Refrigerant Flow Through Flexible Short Tubes

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ABSTRACT

Flexible short tubes were designed to decrease their diameter as the pressure differential increases under high outdoor temperatures. A series of tests for a R-22/lubricant mixture (mass fraction of oil, 1.2 %) were performed with two flexible short tubes to develop flow data over a range of typical air conditioner operating conditions. One short tube had a modulus of elasticity of 7063 kPa and the other a value of 9860 kPa. Both short tubes had identical lengths (14.5 mm), entrance diameters (2.06 mm), and exit diameters (2.46 mm). The tests included both single and two-phase flow conditions at the inlet of the flexible short tube. Upstream pressures were varied from 1179 kPa to 2144 kPa, which corresponded to saturated condensing temperatures of 29.4 °C to 54.4 °C. Experimental results were presented as a function of pressure, subcooling/quality, evaporating pressure, and modulus of elasticity. Mass flow rates were compared with those of a rigid short tube. The flow rate through the flexible short tubes was strongly dependent on the condensing pressure, subcooling/quality, and modulus of elasticity (which in effect changed diameter at different upstream pressures) of the short tube material. However, the flow rate showed little dependence to the evaporating pressure.

An empirical flow model was developed using the experimental data. This flow model was then combined with an air conditioner system simulation model. Results from the model indicated that the flexible short tube provided approximately 2-3 % higher capacity than the fixed expansion devices at temperatures lower than 25 °C and higher than 45 °C. However, the flow control with the flexible short tube was not as good as with a thermal expansion valve.

Nomenclature

A_s	entrance cross-sectional area of a flexible short tube (m^2)
D_1	flexible short tube entrance diameter (mm)
D_2	flexible short tube exit diameter (mm)
$EVAP$	normalized downstream pressure, $(P_c - P_{down})/P_c$
γ_c	dimensional constant, 1.296×10^{10} ($s^2\cdot N/(h^2\cdot kN)$)
$\dot{m}$	mass flow rate (kg/h)
P_c	critical pressure, 4977.4 kPa
P_{down}	downstream pressure (kPa)
P_f	adjusted downstream pressure (kPa)
P_{sat}	upstream liquid saturation pressure (kPa)
P_{up}	upstream pressure (kPa)
PRA	ratio of upstream pressure to critical pressure, P_{up}/P_c
SUB	normalized subcooling, $(T_{sat} - T_{up})/T_c$
T_c	R-22 critical temperature, 369.17 K
T_{sat}	liquid saturation temperature (K)
T_{up}	temperature of upstream fluid (K)
ρ	density (kg/m^3)

Introduction

The expansion device controls refrigerant flow through the system by restricting the flow of refrigerant pumped by the compressor. The performance of an expansion device not only helps determine the capacity of the refrigeration system but can also affect system reliability.

The most commonly used expansion devices in small air conditioners are capillary tubes, short tubes and thermostatic expansion valves (TXVs). Both capillary tubes and short tubes are constant flow area expansion devices, and control the flow rate based on the flow conditions entering the expansion devices such as condensing pressure and the amount of subcooling/quality. Thermostatic expansion valves control refrigerant flow rate by varying the flow area as a response to the amount of superheat at the exit of the evaporator. Because of their low cost and easy installation, capillary tubes or short tubes are often preferred in lower cost air conditioners.

Constant flow area expansion devices have performance limitations. As the outdoor temperature increases, the pressure differential between condenser and evaporator increases for an air conditioner. The increase in pressure differential decreases the pumping capacity of the compressor, but increases the capacity of a constant flow area expansion device. The result of this imbalance is that more refrigerant is directed to the evaporator, resulting in possible flooding, and under extreme conditions, two-phase (liquid plus vapor) flow enters the suction of the compressors. This condition could reduce the life of the compressor. One alternative to the constant area expansion device has been the thermal expansion valve. It adjusts refrigerant flow (within its operating range) so that a fixed superheat is maintained at the exit of the evaporator. The flexible short tube was invented to provide flow control similar to that provided by a TXV, but at a cost nearer that of conventional fixed area expansion devices (Drucker and Cann, 1991; Drucker, 1992; Drucker and Abbot, 1993). Designing a system with a flexible short tube requires knowledge of how the flow is affected by system variables such as evaporator and condenser pressure and upstream subcooling/quality.

The flexible short tube has the same shape as a rigid short tube, but it is made from materials that deform as pressure is applied. Flexible short tubes are made from elastomeric materials and are designed so that the flow area decreases as the short tube is compressed by increasing pressure upstream of the short tube (Drucker and Cann, 1991; Drucker, 1992). No work has been conducted on the flow of R-22 through flexible short tubes prior to the present work. Most of the previous work on short tubes was focused on flow through the rigid short tubes (Mei, 1982; Krakow and Lin, 1988; Aaron and Domanski, 1990; and Kim and O'Neal, 1994).

This paper presents experimental results for flow of an R-22/lubricant (mass fraction of 1.2 %) through flexible short tubes that have two different moduli of elasticity. The experimental results were used to develop an empirical flow model that was incorporated into an air conditioner simulation model. The performance of an air conditioner with the flexible short tube was compared to the same system with a capillary tube, short tube, and TXV.

Experimental Setup

A schematic diagram of the experimental setup is shown in Figure 1. The test loop was designed to allow easy control of each operating parameter such as upstream subcooling or quality, upstream pressure, and downstream pressure. It also allowed for changing of the oil concentration by injection of oil into the system. The test rig consisted of three major flow loops: (1) a refrigerant flow loop containing a detachable test section, (2) a hot water flow loop used for the evaporation heat exchanger and (3) a chilled water-glycol flow loop used for the condensation heat exchanger.

A diaphragm liquid pump with a variable speed motor was used to provide a wide range of refrigerant mass flow rates. The pressure entering the test section (upstream or condenser pressure) was controlled by adjusting the speed of the refrigerant pump. The refrigerant subcooling or quality entering the test section was set by a water-heated heat exchanger (evaporation heat exchanger) and a heat tape. For single-phase conditions at the inlet of the test section, most of the energy transfer to the refrigerant was supplied by the evaporation heat exchanger. A heat tape with adjustable output from 0 kW to 0.9 kW was utilized to provide precise control of upstream subcooling. For two-phase flow conditions at the inlet of the test section, the flow from the pump was heated by the evaporation heat exchanger to 1 °C of subcooling, and a heat tape was used to reheat the refrigerant to the desired inlet quality. The quality of the refrigerant flow entering the test section was calculated from the enthalpy and the measured pressure at the inlet of the test section.

After all upstream conditions were established, the flow entered the test section. The pressure and temperature were measured upstream and downstream of the short tube. Two-phase refrigerant exiting the test section was condensed and subcooled in the water/glycol cooled heat exchanger (condensation heat exchanger) so that the refrigerant pump had only liquid at its suction side. A liquid receiver was used before the refrigerant pump to ensure only liquid entered the pump. The pressure at the exit of the test section (downstream or evaporator pressure) was controlled by adjusting the temperature and flow rate of chilled water/glycol entering the heat exchanger.

The flexible short tube test section was located between the heat tape and condensation heat exchanger. The current testing utilized flexible short tubes having a length of 14.5 mm and no entrance chamfering. The short tubes were made with flexible material with different moduli of elasticity. Figure 2 shows the schematic of the flexible short tube test section. The flexible short tubes used in this investigation are listed in Table 1.

The lubricant was injected into the suction side of the refrigerant pump using an air-cylinder in a batch process. The amount of the lubricant injected was calculated from the rod displacement and the diameter of the cylinder. Oil concentration was determined by sampling. The procedure for sampling and calculating the oil concentration was based on ASHRAE(1984).

A series of experiments for each short tube were run to investigate the influence of the operating parameters on the mass flow rate though the flexible short tubes. Experiments were performed with an oil concentration of approximately 1.2 %. Conditions were chosen to cover a wide range of operating conditions for a short tube expansion device found in a typical residential air-conditioner.

The experimental variables controlled included: (1) upstream subcooling or quality, (2) upstream pressure and (3) downstream pressure. Operating pressures for the flexible short tubes tested were selected based upon condensing saturation temperatures. In the present study, nominal condensing saturation temperatures of 29.4 °C, 37.8 °C, 46.1 °C, and 54.4 °C were selected. The corresponding upstream saturation pressures were 1179 kPa, 1455 kPa, 1779 kPa, and 2144 kPa. Downstream pressures were selected based upon evaporating temperatures of -1.1 °C, 4.4 °C, and 10.0 °C. Upstream conditions were varied by altering the degree of subcooling from 16.7 °C to 0 °C and altering the quality from 0 % to 5 %. Data were taken only at steady state. Several criteria had to be met to ensure the reliability of the test data. The limits of controlling parameters were set as follows: upstream pressure, ± 7 kPa; downstream pressure, 14 kPa; subcooling, 0.3°C; and quality, ±0.3%.

Temperatures, pressures, flow rate, and power input were monitored in the test loop using a computer and data acquisition system. The temperature at each measuring station was monitored using a thermocouple with a standard uncertainty of 0.4 °C. The pressure at each measuring station was measured using a pressure transducer that was calibrated with a standard dead weight tester. The estimated standard uncertainty of the pressure measurements was ±0.2 % of full scale (3447 kPa). The primary device used to measure mass flow rate was a Coriolis effect mass flow meter. The estimated standard uncertainty of the flow meter was ±0.5 % of full scale (10.9 kg/min). A turbine flow meter was used as a secondary meter for mass flow rate. The estimated standard uncertainty of the turbine meter was ±1.0 % of full scale (0.2 L/s). The two flow measurements showed good agreement each other within 2 %, which is very reasonable tolerance to accept the data as valid in the present experiments. A voltage transformer and a watt transducer were utilized to measure the power input into the heat tape. Estimated experimental standard uncertainty for the power was ±0.5 % full scale (1.5 kW). Table 2 summarizes the standard uncertainties of experimental parameters and inlet quality.

Experimental Results

The flexible short tubes of different moduli were tested under flow conditions encountered by air conditioners. Two different moduli of elasticity flexible short tubes were tested. As upstream pressure was increased, mass flow rate through the flexible short tube increased. The approximately choked conditions seen with rigid short tubes were evident for the flexible short tubes. Once the approximately choked conditions were established, any further decrease in downstream pressure produced minor increases within 5 % in mass flow rate. In general, the flow trends for the flexible short tube were consistent with those of the rigid short tubes, being different only in the magnitude and response to changes in upstream pressure and subcooling.

The experimental results with the flexible short tube were compared with those predicted from an empirical model for a rigid short tube (Kim and O'Neal, 1994). The flexible and rigid short tubes were of slightly different shapes. The flexible short tubes were tapered while the rigid short tubes had a constant cross-sectional diameter along their length. The small diameter side of the flexible short tube was chosen as the diameter for comparison with a fixed short tube.

Generally, the flexible short tubes showed higher flow rates compared with a rigid short tube having the same entrance diameter (D_1=2.06 mm). However, as the condensing pressure increased from 1179 kPa to 2144 kPa, the slope of the flow rate of a flexible short tube with a modulus of 7063 kPa decreased gradually. Thus, for higher condensing pressures the refrigerant flow rate of the flexible short tube was almost the same as that for the rigid short tube. The flow trends would suggest that the pressure around the flexible tube forced the flexible short tube to contract, which would decrease the flow through it.

If the flow rate through the flexible short tube at the design point was set to the same as that of the rigid short tube, the flexible short tube would have a lower flow rate than the rigid short tube at higher ambient temperature above the design point. However, for lower condensing pressure (due to lower ambient temperature) the flexible short tube will have a higher flow rate than the rigid short tube. When a heat pump is operating in the cooling mode with a condensing temperature above the design point, it is desirable to have lower flow rate compared to a capillary tube to balance the lower flow pumped by the compressor. As the condensing temperature decreases, it is desirable to have higher flow rate through the expansion device due to higher compressor pumping rate at the lower condensing pressure.

Effect of Condensing Pressure

Figure 3 shows the effects of upstream pressure on refrigerant mass flow rate through a rigid short tube and flexible short tubes with two different moduli of elasticity. The mass flow rates for the rigid short tube were predicted values using a semi-empirical model developed in an earlier study (Kim 1993).

Figure 3 shows that the 9860 kPa modulus short tube produced higher flow rates for all upstream pressures than that of the 7063 kPa modulus short tube. The 7063 kPa and 9860 kPa modulus short tubes produced mass flow rates of 320 kg/h and 339 kg/h, respectively, at the lowest upstream pressure. As the upstream pressure increased to 2144 kPa, the 7063 kPa and 9860 kPa modulus short tubes produced mass flow rates of 375 kg/h and 399 kg/h, respectively.

It was hypothesized that the exterior of the flexible short tube was exposed to upstream conditions because an axial compression of the flexible short tube provided a small gap between the fixture and the front side of the short tube. However, there was no flow passing through the exterior of the flexible short tube due to sealing at the rear. The combination of low pressure on the interior of the flexible short tube with high pressure on the exterior could result in a constricted inside diameter, which would be smaller than the diameter measured prior to installation. It was also hypothesized that once deformed, the entrance of the flexible short tubes was not sharp edged like the rigid short tube. This would result in higher refrigerant flow rates as compared to the rigid short tube (Kim 1993). The degree of constriction for the flexible short tube would depend upon the modulus of elasticity of the short tube material. The higher modulus material would constrict less than the lower modulus material for a given upstream pressure. The lower modulus short tube showed a smaller increase in flow as the upstream pressure was increased. This behavior would be expected to result from the increased restriction with increased pressure provided by the more flexible, lower modulus short tube.

An aspect of the design of the flexible short tube that would affect mass flow rate was the tapered cross-section of the interior. The progression from a small diameter at the entrance to a larger diameter at the exit would influence the effectiveness of constriction in changing the mass flow rate through the short tube. The greater the difference in the two diameters, the larger the diameter could be after the short tube was compressed. During the reinstallation of the test section, it was also noticed that the compression of the short tube installed in the fixture could result in a bulging or increase in the diameter of the short tube. This could produce an increase in mass flow rate. The holding fixture needs to be redesigned to attain reliability by reducing the compression of the flexible short tube.

Effects of Subcooling/Quality

Figure 4 show the effects of upstream subcooling on mass flow rate for the 7063 kPa modulus short tube. The line for upstream pressure of 1179 kPa in the figure corresponds to a flow model developed for rigid short tubes with the same inlet diameter as the flexible short tube. The general trends seen for the rigid short tube were also seen for the flexible short tube; an increase in subcooling resulted in an increase in mass flow rate. The one difference was the larger decrease in flow rate seen by the flexible short tubes as upstream subcooling was reduced. Rigid short tubes tended to have a fractional flow rate drop of approximately 20 % whereas the flexible tubes showed decreases of 25 % to 30 % in going from subcooling of 16.7 °C to 0 °C. At the upstream pressure of 1179 kPa and subcooling of 16.7 °C, the mass flow rate peaked at 321 kg/h. As the upstream subcooling was decreased to 0 °C, the mass flow rate decreased by 35 % to 208 kg/h. The transition from single-phase at the short tube entrance to two-phase conditions caused a sharp decrease in mass flow rate. Mass flow rate decreased from saturated levels by 25 % to 155 kg/h. The same trends occurred for the other upstream pressures and subcoolings with a maximum mass flow rate of 366 kg/h occurring at a subcooling of 16.7°C and 2144 kPa.

Effect of Evaporating Pressure

Figure 5 shows the effects of downstream pressure on mass flow rate through a flexible short tube with a modulus of 7063 kPa. Approximately choked flow conditions existed for all pressures tested with flow rates decreasing by a maximum of 4 %. The figure shows that at higher upstream pressures, there was less of a tendency to increase flow rate as the downstream pressure was lowered. For an upstream pressure of 1779 kPa the drop in mass flow rate with an increase in downstream pressure was less than 1 %.

Mass Flow Model

A modified single-phase flow model has been successfully used in correlating two-phase flow data through short tubes (Kim 1993). Data are typically correlated with a "modified" downstream pressure and a series of constants. The present flow model was derived from the single-phase orifice equation with adequate modifications to capture the flow characteristics through flexible short tubes. The single-phase orifice equation used for short tubes can be derived from equations of continuity and energy equation with the given assumptions (ASME 1971). The single-phase orifice equation for single-component, single-phase substance is given as:

$$\dot{m} = CA_s\sqrt{2\gamma_c\rho\left(P_{up} - P_{down}\right)/\left(1-\beta^4\right)} \quad (1)$$

where β is the ratio of orifice throat diameter to upstream tube diameter.

After dropping the term $(1-\beta^4)$ from equation (1) due to small values of $(1-\beta^4)$ compared with unity, the term P_{down} was replaced by P_f, the pre-flashing pressure, to satisfy the pressure conditions at the downstream flow conditions. The adjusted downstream pressure, P_f, covered the assumptions of incompressible flow and choked flow conditions. In this study, the orifice constant, C, was set equal to unity and P_f was correlated with the experimental data. The flow model is given by:

$$\dot{m} = CA_s\sqrt{2\gamma_c\rho\left(P_{up} - P_f\right)} \quad (2)$$

$$P_f = P_{sat}\left[b_1 + b_2 \cdot PRA^{b_3} \cdot SUB^{b_4} + b_5 \cdot EVAP^{b_6}\right] \quad (3)$$

where,

$EVAP$ = $(P_c - P_{down})/P_c$ (P is in absolute pressure)

PRA = P_{up}/P_c

SUB = $(T_{sat} - T_{up})/T_c$

P_c = critical pressure for R-22, kPa

P_{down} = downstream pressure(evaporating pressure), kPa

T_{sat} = saturated liquid pressure corresponding to upstream temperature, kPa

T_c = critical temperature for R-22, K

A_s = area calculated with inlet diameter, m^2

Based on the measured data, the adjusted downstream pressure, P_f, was correlated with a normalized form of inlet subcooling, upstream pressure, downstream pressure, and flexible short tube diameter. The liquid saturation pressure, P_{sat} , was used as a reference value for P_f , because flashing occurred when the pressure was near P_{sat}. A correlation between the adjusted downstream pressure of P_f and normalized parameters was determined using a non-linear regression technique along with experimental data. All coefficients included in the flow model are given at Table 3. Due to limited data (two moduli of elasticity were tested in the present study), the present flow model was developed separately for each flexible short tube.

Using Equations (2) and (3), and the coefficients in Table 3, the mass flow rate can be estimated for a given set of operating conditions and with either modulus of elasticity. When applying the empirical flow model, it should be understood that the application of the model has a limited range due to the limited test range of the experimental data (Table 4). Approximately 95 % of the measured data were within ±5 % of the model's prediction. The maximum difference between the measured data and the model's prediction was within ±8 %.

Flexible Short Tube Performance Simulation

A nominal 5.3 kW capacity room air conditioner with a coefficient of performance of 2.34 at 35 °C was simulated with the flexible short tube and three other expansion devices (capillary tube, rigid short tube, and thermal expansion device). Performance estimates were made for outdoor temperatures ranging from 20 to 46 °C.

The proprietary simulation model had several key features, including: (1) tube-by-tube model of heat exchangers, (2) refrigerant inventory estimation, (3) compressor map or first principles compressor model, (4) models for four types of expansion valves (TXV, capillary tube, rigid short tubes and flexible short tubes). The empirical equations describing flow for the flexible short tubes in the previous section were used to model these types of short tubes. The condensate from the evaporator was sprayed over the condenser coil using the condenser fan. The system's basic physical descriptions are provided in Table 5. The unit had a reciprocating compressor with a charge of 680 g of R-22 refrigerant. Each expansion device was sized to give the same refrigerant mass flow rate, evaporator superheat, capacity, and COP at 35 °C outdoor air temperature.

The results of the simulation are provided in Figures 6 through 9. For the mass flow rate, capacity, and COP, the results were normalized to 35 °C outdoor temperature. The original intent of the design of the flexible short tubes was that they would provide control better than a fixed area mass flow expansion device. Figure 6 shows that the flexible short tube provides better flow control than either the rigid short tube or capillary tube, but not as good as the TXV. At outdoor temperatures above 45 °C, the refrigerant leaving the evaporator became saturated for both the rigid short tube and the capillary tube (Figure 7). In addition, a dirty condenser, which is very common at outdoor temperatures above 45 °C, can cause the system to operate at these extreme conditions. As shown in Figure 7, the flexible short tube can prevent liquid flood back at high outdoor temperatures. For outdoor temperatures below 25 °C, the flexible short tube provided 2 °C to 4 °C less superheat than the fixed expansion devices.

The flexible short tube provided for 2 % to 3 % higher capacity relative to the capacity of the fixed expansion device system at lower (<25 °C) outdoor temperatures and higher (>45 °C) outdoor temperatures (Figure 8). Capacities with the TXV controlled system were about 1 % higher relative to the flexible short tube controlled system. Normalized COPs followed a similar trend except the percentage improvements were larger.

Conclusions and Recommendations

The flexible short tube provided better control of an air conditioner system than conventional fixed short tubes or capillary tubes. The general trends, such as mass flow rate versus upstream subcooling/quality and pressure, observed in the flexible short tubes were consistent with results for rigid short tubes (Aaron and Domanski, 1990; and Kim and O'Neal, 1994). For the flexible short tubes the slope of mass flow rate versus upstream pressure was dependent on the modulus of elasticity of the short tube material. The higher upstream pressures forced the flexible short tube to change shape and restrict flow. The restriction in flow provided better control of evaporator outlet superheat than a rigid short tube at high outdoor temperatures when the condenser pressure was high. Short tubes with two different moduli of elasticity were evaluated. The data indicated that the more flexible short tube (the one with the modulus of 7063 kPa) provided better flow control than the stiffer short tube (modulus of 9860 kPa).

There are a number of important engineering issues that must be addressed if a flexible short tube is ever to be used in air conditioning systems. These include: optimum flexibility of the short tube, long-term operating performance and reliability compared to conventional rigid short tubes, plus material compatibility with current and future refrigerants and lubricants. Research will need to be conducted to evaluate the optimum flexibility that would provide the best control characteristics without risking a total collapse of the short tube.

Long-term performance and reliability of flexible short tubes should be evaluated. While this study demonstrated their performance benefits in the laboratory, it is not the same as a long term evaluation in the field. Rigid short tubes are typically made of a metal, such as brass. The flexible short tubes used in this study were made from elastomeric materials that provide the flexibility. Long-term tests would need to be conducted to determine if these materials maintain their resiliency and do not erode after years of use.

Material compatibility of flexible short tubes relates to long-term performance. There are a wide variety of refrigerants and lubricants currently in use. Short tube materials that work for one refrigerant may not work for another. Implementing flexible short tubes will require a comprehensive study of material compatibility.

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Table 1 Characteristics of the flexible short tube used in the test

Tube No.	**Uncontracted Diameter, mm**	**Modulus of Elasticity**
1	D_1=2.06 D_2=2.46	7063 kPa
2	D_1=2.06 D_2=2.46	9860 kPa

Table 2 Uncertainties of experimental parameters

Parameters	Standard Uncertainty
Temperature	±0.4 °C
Pressure	±6.9 kPa
Mass flow rate	±0.5 % of reading
Power input	±0.5 % of rated output
Quality (estimated)	±2.5 % of calculated value

Table 3 Coefficients for the flow model

Coefficients	Modulus of Elasticity	
	7063 kPa	9860 kPa
B_1	0.9187	1.0434
B_2	10.4091	7.4048
B_3	-0.0500	0.0191
B_4	1.1787	1.0421
B_5	-0.0276	-0.1683
B_6	0.9241	-0.7797

Table 4 Limitations on the application of the flow model

Parameters	Minimum	Maximum
L	14.5 mm	14.5 mm
D_1	2.06 mm	2.06 mm
D_2	2.46 mm	2.46 mm
Modulus	7063 kPa	9860 kPa
P_{up}	1176 kPa	2149 kPa
P_{down}	481 kPa	Psat
Subcooling	0 °C	16.7 °C
Oil Concentration	1.2 %	1.2 %

Table 5 Description of the evaporator and condenser used in the window air conditioner

Item	**Evaporator**	**Condenser**
Rows	2	2
Tubes	7 mm enhanced	7 mm enhanced
Fin Density (fins/m)	630	709
Face Area (m^2)	0.155	0.245
Airflow (m^3/min)	13.0	29.8

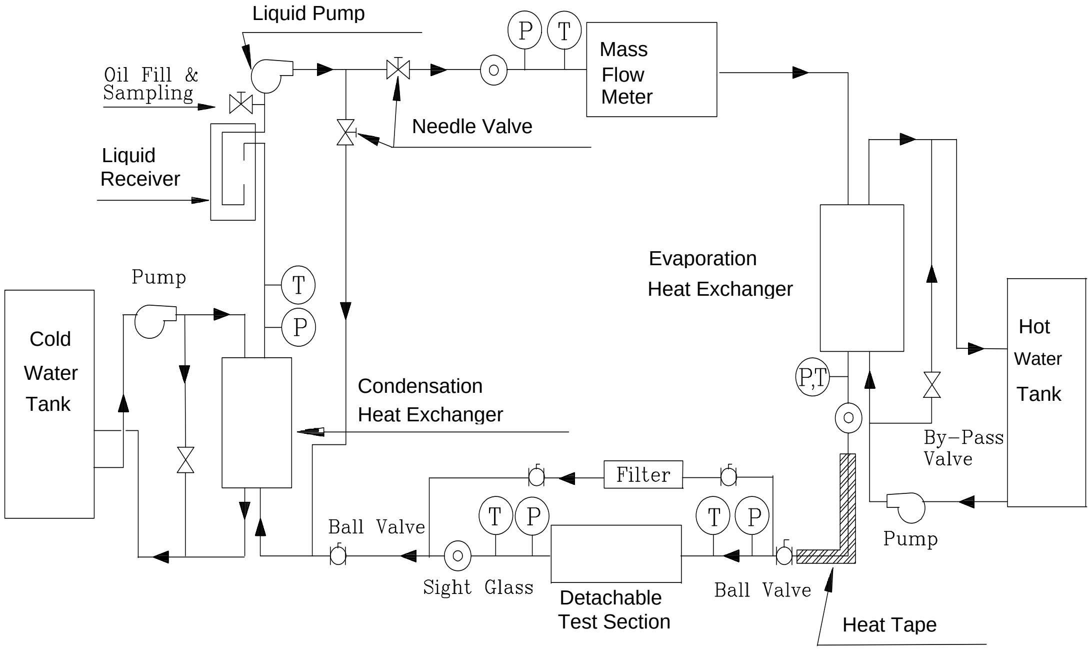

Figure 1 Schematic diagram of the flexible short tube test setup.

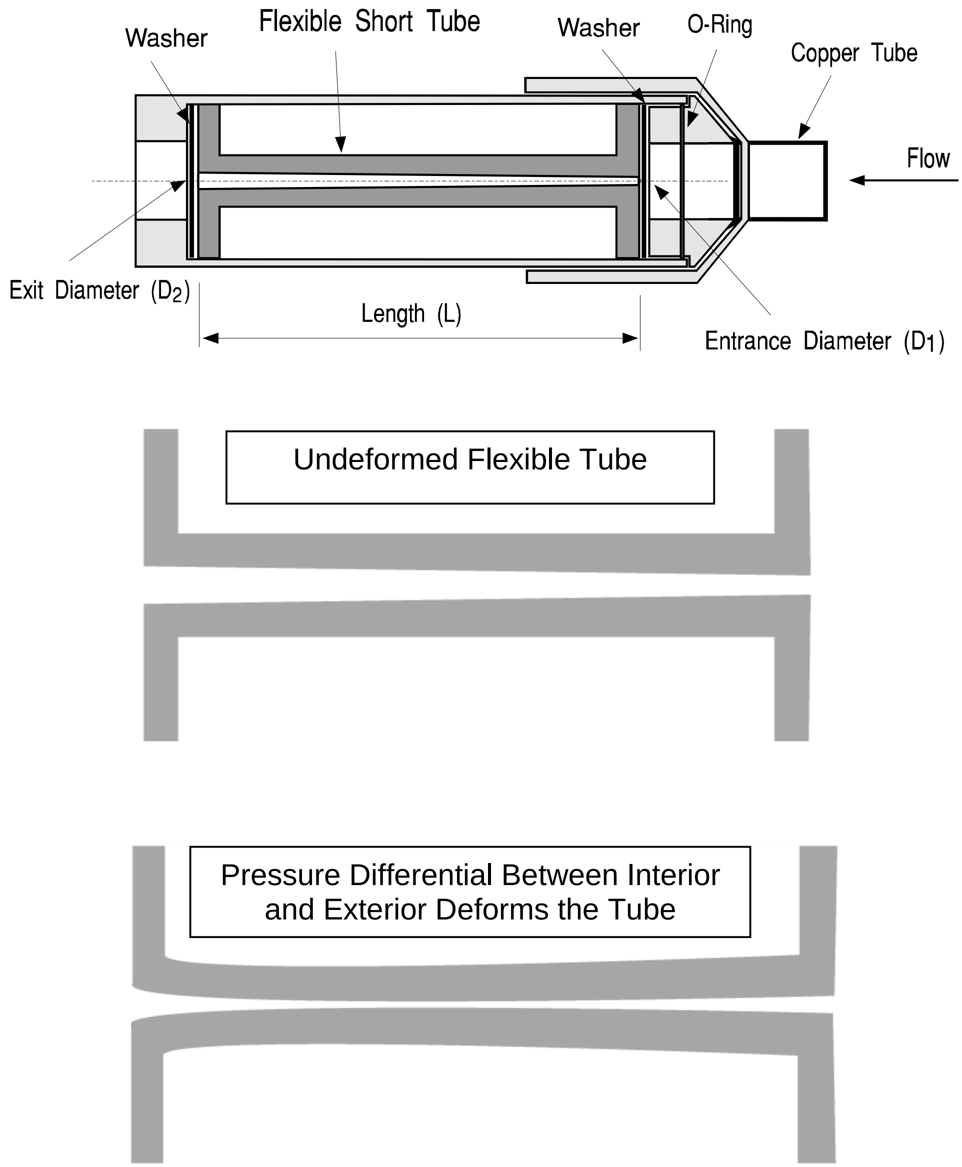

Figure 2 Schematic of a flexible short tube test section.

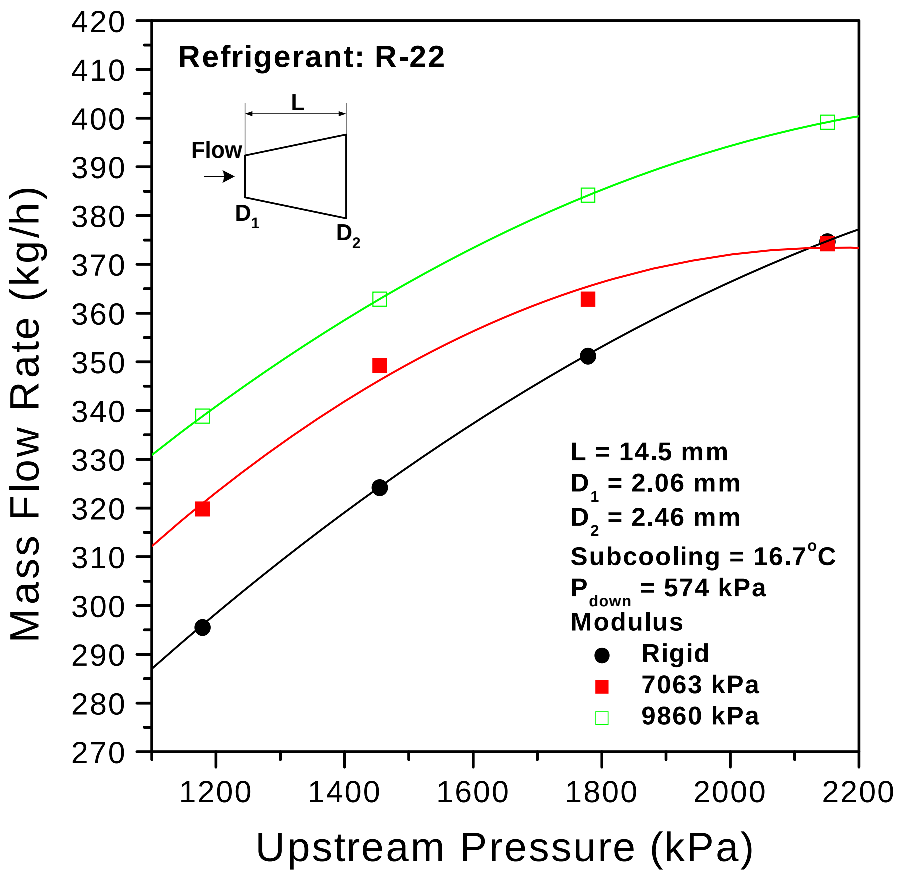

Figure 3 Effect of upstream (condensing) pressure and modulus of elasticity at 16.7 °C

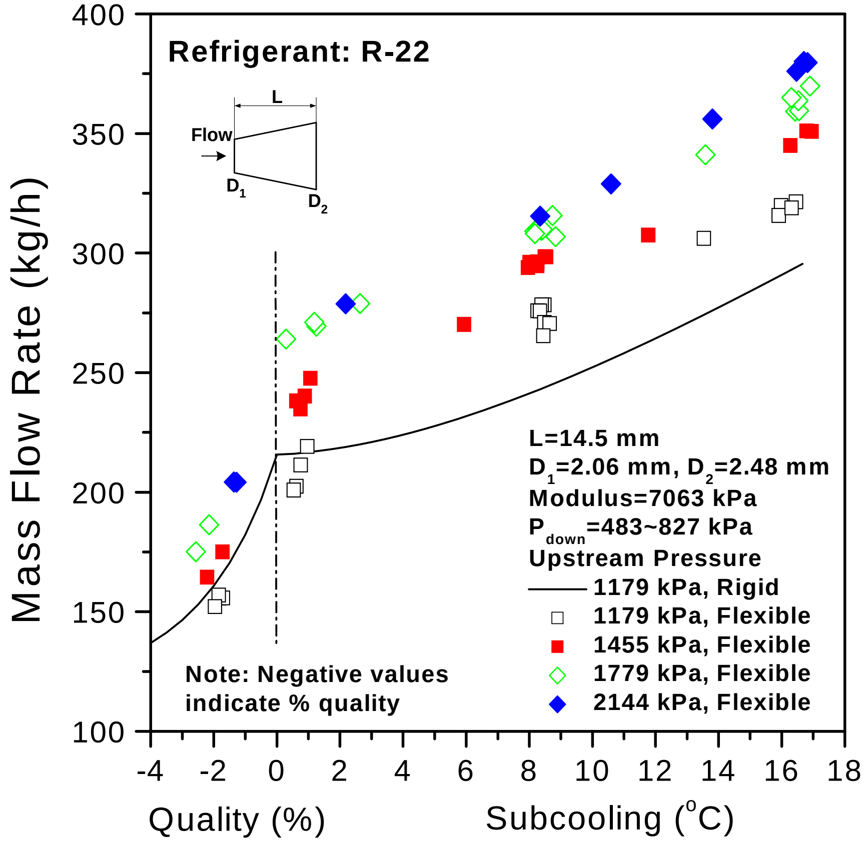

Figure 4 Comparison of the flow rate of a flexible tube to that of a rigid short tube.

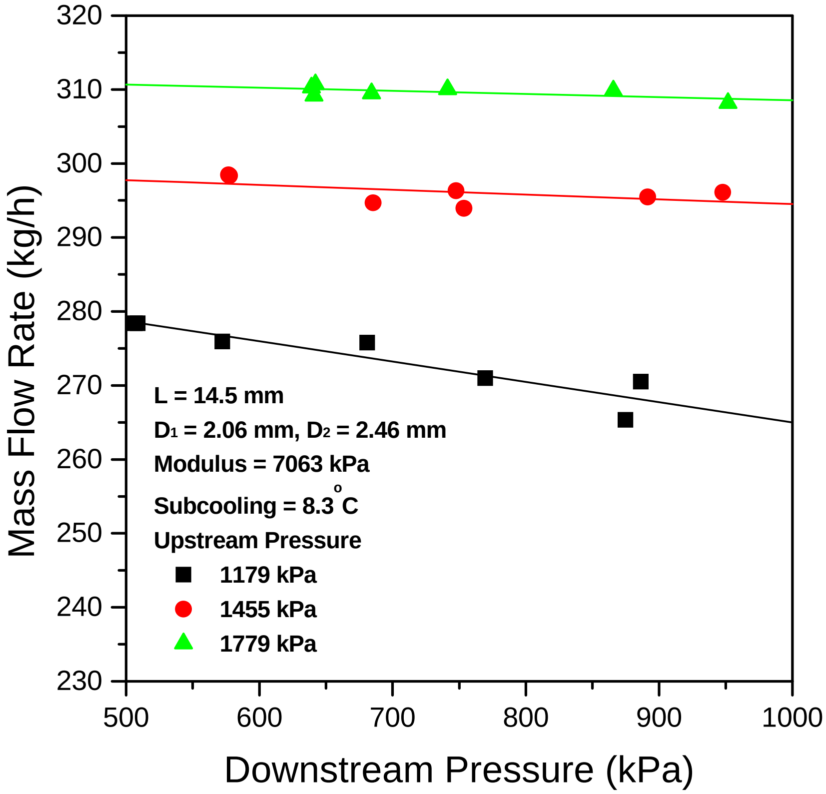

Figure 5 Effect of downstream (evaporating) pressure on flow rate for a flexible short tube with modulus of 7063 kPa.

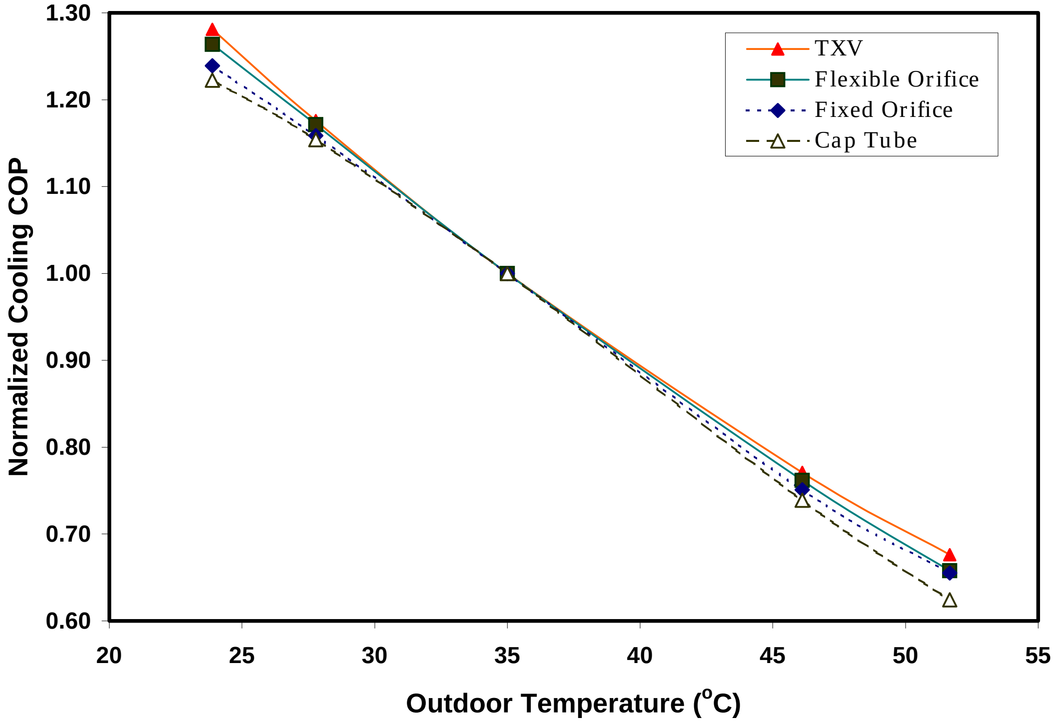

Figure 6 Normalized cooling COP for the window air conditioner as a function of outdoor temperature for four expansion devices.

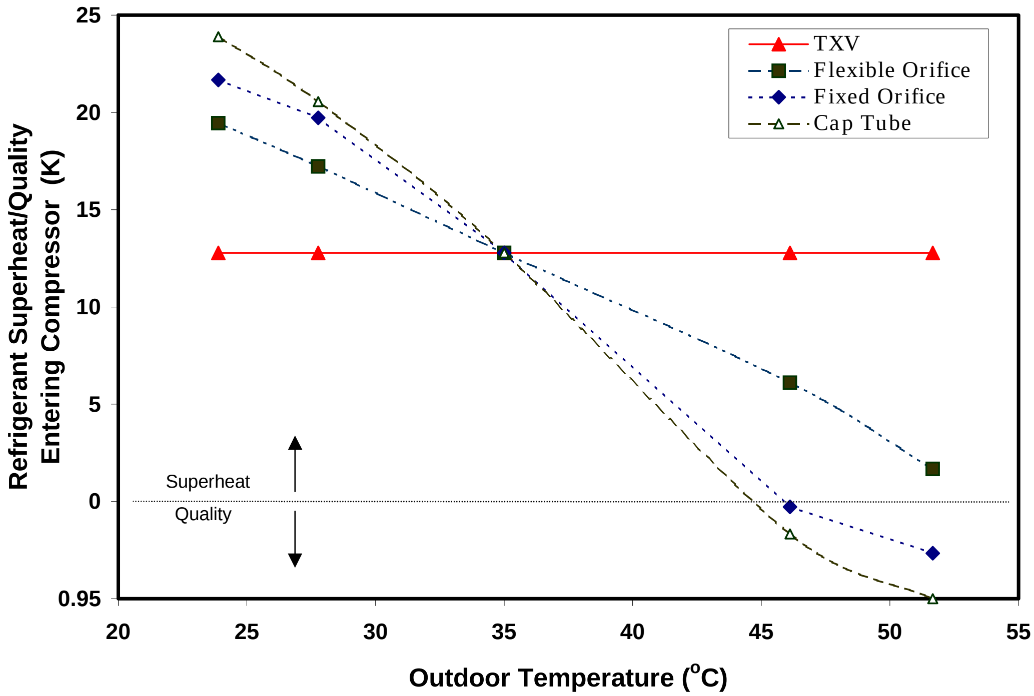

Figure 7 Refrigerant superheat/quality entering the compressor.

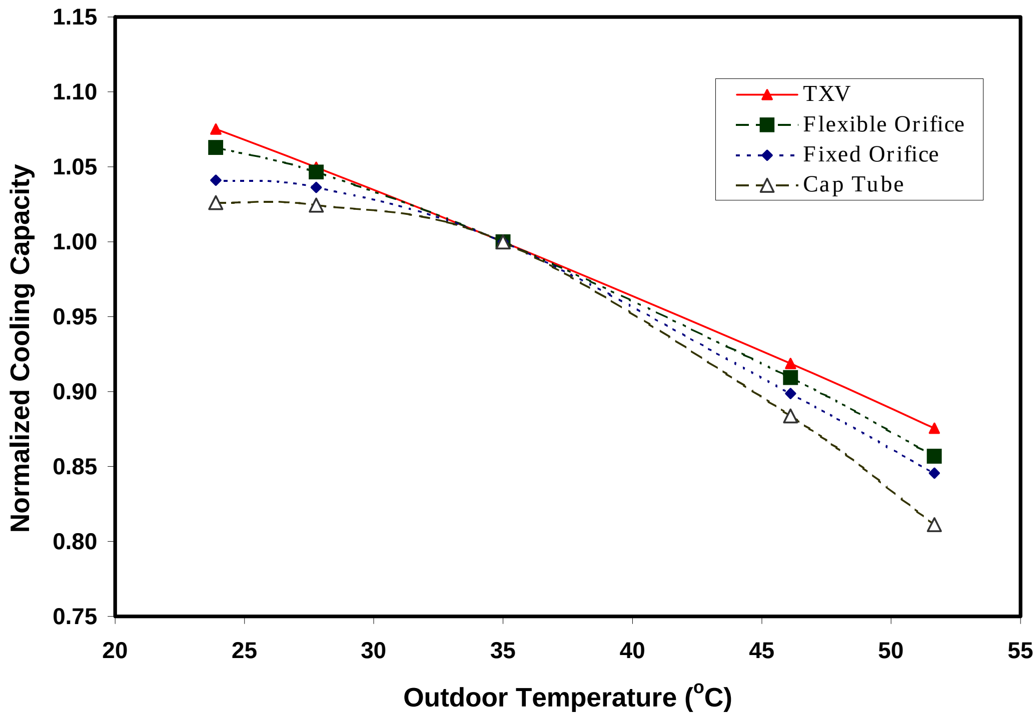

Figure 8 Estimated normalized cooling capacity for the window air conditioner as a function of outdoor temperature for four expansion devices.

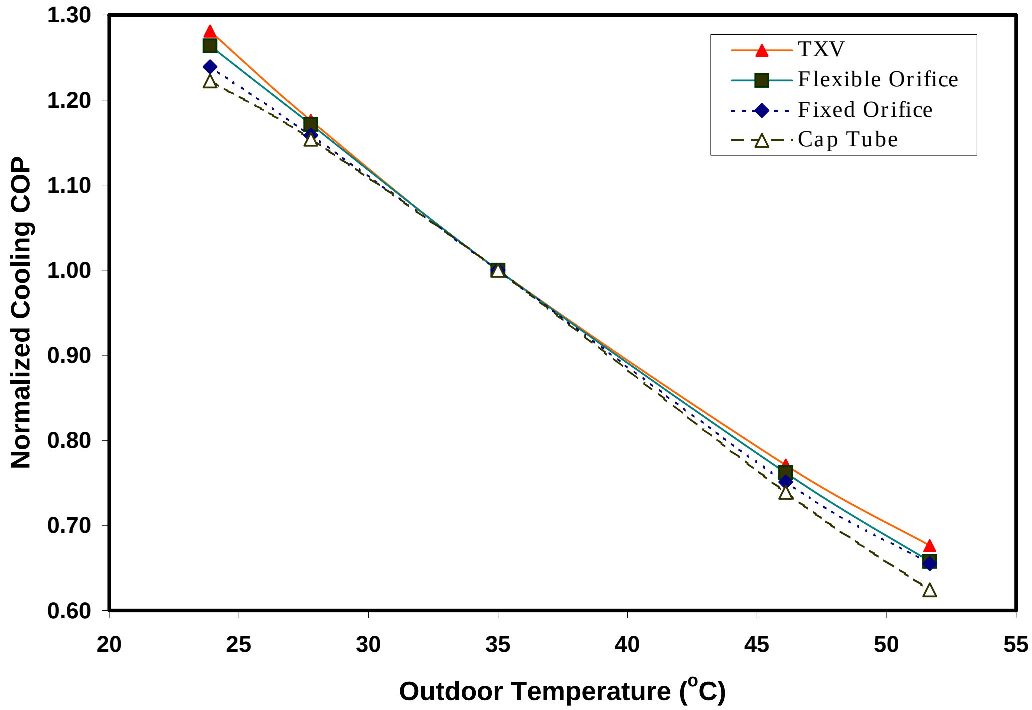

Figure 9 Normalized cooling COP for the window air conditioner as a function of outdoor temperature for four expansion devices.