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A Generalized Correlation for Pressure Drop of Refrigerant R-134a through Helical Capillary Tubes

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ABSTRACT

A capillary tube is a common expansion device used in small sized refrigeration and air-conditioning systems. The objective of this study is to present test results and to develop a dimensionless correlation on the basis of the experimental data of helical capillary tube for R134a. A generalized correlation for Pressure drop of refrigerant through helical capillary tubes is developed by implementing Buckingham Pi theorem and regression analysis. Several capillary tubes with different length and inner diameter were selected as test sections. Mass flow rate and Pressure drop through capillary tube was measured for several inlet pressures for each capillary tube. The generalized correlation yields good agreement with the present experimental data for R134a. The Predicted results and experimental data within relative deviations ranging from ±5% for 93% of experimental reading.

Keywords—**Capillary tube, Refrigerant pressure drop, Mass flow rate, R134a, Expansion device, Compressor Calorimeter test rig**

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I. INTRODUCTION

Every refrigeration system requires a pressure reducing device to meter refrigerant flow from high pressure side to low pressure side as per the load requirements. The capillary tubes especially popular in smaller vapor compression systems such as household refrigerators, freezers and room air conditioners. The capillary operates on the principle that liquid passes through it much more readily than vapors; as a result, it is practical metering device. It is a length of drawn copper tubing with small inner diameter. Generally, inner diameters are from 0.5 mm to 1.8 mm and length up to 6 meter can be used. Due to the simplicity, low cost, zero maintenance and requirement of a low starting torque motor to run the compressor, capillary tubes are widely used in vapor compression systems. When a capillary tube is sized to permit the desired refrigerant flow, the liquid refrigerant seals its inlet. If the system becomes unbalanced some uncondensed refrigerant enters the capillary tube. This vapor reduces the mass flow of refrigerant considerably, which increases condenser pressure and causes sub-cooling at condenser outlet means at capillary tube inlet. The result is increases refrigerant mass flow rate through capillary tube. If properly sized for the application, the capillary tube compensates automatically for load and systems variations and gives acceptable performance over a limited range of operating conditions.

Walf et al. 1995 [1] had developed generalized correlation for prediction of mass flow through adiabatic straight capillary tube for refrigerants R-134a, R-22 and R-410A. In this method the Buckingham π theorem and regression was applied to the physical factor and fluid factors that affect capillary tube flow rate. Khan et al. [2] carried out experimental study on performance of spirally coiled and straight capillary tube for R-134a refrigerant. He found that the effect of coiling reduces mass flow through capillary by 10% to 15% than straight capillary tube. It is considered that pressure drop of coiled tube was increased due to an existence of secondary flow generated by centrifugal force. Further, generalized correlation was developed for spirally coiled capillary tubes. Mittal et al. [3] experimentally investigated performance of helically coiled capillary tube for R-407C refrigerant which is alternate refrigerant for HCFC R-22. The mass flow rate through coiled capillary with coil diameter of 60 mm, 100 mm and 140 mm was reduced by an average of 10%, 7% and 5% respectively than straight capillary. He also found that increase in degree of sub-cooling from 5ºC to 10ºC results in increase of mass flow rate by 15% for 1.52 mm inner diameter tube. Further, he found effect of sub-cooling on mass flow rate was more on larger inner diameter of capillary tubes due to increase in liquid length. S. Kim et al. [4] experimentally studied the performance of capillary tube for HCFC R-22, HFC R-407C and R-410A. The mass flow rates of R-407C were greater by 4.0%, and those of R-410A were greater by 23% as an average, than those of R22. Choi et al. [5] has experimentally investigated performance of capillary tube for R-22, R-290 and R-407C refrigerant and developed generalized correlation.

The flow characteristics of refrigerants through straight capillary tubes have been studied extensively in the past six decades, both experimentally and numerically. Experimental studies on helically coiled capillary tubes are very limited. Actually, in all practical applications, the capillary tubes are helically coiled to make it compact in size. Further, the practical capillary sizing problem is to determine required length at given pressure drop, mass flow rate, inner diameter and given operating conditions. However in most of the generalized correlations pressure drop term were not found and mostly focused on determination of mass flow rate through capillary tube. Therefore, considering practical sizing problem of capillary, the experimental investigation has been undertaken to study the flow of R-134a through helically coiled capillary tube at various inner diameter, length and operating conditions. The generalized correlation is developed using 68 experimental readings with different diameter, length and operating conditions. A pressure drop term also included in dimension analysis in the point of view of practical sizing of capillary.

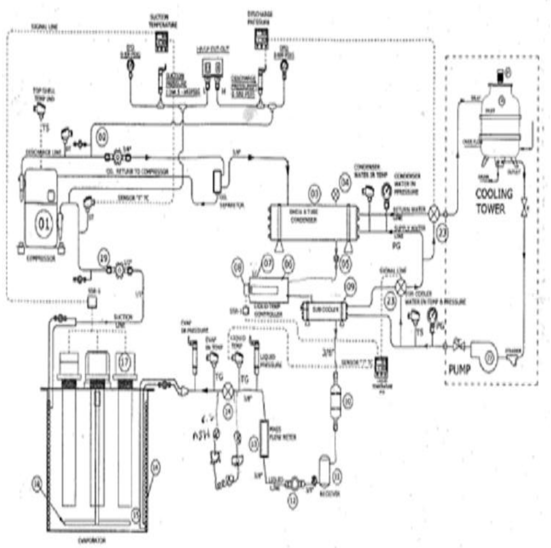

Fig. 1 Schematic diagram of experimental setup

II. EXPERIMENTAL SET-UP AND PROCEDURE

The schematic diagram of experimental set-up is shown in Fig. 1. The test-section was a copper capillary tube, in which the refrigerant expands from high pressure side to low pressure side. The helically coiled tube test-section in the experimental set-up was put in horizontal position. Suitable isolation valves are installed for easy replacement of test pieces. The low pressure refrigerant coming out from capillary tube entered the evaporator consisting of a copper coil submerged in a glycol tank. An electric heater was fitted in the evaporator tank to provide a heat load to the evaporator. The heating load was varied through a Solid state rely and PID controller. An agitator was also fitted in the tank to maintain the uniform bulk temperature of glycol. The refrigerant vapors emerging from the evaporator were sent to suction port of compressor after assurance of achieving desired suction superheat of refrigerant. The hermetically sealed compressor was run by means of single phase electrical supply. The high pressure superheated vapors emerging from the compressor entered the oil separator. The oil free vapors from separator were condensed in the water cooled condenser. The desired condensing pressure/temperature was achieved by varying mass flow rate through condenser by using three way proportionate control valve and PID controller. The cooling tower was used to reject heat in to the atmospheric. The provision of on-off of cooling tower fan was made to control the cooling tower water temperature. To vary the degree of sub-cooling, a pre-heater followed the sub-cooler was installed. In the pre-heater, liquid refrigerant heating was done by electrical heater and controlled by Solid state relay and PID controller. The high pressure saturated liquid from the pre-heater was made to flow through sub-cooler to control the degree of sub-cool. The same arrangement like condenser cooling water circuit was made to achieve desired sub-cooling in sub-cooler heat exchanger. The cold water from cooling tower flows through sub-cooler and then flows through condenser. The sub-cooled liquid refrigerant collected in a receiver to ensure a continuous supply of refrigerant to the capillary tube. The unwanted solid particles and moisture in refrigerant were removed through a filter-drier. A hand operated expansion valve was also provided in parallel to the capillary tube. A centrifugal pump was used to circulate cooling water through the sub-cooler and condenser. The mass flow rate of high pressure liquid refrigerant was measured by magnetic mass flow meter having an accuracy of ±0.25% of indicated value. A sight glass was also provided to visualize the state of refrigerant entering into the capillary tube. A number of hand shutoff valves were provided in between the major components of the experimental set-up. Therefore, in case of leakage or any repair, the damaged component was retrieved with ease. The temperature at different locations of the set-up and the test section was measured by means of T-type thermocouples with an accuracy of ±0.2. The pressure of the refrigerant was measured by pressure transducers having an accuracy of ±0.25%. The pressure at capillary tube inlet was adjusted from 915 kPa to 1350 kPa. The degree of sub-cooling and degree of superheating was maintained constant throughout the experimentation to 3ºC and 5ºC respectively. The capillary inner diameters, 1.52 mm, 1.63 mm and 1.78 mm, and coil diameter 50 mm were selected for this study. All the test data were collected under steady state conditions. At the start up, the system typically takes about 1 h to reach the steady state and thereafter, it takes only 15 min for any subsequent setting.

III. RESULT AND DISCUSSION

The test results on the performance of various capillary tubes are presented for various geometric and operating conditions. The generalized correlation is developed based on extensive experimental readings.

A. Pressure Drop through Capillary Tube:

Fig. 2 shows variation of pressure drop with capillary inlet pressure. Pressure drop curve of all six capillary tube samples are plotted. The pressure difference across capillary tube increases as the capillary inlet pressure increases due to higher frictional resistance offered by capillary at higher inlet pressure. For capillary tube of 1.625 mm diameter and 1.59 meter length, the pressure difference across capillary increases by 26.7% and 48.5% after increasing inlet pressure from 9.2 bar to 11.53 bar and 13.53 bar respectively. The Fig. 2 also indicates that pressure difference across capillary decreases as capillary inner diameter increases and pressure drop increases as length of capillary increases.

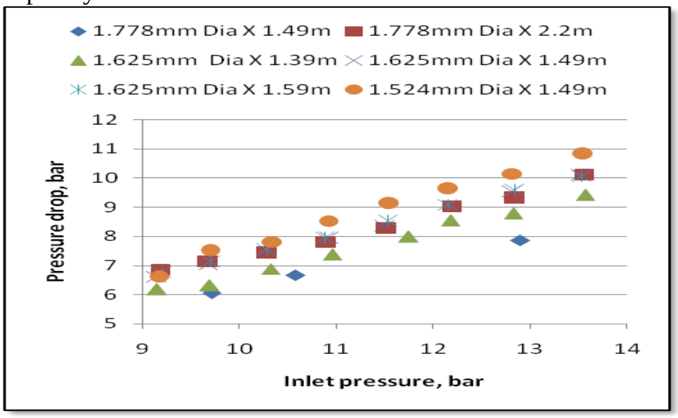

Fig. 2 Variation of Pressure drop with inlet pressure for different capillary

Fig. 3 shows variation of mass flow with capillary inlet pressure for all six capillary samples. The refrigerant mass flow rate increases as the capillary inlet pressure increases due to higher driving force at the inlet of capillary tube at higher capillary inlet pressure. For capillary tube of 1.625 mm diameter and 1.59 meter length, the mass flow rate increases by 19% and 31% after increasing inlet pressure from 9.2 bar to 11.53 bar and 13.53 bar respectively. The Fig. 3 also indicates that mass flow increases as capillary inner diameter increases and mass flow decreases as length of capillary increases. Fig. 4 shows that variation of pressure drop with capillary inlet diameter and constant capillary length of 1.49 meter. The refrigerant pressure drop decreases as the capillary inner diameter increases due to lower frictional resistance. For capillary tube inlet pressure 9.7 bar corresponding to 42ºC condensing pressure, the pressure drop decreases by 6.85% and 24.33% after increasing the inner diameter 1.524 mm to 1.625 mm and 1.778 mm respectively.

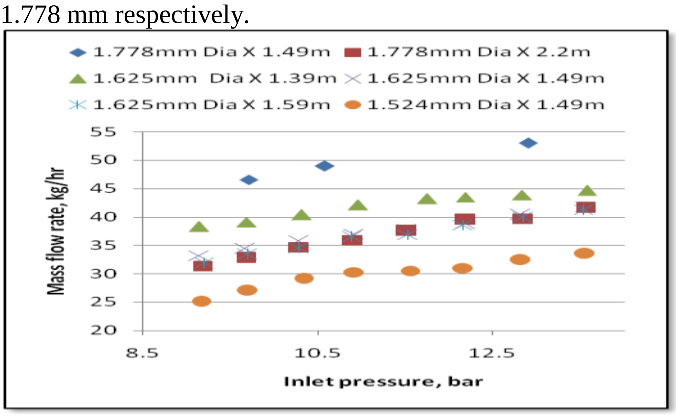

Fig. 3 Variation of Mass flow rate with inlet pressure for different capillary

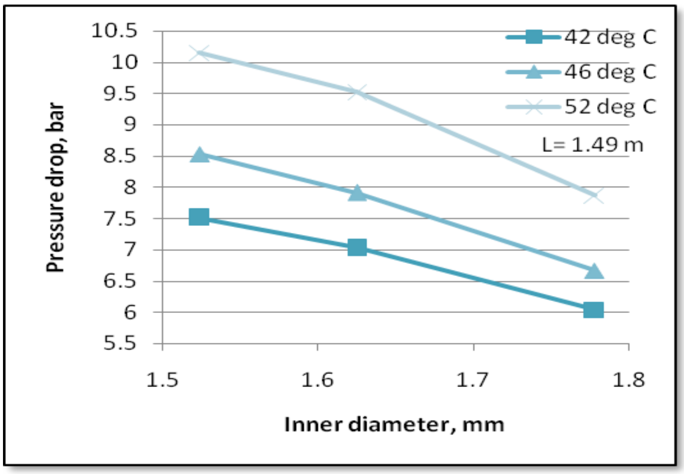

Fig. 4 Variation of pressure drop with inner diameter at constant length

Fig. 5 shows variation of pressure drop with capillary tube length at constant inner diameter 1.625 mm. The refrigerant pressure drop increases as the capillary length increases due to higher frictional resistance. For capillary tube inlet pressure 9.15 bar corresponding to 40ºC condensing pressure, the pressure drop increases by 7.12% and 9.7% after increasing the capillary length from 1.39 meter to 1.49 meter and 1.59 meter respectively. From Fig. 3 and Fig. 4, we can conclude that inner diameter has significant effect on pressure drop through capillary than capillary tube length. In trial and error method for optimization of capillary tube, it is always advisable to alter capillary tube length than diameter to smooth increment or decrement in pressure drop.

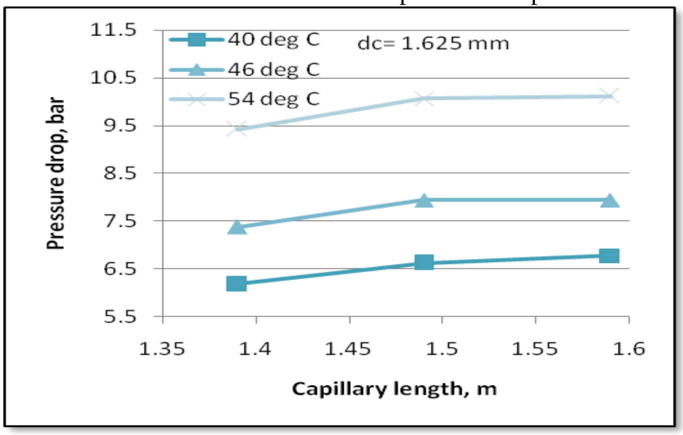

Fig. 5 Variation of mass flow with capillary length for constant inner diameter

From above Fig.2 to Fig.5, we can say the refrigerant pressure drop and mass flow rate in adiabatic capillary tube depends upon capillary tube diameter, length, coil diameter and pressure at capillary inlet. Other parameters like liquid specific volume, vapor specific volume, liquid and vapor viscosity and latent heat of vaporization also have influence on the mass flow of the refrigerant. Hence these parameters are taken into consideration while preparing generalized correlation for prediction of mass flow rate and pressure drop across the capillary tube.

A. Development of Generalized Correlation:

$\Delta P = f\,(V_{f},V_g, m, L, D, g, h_{fg}, P_s, P_d, \mu_f, \mu_g)$

The non dimensional π-groups represented in table-1 have been derived from Buckingham-π theorem. The equation-1 is developed using Buckingham-π theorem and multiple linear regression. The refrigerant properties are obtained from REFPROP.

$\pi 1 = f\ (\pi 2, \pi 3, \pi 4, \pi 5, \pi 6)$

$$\pi 1 = 1.0002\ \pi 2^{1.0745}\ \pi 3^{0.6115}\ \pi 4^{0.3418}\ \pi 5^{0.659}\ \pi 6^{-2.6202} \quad - (1)$$

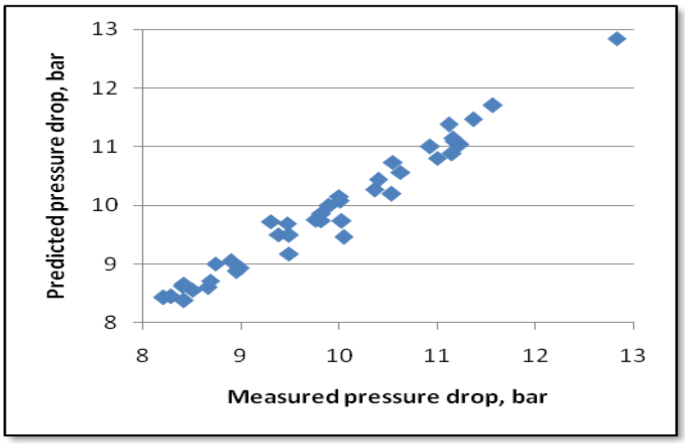

Fig. 6 Comparison of predicted pressure drop with measured pressure drop data with this study

Table-1

Non dimensional π-groups

π Group	Correlation	Consideration
π1	$\frac{\Delta P . dc^{3/2}}{\dot{m}\sqrt{g}}$	Mass flow rate and pressure drop
π2	$\frac{V_g}{V_f}$	Specific volume
π3	$\frac{L}{dc}$	Tube geometry
π4	$\frac{h_{fg}}{g.dc}$	Heat of vaporization
π5	$\frac{P_d}{P_s}$	Pressure ratio
π6	$\frac{\mu_f}{\mu_g}$	Viscosity

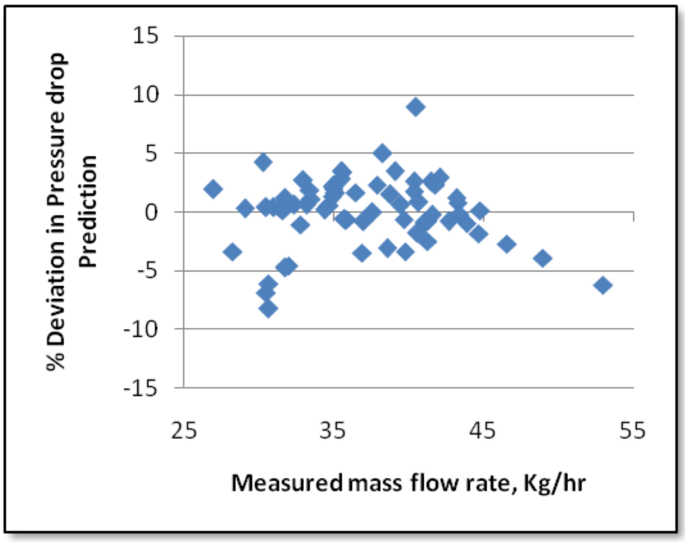

Fig. 7 Deviation in predicted mass flow rate from measured data for different mass flow rate

The proposed correlations for helical capillary tubes have been verified by comparing the measured mass flow rates with those predicted by the proposed correlations in Fig. 6 for all experimental reading. Fig.7 shows deviation in predicted mass flow rate from measured mass flow rate at different mass flows of experimental data.

$$\%\ \text{deviation} = \frac{\dot{m}\ \text{predicted} - \dot{m}\ \text{measured}}{\dot{m}\ \text{measured}}$$

All the points are inside the 10% deviation line. The generalized correlation yields good agreement with the present experimental data. The Predicted results and experimental data within relative deviations ranging from ±5% for 93% of experimental reading.

IV. CONCLUSION

The following conclusions can be drawn from the present study:

1. For a given capillary, the pressure drop is function of mass flow rate, capillary length, inlet pressure and coil diameter.
2. In proposed correlation, capillary length can be determined at given pressure drop across capillary tube, mass flow rate and operating conditions.
3. Non-dimensional correlations to predict the refrigerant pressure drop through helical capillary tube have been proposed. The developed correlation for pressure drop yield good agreement with the measured data of present study with deviations of ±5 percent for 93 percent of experimental reading.

NOMENCLATURE

dc	Capillary tube internal diameter, m
L	Capillary tube length, m
$\dot{m}$	Mass flow rate, kg/sec
Pd	Discharge pressure, Pa
Ps	Suction pressure, Pa
Vg	Specific volume of vapor, m3/kg
Vf	Specific volume of liquid, m3/kg
µg	Dynamic viscosity of vapor, pa-s
µf	Dynamic viscosity of liquid, pa-s
g	Gravitational acceleration, m/sec2
ΔP	pressure difference across capillary, pa
µ	Dimensionless parameter group

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