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# Short-Tube Refrigerant Flow Restrictors

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## ABSTRACT

Five short-tube refrigerant expansion devices, with L/D ratios from 7.5 to 11.9, were tested on a three-ton split-system heat pump. The test results show that first-stage choking has occurred at the liquid subcooling temperature of 22.2 C° (40.0 F°). However, there is no indication that second-stage choking has ever occurred.

Based on the test results, two correlations were formed. One is an orifice model with the orifice constant as a function of the pressure differential and the level of liquid subcooling. This model is for liquid subcooling less than 22.2 C° (40.0 F°). The other is a first-stage choking model, which is for a level of liquid subcooling above 22.2 C° (40.0 F°).

## INTRODUCTION

For air-conditioning and refrigerating systems, capillary tubes[1-7] and expansion valves[8-10] have long been regarded as the standard refrigerant expansion devices. The capillary tube operates on the principle that liquid passes through it much more readily than does gas. The expansion valve consists basically of a sharp-edged orifice whose throat area can be varied. A third type of expansion device, the short tube, has recently been adopted by some heat pump manufacturing companies, and because of its low cost and high reliability, this kind of flow restrictor probably will be more widely used in the future. Because the use of short tubes is relatively new in the air-conditioning field, however, quantitative knowledge about their application is inadequate. Furthermore, the information often is considered proprietary.

Previous research activities concerning flow in short tubes were mainly conducted by those involved with steam-water two-phase flow, basically used for nuclear reactor cooling systems analysis.[11-14] In the findings, the characteristics of such flow were clearly defined;[13] however, such correlations cannot be applied directly to heat pump operation, because the operating conditions are widely different and also because the physical properties of the refrigerant are not comparable to those of water. Pasqua discussed the flow of saturated and subcooled liquid refrigerant R-12 through small-diameter orifices, short tubes, and nozzles.[15] He also presented a correlation to calculate the mass flow rate of saturated liquid, yet he did not quantify the influence of liquid subcooling. Davis and Daniels have investigated single- and two-phase flow of refrigerant R-12 through sharp-edged orifices.[16] However, they noted that the correlation they used to calculate the liquid mass flow rate through orifices cannot be applied in the design of expansion devices for refrigeration and air-conditioning systems.

In this study, five short tubes, with length-to-diameter (L/D) ratios from 7.5 to 11.9, were tested on a heat pump system with R-22 as the fluid. While the test results for the subcooled liquid have the same general flow characteristics as those presented by Pasqua[15] and Zaloudek,[13] they also show that the fluid mass flow rate is affected not only by the pressure differential across the tube, but by the level of liquid subcooling as well. The results also show that first-stage choking[13] (caused at the vena contracta by liquid evaporation, which is then recondensed by and reattached to the tube wall) will happen if the level of liquid subcooling is high. However, no evidence has been found that second-stage choking[13] (caused by liquid flashing into vapor near the exit of the tube) has ever occurred at conditions obtainable in the test facility.

Based on the test results, two correlations were formed. One is the orifice model with the orifice constant as a function of (1) the pressure differential across the short tube and (2) the level of liquid subcooling. This model is designed for liquid subcooling less than 22.2 C° (40.0 F°). The other model is formed based on the first-stage choking model[13] with the constant in the model correlated to the level of liquid subcooling only. This correlation is suggested for levels of liquid subcooling above 22.2 C° (40.0 F°).

For heat pump operation, the level of refrigerant subcooling is always an important factor to be considered, but this factor has been generally neglected by most previous researchers. This paper presents a simple method to account for the influence of the level of liquid subcooling. The correlations presented in this paper should be particularly useful to heat pump design engineers in sizing short tubes as refrigerant expansion devices and in predicting off-design operating conditions.

## THEORETICAL ANALYSIS

In 1951 Bailey found that there was a transition zone between the full tube and free jet-type metastable flow for the short-tube water stream.[12] In 1953 Pasqua concluded experimentally that short-tube flow of subcooled liquid exhibited four patterns[15] (Fig. 1). In 1963 Zaloudek reported the same findings.[13] Zaloudek found (Fig. 1) that, at the very low pressure differential along region AB, the fluid flow exhibited the usual $G \sim (\Delta P)^{1/2}$ relationship commonly observed for such flow geometries conveying a single-component, single-phase substance.

$$G = \frac{\dot{m}}{A} = C[2g_c\rho_l\ (P_{up} - P_{down})/(1 - \beta^4)]^{1/2} \tag{1}$$

Because the diameter ratio of the short-tube opening over the liquid line diameter is very small, $\beta^4$ can be neglected so that

$$G = \frac{\dot{m}}{A} = C[2g_c\rho_l\ (P_{up} - P_{down})]^{1/2} \tag{1a}$$

Starting at point B (Fig. 1), a discontinuity in slope occurs because the pressure at the vena contracta has reached the saturation pressure. Moving from point B to point C, more and more liquid evaporation occurs, but the tube wall is able to condense the vapor. The region BC is the first-stage choking region. The pressure at the vena contracta near the entrance of the tube is nearly equal to the saturation pressure of the water. Based on his test results, Zaloudek suggested that the mass flow rate per unit area could be calculated by a form of the formula

$$G = C_1[2g_c\rho_l\ (P_{up} - P_{sat})]^{1/2} \tag{2}$$

where $C_1$ is the adiabatic coefficient of discharge and is approximately 0.61 to 0.64.

After point C (Fig. 1), the tube wall length was no longer able to condense the vapor, which resulted in a free jet-type discharge and reestablished the $G \sim (\Delta P)^{1/2}$ relationship. In the region CD, the flow pattern was in the form of a ring of steam-water mixture surrounding a core of liquid water. The water core was at a sufficient velocity to preclude the formation of steam within the length of the tube, which was at or even below the saturation pressure. It is clear that the water in the core was in a metastable or nonequilibrium state. The pressure along the tube assumed a value that closely approximated the downstream pressure. When the back

pressure was further lowered, a second or downstream choking condition was generally established. This condition is characterized by the line DE. The location of choking is at the terminus of the tube, since the pressure along the tube had no measurable change, even when the downstream pressure was varied below the critical pressure.

Burnell hypothesized that, under second-stage critical-flow conditions, the surface tension of the water delays the formation of vapor bubbles.[11] He proposed the following expressions for initially compressed water:

$$G = 2g_c\rho_l[P_{up} - (1 - C_2)P_{sat}]^{1/2} \tag{3}$$

where $C_2$ is a surface-tension-dependent constant; that is,

$$C_2 = K\frac{\sigma_{sat}}{\sigma_{175}} \tag{4}$$

where $K$ is an experimentally determined coefficient equal to 0.264.

Using more accurate values of water surface tension, Kinderman and Wales proposed the following modification to Eq 4:[17]

$$C_2 = K\frac{\sigma_{sat}}{\sigma_{200}} \tag{5}$$

where $K = 0.284$.

This model considers only the influence of surface tension on the bubble growth process and does not recognize the effects of heat and mass transfer; neither does it predict any geometrical dependency.

The above models clearly defined the short-tube flow patterns. However, since most of the tests used steam and water as the two-phase fluid, their results could not be applied for air-conditioning purposes because the properties of water were not the same as those of refrigerants. Eqs 1a, 2, and 3 also do not include the influence of the level of liquid subcooling, which is very important in determining the performance of the air-conditioning systems.

The limited data derived from Domingorena's[18] (the original mass flow rates were corrected, using a turbine meter calibration) heat pump tests (Fig. 2), with L/D = 9.1 and with R-22 as the fluid, indicated that for the pressure differentials from 1151 to 1551 kPa (167.0 to 225.0 psi) the liquid subcooled from 8.3 to 19 C° (15.0 to 34.2 F°); the value of the orifice constant, $C$, varied from 0.37 to 0.54; and the value of $C_1$ of the first-stage choking model, Eq 2, varied from 0.77 to 0.90. The values of $K$, calculated from Eqs 3 and 5, were all negative, which indicated that the fluid flow did not exhibit any second-stage choking. The variation of $C$ and $C_1$ indicates that the simple orifice and first-stage choking models are inadequate for the present analysis.

Because the pressure differential and level of liquid subcooling are the two most important factors in determining short-tube performance, the effect of each factor will have to be characterized before correlations including these two factors can be formed. From the variation of $C$ and $C_1$, one must correlate the pressure differential and liquid subcooling to $C$ and $C_1$ in order to use the orifice and first-stage choking models.

## TEST SETUP AND TEST PROCEDURE

The tests were performed on a three-ton split-system heat pump with R-22 as the refrigerant. Fig. 3 is the schematic of the setup. Two turbine-meter-calibrated rotometers were used for flow rate measurement. Pressure gauges were added before and after the refrigerant expansion devices for upstream and downstream pressure measurements. Chromel-Alumel thermocouples were used as the temperature-sensing elements. In addition, a concentric-tube heat exchanger was installed on the refrigerant liquid line. This 1.8-m-long (6-ft-long) heat exchanger, which could either add heat to or take heat from the liquid refrigerant by passing hot steam or

cool water through it, provided control of the liquid subcooling before the expansion device. Five 1.27-cm-(1/2-in.-) long short tubes with L/D ratios from 7.5 to 11.9 were tested under cooling-mode operation. The L/D ratio was varied by changing tube diameters.

The indoor and outdoor unit air-circulation loops were the same as those used by Domingorena.[18] By controlling the amount of the condenser air recirculation, the system's high-side pressure could be varied, but not without also affecting the downstream pressure level. Thus, because the tests of the short tubes were conducted on an actual heat pump, the upstream pressure also could not be held fixed while varying the downstream pressure. However, Zaloudek has noted that the characteristic flow regions can be approached from either direction without affecting the point at which transition occurs from one type of region to the next.[13] Also, in actual heat pump operations, increasing the level of liquid subcooling will reduce the high-side-to-low-side pressure differential. Theoretically, if one increases the level of liquid subcooling, one should encounter the first-stage choking region, BC, and eventually the region AB, as shown in Fig. 1.

All the data were gathered under steady state operating conditions. The data collected for each test were the pressures on both sides of the expansion device, the upstream liquid temperature, and the refrigerant mass flow rate.

The heat pump was first operated at very low high-side pressure, with low temperature airflows through the condenser. After collecting the data from a very low to a very high degree of liquid subcooling, part of the air in the condenser loop was recirculated to boost the high-side pressure. The data were again collected for various levels of liquid subcooling.

To examine separately the influence of the pressure differentials across the tube, parts of the tests were performed under constant liquid subcooling conditions. This was achieved by adjusting the level of liquid subcooling, at different heat pump operating conditions, with the liquid-line heat exchanger. Since it was very time-consuming to perform the tests at various pressure differentials with the same liquid subcooling, only enough tests were performed to establish the effect of the pressure differentials on the flow characteristics. The rest of the tests were performed from very low to very high levels of liquid subcooling for various pressure differentials. With the effect of pressure differentials already determined from the constant liquid subcooling tests, the effect of the level of liquid subcooling could be identified and characterized from the remaining tests.

## TEST RESULTS AND DISCUSSION

Fig. 4 represents the tests at different levels of liquid subcooling for three short tubes with different L/D ratios. The results show that pressure drops across the tube do affect the value of the orifice constant, $C$, for the general orifice model, Eq 1a. Under a specific degree of liquid subcooling and the influence of the pressure differential, $\Delta P$, an orifice constant, $C$, can be represented by the following linear equation.

$$C = B_1\sqrt{\Delta P} + B_2 \tag{6}$$

where $B_1$ is the slope of the straight lines, which turns out to be constant for all the tests and is equal to $-0.007364\ \mathrm{kPa}^{-1/2}$ ($-0.019337\ \mathrm{psi}^{-1/2}$).

The negative slope means that the short tube will be less effective when the pressure differential increases. $B_2$ in Eq 6 represents the influence of the liquid subcooling.

However, for the first-stage choking model, Fig. 5 shows that, at a specific level of liquid subcooling, the value of $C_1$ is not dependent on the values of differentials of the upstream pressure and liquid saturation pressure. It is a function of the level of liquid subcooling only.

For all of the tests, the values of $K$ for the second-stage choking model, Eqs 3 and 5, are either negative or very small, which indicates that second-stage choking does not occur.

For most of the tests, both the level of liquid subcooling and pressure differentials are variables. To separate the effects of liquid subcooling on the fluid flow, the following method is adopted.

For each test, the orifice constant, $C$, is calculated from Eq 1a with measured $\Delta P$ and mass flow rate; $B_1$ is found from Fig. 4; $B_2$ can then be calculated from Eq 6. If one defines $C_{eq}$ as the equivalent orifice constant at, $\Delta P$ = 1034.2 kPa (150.0 psi), $C_{eq}$ can be calculated for each test from Eq 6:

$$C_{eq} = B_1\sqrt{1034.2} + B_2 \tag{7}$$

or, in psi,

$$C_{eq} = B_1\sqrt{150.0} + B_2 \tag{7a}$$

All $C$ values were converted into $C_{eq}$ values, because $C$ was obtained at different levels of liquid subcooling and at different pressure differentials, while the values of $C_{eq}$ are calculated at different levels of liquid subcooling but at a constant pressure differential of 1034.2 kPa (150.0 psi). In other words, $C_{eq}$ is a function of the level of liquid subcooling only. If one plots $C_{eq}$ as a function of liquid subcooling, one can study the effect of liquid subcooling alone on the fluid flow characteristics.

Figs. 6 and 7 show $C_{eq}$ and $C_1$ as functions of liquid subcooling, $\Delta T$. The figures indicate that for all the short tubes, with L/D from 7.5 to 11.9 and mass flow rate of 68.9 to 213.2 kg/h (151.0 to 470.0 lb/h), $C_{eq}$ can be expressed as a linear function of liquid subcooling for liquid subcooling less than 19.4 C° (35.0 F°).

$$C_{eq} = B_3\Delta T + B_4 \tag{8}$$

where $B_3$ = 0.0108 $(C°)^{-1}$ [or $B_3$ = 0.006 $(F°)^{-1}$ ] and $B_4$ is the projected value of $C_{eq}$ at zero degree of liquid subcooling and is equal to 0.40.

The above equation will have a maximum potential error of around 7.5%, which occurs at a very low level of liquid subcooling.

Substituting Eq 8 into Eq 7,

$$B_2 = B_3\Delta T + B_4 - B_1\sqrt{1034.2} \tag{9}$$

or, in psi,

$$B_2 = B_3\Delta T + B_4 - B_1\sqrt{150.0} \tag{9a}$$

Eq 9 represents the effect of liquid subcooling on the fluid flow. Substituting Eq 9 into Eq 6, the correlation of the orifice constant, $C$, is a function of both $\Delta T$ and $\Delta P$,

$$C = -0.007364(\sqrt{\Delta P} - \sqrt{1034.2}) + 0.0108\Delta T + 0.40 \tag{10}$$

or, in psi,

$$C = -0.019337(\sqrt{\Delta P} - \sqrt{150.0}) + 0.006\Delta T + 0.40 \tag{10a}$$

Fig. 7 shows that $C_1$ is constant for 19.4 C° < $\Delta T$ < 22.2 C° (35.0 F° < $\Delta T$ < 40.0 F°), which is the first-stage choking region. For $\Delta T$ > 22.2 C°, the value of $C_1$ drops and the slope of $C$ flattens slightly (Fig. 6). This effect possibly indicates the transition to the region AB of Fig. 1, where the fluid flow again reestablishes the $G \sim (\Delta P)^{1/2}$ relationship.

From the test results, one can summarize that for 19.4 C° < $\Delta T$ < 22.2 C° (35.0 F° < $\Delta T$ < 40.0 F°), the flow is in the first-stage choking region, BC (Fig. 1). For $\Delta T$ > 22.2 C° (40.0 F°), it will be in the region AB. Second-stage choking (region DE) did not occur at all even at very low levels of liquid subcooling.

For engineering applications, use Eq 10 or 10a, together with Eq 1a for liquid subcooling up to 22.2 C° (40.0 F°); however, in the region 19.4 C° < $\Delta T$ < 22.2 C° (35.0 F° < $\Delta T$ < 40.0 F°), Eq 2 would be theoretically more accurate. For most heat pump operations, Eqs 1 and 10 or 10a should be adequate to estimate the design and off-design conditions. However, for those who want to operate the heat pump at a liquid subcooling higher than 22.2 C° (40.0 F°), use Eq 2 instead of a formula similar to Eq 1, with

$$C_1 = 0.9175 - 0.00585\Delta T \qquad (22.2\ \text{C}° < \Delta T < 27.8\ \text{C}°) \tag{11}$$

or

$$C_1 = 0.9175 - 0.00325\Delta T \qquad (40.0\ \text{F}° < \Delta T < 50.0\ \text{F}°) \tag{11a}$$

The reason for this suggestion is that for $\Delta T$ > 22.2 C° (40.0 F°), $C$ varies with the short-tube diameter. This is probably because the refrigerant remains in the liquid phase all the way through the tube. For such a situation, frictional pressure drops become more significant and will depend on the tube L/D ratio. The value of $C$, therefore, will be slightly different for each short tube, as shown in Fig. 6. There is, however, no apparent difference in $C_1$ values in this region (Fig. 7).

Mathematically, Eq 2 could be used as the correlating equation for the entire heat pump operating range, with $C_1$, say, as an exponential function of the level of liquid subcooling. Since Eq 2 does not include the downstream pressure, it is easier, particularly for heat pump system analysis, to apply. However, $C_1$ is very sensitive to $(P_{up} - P_{sat})$ at a low level of liquid subcooling; thus, larger errors in $C_1$ could result from a small deviation in the level of liquid subcooling.

From Figs. 6 and 7, it is clear that for liquid subcooling less than 22.2 C° (40.0 F°) the flow characteristics for all five tested short tubes are identical, which means that the ratio L/D is adequately accounted for by the cross-sectional area term in $G$ in Eq 1a.

For most heat pump operations, the level of liquid subcooling is less than 22.2 C° (40.0 F°). For this application range, Eq 10 in conjunction with Eq 1a is recommended.

In order to size the short tubes, one has to know the upstream-to-downstream pressure differential. One also has to know the level of liquid subcooling at the entrance to the tube (as opposed to condenser subcooling) and liquid flow rate. With the above information, the value of $C$ can be found from Eq 10, and the tube opening can be calculated from Eq 1a.

It was found that, when the short-tube L/D ratio was lower than 7.5, the fluid before the expansion device became two-phase. An attempt to increase the liquid subcooling only resulted in an unstable flow rate. Since the heat pump in use in the tests was a three-ton unit with a short tube, L/D = 7.5, as its original component for cooling mode operation, the above correlations have only been validated for heat pumps with a capacity of three tons or less.

## CONCLUSION

Five short-tube refrigerant expansion devices with L/D ratios from 7.5 to 11.9 were tested on a three-ton air-to-air split heat pump system to obtain representative flow characteristics. The test results showed that the level of the upstream liquid subcooling and the pressure differential across the short tube are the two major factors affecting the flow characteristics.

Within the normal heat pump operating range, there was no indication that downstream choking occurs. There was evidence that first-stage choking occurs at around 19.4 to 22.2 C° (35.0 to 40.0 F°) of liquid subcooling.

Because the flow characteristics followed the general $G \sim (\Delta P)^{1/2}$ relationship for most operating conditions, a general orifice-type model was adopted. However, the conventional orifice constant, $C$, was found to be a function of both the pressure differential across the tube and the level of liquid subcooling. A correlation to calculate $C$ from the pressure differential and degree of liquid subcooling was formed based on the test results. This mathematical model should be used only for the liquid subcooling of 22.2 C° (40.0 F°) or less. For subcooling higher than 22.2 C° (40.0 F°), a modified first-stage choking model gives better results. With these two mathematical models, one will be able to size short tubes as the refrigerant expansion devices for heat pump systems and predict off-design performance. However, the application of these models should be within the specified L/D ratios. These models should be adequate for most residential heat pump applications.

Because the short-tube expansion devices are rugged, are simple to install, and have low costs, their acceptance by the heat pump industry might be accelerated if more knowledge about their application were available. This paper provides design equations for direct use by engineers of heat pump systems to study the relative merits of short tubes versus capillary tubes or expansion valves.

This study is limited to refrigerant R-22 and heat pumps up to three tons. For bigger heat pump systems and different kinds of refrigerants, further research work will be needed before a complete comparative study of expansion valves, capillary tubes, and short tubes is possible.

## NOMENCLATURE

| | |
|---|---|
| $A$ | Area of the tube opening |
| $B_1$, $B_2$, $B_3$, $B_4$ | Factors defined in Eqs 6, 7, and 8 |
| $C$ | Orifice constant as defined in Eq 1 |
| $C_1$ | Short-tube first-stage choking model constant as defined in Eq 2 |
| $C_2$ | Short-tube second-stage choking model surface-tension-related constant as defined in Eq 3 |
| $C_{eq}$ | Equivalent orifice constant as defined in Eq 7 |
| $D$ | Diameter of the short-tube opening |
| $G$ | Refrigerant mass flow rate per unit area |
| $g_c$ | Dimensional constant |
| $K$ | Constant defined in Eq 4 |
| $L$ | Length of the short tube |
| $\dot{m}$ | Refrigerant mass flow rate |
| $P_{up}$ | Tube upstream pressure |
| $P_{down}$ | Tube downstream pressure |
| $P_{sat}$ | Refrigerant liquid saturation pressure |
| $\Delta P$ | $P_{up} - P_{down}$ |
| $\Delta T$ | Level of liquid subcooling |
| $\beta$ | Ratio of short-tube diameter over liquid line diameter |
| $\sigma_{sat}$ | Liquid surface tension at saturation pressure |
| $\sigma_{175}$ | Liquid surface tension at 12.3 kg/cm$^2$ (175 psia) saturation |
| $\sigma_{200}$ | Liquid surface tension at 14.1 kg/cm$^2$ (200 psia) saturation |
| $\rho_l$ | Liquid density |

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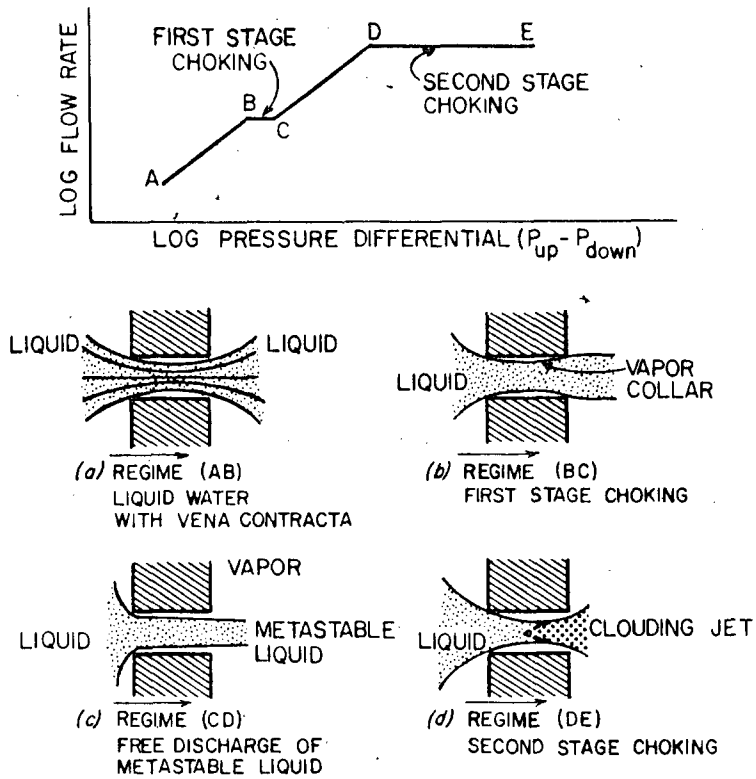


*Figure 1. Short-tube flow patterns* [13]

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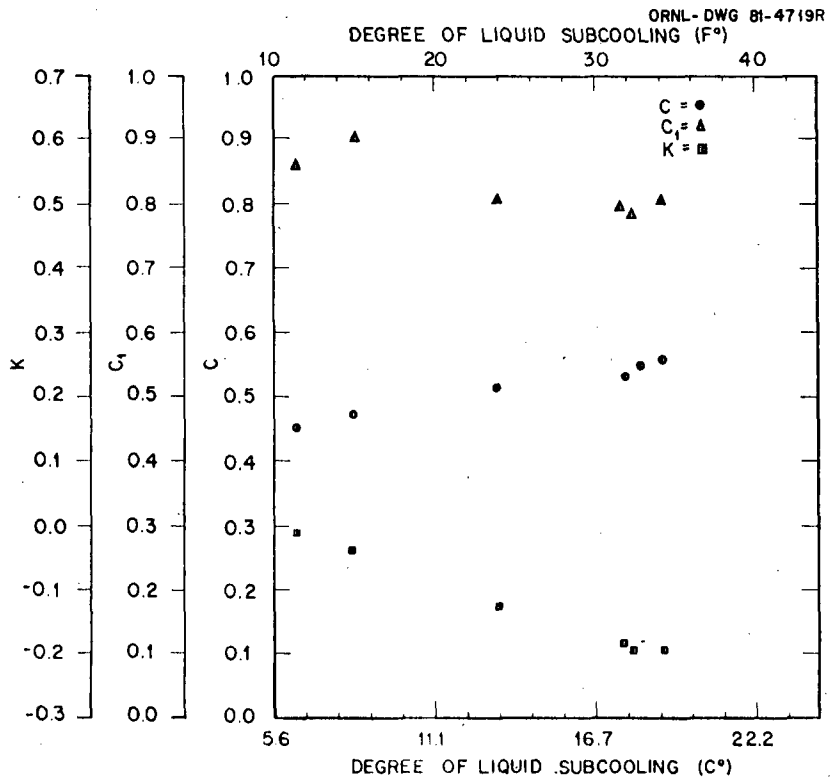


*Figure 2. Values of C, $C_1$, and K calculated from Domingorena's*[18] *steady state heat pump tests*

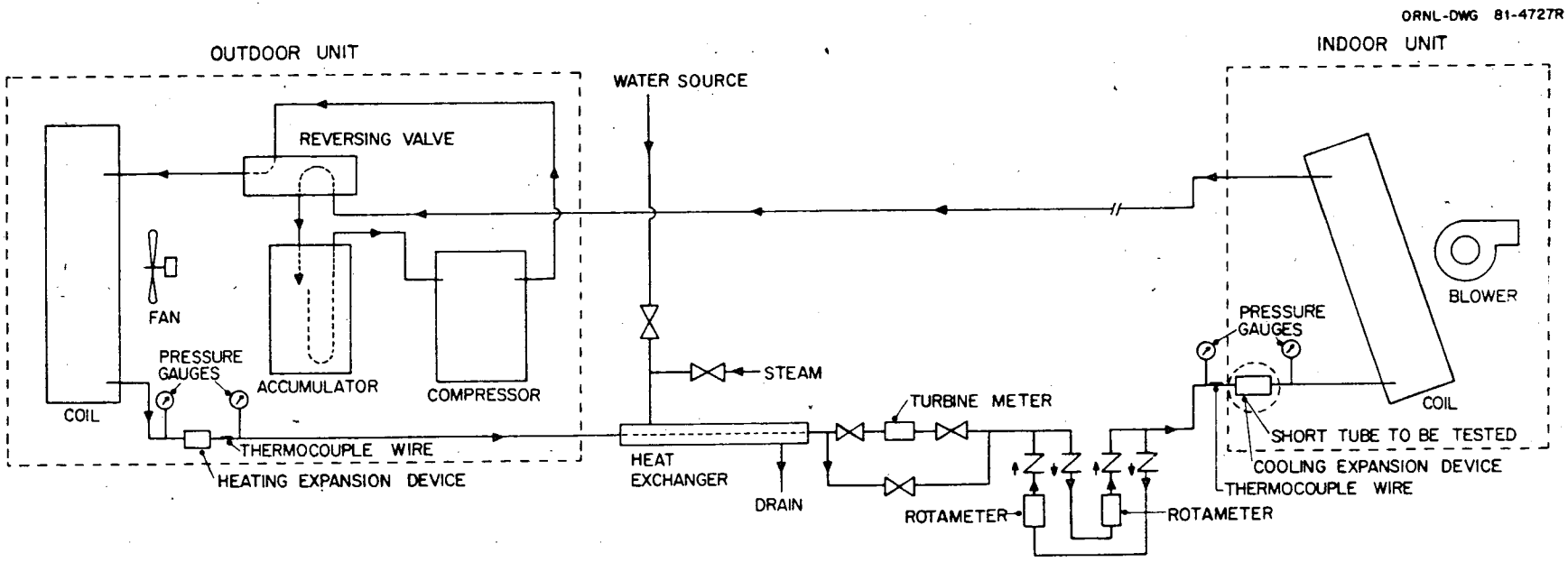


*Figure 3. Schematic of the test setup*

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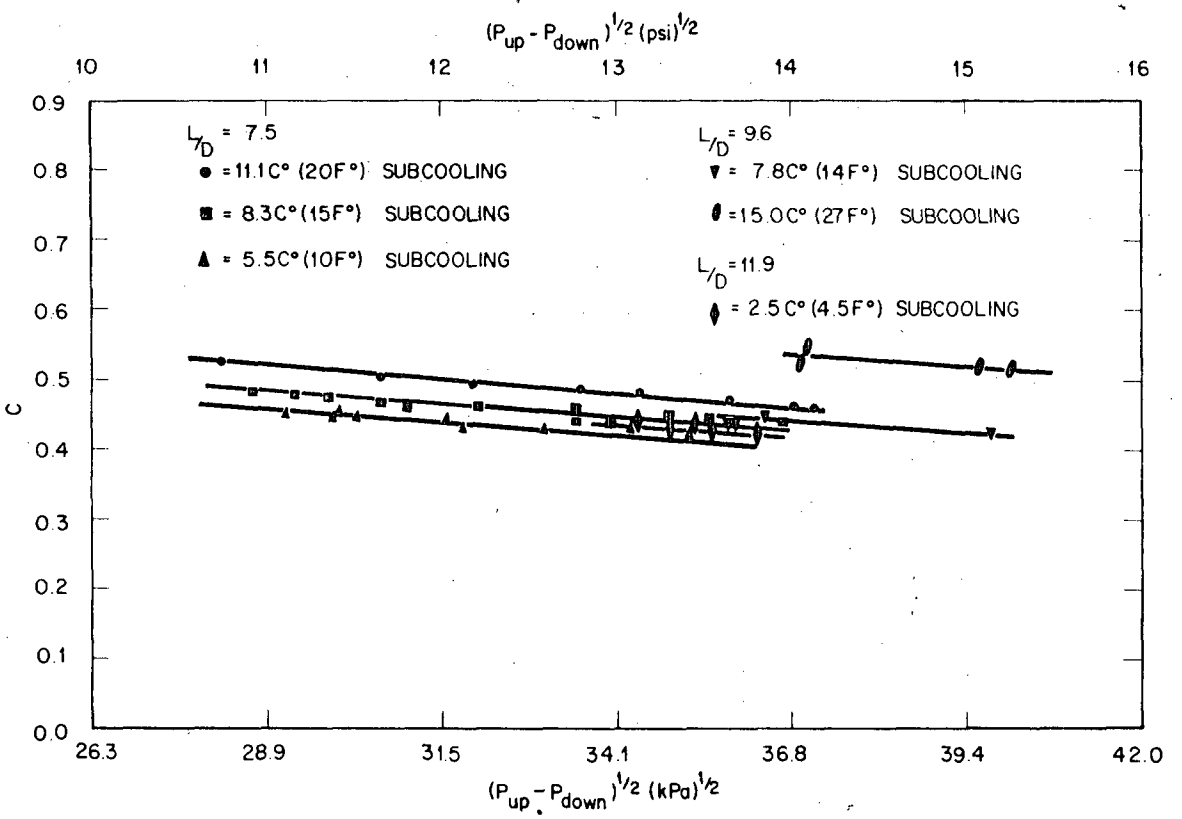


*Figure 4. Effect of pressure differential on C with liquid subcooling levels as the parameter*

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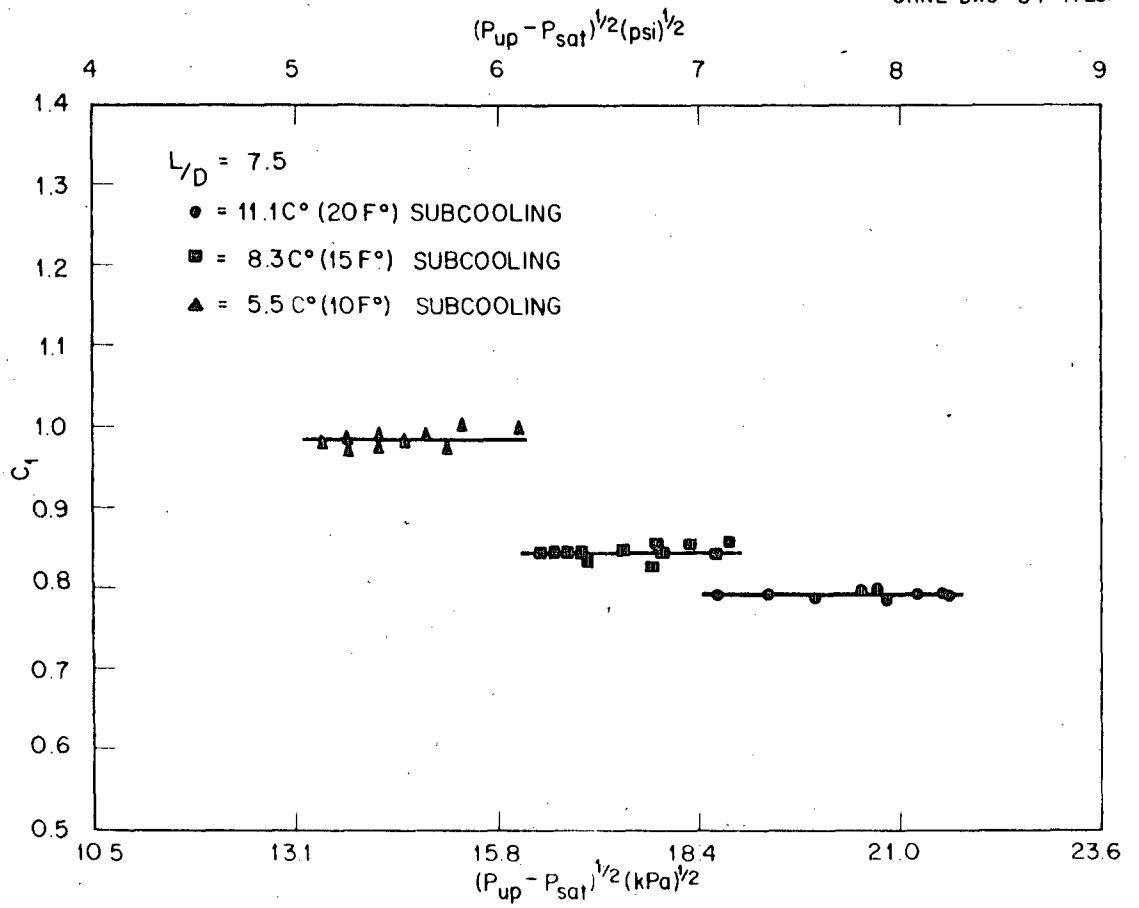


*Figure 5. Effect of pressure differential on $C_1$ with liquid subcooling levels as the parameter*

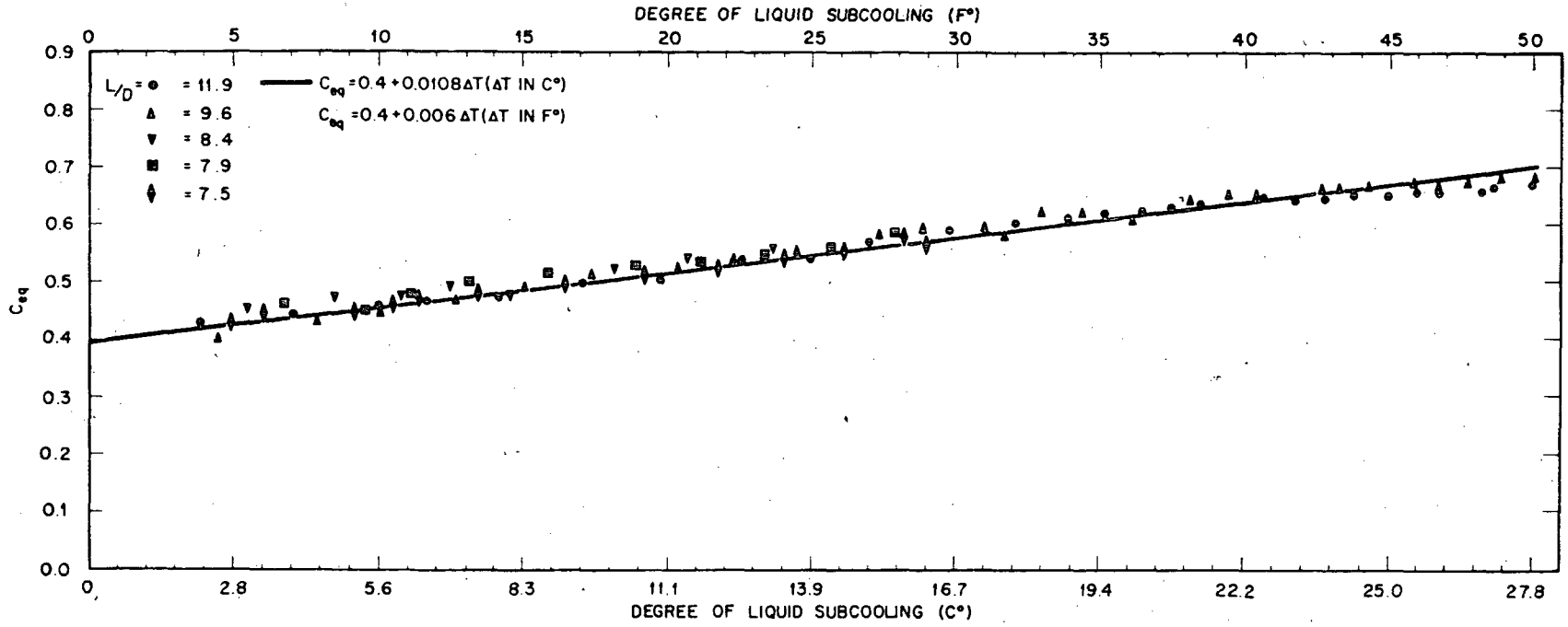


*Figure 6. Effect of the level of liquid subcooling on $C_{eq}$*

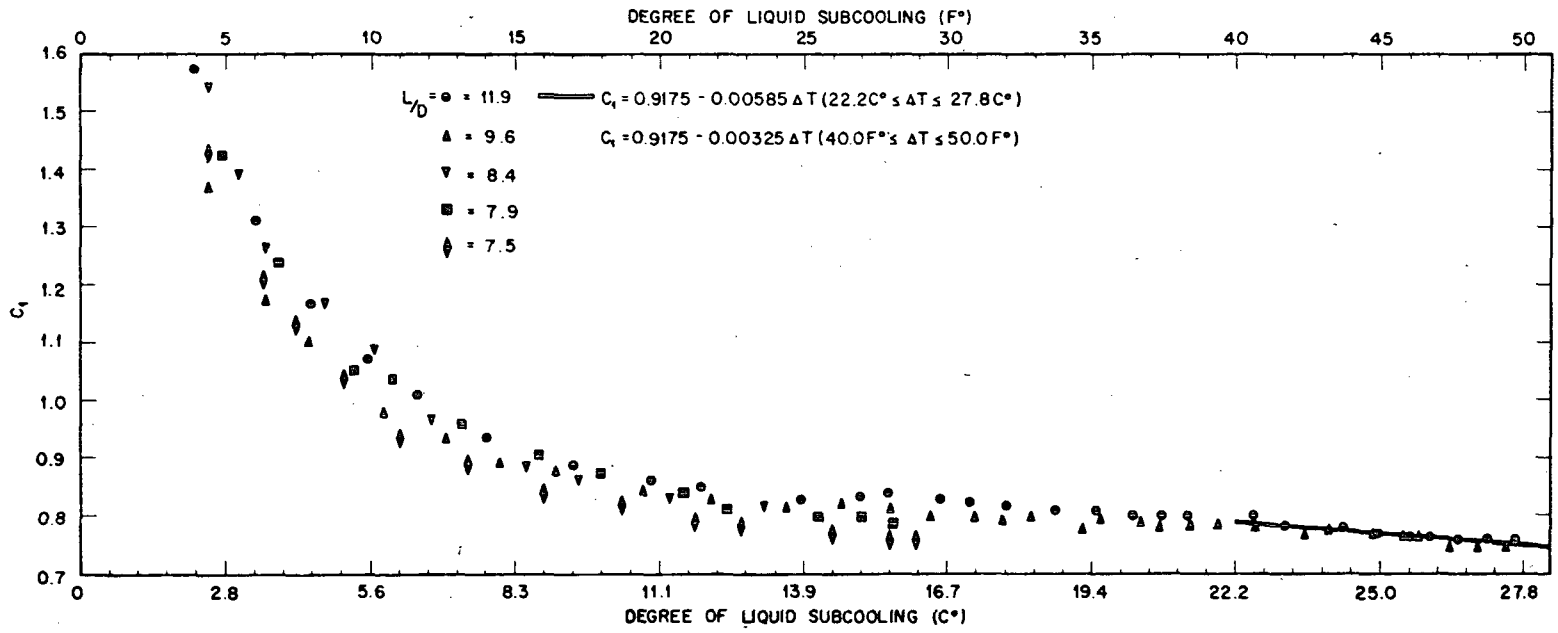


*Figure 7. Effect of the level of liquid subcooling on $C_1$*

# DISCUSSION

R.T. KOHLI, Stone & Webster, Boston, MA: What is the typical length of the short tube?

What is the pressure drop across the tube and its comparison to T-X value and capillary tube?

V.C. MEI: There is no typical length of the short tube. A long pipe can be classified as a short tube if its inside diameter is large enough so that its length-over-diameter ratio, L/D, falls into the short tube range--up to 11.9 in our experiment. The short tube length in this test was 1.27 cm (0.5 inch).

The pressure drop across the short tubes in this test ranged from 744 to 1517 kPa (108 to 220 psi) for cooling mode operation, which was purposely extended beyond the normal heat pump operating pressure drop range across the refrigerant expansion devices. The pressure drop across the short-tube type expansion devices should be compatible to those of T-X valves and capillary tubes.