. Temperature and pressure measurements were performed when the system reached a steady state. When the heat exchanger efficiency of the evaporator is 85%[34], the cooling capacity from the evaporator coil is 0.546 TR. These parameters, cooling capacity, and Pressure-Enthalpy diagram can be used to analyze the refrigerant mass flow rate at the evaporator. The change in refrigerating effect is 159.7 kJ/kg. the cooling capacity from the evaporator coil is calculated as 0.546 Ton of Refrigeration or 1.923 kW. And the mass flow rate is calculated to be 0.012 kg/s. The experiment was conducted with two capillary tubes of diameter 0.055 inches, with a length of 4.5 meters per line. Hence, each line of the capillary tube supported a flow rate of 0.006 kg/s.[35] And the results from the analysis by mathematical modeling were consistent in the same direction of 0.006 kg/s refrigerant flow rate. This validates the mathematical model.

The observations from the experiment revealed the following discussions,

The selection of capillary tubes from the test was based on the available sizes on the market. The first selection was not analyzed by a mathematical model.[36] As a result, the flow of the capillary tube is not following the specification it should. The mass flow rate and cooling capacity are lower than they should be. In this case, the cooling capacity is reduced from 1 TR down to 0.546 TR, or 54.6% of the capacity according to the specification. This reflects the selection of unanalyzed capillary tubes. This results in problems with cooling capacity and COP. However, the mathematical model in this study is also possible to analyze parameters related to capillary tubes precisely. This ensures further use in design and research.

**Variation of parameters on any length of a capillary tube**

An analysis of parameter changes over the length of the capillary tube is presented in this section. An air conditioner with a cooling capacity of 1 TR or 12,000 BTU/hour was used in this research. Evaporator's operating conditions are designed at 5°C due to its suitability for the condition of supplying air at the front of the cooling coil in the range of 10-15°C. At this condition, thermal comfort of 25°C and 50% RH in a typical room can be optimally controlled. The condenser operating conditions are designed at 50°C due to its suitability with the air-cooled condenser air conditioner. The research was conducted in Thailand with an average ambient temperature of 35°C. The operating conditions of the evaporator and condenser were analyzed in the P-h chart to obtain the properties of the R-410A refrigerant system. This makes it possible to analyze that the 1 TR cooling system requires a refrigerant flow rate of 0.0256 kg/s.

**Table 3** Parameters for refrigerant mass flow rate analysis

| Initial parameters | unit | value |
|---|---|---|
| Evaporator temperature | °C | 5.00 |
| Condensing temperature | °C | 50.00 |
| Evaporator absolute pressure | kPa | 932.45 |
| Condenser absolute pressure | kPa | 3,056.40 |
| Cooling capacity | kW | 3.52 |
| Refrigerant mass flowrate | kg/s | 0.0256 |

The refrigerant flow rate and capillary tube diameter data were fed into the input of the mathematical model. A computer simulation requires that each control volume has a refrigerant flow and a decrease in temperature of 1°C per control volume. Point 1 is the saturated temperature of 50°C. Point 2 has a temperature of 49°C and point 3 has a temperature of 48°C, respectively. Model analysis was performed continuously at 45 control volumes to reduce the temperature from 50°C to 5°C.

In addition to the study of parameter changes, a comparative study of capillary tube sizes was also performed. Three sizes of capillary tubes were selected for analysis and were compared with the results of various parameters. Capillary tubes come in three sizes with internal diameters of 0.080, 0.087, and 0.094 inches. Since the velocity of the flow is high and the choked flow conditions easily occur[37], the design, therefore, requires caution for chocked flow to prevent such conditions from occur[38]. The choked flow condition can be verified by calculating the incremental length of the capillary tube to be negative. This phenomenon is impractical and manifests itself in the non-conforming analysis of the laws of thermodynamics and the principles of conservation of momentum.

If a choked flow condition occurs, the operation of the capillary tube will not comply with the laws of thermodynamics and harm the system's efficiency.

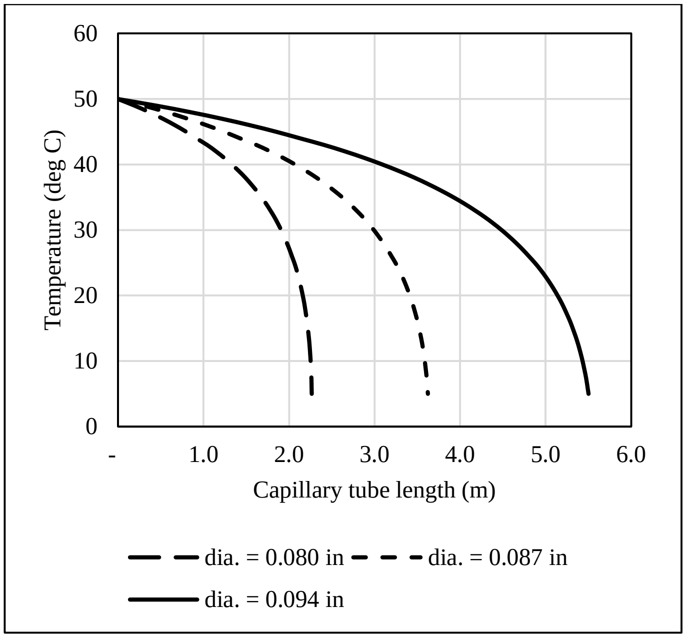


**Figure 6.** The relationship between the temperature and the length of the capillary tube

From **figure 6**, it can be seen that the refrigerant temperature decreases at a higher rate when it flows closer to the outlet. Data of the 0.094 inches tube are, from 0 to 1 m, the temperature drops from 50 °C to 47.5 °C with a rate of -2.5 °C/m. While from 4 to 5 m, the temperature was reduced from 34 °C to 22 °C with a rate of -12 °C/m. Which is 4.8 times greater.

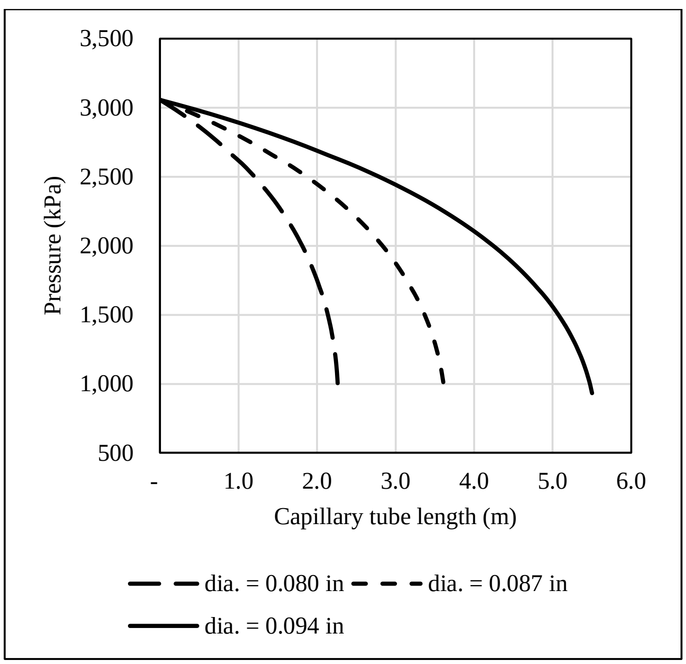


**Figure 7.** The relationship between the pressure and the length of the capillary tube

From **figure 7**, the refrigerant pressure decreases at a higher rate when it flows closer to the outlet. Data of the 0.094 inches tube are, from 0 to 1 m, the pressure drops from 3,056 kPa to 2,891 kPa with a rate of -165 kPa/m. While from 4 to 5 m, the pressure was reduced from 2,083 kPa to 1,522 kPa with a rate of -561 kPa/m. Which is 3.4 times greater.

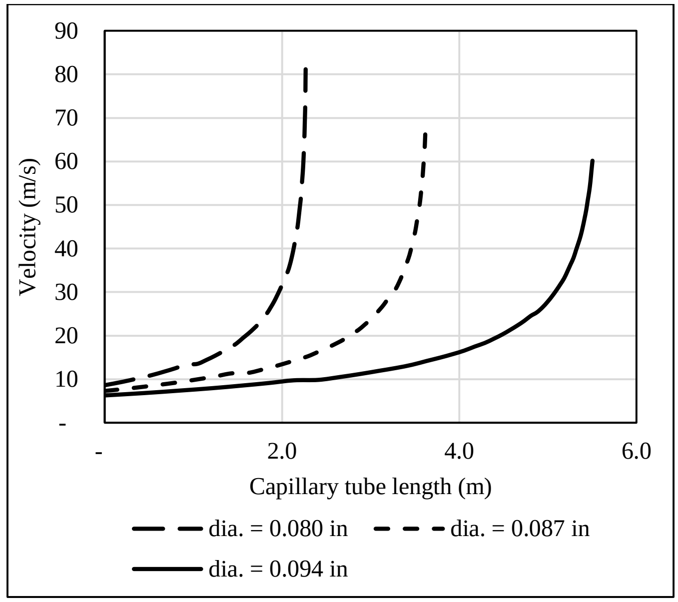


**Figure 8.** The relationship between the velocity and the length of the capillary tube

From **figure 8**, the refrigerant velocity increases at a higher rate when it flows closer to the outlet. Data of the 0.094 inches tube are, from 0 to 1 m, the velocity increases from 7.33 m/s to 9.23 m/s with a rate of 1.90 m/s/m. While from 4 to 5 m, the velocity increased from 17.38 m/s to 38.41 m/s with a rate of 21.03 m/s/m. Which is 11.06 times greater.

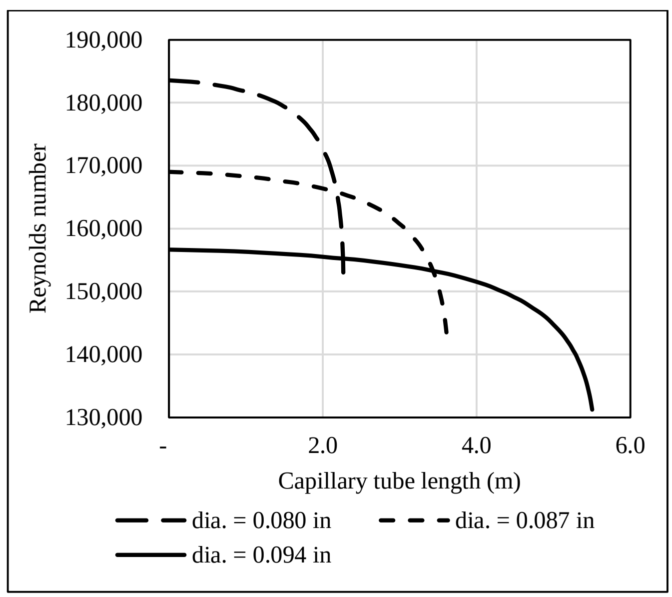


**Figure 9** The relationship between the Reynolds number and the length of the capillary tube

From **figure 9**, the Reynolds number decreases at a higher rate when it flows closer to the outlet. Data of the 0.094 inches tube are, from 0 to 1 m, the Reynolds number drops from 156,643 to 156,288 with a rate of -355/m. While from 4 to 5 m, the Reynolds number was reduced from 151,316 to 144,134 with a rate of -7,182/m. Which is 20.23 times greater.

Reynolds number is a dimensionless combination of variables. It is important in the analysis of viscous flow through capillary tubes. Reynolds number is derived from the ratio between the inertia force on an element of fluid to the viscous force on it. When these two types of forces are important, the Reynolds number will play an important role. When the Reynolds number is large, this is an indication that the viscous effects are small relative to inertial effects. In this case, the flow is in the range of moderate values of the Reynolds number. According to Moody's chart, the friction factor is dependent on both the Reynolds number and relative roughness.

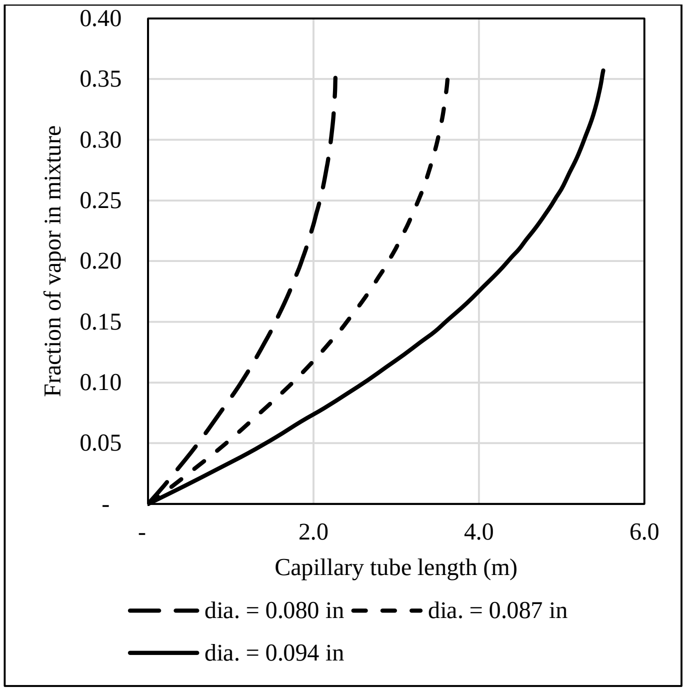


**Figure 10.** The relationship between the fraction of vapor and the length of the capillary tube

From **figure 10**, the fraction of vapor in mixtures increases at a higher rate when it flows closer to the outlet. Data of the 0.094 inches tube are, from 0 to 1 m, the fraction of vapor in mixtures increases from 0 % to 3.50 % with a rate of 3.50 %/m. While from 4 to 5 m, the fraction of vapor in mixtures increased from 17.90 % to 26.60 % with a rate of 8.70 %/m. Which is 2.49 times greater.

From **figure 11**, the specific volume increases at a higher rate when it flows closer to the outlet. Data of the 0.094 inches tube are, from 0 to 1 m, the specific volume increases from 0.00110 m3/kg to 0.00124 m3/kg with a rate of 0.00014 m3/kg/m. While from 4 to 5 m, the specific volume was increased from 0.00290 m3/kg to 0.00508 m3/kg with a rate of 0.00218 m3/kg/m. Which is 15.57 times greater.

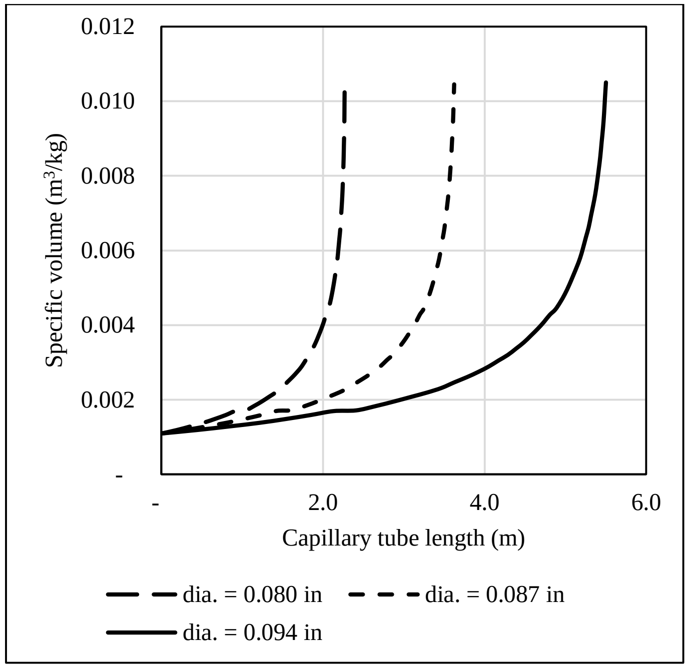


**Figure 11.** The relationship between the specific volume and the length of the capillary tube

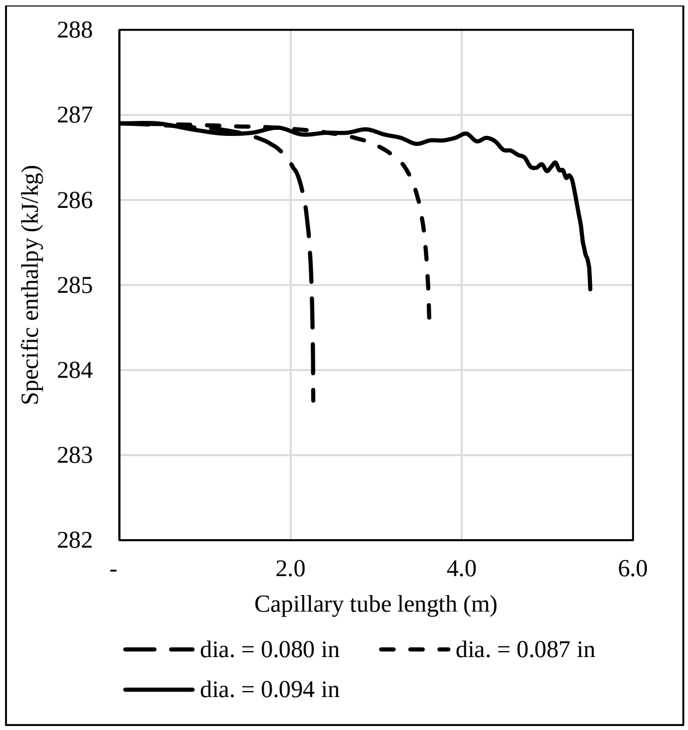


**Figure 12.** The relationship between the specific enthalpy and the length of the capillary tube

From **figure 12**, the specific enthalpy decreases at a higher rate when it flows closer to the outlet. Data of the 0.094 inches tube are, from 0 to 1 m, the specific enthalpy drops from 286.90 kJ/kg to 286.81 kJ/kg with a rate of -0.09 kJ/kg.m. While from 4 to 5 m, the specific enthalpy was reduced from 286.78 kJ/kg to 286.39 kJ/kg with a rate of -0.39 kJ/kg/m, 4.33 times greater.

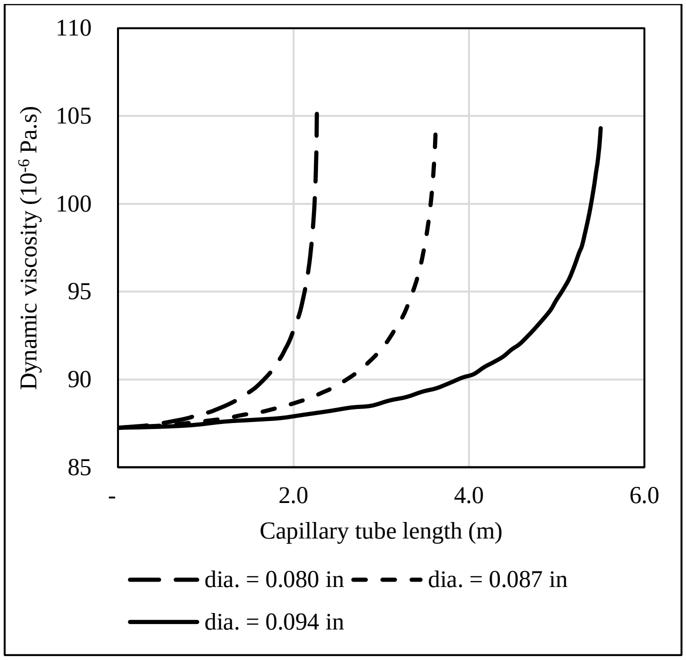


**Figure 13.** The relationship between the dynamic viscosity and the length of the capillary tube

From **figure 13**, the dynamic viscosity increases at a higher rate when it flows closer to the outlet. Data of the 0.094 inches tube are, from 0 to 1 m, the dynamic viscosity increases from 87.25 x $10^{-6}$ Pa.s to 87.50 x $10^{-6}$ Pa.s with a rate of 0.25 x $10^{-6}$ Pa.s/m. While from 4 to 5 m, the dynamic viscosity was increased from 90.30 x $10^{-6}$ Pa.s to 94.90 x $10^{-6}$ Pa.s with a rate of 4.60 x $10^{-6}$ Pa.s/m. Which is 18.4 times greater.

The value of the dynamic viscosity depends on temperature. In the general case of a single-phase fluid with constant pressure, the viscosity of the substance in the liquid state is inversely proportional to the temperature of the substance. Lower temperatures for liquid result in higher viscosity. This phenomenon can be described. Liquid molecules are linked by cohesive force. The lower the temperature, the higher the cohesive force. An increase in cohesive force creates a greater resistance to flow. This results in increased viscosity. On the contrary for the single-phase fluid with constant pressure, the viscosity of substance in the gaseous state varies with the temperature. Lower temperatures for gas result in lower viscosity. This phenomenon can be described. Gas molecules are positioned farther apart than liquid molecules. The exchange of momentum of the molecules produces the resistance of the flow. Lower temperatures result in the lower exchange of molecular momentum and resistance to flow. This results in a decrease in viscosity.

In this research, the refrigerant flow is a two-phase flow with a continuous state change. There is a change in pressure and temperature along the flow line through the capillary tube. A refrigerant transition causes the substance to form a mixture of liquid and gas. At the beginning of the capillary tube, the liquid refrigerant has a higher mass ratio than the gas refrigerant, viscosity is dominated by liquid only. At the end of the capillary tube, liquid refrigerant and gas refrigerant are mixed and the viscosity is dominated by both liquid and gas. As a result, the end of the capillary tube has a much higher viscosity. The results of the analysis showed that the two-phase flow behavior was complex. Single-phase flow analysis was insufficient to analyze this phenomenon. Complexity increases with changes in pressure and mass ratio along the flow line. Therefore, this model is very useful in two-phase flow analysis.

**Optimum Sizing of Capillary Tube**

The mathematical model can be used to select a suitable capillary tube. This method differs from the ASHRAE standard method and has no defined diameter or length restrictions. The choked flow can be monitored and prevented at the design stage. Choosing a size that is suitable for the commercial size and economical price can be performed. This research is scoped to the cooling capacity of 0.75 - 5 TR. Initial data of capacity, operating conditions, and refrigerant properties were analyzed for refrigerant mass flow rate. Evaporator operating conditions are designed at 5°C, condenser operating conditions are designed at 50°C. The analyzed data are shown in **Table 4**. The mass flow rate and working conditions of the system were used as inputs to the mathematical model. For each cooling capacity, the results from the simulation are a data set consisting of the diameter and length of the capillary tube. The optimum size for various cooling capacities is shown in **figure 14**.

**Table 4**: Mass flowrate for various cooling capacity

| Cooling Capacity | | | Mass Flowrate | |
|---|---|---|---|---|
| **TR** | **BTU/hr** | **kW** | **kg/s** | **TR** |
| 0.75 | 9,000 | 2.64 | 0.0192 | 0.75 |
| 1.0 | 12,000 | 3.52 | 0.0256 | 1.0 |
| 1.5 | 18,000 | 5.28 | 0.0384 | 1.5 |
| 2.0 | 24,000 | 7.03 | 0.0512 | 2.0 |
| 2.5 | 30,000 | 8.79 | 0.0639 | 2.5 |
| 3.0 | 36,000 | 10.55 | 0.0767 | 3.0 |
| 3.5 | 42,000 | 12.31 | 0.0895 | 3.5 |
| 4.0 | 48,000 | 14.07 | 0.1023 | 4.0 |
| 5.0 | 60,000 | 17.58 | 0.1279 | 5.0 |

For each cooling capacity and each capillary tube diameter, there is only one optimal value of the capillary tube length. An improper value of a capillary tube is either too short or too long. The capillary tube that is too short will not produce enough pressure drop. The capillary tube that is too long creates an excessive pressure drop. From the analysis, it can be seen that smaller pipes use shorter distances and larger pipes use longer distances. An example of how to use this data is shown below. In the case of cooling capacity size 1 TR, choosing a capillary tube with a diameter of 0.080 inches will require a length of 2.2 m. Alternatively, choosing a capillary tube with a diameter of 0.087 inches will require a length of 3.7 m. The selection is flexible. That is, if choosing a small tube diameter and short tube length or if choosing a large tube diameter and long tube length. This method is more convenient and quicker than using charts from the ASHRAE standard, which has more steps and procedures. This approach can be applied to any size of diameter

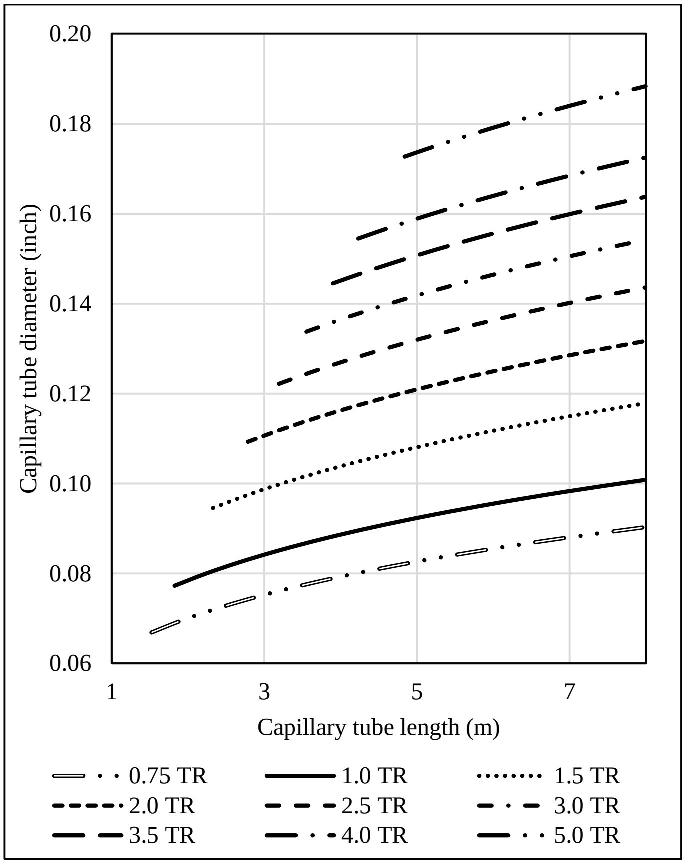


**Figure 14**. The optimum size (diameter and length) of capillary tubes for various cooling capacity

and length of capillary tube whereas the ASHRAE chart is limited to some capillary tube sizes only. In this way, choked flow can be monitored at the design stage while using the ASHRAE chart is not feasible.

## CONCLUSIONS

The model developed from the concept that the refrigerant flow through the capillary tube changes its properties along the length of the tube through it. The capillary tube is divided into smaller sections, each of which is considered by a small control volume. Thermodynamic principles are applied to each control volume. The initial parameters consist of the size of the air conditioner, the liquid flow rate, and the diameter of the capillary tube. The model can analyze continuously changing refrigerant properties. The properties of refrigerants include vaporization ratio, specific density, specific enthalpy, and viscosity. In addition to the refrigerant properties, the model can continuously analyze flow velocity, Reynolds number, and friction inside the tube. The analyzed output from the control volume initially becomes the input for analysis in the next control volume. Continuous parameter updates result in accurate, engineering-based analysis results.

Since the refrigeration and air conditioning system consumes a large portion of the building's energy usage and greenhouse gas emission. Numerous pieces of research are carried out to increase its operational efficiency in many ways to decrease energy consumption and greenhouse gas emission.[39,40] This work also aims for that purpose. A mathematical model is useful to make designs more accurate. The model minimizes trial and error and reduces the possibility of system malfunctions and failures. Model analysis results in air conditioning systems operating more efficiently, saving energy and hence, reducing greenhouse gas emissions.

## NOMENCLATURE

| | |
|---|---|
| $A$ | = Cross-sectional area of the inside of the tube, m$^2$ |
| $D$ | = ID of the tube, m |
| $f$ | = Friction factor, dimensionless |
| $h$ | = Enthalpy, kJ/kg |
| $h_f$ | = Enthalpy of saturated liquid, kJ/kg |
| $h_g$ | = Enthalpy of saturated vapor, kJ/kg |
| $v$ | = Specific volume, m$^3$/kg |
| $v_f$ | = Specific volume of saturated liquid, m$^3$/kg |
| $v_g$ | = Specific volume of saturated vapor, m$^3$/kg |
| $\mu$ | = Viscosity, Pa.s |
| $\mu_f$ | = Viscosity of saturated liquid, Pa.s |
| $\mu_g$ | = Viscosity of saturated vapor, Pa.s |
| $\Delta L$ | = Length of increment, m |
| $p$ | = Pressure, Pa |
| $Re$ | = Reynolds number |
| $V$ | = Velocity of refrigerant, m/s |
| $m$ | = Mass rate of flow, kg/s |
| $x$ | = Fraction of vapor in a mixture |
| e | = inner surface roughness of tube, m |

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## CONFLICTS OF INTEREST STATEMENTS

Prakob Surawattanawan declares no conflicts of interest. Juggarin Chutikusol has received a research grant from the Faculty of Engineering, Kasetsart University.

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