Pressure Drop During Two-Phase Flow of Refrigerants in Horizontal Smooth Tubes


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PRESSURE DROP DURING TWO-PHASE FLOW OF
REFRIGERANTS IN HORIZONTAL SMOOTH TUBES


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ABSTRACT

A single tube evaporator test facility capable of measuring pressure drop and heat transfer coefficients inside horizontal tubes has been designed and developed at the Air Conditioning and Refrigeration Center at the University of Illinois. Baseline testing with R-12 and testing with R-134a, an ozone-safe refrigerant, have been completed. Tests were conducted using pure refrigerants and refrigerant-oil mixtures. For the refrigerant-oil testing, PAG and Ester oils were added to R-134a and mineral oils were added to R-12 with the oil concentration varying from 0 to 5% by weight. The observed flow regimes were predominantly annular. A new correlation for two-phase frictional pressure drop inside smooth tubes was developed for pure refrigerants using the Lockhart-Martinelli parameter "Xi" and Froude number "Fr". The two dimensionless parameters predicted the data well within the range of parameters considered in this study. In addition, a functional dependence between the ratio of the pressure drops with and without oil and the oil concentration "ω" was developed.
NOMENCLATURE

B   coefficient proposed by Chisholm (1973)
D   internal tube diameter, ft [m]
f   friction coefficient
g   gravitational acceleration, ft/s² [m/s²]
G   mass flux, lbm/ft²·s [kg/m²·s]
L   total evaporator length, ft [m]
n   number of experiments
P   pressure, psi [kPa]
ΔP  pressure drop along the evaporator, psi [kPa]
x   quality
Δx  quality change along the evaporator

Greek Symbols

α   void fraction
ρ   density, lbm/ft³ [kg/m³]
μ   dynamic viscosity, lbm/ft·s [kg/m·s]
ω   mass fraction of oil

Dimensionless Parameters

\[ \text{Fr}_I = \frac{G^2}{\rho_i^2 g D}, \text{ Froude number} \]
\[ \text{Re}_{LO} = \frac{GD}{\mu_i}, \text{ Reynolds number} \]
\[ X \quad \text{Lockhart-Martinelli parameter, defined by Eq. (1)} \]
\[ X_{LT} \quad \text{Lockhart-Martinelli parameter, defined by Eq. (2)} \]
\[ \Gamma = \left( \frac{\rho_i}{\rho_v} \right)^{0.5} \left( \frac{\mu_v}{\mu_i} \right)^{0.125}, \text{ physical property index} \]
\[ \phi_{L^2} \quad \text{Two-phase multiplier defined by Eq. (3)} \]
\[ \phi_{LO^2} \quad \text{Two-phase multiplier defined by Eq. (4)} \]
**Subscripts**

- acc: acceleration
- f: friction
- calc: calculated from correlation
- exp: experimentally determined
- i: test section inlet
- l: liquid
- L: liquid flows alone in the tube
- LO: both phases flow as liquid
- o: test section outlet
- oil: oil
- pure: pure refrigerant
- v: vapor
- TP: two-phase
- tt: turbulent-turbulent
INTRODUCTION

The prediction of pressure drop during flow boiling or evaporation of new, ozone-safe refrigerants, such as R-134a, is essential for proper evaporator design of refrigerators and air conditioners, among other applications. R-134a is a promising, potential replacement for R-12. Schlager et al. (1988, 1990) present literature reviews on pressure drop and heat transfer characteristics for pure refrigerants and oil-refrigerant mixtures. Due to differences in the physical properties of R-134a and R-12, the evaporator design may need to be reassessed. Therefore, it is very important to create suitable procedures which adequately estimate the two-phase pressure drop in evaporators.

In air conditioning and refrigeration applications, the pressure drop must be limited to within certain design criteria. If the compressor suction line pressure is lowered too much, the volumetric efficiency will be greatly reduced. Knowing the pressure drop also helps to determine the pumping load and enables correct evaluation of a variety of physical properties needed for many transport correlations used in other design equations.

A test apparatus has been constructed and operated [Wattelet (1990), Panek (1992)] at the Air Conditioning and Refrigeration Center at the University of Illinois. The purpose of this apparatus is to experimentally investigate pressure drop and evaporation heat transfer coefficients for ozone-safe refrigerants. This includes testing with both pure refrigerants and refrigerant-oil mixtures.

Despite the considerable progress made to predict pressure drop both experimentally and theoretically [Tong (1967), Wallis (1969), Chisholm (1983)], a need still exists for convenient, rapid and accurate estimation procedures. An experimental study was conducted to provide the local pressure drop during two-phase flow of R-134a and R-12 inside smooth tubes. A correlation was developed based on theoretical pressure drop modeling. The ranges studied were those utilized in residential and automobile air conditioning evaporators. The overall pressure drop can be obtained by a numerical integration of the local pressure drop correlation along the length of the tube, i.e. over the evaporator quality change.

The oil influence on the pressure drop for refrigerant-oil mixtures was also investigated with oil concentrations of up to 5% by weight. PAG and Ester oils were added to R-134a and mineral oils were added to R-12 to obtain the refrigerant-oil mixtures.
EXPERIMENTAL TEST FACILITY

A test apparatus has been constructed for the measurement of pressure drops and evaporative heat transfer coefficients of ozone-safe refrigerants, as shown in Fig. 1. The test apparatus consists of the refrigerant flow loop, a commercial chiller subsystem, and the instrumented test section. A refrigerant loop capable of investigating different refrigerants flowing in horizontal tubes is the main feature. A commercial R-502 chiller subsystem is thermally connected to the refrigerant loop to provide cooling. The test section can be easily isolated from the other test loop components to allow for quick repair or replacement.

Figure 1. Experimental apparatus

Refrigerant Flow Loop

The main component of the test apparatus is the refrigerant flow loop, shown in Fig. 2, which schematically represents the following equipment and systems:
- Test Section,
- Condensers (2),
- Filters and receiver,
- Variable speed gear pump and flowmeter,
- Absolute and differential pressure transducers,
- Thermocouples,
- Preheater,
- Data acquisition system,
- Sampling and oil injection system.

Figure 2. Refrigerant test loop

With the desired test section in place, the loop is charged with the refrigerant or refrigerant-oil mixture to be studied. Subcooled liquid exiting the condenser is pumped by a positive displacement gear pump into a radial piston-type positive displacement flowmeter. A bypass line connects the outlet of the pump to the entrance of the condenser. This valved bypass allows coarse control of the mass flow rate into the test section. Once the refrigerant exits the flow meter, it enters the vertical circulation preheater. A precise amount of heat is added to boil the subcooled liquid. When running evaporation experiments, more heat is added in the test section to further vaporize the refrigerant. The outlet of the test section is connected to two condenser modules where the refrigerant is condensed and subcooled.
Test Section

The test section consists of a 8.0 ft (2.44 m) long, 0.43 in. (10.9 mm) i.d. horizontal smooth copper tube instrumented with 20 thermocouples. The heat flux at the inner surface of the tube is uniform. At the inlet and outlet of the test section, pressure taps are soldered onto the tube. Glass tubes for flow visualization are located at the entrance and exit of the test section outside of the pressure taps. The inside diameter of the glass tubes is within 0.004 in of the inside diameter of the test section, minimizing perturbations in the flow pattern. A stroboscopic light flashing on these sight glasses makes visual observation of the flow patterns clearer. The ends of the test section are connected to the main refrigerant loop by special zero clearance fittings. This arrangement permits quick changes of the test section.

Instrumentation

Instrumentation is present at all important locations to give information on the state of the flowing refrigerant at that point. All temperature measuring devices were calibrated in a controlled temperature bath against NIST-traceable precision mercury thermometers. Uncertainty of temperature measurements after processing in the data acquisition system is considered to be ± 0.36 °F (± 0.2 °C). Refrigerant bulk temperature is measured in five locations including the entrance and exit of the test section using type T copper-constantan thermocouples.

Refrigerant absolute pressures are measured in three locations: at the exit of the condenser, at the inlet of the preheater, and at the inlet of the test section. In addition, differential pressure is measured across the test section. All four pressure sensing devices were calibrated with a dead weight tester. Uncertainty of the pressure measurement reported by the data acquisition system is considered to be ± 0.3 % of the full scale reading.

All absolute pressure transducers are BEC company Models GWP5-46AW. The range of the transducer located at the inlet of the preheater is 0-300 psi (0-2.07 MPa). The range of the other two absolute pressure transducers is 0-50 psi (0-345 kPa). The differential pressure transducer is a Sensotec Model AD111AT, with a range of 0-5 psid (0-34.5 kPa). The output of these transducers is 4-20 mA.

Refrigerant volumetric flow rate is measured by a Max Machinery Model 214-411 four-piston, positive displacement flow meter. The flowmeter is located after the main refrigerant bypass line, and measures the amount of refrigerant flowing through the preheater and into the test section. The capacity of the flow meter is 0-1 gpm (0-0.063 L/s), and the output signal is 0-10 VDC. Uncertainty of the measurement is ± 0.5 % of full scale reading.
Oil Concentration Measurement

The amount of oil dissolved in the refrigerant is measured with a technique similar to the procedure outlined in ASHRAE Standard 41.4. However, only one sample is taken at the beginning of each series of test runs, instead of three samples required by the Standard. All concentrations reported herein use the sample basis. Many potential sources of error exist in this technique if great care is not taken while sampling. Adding air into the initial sample mass, leaving oil behind in the hose, and not removing all the refrigerant from the container for the final mass measurement are some of these potential errors. A more detailed description of the oil sampling technique was given by Panek (1992).

Data Collection

Prior to running, the system loop was evacuated and then charged with pure refrigerant or refrigerant-oil mixture. Saturation pressure for pure refrigerant based on thermocouple readings matched saturation pressure based on pressure transducers within ±0.3 psi (±2.0 kPa). Schrader valves were located in the upper section of the loop for purging purposes if noncondensibles would become a problem.

Data collection was performed using a micro-computer and data acquisition system. Testing was conducted at steady-state conditions, which were monitored and controlled by the above mentioned system. Electric heaters were controlled through 4-20 mA analog output signals sent to the silicone controlled rectifiers (SCR).

The parameters controlled during the tests were mass flux, inlet quality, and saturation temperature. Steady-state conditions, reached in approximately 30 minutes to 2 hours, were assumed when the variation of the saturation temperature was less than 0.18°F (0.1°C) in about five minutes. The controlled parameters also had to be within the following range of target values: mass flux, ±5%; inlet quality, ±1%; saturation temperature, ±0.36°F (0.2°C). Once steady-state conditions were achieved, the transducer signals were logged into a data acquisition output file.

The channels were scanned once every second for a period of 60 seconds. The values were then averaged and reduced using a spreadsheet macro. Refrigerant transport and thermodynamic properties used during data collection and reduction were obtained from Wilson and Basu (1988), McLinden (1989), and Jung and Radermacher (1991).
ANALYSIS

Total pressure drop for two-phase flow in tubes consists of frictional, acceleration, and gravitational components. It is necessary to know the void fraction (the ratio of gas flow area to total flow area) to compute the acceleration and gravitational components. To compute the frictional component of pressure drop, either the two-phase friction factor or the two-phase frictional multiplier must be known [ASHRAE (1989)]. For these experiments the horizontal test section was thermally isolated (adiabatic). Assuming the flow was fully developed, the contribution of gravitational and acceleration components to the pressure drop were small and could be considered negligible. The tests were conducted during adiabatic turbulent flow using the pure refrigerants R-134a and R-12 at the reduced pressures of 0.07 to 0.12 over the following range of parameters: mass flux, 41-123 lbm/ft²-s (200-600 kg/m²-s); quality, 10-90%; saturation temperature, 32-59 °F (0-15 °C). It should be noted that these reduced pressures correspond to a saturation pressure of 50.8 psi (350 kPa) for R-12 and 52.6 psi (363 kPa) for R-134a. The saturation temperature for these pressures is 41 °F (5 °C).

The pressure drop was measured directly from the differential pressure transducer through the data acquisition system. The inlet quality to the test section was determined from an energy balance on the preheater.

Two principal types of flow models were used in developing frictional pressure drop: the homogeneous model and the separated flow model. In the first, the flow of both phases is assumed to be in equilibrium, and the gas and liquid velocities are assumed equal (slip ratio = 1). The frictional pressure drop is computed as if the flow were a single-phase flow, except for introducing modifiers to the properties inside the single-phase friction coefficient. In the separated flow model, the two phases are considered separate and the velocities may differ.

Lockhart-Martinelli (1947) and Martinelli-Nelson (1948) developed a separated flow model with a parameter X defined as

\[ X = \left[ \frac{\frac{\partial p}{\partial z}}{\left(\frac{\partial p}{\partial z}\right)_l} \right]^{0.5} \]

where \( \left(\frac{\partial p}{\partial z}\right)_l \) = frictional pressure gradient, assuming that liquid alone is flowing in the pipe

\( \left(\frac{\partial p}{\partial z}\right)_v \) = frictional pressure gradient, assuming that vapor alone is flowing in the pipe
The frictional pressure gradient due to the single-phase flow of the liquid or the vapor depends on the type of flow for each phase, which can be laminar or turbulent. Assuming turbulent flow for both phases and using a Blasius type correlation [Wallis (1969)] for smooth tubes [Eq. (8) below] to calculate the friction factor for the liquid and vapor phases, the parameter \( X \) is replaced by \( X_u \) that can be defined as:

\[
X_u = \left( \frac{1}{x} \right)^{0.875} \left( \frac{\mu_v}{\mu_l} \right)^{0.5} \left( \frac{\rho_v}{\rho_l} \right)^{0.125}
\] (2)

Lockhart-Martinelli-Nelson also defined the two-phase multipliers \( \phi_L^2 \) and \( \phi_L^2 \) as

\[
\phi_L^2 = \text{ratio of two-phase friction pressure gradient to the friction pressure gradient if liquid flows alone in the tube}
\]

\[
\phi_L^2 = \text{ratio of two-phase friction pressure gradient to the friction pressure gradient if total mixture flows as a liquid}
\]

These can be expressed by the following equations:

\[
\phi_L^2 = \frac{\left( \frac{\partial p}{\partial z} \right)_L}{\left( \frac{\partial p}{\partial z} \right)_L} \quad (3)
\]

\[
\phi_L^2 = \frac{\left( \frac{\partial p}{\partial z} \right)_L}{\left( \frac{\partial p}{\partial z} \right)_L} \quad (4)
\]

and can be related [Martinelli and Nelson (1948)] as

\[
\phi_L^2 = \phi_L^2 (1 - x)^{1.75} \quad (5)
\]

The overall pressure drop due to friction in two-phase flow can be evaluated by the following integral equation over the quality range as

\[
\Delta p_t = \Delta p_{LO} \left( \frac{1}{\Delta x} \int \phi_L^2 dx \right) \quad (6)
\]

where \( \Delta p_{LO} \) is the frictional pressure drop for the total mixture flowing as a liquid, given by
\[ \Delta p_{LO} = \frac{2f_{LO}G^2L}{\rho_iD} \]  

(7)

and \( f_{LO} \) is the liquid friction factor for smooth tubes calculated from

\[ f_{LO} = \frac{0.079}{Re_{LO}^{0.25}} \]  

(8)

To evaluate the pressure drop due to acceleration, the Zivi (1964) equation for void fraction is recommended, which was developed using the concept of minimum entropy production and is expressed as

\[ \alpha = \frac{1}{1 + \left(\frac{1-x}{x}\right)^{\frac{1}{0.87}}} \]  

(9)

The overall pressure drop due to acceleration can be obtained by the following expression attributed to Martinelli-Nelson

\[ \Delta p_{acc} = G^2 \left\{ \frac{x_o^2}{\rho_i\alpha} + \frac{(1-x_o)^2}{\rho_i(1-\alpha)} \right\} - \left\{ \frac{x_i^2}{\rho_i\alpha} + \frac{(1-x_i)^2}{\rho_i(1-\alpha)} \right\} \]  

(10)

Therefore, the total pressure drop during two-phase flow inside a horizontal tube can be evaluated as follows:

\[ \Delta p_{TP} = \Delta p_{acc} + \Delta p_f \]  

(11)
PRESSURE DROP DUE TO FRICTION

Correlation Development

Using the separated flow model, we developed a correlation of the data for turbulent two-phase flow in smooth tubes with refrigerants R-134a and R-12. In this correlation the frictional pressure drop ratio, $\Phi_{LO}^2$, is a function of the Lockhart-Martinelli parameter, $X_{lt}$, and the Froude number, $Fr_l$. When body forces and inertia forces are significant in the flow, i.e. for stratified or wavy flow regimes, the Froude number plays an important role in the correlation. As in the case of two-component flow, a strong influence of the mass flow rate was demonstrated by Isbin et al. (1959), Baroczy (1966), Chisholm (1968, 1973) and others, which can be well represented by the Froude number as shown in Eqs. (13), (14), and (15) below.

Using 52 data points for R-134a and 45 data points for R-12, a regression analysis was carried out to determine the functional dependence of $\Phi_{LO}^2$ on $X_{lt}$ and $Fr_l$ with a correlating coefficient of 0.985. The mean deviation was defined as

$$\text{Mean Dev.}(\%) = \frac{100}{n} \sum \frac{|\Delta p_{calc} - \Delta p_{exp}|}{\Delta p_{exp}}$$  \hspace{1cm} (12)

Using the experimental data for both refrigerants R-134a and R-12, the mean deviation was determined to be 4.6 %. The final correlating form is:

$$\Phi_{LO}^2 = (1.376 + c_1 X_{lt}^{-0.5})(1 - x)^{1.75}$$  \hspace{1cm} (13)

$$c_1 = 4.172 + 5.480 Fr_l - 1.564 Fr_l^2$$  \hspace{1cm} (14)

$$c_2 = 1.773 - 0.169 Fr_l$$  \hspace{1cm} (15)

Figure 3 is a plot of this correlation compared with the experimental data points for $\Phi_{LO}^2$. As can be noted, the proposed correlation predicts the pressure drop for both R-134a and R-12 within ± 10 %.

The following flow regimes were observed through the sight glasses during the experiments: for lower qualities and lower mass fluxes more wavy-stratified type of flow was observed; for higher qualities and higher mass fluxes more misty-annular type of flow was observed; in the intermediate region, the observed flow regimes were annular. Although many types of flow regimes were observed, for most experiments conducted the flow regimes were annular.
Figure 3. Comparison of two-phase multipliers from Eq. (13) with the experimental ones

Comparisons with Other Correlations

Several studies on two-phase pressure drop have developed correlations using graphical methods [Lockhart and Martinelli (1947), Martinelli and Nelson (1948), Baroczy (1966), Thom (1964), and Isbin (1959)]. Due to the cumbersome and less practical nature of these methods compared to equation-based ones, these graphical correlations will not be considered in this work. However, these studies provided insights and understanding of the physical phenomena during two-phase flow and helped finding the appropriate non-dimensional parameters associated with pressure drop.

Chisholm (1968, 1973, 1983) has published important papers and books on pressure drop and has improved several correlations that predicted frictional pressure drop during two-phase flow for many different fluids. The most general form of his correlation can be expressed as

\[
\phi_L^2 = 1 + (\Gamma^2 - 1)\left[\beta \left(1 - x\right)\right]^{0.875} + x^{1.75}\]

Figure 4 is a plot of this correlation compared with the experimental data points for the frictional pressure drop ratio \(\phi_L\). As can be noted, Chisholm's correlation predicts frictional pressure drop for both R-134a and R-12 between +10% and -40%. Values for small qualities were slightly overpredicted while those for high qualities were underpredicted. Chisholm (1973) also reported considerable underestimation using his procedures on the data of Petrick (1958) for
horizontal tubes where the mass flux was below 143.5 lbm/ft²-s (700 kg/m²-s). However, for higher mass fluxes this correlation showed good agreement with Baroczy's data (1966).

Jung and Radermacher (1989) developed a correlation for $\phi_{L0}^2$ using the refrigerants R-22, R-114, R-12 and R-152a, in which the two-phase multiplier $\phi_{L0}^2$ based on the total mixture flowing as liquid is expressed as

$$\phi_{L0}^2 = 12.82X_a^{-1.47}(1-x)^{1.8}$$

(17)

Figure 5 is a plot of this correlation compared with the experimental data points. The Jung-Radermacher correlation overpredicts the experimental data by approximately 15% on the average. The disagreement between their correlation and the experimental data results from the development of the correlation using data on flow with heat transfer. Acceleration pressure drop was not separated out of the correlation based on the premise that it could also be correlated as a function of $X_H$. Therefore, the correlated equation overestimates the pressure drop due to friction alone.
Figure 5. Comparison of the two-phase multipliers from Eq. (17) with the experimental ones
**OIL INFLUENCE**

Adding oil to a refrigerant changes the properties of the mixture and affects the pressure drop during two-phase flow. The addition of oil to a refrigerant generally increases the viscosity and surface tension. By increasing the viscosity of the mixture, the pressure drop will increase due to greater shear stresses between the wall and liquid film as well as between the liquid film and the vapor core.

Tichy et al. (1985) multiplied a smooth-tube, pure-refrigerant pressure drop correlation by a polynomial function of oil concentration to estimate evaporation pressure drop. They used a quadratic function of oil concentration and the Dukler II/Hughmark correlation [Dukler et al. (1964)] together with the homogeneous void fraction model for the momentum pressure drop. The obtained expression was corrected by Schlager et al. (1990) to

\[
\Delta p_{\text{oil}} = \Delta p_{\text{calc}}(1 + 41.3 \omega - 479 \omega^2)
\]  

Using the same procedure described above, we correlated the pressure drop data for evaporation in smooth tubes with mixtures of R-134a and three oils (PAG 0332, PAG 0354, and Ester) and of R-12 and two mineral oils (3GS and 4GS). To evaluate the oil influence on two-phase evaporation pressure drop for refrigerant-oil mixtures, a second set of tests were conducted over the following range of parameters: mass flux, 20.5-102.5 lbm/ft²-s (100-500 kg/m²-s); heat flux, 1,585-9,510 Btu/hr-ft² (5,000-30,000 W/m²); test section inlet quality, 20-60%; saturation temperature 41°F (5°C).

Using 157 experimental data points, a regression analysis was carried out to determine the functional dependence of the pressure drop ratio, \( \Delta p_{\text{oil}}/\Delta p_{\text{pure}} \), to the oil concentration, \( \omega \), for each oil-refrigerant mixture. The calculated correlating coefficient was 0.985 and the mean deviation was 3.6%. The final form of the correlation can be expressed as follows:

\[
\Delta p_{\text{oil}} = \Delta p_{\text{pure}}(1 + 12.4 \omega - 110.8 \omega^2)
\]  

Figure 6 is a plot of the average \( \Delta p_{\text{oil}}/\Delta p_{\text{pure}} \) against the oil concentration, \( \omega \), for various refrigerant-oil mixtures. Figure 7 is a plot of this correlation compared with the experimental data points for the pressure drop ratio \( \Delta p_{\text{oil}}/\Delta p_{\text{pure}} \). As can be noted, the proposed refrigerant-oil pressure drop correlation was able to predict the pressure drop for both R-134a and R-12 within ± 7.5%.
Figure 6. Average pressure drop ratio versus oil concentration for various refrigerant-oil mixtures

Figure 7. Comparison of the predicted pressure drop ratios from Eq. (19) with the experimental average pressure drop ratios
CONCLUSIONS

For the experiments conducted, the observed flow regimes were predominantly annular. In addition, wavy-stratified flow was observed for lower qualities and lower mass fluxes while misty-annular type of flow was observed for higher qualities and higher mass fluxes.

A new correlation for two-phase frictional pressure drop was developed by modifying the Lockhart-Martinelli (1947) and Martinelli-Nelson (1948) separated flow model to include a Froude number dependence. This takes into account the influence of mass velocity described by Baroczy (1966) and Chisholm (1968), among others. The correlation, represented by Eqs. (13), (14), and (15), predicts the frictional pressure drop for R-134a and R-12 data as a function of $X_u$ and $F_r$ with a mean deviation of 4.6%. Although this correlation predicted the data well, the effect of Froude number on two-phase multiplier $\phi_{LO^2}$ can be neglected for annular and misty flows. This should be expected since the body forces acting on the refrigerant are negligible for annular and misty flows.

For refrigerant-oil mixtures, the pressure drop increases as the oil concentration increases. A correlation was developed for the pressure drop ratio $\Delta P_{oil}/\Delta P_{pure}$ as a function of the oil concentration, $\omega$. The correlation, expressed by Eq. (19), predicts the present data for both R-134a and R-12 and five different oils with a mean deviation of 3.6%.
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