International Refrigeration and Air Conditioning Conference
School of Mechanical Engineering

2000

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# A STUDY OF REFRIGERANT PRESSURE DROP AND VOID FRACTION IN FLATTENED COPPER TUBES

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## ABSTRACT

This study examines the refrigerant side effects of flattened round tubes. The tubes being investigated are an internally smooth tube, and two microfin tubes, one with no helix angle and the other with an 18° helix angle. All of the tubing is nominal 3/8" o.d. copper refrigerant tubing. The tubes will then be successively flattened so that the inside wall to wall height is 5.74 mm, 4.15 mm, 2.57 mm, and 0.974 mm (0.226",0.163",0.101", 0.0383"). Refrigerant void fraction and pressure drop data will be discussed and correlations used to predict these refrigerant behaviors will be given.

## INTRODUCTION

The purpose of this project is to examine refrigerant behavior (pressure drop and void fraction) in flattened copper passageways. This study is motivated by the desire to explore the advantages and disadvantages of refrigerant flow through small copper passageways. Potential advantages are reduced air side pressure drop and increased air side heat transfer. Effects on the refrigerant side are unknown, but an educated guess would be a reduction in refrigerant charge with an increase in refrigerant side pressure drop.

Twelve different channel configurations are investigated using two different refrigerants, R134a and R410A. The tubes used are a smooth tube and two microfin tubes, one with an 18° helix angle and one with a 0° helix angle (axially grooved). The tubes are initially round with a 9.52 mm (3/8") outer diameter, and 8.92 mm (.351") inner base diameter. The tubes are then flattened into an oval shape with inside heights of 5.74 mm, 4.15 mm, 2.57 mm, and 0.974 mm (0.226", 0.163", 0.101", 0.0383"). These heights correspond to using spacers with heights of 1/4", 3/16", 1/8" and 1/16". Please see Figure 1 for a diagram of the geometries involved.

## LITERATURE REVIEW

A literature review was done on flattened tubes with little success, thus a review of round tubes (both smooth and microfin) as well as microchannel heat exchangers will be given.

### Flow Regime Maps

One of the most important characteristics of a two-phase flow is its flow regime. Many methods exist for determining flow regime, one of the more well known methods is the use of the Taitel-Dukler (1976) flow regime map. One of the features of the Taitel – Dukler map is that boundaries are given between the stratified, stratified-wavy, annular, and slug-plug flow regimes Wattelet (1994) found that the Taitel – Dukler map worked well for refrigerants. The Taitel-Dukler map predicts that most of our data range will be in the annular regime with some low mass flow rate data points in the stratified flow regime and some low quality points in the intermittent flow regime.

### Theoretical Flow Regimes

Two flow regimes that may bound the expected range of results are the "homogenous" and the "separated" flow regimes. These ideal flows form conceptual limits for understanding the actual flow field characteristics.

In the homogenous regime the liquid and vapor phases flow as if they were a single fluid. This may occur in the slug-plug flow, mist flow, or bubbly flow where the two phases have equal velocities. The equations governing this regime are the single-phase equations with the density and viscosity weighted by the void fraction.

An ideal homogenous flow is very efficient due to the lack of liquid-vapor interactions. They are characterized by low pressure drops and high void fractions.

The separated flow regime occurs when the liquid and vapor flow independently of each other. Mathematically this flow can be described by equating the pressure gradients driving each flow. Separated flow conditions result in higher pressure drops and lower void fractions than other regimes.

## Void Fraction

A complete review of refrigerant void fraction ($\alpha$) in smooth tubes appears in a University of Illinois Air Conditioning and Refrigeration Center (UI ACRC) technical reports by Graham et al. (1998) and Yashar et al. (1998). Refrigerant void fraction data was collected for round tubes with diameters from 4 mm to 9 mm for both evaporation and condensation. The UI ACRC group found a void fraction relation of

$$\alpha = (1 + X_{tt} + 1/Ft)^{-0.321} \tag{1}$$

for evaporation in all tubes, and condensation in microfin tubes. For condensation in microfin tubes a similar relation is used with a different exponent,

$$\alpha = (1 + X_{tt} + 1/Ft)^{-0.375} \tag{2}$$

where $X_{tt}$ is the Lockhart-Martinelli Parameter and Ft is a modified Froude number derived by Hurlburt and Newell (1997).

$$X_{tt} = \left(\frac{1-x}{x}\right)^{0.9}\left(\frac{\rho_v}{\rho_l}\right)^{0.5}\left(\frac{\mu_l}{\mu_v}\right)^{0.1} \tag{3}$$

$$Ft = \left[\frac{x^3 G^2}{\rho_v{}^2 gD(1-x)}\right]^{\frac{1}{2}} \tag{4}$$

x is the quality, $\rho$ the density, $\mu$ the viscosity, G the mass flux, D diameter, and g acceleration due to gravity. The subscripts l and v refer to the liquid and vapor phases respectively

## Pressure Drop

Several models exist to predict two-phase frictional pressure drop in both smooth and microfin tubing. Moser (1998) suggests using a correlation developed by Friedel (1979). Friedel's correlation for the two phase multiplier is:

$$\phi_{LO}^2 = A_1 + \frac{3.24 \cdot A_2}{Fr^{0.045} We^{0.035}} \tag{5}$$

$$A_1 = (1-x)^2 + x^2\left(\frac{\rho_l}{\rho_v}\right)\left(\frac{f_{vo}}{f_{lo}}\right) \tag{6}$$

$$A_2 = x^{0.78}(1-x)^{0.24}\left(\frac{\rho_l}{\rho_v}\right) \tag{7}$$

$$\rho_{tp} = \left(\frac{x}{\rho_v} + \frac{1-x}{\rho_l}\right)^{-1} \tag{8}$$

$$We = \frac{G^2 D}{\rho_{tp}\sigma} \tag{9}$$

$$Fr = \frac{G^2}{\rho_{tp} Dg} \tag{10}$$

where $f_{lo}$ and $f_{vo}$ are the single phase liquid only and vapor only friction factors and $\sigma$ is the surface tension. Fr and We and two-phase Froude and Weber numbers. Cavallini et al. (1999) proposed using the Friedel correlations for two-phase flow with an adjustment for the single-phase friction component.

$$\frac{e}{d} = 0.18\left(\frac{h}{d_i}\right)\left(\frac{1}{0.1+\cos\beta}\right) \tag{11}$$

where h is the fin height, $d_i$ is the nominal tube diameter and $\beta$ is the swirl angle of the fins.

## CURRENT RESULTS

The primary purpose of an initial data collection phase was to compile adiabatic void fraction and pressure drop data for the flattened tubes. The refrigerants used in this study represent a mid pressure refrigerant (R134a) and a high pressure refrigerant (R410A). The pressure primarily effects the vapor density which plays a primary role in the flow regime. It is believed that these two refrigerants are the limits of the vapor density bands for other refrigerants (R22, R404A, and R407C) and that these results for R134a and R410A can be applied to other refrigerants.

### Void Fraction Results

The void fraction data is plotted against the ($X_{tt}$ + 1/Ft) grouping discussed earlier in the literature review with the diameter term in the Ft being replaced with the channel height. The void fraction data appears to correlate well with this grouping as evident in Figures 2 through 6. Figures 2, 3 and 4 are for smooth tubes, Figure 5 for axially grooved tubes, and Figure 6 for helically grooved tubes. Some characteristics that appear in these plots:

1. As spacing height decreases, the void fraction decreases at the same value of $X_{tt}$+1/Ft.
2. At higher values of $X_{tt}$+1/Ft as the mass flux decreases the void fraction will approach the homogenous flow limit.
3. At lower values of $X_{tt}$+1/Ft in the axially grooved and smooth tubes when flattened to 1/16", the void fraction appears to possibly shift towards the separated flow limit.

In the helically grooved tube, the void fraction data does not shift as severely as in the other tubes. An explanation of this could be the ability of the helically grooved tube to promote annular flow at lower qualities and mass flux rates.

The void fraction data was curve fit to the form:

$$\alpha = \left[1 + a\cdot\left(X_{tt} + \frac{1}{Ft}\right)\right]^{n} \tag{12}$$

The values of 'a' and 'n' are given in Table 1. In general as the spacing height decreases, the constant multiplier 'a' increases, and the exponent 'n' decays slightly. Continued efforts in modeling will hopefully explain these trends.

### Pressure Drop Results

Pressure drop data was collected during the initial experimental activities. Only tests in the smooth tube at 1/16" and both the microfin tubes at 1/8" and 1/16" had pressure drops at a high enough level to appear on our pressure drop level. Figure 7 shows the accuracy of the Friedel correlation in smooth tubes flattened to 1/16" and the Cavallini correlation in microfin tubes flattened to 1/8" and 1/16". Fair agreement is obtained using the Friedel and Cavallini correlations.

## SUMMARY

Heat transfer coefficient data will be collected in the next phase of the investigation using the techniques suggested by Dobson (1994). Additional diabatic pressure drop data will also be collected. All of the geometric configurations previously mentioned will be tested using R134a and R410A.

### General Observations

Examinations of the void fraction and pressure drop characteristics lead to the conclusion that most of the data points lie in the annular flow regime and behave similar to annular flow in round tubes:

1. This void fraction data falls in between the homogenous and separated limits and obeys a general trend seen in round tubes.
2. $X_{tt}$, a parameter devised for annular flow, predominates in property calculations.
3. Pressure drop relations that appear to correlate the data well (Friedel for smooth tubes and the Cavallini adjustment for microfin tubes) are generally based on annular flow analysis.

Another observation is the shifting of void fraction data from the annular flow regime:

1. In the axially grooved and smooth tube at lower void fractions (and higher values of ($X_{tt}$+1/Ft)), at the lower mass flow rates the void fraction data will tend toward the homogenous flow solution
2. At the 1/16" flattening height for smooth and axially grooved tubes at lower values of ($X_{tt}$+1/Ft), the void fraction data shifts toward the separated flow limit. The void fraction flow regime change is not seen as strongly in the helically grooved tubes.

## Conclusion

There are two issues that must be resolved with regard to the current void fraction results. First, the void fraction data shifting from the annular flow must be explained. And second a simple model, hopefully using the ($X_{tt}$ + 1/Ft) grouping and explaining the shift in coefficients in Table 1, will be devised.

The Friedel correlation and the Cavallini adjustment for microfin tubes accurately predict the pressure drop data. More data at higher qualities and at lower pressure drop conditions, will be taken to verify the previous results over a broader range of conditions.

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| Tube | a | n |
|---|---|---|
| Smooth – Round | 1.63 | -0.258 |
| h=5.74 mm | 2.37 | -0.231 |
| h=4.15 mm | 2.68 | -0.214 |
| h=2.57 mm | 3.05 | -0.227 |
| h=0.974 mm | 11.5 | -0.166 |
| Axial – Round | 1.73 | -0.290 |
| h=5.74 mm | 3.23 | -0.220 |
| h=4.15 mm | 2.61 | -0.254 |
| h=2.57 mm | 3.48 | -0.231 |
| h=0.974 mm | 17.8 | -0.146 |
| Helical – Round | 1.5 | -0.333 |
| h=5.74 mm | 1.63 | -0.349 |
| h=4.15 mm | 2.23 | -0.321 |
| h=2.57 mm | 6.27 | -0.217 |
| h=0.974 mm | 7.53 | -0.201 |

Table 1 Coefficients for Void Fraction Correlations

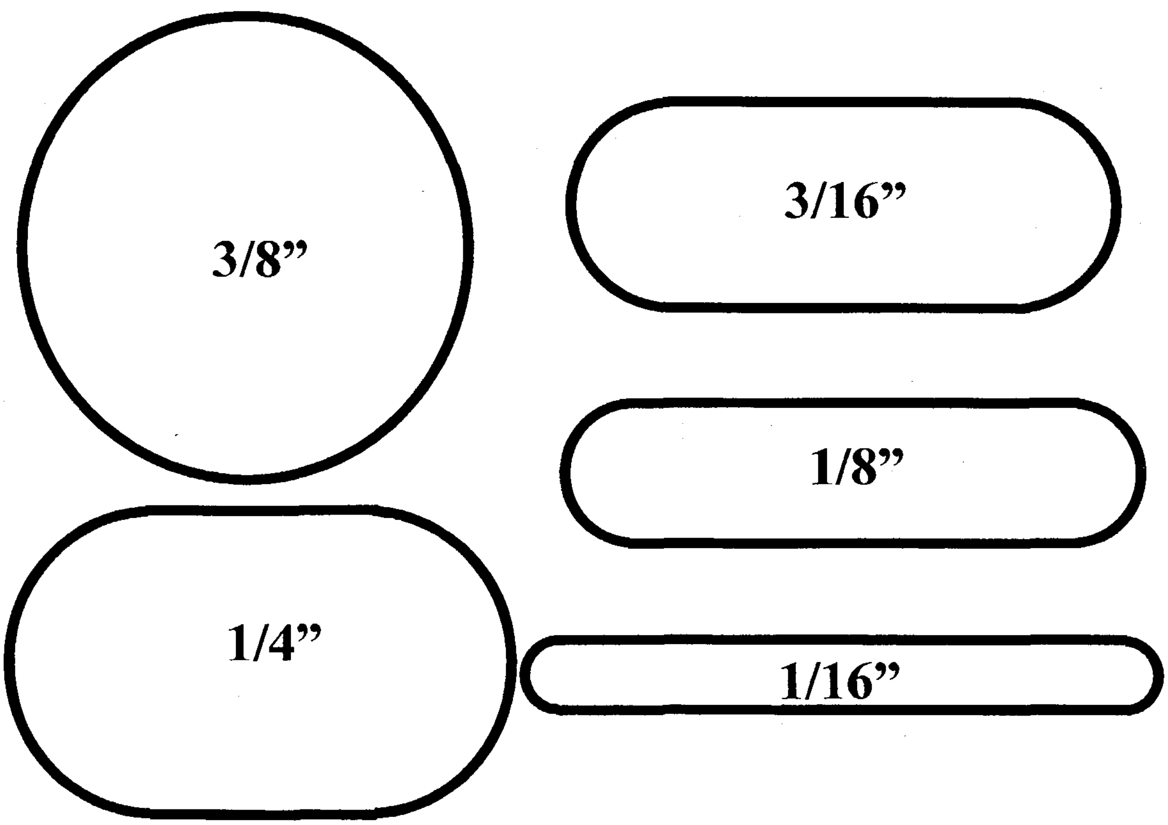


Figure 1 Detail of Geometries of Flattened Tubes

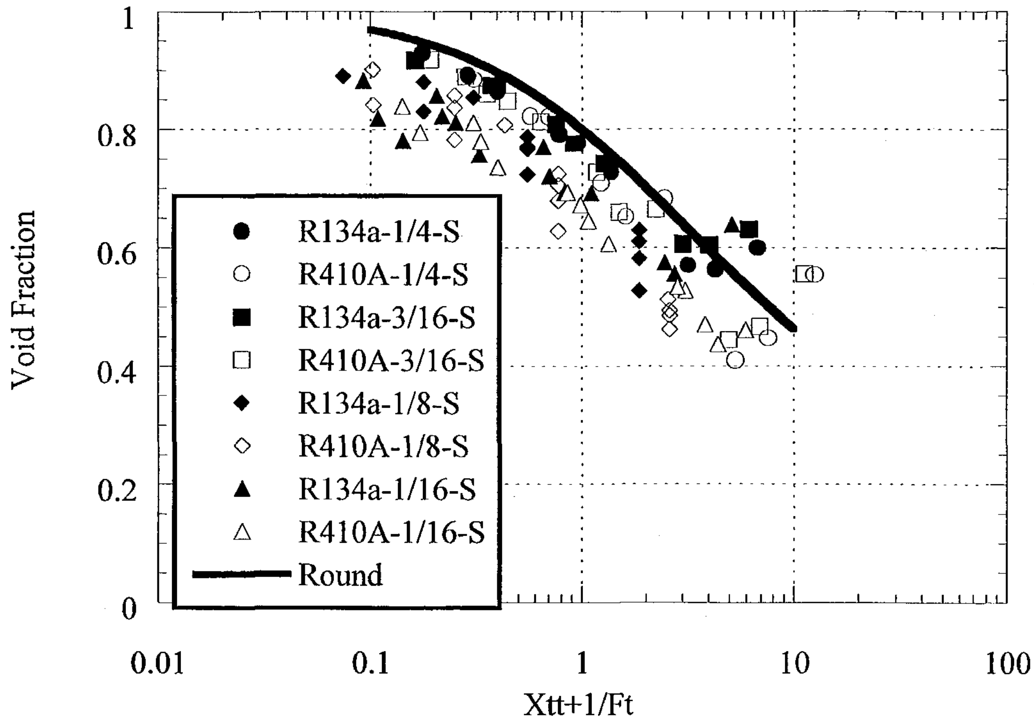


Figure 2 Void Fraction for Smooth Tubes with no Corrections

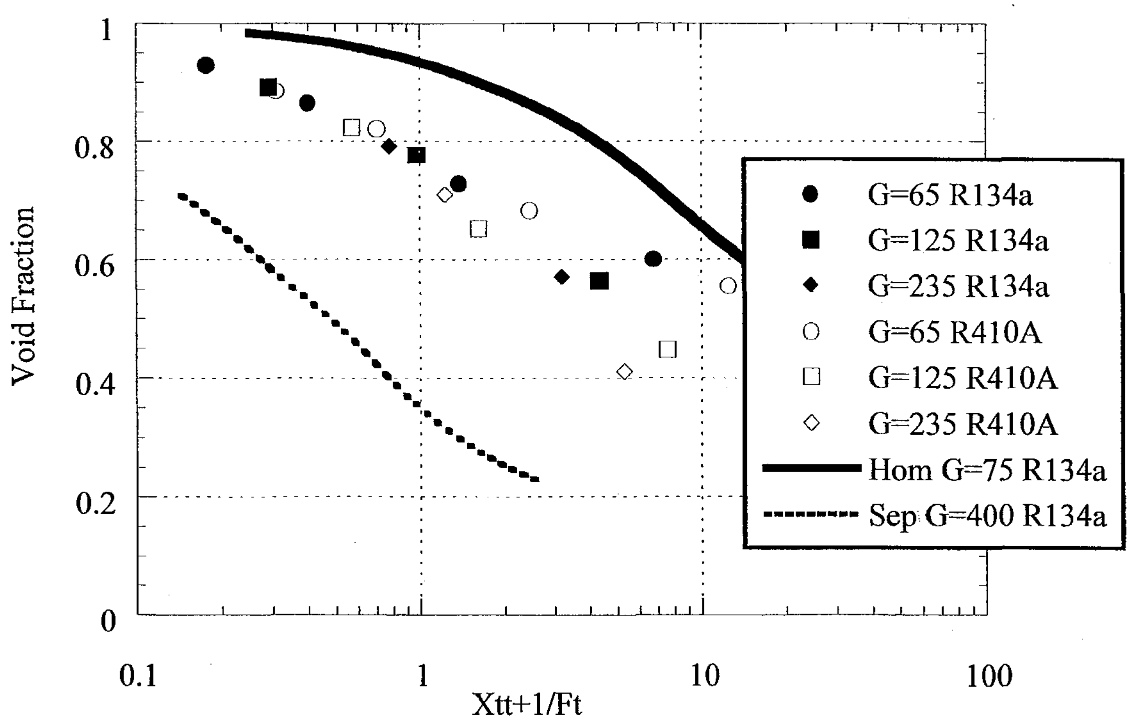


Figure 3 Void Fraction for Smooth Tube Flattened to 1/4" (mass flux "G" units are $kg/m^2$-s)

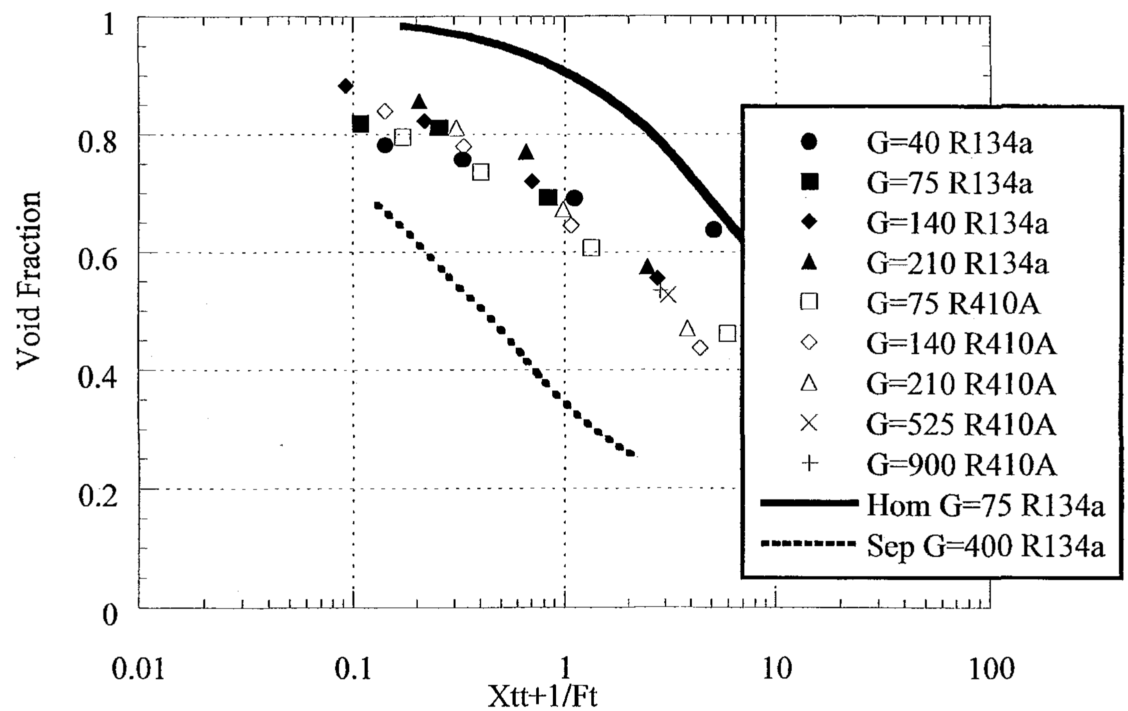


Figure 4 Void fraction in Smooth Tubes Flattened to 1/16" (mass flux "G" units are $kg/m^2$-s)

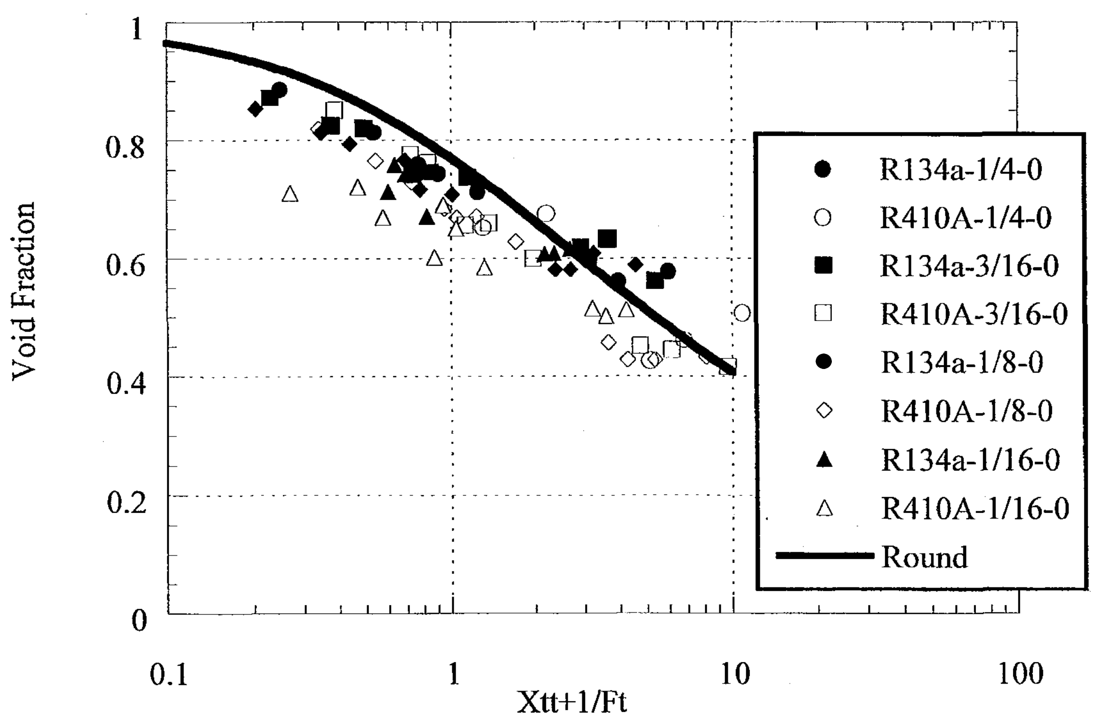


Figure 5 Void Fraction in Axially Grooved Tubes

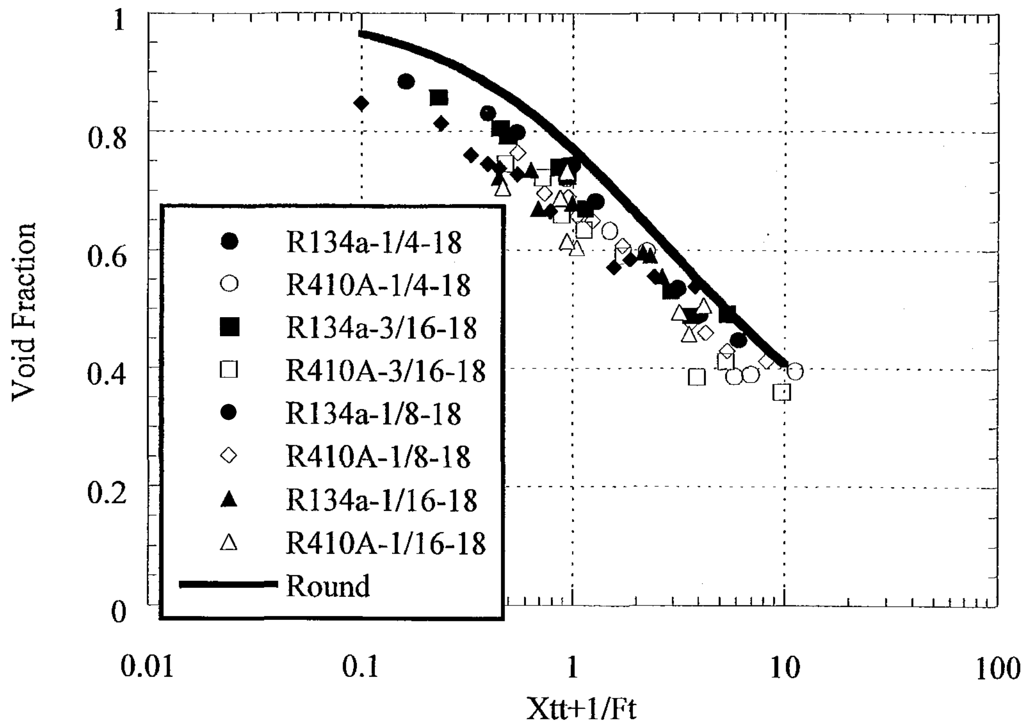


Figure 6 Void Fraction in Helically Grooved Tubes

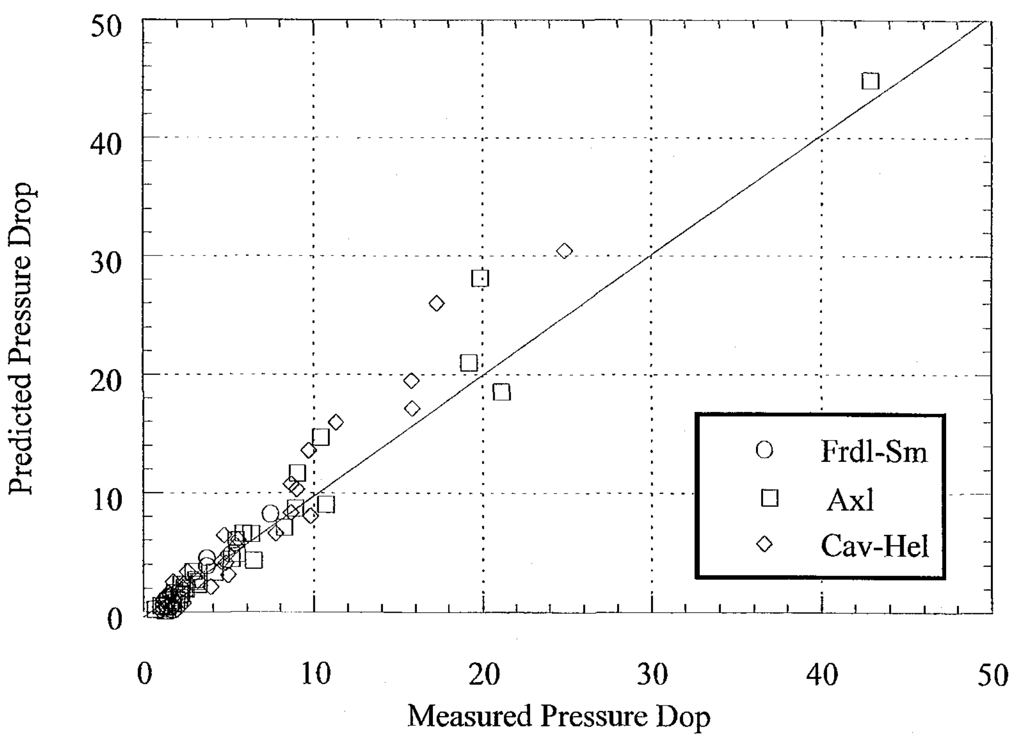


Figure 7 Pressure Drop Comparison