CHAPTER 5

EXPERIMENTAL INVESTIGATION

5.1 INTRODUCTION

The performance of the considered refrigerant mixtures was experimentally studied as against that of R-22 in the conventional window air conditioner. The tests were performed in a calorimetric room. After the system performance study, the capillary tube flow characteristics study was carried out for the optimal composition of the refrigerant mixture and compared with that of R-22. The experimental set-up and procedure used for the above studies are discussed in this chapter.

5.2 SYSTEM PERFORMANCE STUDY - EXPERIMENTAL SET-UP

An experimental set-up was constructed to facilitate the study on the performance of a window air-conditioner with the new HFC/HC refrigerant mixtures. The photographic view of the experimental facility used to test a domestic window air-conditioner is shown in Figure 5.1. The specifications of the components used in the test set-up are given in Appendix 3. Figure 5.2 shows the schematic diagram of the experimental set-up used for this performance study. The experimental set-up consists of a room calorimeter, a window air-conditioner of 1050 W capacity, instruments and accessories fitted to facilitate performance study as detailed below.
Figure 5.1  Photographic view of the experimental facility used to test a domestic window air-conditioner

Figure 5.2  Schematic diagram of the experimental facility used to test a domestic window air-conditioner
5.2.1 Room Calorimeter

The outer dimensions of the room calorimeter are 2300 mm x 2300 mm x 2800 mm. The walls of the room were insulated with glass wool of thickness 200 mm in order to maintain the heat infiltration to be less than 5% of the evaporation capacity (BIS 1998). A 2000 W air heater was placed inside the room calorimeter to supply heat to the room. The heater was connected to the mains through a variac and wattmeter (±0.25% accuracy), to regulate and measure the heat load. In order to have a uniform temperature throughout the calorimeter room, a fan (40 W) was used to circulate the air inside the calorimeter. The air-conditioner was fixed in one of the walls of the calorimeter in such a way that the evaporator coil takes the heat from the room calorimeter.

5.2.2 Window Air-Conditioner

To periodically check the oil level in the compressor an oil level indicator was attached suitably to the compressor as shown in Figure 5.2. To optimise the capillary, eight capillaries of different diameters and lengths viz., 1.1176 mm diameter; 1.25 m, 1.5 m, 1.75 m, 2 m, 2.25 m long and 1.27 mm diameter; 1.5 m, 1.75 m, 2 m long were fixed to a header. Suitable ball valves were used to select the required capillary to be included in the circuit. A thermally insulated duct was used to control the temperature of air passing over the condenser to simulate various ambient conditions without obstructing the flow of air (Aprea et al 2004).
5.2.3 Instrumentation

To monitor the mass flow of the refrigerant in the system, a mass flow meter (0–1090 kg/h) with ± 0.25% accuracy was installed next to the condenser as shown in the Figure 5.2. To measure the compressor power a wattmeter with ±0.5% accuracy was used. The energy consumption per day was also measured with an energy meter with 0.5% accuracy. Pressure transducers with ±0.25% accuracy and film type PT100 RTD temperature sensors with ± 0.1°C accuracy were fixed appropriately to measure the respective parameters across each component. Since the mixture is zeotropic in nature, to measure the temperature distribution along the evaporator coil eight-temperature sensors were fixed suitably. Computerized data acquisition system (Agilent 34970A, polling frequency 60 channels / second) was used to record the entire temperatures (T) and pressures (P). Five temperature sensors were fixed at strategically selected locations inside the room calorimeter to ensure that the variations in the temperature inside is not exceeding 1°C (Aprea et al 2004) at steady state conditions before making observations.

5.2.4 Experimental Procedure For Performance study

The total experimentation consisted of the following tests

i. Heat infiltration test.

ii. Capillary optimisation test.

iii. Charge optimisation test.

iv. Actual refrigeration capacity and COP.

v. Pull down test.

vi. Per-day energy consumption test.
5.2.4.1 Heat Infiltration Test

The entire test was conducted as per standards (BIS 1998). Before determining the actual performance parameters it was essential to assess the heat infiltration in the calorimeter room. The heat infiltration for a temperature differential ranging from 0 to 15 degrees between the evaporator inlet air temperature and the ambient temperature was calculated. To carry out the test at no load the system was switched ‘ON’ and when the room temperature got stabilized at 20°C at the end of pull down the system was put ‘OFF’. Now the calorimeter temperature started rising up due to the heat infiltration. During this period the time taken (Δts) for a temperature rise of 2°C inside the room calorimeter was noted continuously till the inside temperature became equal to the ambient temperature. The heat infiltration at any indoor temperature was calculated using the following equation and plotted in Figure 5.5.

\[ Q = \frac{m_a C_p (T_{\text{initial}} - T_{\text{final}})}{\Delta t_s} \]  \hspace{1cm} (5.1)

In the above expression \( T_{\text{initial}} \) and \( T_{\text{final}} \) are temperatures one degree above and one degree below the temperature considered. Based on the density of air and the known volume of room calorimeter \( m_a \) was calculated. From the figure it was found that for the maximum temperature differential of 15°C, the heat leak was 47.1 W, which was lesser than 5% of the evaporator capacity.
Figure 5.3  Variation of heat infiltration with temperature differential between ambient and calorimetric room temperatures

5.2.4.2  Capillary and Charge Optimisation

Optimization is the process by which the charge quantity as well as the capillary would result in maximum COP. To have a realistic comparison of the performance of the proposed mixtures with conventional refrigerant the experiment was carried out initially with the conventional refrigerant HCFC22. The capillary tube diameter, length and the refrigerant charge were optimised as the refrigerant flow volume in the system had changed due to alterations made to fix instruments, receiver etc. To start with, the system was properly leak tested, evacuated and flushed with R-22 three times to remove air and moisture from the system. During capillary tube optimisation, the system was initially charged with 750 g of R-22 (as per the manufacturer’s catalogue). To optimise the capillary tube and charge quantity the COP of the system at evaporator inlet
an air temperature of 27°C and a condenser inlet air temperature of 35°C was studied. For the selected condenser inlet air temperature the evaporator inlet air temperature was maintained at 27°C by suitably regulating the room heater load through the dimmerstat. All the capillary tubes mentioned in section 5.2.2 were considered for the optimisation. The variation of COP with capillary tube length is plotted in Figure 5.4. As shown in figure the COP of the system was maximum with 1.1176 mm diameter capillary tube at 1.75 m length and it was selected for the system performance study. Subsequently for the selected capillary tube the charge quantity of R-22 was optimised for maximum COP by varying charge from 600 g to 1100 g in steps of 50 g. The variation of COP with charge quantity is plotted in Figure 5.5. From that it was found that the COP was good for a charge varying from 800 g to 1000 g. Since receiver was used in the system, better performance was observed for a range of charge quantity. However, 950 g was considered as the optimum charge quantity for the present study.

![Figure 5.4 Variation of COP of R-22 with capillary length](image-url)
After that, the performance study of the system was carried out at various condenser and evaporator inlet air temperatures for the optimal capillary and optimal charge. During experimentation, for each test condition the performance parameters of the refrigerant such as refrigeration capacity, compressor power, discharge pressure, suction pressure, mass flow rate, discharge temperature, dome temperature and evaporator coil temperature distribution were measured.

### 5.2.4.3 Actual Refrigeration Capacity and COP

During this test the thermostat was totally disconnected, using the duct arrangement as shown in Figure 5.2, the condenser inlet air temperature was varied from 30°C to 45°C in steps of 5°C, whereas evaporator inlet air...
temperature was varied from 21°C to 29°C in steps of 2°C for each condenser inlet air temperature. Thus the evaporator inlet air temperature and condenser inlet air temperature were the main variables in the test matrix. For a given condenser inlet air temperature the different evaporator inlet air temperatures were achieved by suitably regulating the heater load through the dimmerstat. It is to be mentioned that to maintain steady power supply a servo power stabilizer was used. The heater load with which the system stabilises to prevail at the required evaporator inlet air temperature was noted. At each test condition with reference to the ambient and evaporator inlet air temperatures the corresponding heat leak was added to the heater load to get the actual refrigeration capacity. It is to be mentioned that this stabilization took 3 to 4 hours for each case. After stabilization the difference between the average values of temperature indicated by PT100 sensors inside the calorimeter room and the evaporator inlet air temperature was only 0.5°C. At the stabilized condition the system was monitored for an hour and the readings were scanned for all the state points every ten seconds and stored every ten minutes with the help of data acquisition system and the average result over the period of one hour was taken for the calculations.

5.2.4.4 Pull Down Test

A pull down test to find the cooling rate of the system for the refrigerants was also conducted during the experimentation. Initially at no load, both the evaporator and the condenser inlet air temperatures were ensured to be at 35°C. After the attainment of equilibrium, the system was made to run and the power consumption for every 10 seconds was manually noted while the temperatures were recorded using the data acquisition system. The readings
were taken until the system reached its cut-off for an evaporator inlet air temperature of 27°C.

5.2.4.5 Per-day Energy Consumption

The per-day energy consumption was conducted for an evaporator inlet air temperature of 27°C and for all condenser inlet air temperature varying from 30°C to 45°C in steps of 5°C. To measure the per day energy consumption of window air conditioner, the thermostat inside the room was adjusted to cut-in at 28°C and cut-off at 26°C. For any selected condenser inlet air temperature, once the system reached steady cut-in and cut-off cycles, the energy consumption was noted for 24 hours using the energy meter.

5.2.4.6 Testing With Mixtures

After completing all the above said performance tests with R-22, the refrigerant was recovered and equivalent quantity of mixtures M10, M15, M20 and M25 were charged and tests were repeated as discussed already. Of the four mixtures, M10 was initially selected for the performance analysis. The capillary tube optimisation for the mixture was carried out as mentioned earlier. In order to optimise the capillary an optimal reference charge quantity is required. Hence the charge quantity of M10 equivalent to 950 g of R-22 was obtained as per the procedure discussed in section 3.4. This equivalent charge of 816 g was considered to optimise the capillary tube for M10. Figure 5.6 shows the variation of the COP with capillary tube length for M10. From the graph it was observed that the COP of the system was found to be highest for a capillary tube diameter of 1.1176 mm and length 1.5 m. Having found the optimal
capillary tube; the next step was to confirm the optimality of the charge quantity for the selected capillary.

The COP of the system was studied as the M10 charge quantity was varied from 500 g to 1000 g in steps of 50 g. Figure 5.7 shows the variation of the COP of M10 with charge quantity. From the figure it was observed that the COP of the system is maximum for the optimal charge of 800 g for the mixture, which was closer to its equivalent charge of 816 g. This confirmed that 816 g could be considered as the optimal charge quantity. Hence further the system performance study was repeated with the equivalent charge of the respective mixtures M10 (816 g), M15 (774 g), M20 (736 g) and M25 (702 g) evolved as per the procedure in section 3.4.

Before doing the performance test with M15, M20 and M25 the suitability of the capillary tube was confirmed with their equivalent quantity of refrigerant charge. Figure 5.8 shows the variation of the COP with capillary tube length (1.1176 mm diameter) for the mixtures M15, M20, M25. From the graph it is observed that the COP of the system is found to be the highest for the length of 1.5 m for all the refrigerants. It has to be mentioned that no oil was changed during the entire test. Special care was taken during the experimentation to check the oil level in the compressor crankcase and thereby ensure oil miscibility. The condensing pressure shoot up to around 26.5 bar with the mixtures for the tested conditions of 35°C condenser inlet temperature and 27°C evaporator inlet temperature. To decrease the discharge pressure of the mixtures the condenser length had to be altered. The simulation study predicted a 19% increase in the condenser length that brings in reduction in discharge pressure to 24.2 bar without affecting the evaporator temperature.
Therefore in this study, the condenser surface area was increased by 19% for the mixtures to control the increase in discharge pressure. This resulted in reduced pressure ratio for the mixtures as compared to R-22 and also to realize superior heat transfer in the condenser (Kuijpers et al 1988; Wei et al 1997).

![Figure 5.6 Variation of COP of M10 with capillary length](image-url)

**Figure 5.6** Variation of COP of M10 with capillary length
Figure 5.7 Variation of COP of M10 with charge quantity

Figure 5.8 Variation of COP of mixtures with capillary length
5.3 UNCERTAINTY ANALYSIS

The uncertainty in measurement is expressed in the following way. Consider a variable $Z_i$, which has a known uncertainty of $\delta Z_i$. The variable and its uncertainty is expressed as

$$Z_i = Z_i \pm \delta Z_i \quad (5.2)$$

where $Z_i$ is the arithmetic mean of the measured value $\delta Z_i$ is evaluated based on the standard deviation, the number of samples and the confidence level. In a single experiment, the result $R$ is a function of ‘n’ independent variable and it may be represented by

$$R = R(Z_1, Z_2, Z_3, \ldots, Z_n) \quad (5.3)$$

The combined uncertainty $\delta R$ is given by the mean square method (Klien and McClintok 1953; Holman 2000).

$$\delta R = \left[ \left( \frac{\partial R}{\partial Z_1} \right)^2 + \left( \frac{\partial R}{\partial Z_2} \delta Z_2 \right)^2 + \cdots + \left( \frac{\partial R}{\partial Z_n} \delta Z_n \right)^2 \right]^{\frac{1}{2}} \quad (5.4)$$

Further if the variables are having linear relation, the uncertainty is expressed as (Holman 2000)

$$\delta R = \left[ \left( \frac{\delta Z_1}{Z_1} \right)^2 + \left( \frac{\delta Z_2}{Z_2} \right)^2 + \left( \frac{\delta Z_3}{Z_3} \right)^2 + \cdots + \left( \frac{\delta Z_n}{Z_n} \right)^2 \right]^{\frac{1}{2}} \quad (5.5)$$
The actual COP of the system was normally calculated from the equations listed below:

\[ COP_{ac} = \frac{Q_{he} + Q_{hl}}{W_{com}} \]  \hspace{1cm} (5.6)

\[ Q_{hl} = UA(T_{CIAT} - T_{EIAT}) \]  \hspace{1cm} (5.7)

\[ Q_{he} = f(Q_{he,md}, T_{EIAT}, T_{CIAT}, T_{ein}, T_{eout}, P_{ein}, P_{eout}) \]  \hspace{1cm} (5.8)

\[ W_{com} = f(W_{com,md}, T_s, T_d, T_{CIAT}, P_s, P_d) \]  \hspace{1cm} (5.9)

\[ \delta Q_{hl} = \left[ \frac{\delta L}{L} + \frac{\delta He}{He} + \left( \frac{\delta T_{CIAT}}{T_{CIAT}} \right)^2 + \left( \frac{\delta T_{EIAT}}{T_{EIAT}} \right)^2 \right]^{1/2} \]  \hspace{1cm} (5.10)

\[ \delta Q_{he} = \left[ \frac{\delta Q_{he-md}}{Q_{he-md}} \right]^2 + \left( \frac{\delta T_{EIAT}}{T_{EIAT}} \right)^2 + \left( \frac{\delta T_{CIAT}}{T_{CIAT}} \right)^2 + \left( \frac{\delta T_{ein}}{T_{ein}} \right)^2 + \left( \frac{\delta T_{eout}}{T_{eout}} \right)^2 + \left( \frac{\delta P_{ein}}{P_{ein}} \right)^2 + \left( \frac{\delta P_{eout}}{P_{eout}} \right)^2 \right]^{1/2} \]  \hspace{1cm} (5.11)

\[ \delta W_{com} = \left[ \frac{\delta W_{com-md}}{W_{com-md}} \right]^2 + \left( \frac{\delta T_{CIAT}}{T_{CIAT}} \right)^2 + \left( \frac{\delta T_s}{T_s} \right)^2 + \left( \frac{\delta T_d}{T_d} \right)^2 + \left( \frac{\delta P_s}{P_s} \right)^2 + \left( \frac{\delta P_d}{P_d} \right)^2 \right]^{1/2} \]  \hspace{1cm} (5.12)
From Equation 5.5

\[
\delta \text{COP}_{ac} = \left[ \left( \frac{\delta Q_{he}}{Q_{he}} \right)^2 + \left( \frac{\delta Q_{hl}}{Q_{hl}} \right)^2 + \left( \frac{\delta W_{com}}{W_{com}} \right)^2 \right]^{1/2}
\] (5.13)

Calculation of uncertainty in COP is given in Appendix 4

5.4 CAPILLARY FLOW CHARACTERISATION

This study will provide a set of capillary tube performance data in an air-conditioning system for R-22 and M20. Based on the experimental data, a correlation to predict the mass flow rate for a wide range of geometric and operating conditions using non-dimensional parameters is developed. The experimental set-up, procedure and development of correlation for the above studies are discussed in the following sections.

5.4.1 Experimental Set-up

An experimental set-up was constructed to study the flow characteristics of various adiabatic capillary tubes. The photographic view of the experimental facility used in the study is shown in Figure 5.9. The specifications of the components used in the test set-up are given in Appendix 3. The schematic of the experimental facility is shown in Figure 5.10. The test facility was designed to allow easy control of each operating parameter, such as pressure and temperature in the upstream and downstream of the test section. The test rig consists of three major flow loops: (1) a refrigerant flow loop containing a fixed test section, (2) an electrically heated hot water
Figure 5.9  Photographic view of the experimental facility used to test the flow characteristics in adiabatic capillary tubes

Figure 5.10  Schematic diagram of the experimental facility used to test the flow characteristics in adiabatic capillary tubes
flow loop used for the upstream heat exchanger, and (3) a chilled water-glycol flow loop used for the downstream heat exchanger.

5.4.1.1 Refrigerant Flow Loop

It consists of a diaphragm pump, test section and measuring instruments such as mass flow meter, temperature sensors and pressure gauges. A diaphragm pump with a pressure relief valve was used to provide a wide range of refrigerant mass flow rates. The pressure relief valve in the pump was adjusted to control the inlet pressure of the test section. The refrigerant at saturated or subcooled condition entered the test section, which was regulated by the water heated upstream heat exchanger. The two phase refrigerant exiting the test section was condensed and subcooled in a water/glycol cooled downstream heat exchanger and sent into a receiver, so that the refrigerant pump had always liquid at its suction side. To monitor the mass flow of the refrigerant in the system, a Coriolis mass flow meter with ± 0.25% accuracy was installed between the diaphragm pump and the upstream heat exchanger. The pressure of refrigerant at the inlet and exit of the test section was recorded manually using bourdon tube master pressure gauges with an accuracy of ± 0.5% of full scale. The film type PT100 RTD sensors with ± 0.1°C accuracy were used to measure the temperature of the refrigerant at inlet and outlet of the test section.

5.4.1.2 Hot Water Flow Loop

It consists of a 30 litres capacity stainless steel tank to store water, 0.5 HP pump to pump the hot water to upstream heat exchanger and two 2 kW heaters. Heaters are placed inside the tank to heat the water, which in turn heats
the refrigerant in the refrigerant flow loop to control the inlet temperature at the test section. For each heater there is a separate thermostat for precised control of temperature at the inlet of the test section.

5.4.1.3 Chilled Water-glycol Flow Loop

It consists of a 30 litres capacity stainless steel tank to store the water-glycol, 0.5 HP pump to pump the water-glycol to downstream heat exchanger and two refrigeration units of each 1.5 TR. Refrigeration units are used to cool the water-glycol, which in turn cools the refrigerant in the refrigerant flow loop to control precisely the temperature at the test section exit. To maintain the set temperature in the cold bath, a temperature controller of 1°C accuracy is used. Initially it is set to a desired temperature of the cold bath. If the set temperature is achieved, automatically the controller will cut off the refrigeration system.

5.4.2 Experimental Conditions

R-22 and the refrigerant mixture ‘M20’ were studied with different adiabatic capillary tubes made of copper. The capillary tubes were selected based on a refrigerant compressor manufacturer’s catalogue (Kirlosker 1996) and also to have sufficient geometric variation in terms of diameter and length to accommodate different tonnages. Ten straight capillaries of different diameter and length viz. 1.1176 mm diameter; 0.75 m, 1.25 m, 1.5 m, 1.75 m, long and 1.27 mm diameter; 0.75 m, 1.25 m, 1.5 m long and 1.397 mm diameter; 0.75 m, 1.25 m, 1.5 m long were considered. All the capillaries were fixed to a header and suitable valves were used to select the required capillary
to be included in the circuit. The experimental conditions were selected based on the extensive range of operating conditions for capillary tubes in window air-conditioners. Capillary tube inlet pressure was adjusted to the saturation pressure corresponding to the condensing temperatures of 37°C, 42°C, 47°C and 52°C. For each condensing temperature, five degree of subcooling viz., 2°C, 5°C, 8°C, 11°C and 14°C were considered at the capillary tube inlet. The exit pressure of the capillary tube was maintained at the saturation pressure corresponding to the evaporating temperature of 7°C. The condensing temperature of the refrigerant M20 mixture was determined as an average of dew point and bubble point temperatures at constant pressure. The degree of subcooling for M20 was calculated as the difference between the refrigerant temperature and bubble point temperature at the same pressure.

5.4.3 Experimental Procedure

Initially, the discharge valve of the diaphragm pump was kept opened and the stroke length was adjusted to be zero. Once the pump was started, the stroke length was increased to 25% of the total stroke length. Allow the pump to run for about 10 to 15 minutes to achieve stabilization. After the stabilization, the stroke length was increased to 100% for the maximum flow rate. Then, the discharge pressure was raised to the required level by adjusting the relief valve. The high-pressure refrigerant was then heated in a tube-in-tube upstream heat exchanger. The temperature and flow rate of hot water entering the heat exchanger was regulated by adjusting the heater and flow control valve in the circuit. Thus the required upstream temperature and pressure of the test section was achieved and the refrigerant was allowed to expand through the capillary. The low pressure and low temperature two-phase refrigerant enters
the tube-in-tube downstream condenser, where it was cooled to the exit conditions by circulating water-glycol solution in the downstream heat exchanger. Its temperature and flow rate could be regulated. Thus the test section downstream temperature and pressure are achieved. The expanded refrigerant enters the receiver, which was again pumped to repeat the above cycle. During experimentation, the mass flow rate, pressure and temperature across the test section were measured after the system attained steady state. The experiment was repeated for different capillaries and upstream conditions in respect of temperature and pressure. While conducting the study with M20, it was observed that to reach steady state, the system takes 1.5 to 2 hours for higher condensing temperatures (47°C and 52°C) and for subsequent subcooling it takes only 15 minutes to 20 minutes. For lower condensing temperatures (37°C and 42°C) it takes 2 hours to 2.5 hours and subsequent subcooling it takes 30 minutes to 40 minutes. While increasing the degree of subcooling, setting the capillary tube inlet temperature and pressure precisely to predetermined values was quite difficult. Therefore, the system was considered to be in steady state when the variations in the capillary tube inlet temperature and pressure were within ±0.3°C and ±15 kPa respectively.

Thus an experimental database was created to develop a non-dimensional correlation for the above mentioned selected capillary tube lengths, diameters, various operating conditions and working fluids. In the database with all the combinations of operating conditions and working fluids there were 400 sets of data for mass flow rate.
5.4.4 Development of Non-dimensional Correlation

Based on the measured mass flow rate for R-22 and M20 a non-dimensional correlation has been developed in order to predict the refrigerant mass flow rate through a capillary tube operating at different conditions. The non-dimensional parameters were selected based on the variables influencing mass flow rate through capillary tubes by using Buckingham \( \pi \) theorem (White 1994). Operating parameters chosen in this study were capillary tube diameter \( D \), length of capillary \( L \), inlet pressure \( P_{in} \), saturation pressure \( P_{sat} \) corresponding to the inlet condensing temperature, degree of subcooling \( \Delta T_{sub} \) and the refrigerant properties which includes the liquid density, vapor density, liquid viscosity, vapor viscosity, liquid specific heat, surface tension and enthalpy of formation. The effect of changing various parameters that influence refrigerant mass flow rate through the capillary tube is shown in Table 5.1.

The functional equation may be written as

\[
\dot{m} = f_1 ((P_{in}-P_{sat}), \Delta T_{sub}, L, D, \rho_l, \rho_g, \mu_l, \mu_g, \sigma, h_{lg}, C_{pl})
\] (5.18)

Eight non-dimensional \( \pi \) groups represented in Table 5.2 are derived based on the Buckingham \( \pi \) theorem with four repeating variables \( D, \rho_l, \mu_l \) and \( C_{pl} \). Many researchers have applied this procedure to the empirical correlations for capillary tubes (ASHRAE Handbook 1994; Melo et al 1999; Kim et al 2002; Jongmin et al 2002). Buckingham \( \pi \) theorem guarantees that the functional relationship must be of the following equivalent form shown in Equation 5.19.
Table 5.1  Effect of various parameters on mass flow rate

<table>
<thead>
<tr>
<th>S. No.</th>
<th>Parameter</th>
<th>Increases / Decreases</th>
<th>Effect on mass flow rate</th>
</tr>
</thead>
<tbody>
<tr>
<td>1.</td>
<td>Condensing Temp. (Tc)</td>
<td>Increases</td>
<td>Increases due to the increase in pressure difference</td>
</tr>
<tr>
<td>2.</td>
<td>Subcooled pr. (P_{sat})</td>
<td>Decreases</td>
<td>Increases due to the delay in flashing.</td>
</tr>
<tr>
<td>3.</td>
<td>Liquid density (\rho_l)</td>
<td>Decreases</td>
<td>Decreases due to the increase in frictional pressure drop</td>
</tr>
<tr>
<td>4.</td>
<td>Liquid viscosity (\mu_l)</td>
<td>Increases</td>
<td>Decreases due to the increase in frictional pressure drop</td>
</tr>
<tr>
<td>5.</td>
<td>Latent heat (h_{lg})</td>
<td>Increases</td>
<td>Increases due to less vapor formation</td>
</tr>
<tr>
<td>6.</td>
<td>Surface tension (\sigma)</td>
<td>Increases</td>
<td>Decreases due to the increase in adhesive force, hence more pressure drop</td>
</tr>
<tr>
<td>7.</td>
<td>Capillary diameter (D)</td>
<td>Increases</td>
<td>Increases due the less restriction on flow</td>
</tr>
<tr>
<td>8.</td>
<td>Capillary length (L)</td>
<td>Increases</td>
<td>Decreases due to more vapour formation because of increased pressure drop</td>
</tr>
</tbody>
</table>


### Table 5.2 Non-dimensional parameters

<table>
<thead>
<tr>
<th>Pi-group</th>
<th>Definition</th>
<th>Effect / Consideration</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\pi_1$</td>
<td>$\frac{\dot{m}}{D \mu_1}$</td>
<td>Mass Flow Rate</td>
</tr>
<tr>
<td>$\pi_2$</td>
<td>$\frac{L}{D}$</td>
<td>Tube Geometry</td>
</tr>
<tr>
<td>$\pi_3$</td>
<td>$\frac{\mu_g}{\mu_1}$</td>
<td>Friction, Bubble Growth</td>
</tr>
<tr>
<td>$\pi_4$</td>
<td>$\frac{\rho_g}{\rho_l}$</td>
<td>Density</td>
</tr>
<tr>
<td>$\pi_5$</td>
<td>$\frac{D \sigma}{\mu_l^2}$</td>
<td>Surface Tension, Bubble Formation</td>
</tr>
<tr>
<td>$\pi_6$</td>
<td>$\frac{D^2 \rho_l^2 \frac{2g}{h_1} \mu_1^2}{\mu_1}$</td>
<td>Heat of Vaporization</td>
</tr>
<tr>
<td>$\pi_7$</td>
<td>$\frac{D^2 \rho_l^2 C_p l \Delta T_{sub}}{\mu_1^2}$</td>
<td>Inlet Subcooling</td>
</tr>
<tr>
<td>$\pi_8$</td>
<td>$\frac{D^2 \rho_l^2 (P_{in} - P_{sat})}{\mu_1^2}$</td>
<td>Inlet Pressure</td>
</tr>
</tbody>
</table>

$$\pi_1 = f_2 (\pi_2, \pi_3, \pi_4, \pi_5, \pi_6, \pi_7, \pi_8) \quad (5.19)$$

To obtain the function $f_2$, Equation (5.19) is rewritten in a generic form with unknown constants and exponents as shown in Equation 5.20.

$$\pi_1 = C \cdot \pi_2^{e_1} \cdot \pi_3^{e_2} \cdot \pi_4^{e_3} \cdot \pi_5^{e_4} \cdot \pi_6^{e_5} \cdot \pi_7^{e_6} \cdot \pi_8^{e_7} \quad (5.20)$$
The constant C and exponents of the seven independent $\pi$ groups are determined by using a non-linear regression technique based on the experimental database consisting of mass flow rate data for different conditions. The non-dimensional correlation for $\pi_1$ was developed in a power law form of the remaining $\pi$ groups as shown in Equation (5.21).

The developed correlation is given as follows:

$$\pi_1 = 28.34 \times 10^3 \pi_2^{-0.499} \pi_3^{-1.952} \pi_4^{2.101} \pi_5^{0.179} \pi_6^{0.016} \pi_7^{0.181} \pi_8^{-0.104}$$

(5.21)

All properties for respective $\pi$ terms, both liquid and vapor, are evaluated at the inlet pressure of the capillary tube. The REFPROP software was used to calculate the properties of the refrigerants. The regression analysis was carried out using “Design Expert” software Stat-Ease (2000).