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Characteristics of Refrigerant Film Thickness, Pressure Drop, and Heat Transfer in Annular Flow

Evan T. Hurlburt and Ty A. Newell

Abstract

Common refrigeration and air conditioning cycles are dependent on two-phase flow for efficient heat transfer with minimal pressure drop. Design of heat exchangers for these systems is aided by an understanding of the refrigerant pressure drop and local heat transfer coefficient. An estimate of the liquid fraction is also important for predicting the charge required in a system. The present work develops a semi-analytical model for predicting liquid fraction, pressure drop, and heat transfer for pure refrigerants in the annular flow regime. The model uses the approach of coupling a uniformly thick, turbulent liquid film layer with a turbulent vapor core. Model predictions are compared to experimental evaporation and condensation data for R11, R12, R134a, and R22. These refrigerants represent the low, medium, and high pressure ranges found in common refrigeration systems. The uniform film model, when compared to experimental data, provides a reference for understanding some of the mechanisms that are important to refrigerant two-phase flow.

Introduction

The refrigeration and air conditioning industry has historically been interested in two-phase flow, however, a renewed interest in understanding two-phase flow from a more fundamental basis has developed over the past decade as new candidate refrigerant compounds are examined. Conversion from common refrigerants to lesser known compounds causes significant concern among manufacturers in terms of system performance, reliability, liability, and consumer acceptance. Continued research into two-phase flow related to refrigerants will continue to be important in order to model and design "enhanced" surfaces, determine the effects and predict the movement of lubricating oil in refrigerant vapor lines, and predict the effects of zeotropic refrigerant mixtures that may offer performance advantages. Characteristics of pressure drop, heat transfer, and void fractions of common refrigerants are modeled and compared to experimental data in this study. The semi-analytical modeling approach used is helpful for understanding physical processes that are important over the annular flow region of interest to refrigeration systems.
Environmental concerns have created significant interest in understanding two-phase flow phenomena in the air conditioning and refrigeration community. Two driving forces are ozone depletion and global warming. Ozone depletion concerns have resulted in the phase out of fully halogenated compounds containing chlorine (CFCs). R11 (CCl₃F) and R12 (CCl₂F₂) are two examples of common refrigerants that have been phased out. New refrigeration equipment using partially halogenated compounds containing chlorine, called HCFCs, will not be manufactured after 2010. R22 (CHClF₂) is an HCFC that is commonly used for household air conditioning systems.

Global warming is a concern that has affected the refrigeration industry in two ways. First, many refrigerant compounds are strong absorbers of infrared radiation. When a refrigerant is released into the atmosphere, a "direct" contribution to global warming is realized. R134a (CF₃CH₂F), the common replacement for R12 systems, has a 100 year Global Warming Potential (GWP) of 1200. This represents the amount of carbon dioxide that would cause the same level of infrared radiation absorption as a unit mass of the refrigerant over a 100 year period. GWPs range from 100 to 5000 for most compounds of interest for refrigeration (DOE (1993)).

The second effect relevant to global warming is called the "indirect" effect. Energy consumption in general, may require combustion of fossil fuels with the resulting release of carbon dioxide. Coal produces a significant amount of carbon dioxide relative to natural gas on an equivalent energy basis. Renewable energies (hydropower, solar energy, wind energy) and nuclear energy are examples of processes that result in negligible production of carbon dioxide. From the viewpoint of a refrigeration equipment manufacturer, reducing the indirect effect requires improving a component's energy efficiency. A U.S. refrigerator manufactured in 1970 produced 21,000 kg of carbon dioxide over its lifetime from both direct and indirect effects. Direct effects from the release of refrigerant compounds accounted for 25 percent of this amount. A U.S. refrigerator manufactured after 1995 can expect to have an equivalent carbon dioxide output of 6500 kg. Approximately 1000 kg of carbon dioxide is due to the direct effect of the new replacement refrigerant compounds (if not reclaimed) and the remaining portion is due to the indirect effect of a unit's electrical energy consumption (Newell (1996)).

Background

A wide range of operating conditions and system capacities have resulted in a relatively wide variety of substances used as refrigerants. The present paper focuses on halocarbon refrigerants that are derivatives of methane and ethane. Combinations of fluorine, chlorine, and hydrogen atom arrangements on these one-carbon and two-carbon molecules leads to a large range of properties that are significantly different from properties of saturated steam or air-water two-phase flow properties. While several correlation models exist for common refrigerants, significant uncertainty
exists when highly empirical correlations are used to model new refrigerant candidates. In addition, purely empirical correlations, while employing physically derived parameter groups, do not allow a more refined examination of the details of the flow field.

Table 1 shows density, viscosity, and thermal conductivity data for R22, R134a, and R123 refrigerants relative to air-water. The refrigerant data show saturated liquid and vapor properties at two temperatures that represent variations one may expect to find between the condenser and evaporator of a refrigeration system. R22 is an example of a "high pressure" refrigerant. The high pressure results in relatively high vapor densities. R134a is typical of a medium pressure range compound while R123, one replacement for R11, is a "low pressure" refrigerant that is often working at pressures less than atmospheric pressure. Figure 1 is a schematic showing the range of mass fluxes and heat fluxes that are characteristic of refrigerants for common household applications. Also plotted on Figure 1 are areas that represent validity ranges of some commonly used correlations for two-phase refrigerant flows (Jung and Radermacher (1991), Kandlikar (1990), Pierre (1956), Shah (1976)). Common tube diameters for these refrigeration systems range from 3 mm to 10 mm. Significant activity is occurring with microchannels for refrigeration systems where tube passageways are less than 1 mm in diameter (Huen (1995) and Zietlow (1995)). These systems tend to be in a range where surface tension effects are significant.

Table 1 Comparison of saturation properties for R22, R134a, and R123 to air-water. R22, R134a, and R123 data taken from Gallagher, et al. (1993). Air and water data taken from Incropera and DeWitt (1985).

<table>
<thead>
<tr>
<th></th>
<th>pressure (kPa)</th>
<th>density (kg/m³)</th>
<th>viscosity (mpoise)</th>
<th>thermal conductivity (W/m-K)</th>
</tr>
</thead>
<tbody>
<tr>
<td>R22 vapor (0º C)</td>
<td>497.7</td>
<td>21.11</td>
<td>117.2</td>
<td>0.0100</td>
</tr>
<tr>
<td>R22 liquid (0º C)</td>
<td>497.7</td>
<td>1279.</td>
<td>2205.</td>
<td>0.1034</td>
</tr>
<tr>
<td>R22 vapor (40º C)</td>
<td>1538.</td>
<td>65.72</td>
<td>141.9</td>
<td>0.0126</td>
</tr>
<tr>
<td>R22 liquid (40º C)</td>
<td>1538.</td>
<td>1127.</td>
<td>1424.</td>
<td>0.0793</td>
</tr>
<tr>
<td>R134a vapor (0º C)</td>
<td>292.2</td>
<td>14.23</td>
<td>109.4</td>
<td>0.0119</td>
</tr>
<tr>
<td>R134a liquid (0º C)</td>
<td>292.2</td>
<td>1294.</td>
<td>2857.</td>
<td>0.0939</td>
</tr>
<tr>
<td>R134a vapor (40º C)</td>
<td>1018.</td>
<td>49.09</td>
<td>131.0</td>
<td>0.0155</td>
</tr>
<tr>
<td>R134a liquid (40º C)</td>
<td>1018.</td>
<td>1144.</td>
<td>1775.</td>
<td>0.0736</td>
</tr>
<tr>
<td>R123 vapor (0º C)</td>
<td>32.65</td>
<td>2.24</td>
<td>96.0</td>
<td>0.0083</td>
</tr>
<tr>
<td>R123 liquid (0º C)</td>
<td>32.65</td>
<td>1522.</td>
<td>5920.</td>
<td>0.0858</td>
</tr>
<tr>
<td>R123 vapor (40º C)</td>
<td>154.3</td>
<td>9.59</td>
<td>111.1</td>
<td>0.0104</td>
</tr>
<tr>
<td>R123 liquid (40º C)</td>
<td>154.3</td>
<td>1423.</td>
<td>3623.</td>
<td>0.0743</td>
</tr>
<tr>
<td>Air (20º C)</td>
<td>101.3</td>
<td>1.23</td>
<td>1.8</td>
<td>0.026</td>
</tr>
<tr>
<td>Water (101 kPa, 20º C)</td>
<td>101.3</td>
<td>1001.</td>
<td>10000.</td>
<td>0.60</td>
</tr>
</tbody>
</table>
Fluid Flow and Film Thickness Modeling

In developing a model that predicts refrigerant heat transfer, the physics of the flow field in both the liquid and vapor phases must first be addressed. While flowing through an evaporator or condenser, a refrigerant passes through a series of different two-phase flow regions. For example, in the evaporator there is a relatively large fraction of liquid at the entrance. In a horizontal tube this results in either a slug flow or stratified-wavy flow pattern with most of the liquid flowing on the tube bottom and vapor flowing in the space above the liquid.

At constant total mass flow rate, as the liquid fraction decreases, vapor velocity increases. If the vapor velocity is high enough, the flow pattern will transition to annular flow. In annular flow, the fluid flows in a thin film on the tube wall with the vapor flowing in the core created by the liquid boundary. If the liquid Reynolds number is high enough, periodic collections of waves, called disturbance waves, will form. A fraction of the liquid phase flows in the vapor core in the form of droplets. At high refrigerant quality, the flow field transitions to mist flow in which the liquid phase is carried along primarily as droplets in the vapor (Whalley (1987), Carey (1992)).

At moderate to high refrigerant mass flux and in tubes of hydraulic diameter from 3 mm to 10 mm in diameter, the predominant flow pattern in both the evaporator and condenser is annular flow. This was demonstrated by Wattelet (1994) for evaporator tubes with inner diameters of 7.04 mm, 7.75 mm, 10.21 mm, and 10.92 mm. Annular flow was observed in all four tubes at mass fluxes above 200 kg/m²-s over the quality range of 0.2 < x < 0.9.
The vapor Reynolds number, $Re_v$, is well above 2000 for refrigerant mass fluxes greater than 200 kg/m$^2$-s in tubes of hydraulic diameter greater than 3 mm, and over the quality range $0.2 < x < 0.9$. For example, in Dobson's (1994) condensation experiments with a 3.14 mm diameter tube, at 0.2 quality and at a mass flux of 200 kg/m$^2$-s, the vapor Reynolds number is about 60,000. Therefore, the vapor phase is modeled as a turbulent flow.

The liquid film Reynolds number is defined as

$$Re_{LF} = \frac{2 \dot{m}_L}{\pi r \mu_L}.$$  

Here $\mu_L$ is the liquid dynamic viscosity and $\dot{m}_L$ is the liquid film mass flow rate, which is assumed to be the entire liquid mass flow rate. At mass fluxes greater than 200 kg/m$^2$-s, in tubes of hydraulic diameter between 3 and 10 mm, and over the quality range $0.2 < x < 0.9$, the liquid film Reynolds number varies between about 200 and 10,000. Assuming a transition Reynolds number of similar order to pipe flow, this value suggests the liquid phase to be in both the laminar and turbulent flow regimes. When laminar flow is assumed in the liquid phase, however, model predictions of heat transfer rates have been found to be several times lower than experimental values. Carpenter and Colburn (1951) hypothesize that the shear driven liquid films found in annular two-phase flow transition to turbulence at Reynolds numbers of about 240, well below the typical value of 2000 used for transition in pipe flow.

Similarly, when the present model is used to predict heat transfer coefficients for pure refrigerants and the liquid layer is assumed to be in the laminar flow regime, model predictions are well below the experimental values of Wattelet (1994) and Dobson (1994). Based on these observations, the liquid phase is modeled as turbulent flow unless the liquid layer has a thickness less than that of the viscous sublayer. The liquid film flow is modeled as laminar flow for films thinner than a viscous sublayer.

In air-water modeling, the liquid film thickness is often assumed small relative to the tube radius. This will be assumed to be true for refrigerants as well. Model predictions will be shown later that validate this thin film assumption.
A graph with the distinct regions modeled in the liquid layer is shown in Figure 2. The velocity in this layer is modeled using von Karman's (1939) form of the Law of the Wall velocity profile.

Viscous Sublayer    \( u^+ = y^+ \quad y^+ < 5 \)

Buffer Region       \( u^+ = 5 \ln(y^+) - 3.05 \quad 5 < y^+ < 30 \)

Log Region          \( u^+ = \frac{1}{\kappa} \ln(y^+) + B \quad 30 < y^+ < h_L^+ \)

where \( u^+ = \frac{u}{u^*} \), \( y^+ = \frac{y u^*}{v_L} \), \( h_L^+ = \frac{h_L u^*}{v_L} \), \( u^* = \sqrt{\frac{\tau_i}{\rho_L}} \), \( B = 5.5 \), \( \kappa = 0.4 \)

Here \( y \) is the radial direction, \( u \) is the axial liquid velocity, \( \tau_i \) is the interfacial shear in the axial direction, \( h_L \) is the liquid film thickness, \( \mu_L \) is the liquid viscosity, \( \rho_L \) is the liquid density, and \( v_L \) is the liquid kinematic viscosity (\( v_L = \mu_L / \rho_L \)).

The log region starts at \( y^+ = 30 \) and extends all the way to the liquid surface. This assumes that turbulent eddies are not damped as the interface is approached and that no viscous sublayer exists near the surface. Davies (1972) points out that a free surface differs from a fixed boundary in that the free surface does not fully damp the eddy motion in the liquid. In addition, the waves on the liquid surface and the interaction of the liquid with the vapor phase may induce motion in this region that is similar to turbulence.

Figure 2 Schematic of liquid layer and liquid layer velocity profile.
Extending the Log Law to the liquid surface, the interface velocity, \( u_i \), can be found by evaluating the Log Region profile at \( h_L \).

\[
\frac{u_i}{u^*} = \frac{1}{\kappa} \ln(h_L^+) + B
\]

where \( h_L^+ = \frac{h_L u^*}{v_L} \), \( u^* = \sqrt{\frac{\tau_i}{\rho_L}} \), and \( B = 5.5 \) \hfill (1)

Another useful relation can be found by integrating the Law of the Wall velocity profile. The liquid mass flux for a uniform thin film is defined as

\[
u_L,\text{avg} \ A_c = 2\pi r \int_0^{h_L} \ u \ dy,
\]

where \( A_c = \pi r h_L \) and \( u_L,\text{avg} \) is the average liquid velocity in the axial direction. With the velocity profile a known function of the film thickness and shear stress, integration relates these to the average liquid velocity.

\[
\frac{u_L,\text{avg}}{u^*} = \frac{1}{\kappa} \ln(h_L^+) + B - \frac{1}{\kappa} \frac{64}{h_L^+}
\]

where \( u^* = \sqrt{\frac{\tau_i}{\rho_L}} \) \hfill (3)

When the liquid layer reaches a non-dimensional film thickness less than 30 (\( h_L^+ < 30 \)), the approach taken in modeling the liquid phase is to use the unmodified Law of the Wall velocity profile. An alternative approach would be to scale the viscous sublayer, buffer region, and log region boundaries, thus keeping intact a sublayer to turbulent region characteristic. Scaling of the boundaries has not been found to offer a clear advantage and has therefore not been used. When \( h_L^+ \) is less than 5, the liquid flow is modeled as laminar flow. In a uniform film thickness model, this condition is only reached at very high quality.

The vapor phase Reynolds number, for the flow conditions of interest, is always high enough to result in a turbulent flow. The vapor velocity profile, therefore, can be modeled using the Law of the Wall velocity profile with the friction velocity evaluated using the interfacial shear stress. No viscous sublayer or buffer regions are assumed to be present in the vapor.

Integrating the log law from 0 to \( r-h_L \) to determine the average vapor velocity gives the following equation,

\[
\frac{u_{g,\text{avg}} - u_i}{u_s^*} = \frac{1}{\kappa} \ln \left[ \frac{(r-h_L) u_s^*}{v_g} \right] + B - \frac{3}{2} \frac{1}{\kappa}
\]

where \( u_s^* = \sqrt{\frac{\tau_s}{\rho_g}} \) \hfill (4)
Here, \( u_{g,avg} - u_i \) is the average vapor velocity relative to the liquid film interface, \( \tau_s \) is the smooth tube vapor wall shear stress, \( \rho_g \) is the vapor density, \( r - h_L \) is the radius of the vapor core, and \( v_g \) is the kinematic viscosity of the vapor (\( v_g = \mu_g / \rho_g \), where \( \mu_g \) is the vapor viscosity). Equation (4) predicts values identical to the smooth-tube friction factors in the Moody chart. It is an approximation since its derivation assumes the entire velocity profile to be described by the Log Law. The error is, however, negligible since nearly all the mass is in the Log Region when the Reynolds number is much greater than 2000.

The vapor core in annular two-phase flow is driven by the pressure gradient in the tube. The liquid film, however, is driven primarily by momentum transfer from the vapor. This momentum transfer is very large resulting in high shear stress in the liquid film and from 2 to 10 times higher pressure drop than is found in smooth tube single-phase flow (Asali, et al. (1985)).

A detailed accounting of the momentum transfer requires modeling of the complex interaction between the liquid surface waves and the vapor flow field. An alternate approach is to represent the momentum exchange by an average interfacial shear stress, \( \tau_i \). In the modeling to date, both a momentum exchange model and a prediction based on a correlation have been attempted. The momentum exchange model provides a conceptual guide to help understand the transfer. The correlation-base approach is presented in this work.

The correlation used to predict the interfacial shear is a modified form of the correlation developed by Asali, et al. (1985) for vertical flow.

\[
\frac{\tau_i}{\tau_s} - 1 = 0.45 \operatorname{Re}_g^{-0.3} (\phi h_L^* - 4)
\]  

where \( \phi = \frac{\mu_L}{\mu_g} \left( \frac{\rho_g}{\rho_L} \right)^{0.5} \), \( h_L^* = \frac{h_L}{v_L} \), \( u^* = \sqrt{\frac{\tau_i}{\rho_L}} \), and \( \operatorname{Re}_g = \frac{\rho_g (u_{g,avg} - u_i) (r - h_L)}{\mu_g} \).

The smooth tube shear, \( \tau_s \), can be found from equation (4), and it is the limiting value reached as the film thickness tends toward zero.

In the original form of equation (5) proposed by Asali, et al. (1985), the exponent on the vapor phase Reynolds number, \( \operatorname{Re}_g \), is -0.2. The value of -0.3 is used to improve the match between the model predictions and Wattelet's (1994) R134a experimental data for pressure drop. Figure 3 shows a comparison between the data and model predictions for three mass fluxes over a range of quality. Model predictions with the vapor phase Reynolds number equal to -0.2 is included for comparison.
Figure 3 Comparison of model pressure drop predictions with evaporator data for R134a from Wattelet (1994). Model predictions are at two different values of the exponent on the Reynolds number in the interfacial shear correlation. (Horizontal D = 7.75 mm L = 1.22 m T = 50°C)

Film Thickness Predictions

Equations (1), (3), (4), and (5), can be solved simultaneously at a specified liquid and vapor mass flow rate, tube radius, and refrigerant temperature to predict the thickness of the liquid film. Figure 4 shows this prediction for R-134a at three different mass fluxes over the quality range 0.2 < x < 0.9. Note how thin the film is at high quality (less than 0.2 mm for the chosen range of mass fluxes). Note, also, that the radius is at least 8 times greater than the film thickness at the lowest quality. At higher qualities it is more than 20 to 50 times greater. This is consistent with the thin film assumption.

It is interesting that the film is predicted to become progressively thinner as the total mass flux increases. This is due to the increased shear from the vapor. The higher average vapor velocity increases the average liquid film velocity allowing a higher flux through a smaller cross sectional area.
The model prediction of the non-dimensional film thickness, $h_L^+$, can be compared to the prediction from the correlations developed by Asali, et al. (1985) and Henstock and Hanratty (1976).

$$h_L^+ \text{ Correlation} = [(0.34 \ Re_L^{0.6})^{2.5} + (0.0379 \ Re_L^{0.9})^{2.5}]^{0.4}$$

Table 2 shows this comparison for R-134a at mass fluxes of 200, 300, and 500 kg/m$^2$-s and a temperature of 50°C. The difference between the model and the correlation predictions is never more than 5%. Although this does not prove the Law of the Wall velocity profile is the actual profile within the liquid film, the agreement indicates that the Law of the Wall may be a reasonable means of modeling the film. It is worth noting that R134a has a liquid to vapor density ratio of approximately 90 while the density ratio of liquid water to air, upon which the interfacial shear correlation is based, is approximately 800. Note also that although the film thickness decreases with mass flux, the non-dimensional film thickness increases. This is because the friction velocity is increasing faster than the film thickness is decreasing.
Table 2  Comparison of non-dimensional R134a film thickness prediction over a range of mass fluxes from the present model and from the correlations of Asali, et al. (1985) and Henstock and Hanratty (1976).

<table>
<thead>
<tr>
<th>Quality</th>
<th>Model 200 kg/m²-s</th>
<th>Correlation 200 kg/m²-s</th>
<th>Model 300 kg/m²-s</th>
<th>Correlation 300 kg/m²-s</th>
<th>Model 500 kg/m²-s</th>
<th>Correlation 500 kg/m²-s</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.2</td>
<td>91</td>
<td>87</td>
<td>127</td>
<td>121</td>
<td>193</td>
<td>186</td>
</tr>
<tr>
<td>0.3</td>
<td>81</td>
<td>78</td>
<td>112</td>
<td>108</td>
<td>170</td>
<td>166</td>
</tr>
<tr>
<td>0.4</td>
<td>71</td>
<td>69</td>
<td>97</td>
<td>95</td>
<td>148</td>
<td>146</td>
</tr>
<tr>
<td>0.5</td>
<td>61</td>
<td>60</td>
<td>84</td>
<td>82</td>
<td>126</td>
<td>125</td>
</tr>
<tr>
<td>0.6</td>
<td>51</td>
<td>50</td>
<td>70</td>
<td>69</td>
<td>105</td>
<td>104</td>
</tr>
<tr>
<td>0.7</td>
<td>41</td>
<td>40</td>
<td>55</td>
<td>55</td>
<td>83</td>
<td>82</td>
</tr>
<tr>
<td>0.8</td>
<td>31</td>
<td>30</td>
<td>41</td>
<td>40</td>
<td>60</td>
<td>60</td>
</tr>
<tr>
<td>0.9</td>
<td>20</td>
<td>18</td>
<td>25</td>
<td>24</td>
<td>36</td>
<td>35</td>
</tr>
</tbody>
</table>

The predicted liquid volume fraction can be calculated from the predicted film thickness (Liquid Fraction = \(2h_L/r - (h_L/r)^2\)). Figure 5 shows a comparison of the liquid volume fraction between the uniform film model and experimental data from Sacks (1975) for R11 and R22. As with Wattelet (1994) and Dobson (1994), the quality changes by a small amount from the inlet to the exit of the test section, and the average quality is used. Sacks (1975) paper is especially valuable because it presents tables of raw data and because the R22 data provides a valuable link that can be used in new experiments as newer refrigerant candidates are examined. Generally, liquid fractions are greater for R22 which has a lower liquid to vapor density ratio than R11. Also plotted on Figure 5 for reference is the model prediction for air-water, which has a significantly higher liquid-vapor density ratio than R11. The exponent on the vapor Reynolds number for the air-water curve in the interfacial shear model was left at -0.2 as in the original Asali, et al. (1985) paper.
Figure 5 Comparison of predicted liquid fraction versus quality for air-water, R11, and R22 and Sacks (1975) experimental results for R11. (Horizontal D = 9.58 mm L = 2.24 m T = 30°C) and R22 (Horizontal D = 9.58 mm L = 2.24 m T = 27°C).

Pressure Drop Predictions

Equations (1), (3), (4), and (5) can be solved simultaneously at a specified liquid and vapor mass flow rate, tube radius, and refrigerant temperature to predict the interfacial shear stress, \( \tau_i \). From this, the total pressure drop in a tube of length \( L \) can be calculated

\[
\Delta p = \frac{2 \tau_i L}{r - h_L}. \tag{6}
\]

Figure 6 shows this prediction at a series of fixed qualities for R-134a at 5 °C in a 7.75 mm diameter, 1.22 m long tube at a mass flux of 300 kg/m^2-s. The pressure drop depends primarily on the vapor velocity and the non-dimensional film thickness. It increases with quality as vapor velocity increases, reaching a maximum near a quality of 0.8. Beyond 0.8, pressure drop decreases as the film thins, diminishing its grip on the vapor. The lower curve in Figure 6 shows the pressure drop if the vapor phase were flowing alone in a smooth tube. The largest percentage increase above the smooth tube value occurs at low quality where the liquid Reynolds numbers are largest and the film is thickest.
Figure 6 Comparison of pressure drop for two phase flow and smooth tube vapor flow for R134a over a range of qualities. \(D = 7.75 \text{ mm} \quad L = 1.22 \text{ m} \quad T = 50^\circ \text{C}\)

Figure 7 compares uniform film model predictions for pressure drop with data from Sacks (1975) for two mass fluxes. Also shown are pressure drop predictions from a Lockhart-Martinelli-based empirical correlation for refrigerants (Souza and Pimenta (1995)). (An assumed surface roughness equal to 0.03276 mm was used in Souza's correlation. This is the same value used in Souza and Pimenta (1995).) Sacks (1975) tabular data conditions were used as input for the model and for the Souza correlation. A third order polynomial is fit through the data sets in order to view trends. Both the experimental data and the Souza correlation show a maximum pressure drop at a higher quality than predicted by the uniform film model. A transition from an annular to a droplet entrainment dominated flow may be responsible for this difference.

Figure 8 shows pressure drop comparisons for R11 between data from Sacks (1975), Souza and Pimenta's (1995) correlation, and the uniform film model. The uniform film model tends to underpredict the R11 results, in contrast to the trends observed in Figure 7 for R12. R11 has a relatively low vapor density, resulting in higher vapor velocities at the same mass flux-quality condition of R12. R11 and other low pressure refrigerants, such as R123, may tend to behave more like an annular film because of these higher velocities. Interestingly, the uniform film model tends to have a maximum pressure drop similar to the experimental data while the Souza correlation peaks at a significantly higher level.
Figure 7 Comparison of pressure drop between Sacks' (1975) experimental results, correlation prediction, and uniform film model predictions for R12 at high and low mass fluxes. (Horizontal D = 9.58 mm L = 2.24 m T = 27°C)

Figure 8 Comparison of pressure drop between Sacks' (1975) experimental results, correlation prediction, and uniform film model predictions for R11. (Horizontal D = 9.58 mm L = 2.24 m T = 30°C)
Figure 9 shows a comparison of pressure drop trends for R22 from Sacks experimental results and the Souza correlation. Trends are similar to those seen in the R12 results. Both the uniform film model and Souza's correlation peak at pressure drop levels lower than those observed experimentally, however, the experimental results tend to have some scatter at the high quality conditions.

![Figure 9 Comparison of pressure drop between Sacks' (1975) experimental results, correlation prediction, and uniform film model predictions for R22 at high and low mass fluxes. (Horizontal D = 9.58 mm L = 2.24 m T = 27°C)](image)

**Heat Transfer**

The heat transfer coefficient is defined as

\[ HTC = \frac{q''}{T_{w,\text{out}} - T_b} \]

To determine the heat transfer coefficient experimentally, the wall heat flux, \( q'' \), the bulk temperature, \( T_b \), and an average outside tube wall temperature, \( T_{w,\text{out}} \), must be measured. It is important to note that the difference between these two temperatures is often a small number. The fluid flow model predictions of momentum transport, shear stress, film thickness, and liquid and vapor phase velocity are the starting point for developing a heat transfer model.
The shear in the liquid film is proportional to the velocity gradient and to the transport of momentum due to molecular plus turbulent viscosity,
\[ \tau = \rho_L (v_L + \varepsilon_m) \frac{du}{dy} \]  
(7)

where \( \varepsilon_m \) is the turbulent momentum diffusivity.

Using the assumption of constant shear stress in the liquid, \( \tau = \tau_w = \tau_f \), and the definition of the friction velocity, \( u^* = \sqrt{\tau_w / \rho_L} \), equation (7) can be rewritten in the form
\[ \varepsilon_m = \frac{1}{v_L} \frac{du^*/dy^*}{} \]  
(8)

Solving the Law of the Wall velocity profile for \( du^*/dy^* \), we can find the turbulent momentum diffusivity, \( \varepsilon_m \), in each of the three liquid layers.

Viscous Sublayer
\[ \varepsilon_m = 0 \quad y^+ < 5 \]

Buffer Region
\[ \varepsilon_m = \frac{y^+ - 1}{5} \quad 5 < y^+ < 30 \]

Log Region
\[ \varepsilon_m = \kappa (y^+ - 1) = \kappa y^+ \quad 30 < y^+ < h_L^+ \]

If analogous mechanisms exist for the turbulent transport of momentum and the turbulent transport of thermal energy, the momentum diffusivity can be used as an estimate of the thermal diffusivity \( (\varepsilon_m = \varepsilon_t) \).

The heat flux can be related to the thermal diffusivity and the temperature gradient by an equation analogous to equation (7)
\[ q'' = \left( \frac{q}{A_w} \right) = - \rho_L C_{pl} v_L \left( \frac{1}{Pr_L} + \frac{\varepsilon_t}{v_L} \right) \frac{dT}{dy} \]  
(9)

Here, \( C_{pl} \) is the specific heat at constant pressure of the liquid, \( A_w \) is the tube wall surface area, and \( Pr_L \) is the Prandtl number of the liquid, \( Pr_L = \mu_L C_{pl} / k_L \), where \( k_L \) is the conductivity of the liquid. The thin film assumption, \( h_L \ll r \), allows the heat flux to be treated as constant in the liquid, \( q'' = q_w'' = q_t'' \). With the thermal diffusivities known, Equation (9) can be integrated across each of the liquid layers to find the heat transfer resistance, \( R \), in each layer. The resistance is defined as \( R = \Delta T / q'' \) where \( \Delta T \) is the temperature change across each layer.

\[ R_{Viscous Sublayer} = \frac{5 Pr_L}{\rho_L C_{pl} u^*} \quad y^+ < 5 \]
The liquid resistance network can be used to estimate a heat transfer coefficient based on the average inner wall temperature, $T_{w,in}$, and the liquid-vapor interface temperature, $T_i$.

$$HTC = \frac{q''}{T_{w,in} - T_i}$$

Using the resistance network we find

$$HTC = \frac{\rho_L C_{pL} u^*}{5 \Pr_L + 5 \ln (1 + 5 \Pr_L) + \frac{1}{\kappa} \ln \left( \frac{h_{L+}}{30} \right)}$$

where $u^* = \sqrt{\frac{\tau_i}{\rho_L}}$

The accuracy of this estimate depends on the following assumptions:

1) The tube wall resistance is negligible, i.e. $T_{w,out} = T_{w,in}$.

2) The heat transfer resistance in the vapor is small compared to the resistance in the liquid. This is true if most of the energy transfer at the interface goes into phase change resulting in a very small temperature gradient in the vapor phase.

3) The liquid interface temperature is circumferentially uniform.

4) The tube wall temperature is circumferentially uniform. This is true if the interface temperature, the external boundary conditions, and the liquid film thickness all do not vary circumferentially.

5) The bulk temperature is approximately equal to the interface temperature.

The bulk temperature is defined as,

$$T_b = \frac{\int_0^{h_L} \rho_L u_L C_{pL} T \, dA + \int_{h_L}^r \rho_g u_g C_{pg} T \, dA}{\dot{m}_L C_{pL} + \dot{m}_g C_{pg}}$$
where \( u_g \) is the vapor velocity, \( C_{pg} \) is the specific heat at constant pressure for the vapor, and \( \dot{m}_g \) is the vapor mass flow rate. Assumption 5 depends, therefore, on whether the mass flow averaged temperature is equal to the interface temperature. This is true when the temperature gradient in the vapor is negligible, and when the mass flow passing through the high temperature gradient region near the wall is small compared to the total mass flow.

The heat transfer coefficient estimate outlined above assumes the film thickness is such that \( hL^+ > 30 \). A new heat transfer coefficient relation must be derived for the regions in which \( hL^+ < 30 \). As discussed in the fluid flow model sections, the approach taken is to use the Law of the Wall profile up to the limit of the film thickness. The heat transfer coefficient thus becomes

\[
HTC = \frac{\rho_L C_{pL} u^*}{5 Pr_L + 5 \ln \left(1 + \frac{hL^+}{5 Pr_L - Pr_L}\right)} \quad \text{for } 5 < hL^+ < 30
\]

and

\[
HTC = \frac{\rho_L C_{pL} u^*}{(hL^+) Pr_L} \quad \text{for } hL^+ < 5
\]

where \( u^* = \sqrt{\frac{\tau_i}{\rho_L}} \)

Using equation (10) with the film thickness and friction velocity predictions from the fluid flow model, we can calculate the heat transfer coefficient for pure refrigerant annular two-phase flow. Figure 10 shows the model heat transfer coefficient predictions at a series of fixed qualities for R-134a at 5 °C in a 7.75 mm diameter, 1.22 m long tube at mass fluxes of 200, 300, and 500 kg/m²-s. The heat transfer coefficient increases with quality in a similar manner to pressure drop. This is expected due to heat transfer's strong dependence on shear.
Figure 10  Comparison of evaporation heat transfer coefficients from the uniform film thickness model and Wattelet's (1994) experimental results for R134a. (Horizontal  \(D = 7.75\) mm  \(L = 1.22\) m  \(T = 50\) C)

Table 3  Percentage of thermal resistance in the three liquid film layers for the R134a conditions shown in Figure 10 at a mass flux of 300 kg/m²-s.

<table>
<thead>
<tr>
<th>Quality</th>
<th>(R_{Visc})%</th>
<th>(R_{Buff})%</th>
<th>(R_{Log})%</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.2</td>
<td>51.4</td>
<td>39.3</td>
<td>9.3</td>
</tr>
<tr>
<td>0.3</td>
<td>51.8</td>
<td>39.6</td>
<td>8.6</td>
</tr>
<tr>
<td>0.4</td>
<td>52.3</td>
<td>40.0</td>
<td>7.8</td>
</tr>
<tr>
<td>0.5</td>
<td>52.8</td>
<td>40.4</td>
<td>6.8</td>
</tr>
<tr>
<td>0.6</td>
<td>53.5</td>
<td>40.9</td>
<td>5.7</td>
</tr>
<tr>
<td>0.7</td>
<td>54.3</td>
<td>41.5</td>
<td>4.2</td>
</tr>
<tr>
<td>0.8</td>
<td>55.5</td>
<td>42.4</td>
<td>2.2</td>
</tr>
<tr>
<td>0.9</td>
<td>58.3</td>
<td>41.7</td>
<td>0.0</td>
</tr>
</tbody>
</table>

Table 3 shows the importance of the viscous sublayer and buffer regions as thermal resistance layers. Although significantly thinner than the log region over most of the quality range, the viscous sublayer and buffer region account for over 90 percent of the thermal resistance. This observation is important from a modeling perspective. If the log region can be assumed well
mixed, a more detailed description of the complex flow field in this region may not be necessary for accuracy in heat transfer predictions. Rohsenow, et al. (1956), expressed a similar thought in response to a comment by Seban: "Realizing that the region of expected error is also a region of very small resistance to heat flow when compared to the 'buffer' layer and the laminar sublayer, it is felt that deviations in this area will have but little effect on the predicted values of heat-transfer coefficient for the entire film."

The current model assumes no differences between evaporation and condensation processes. The effects of evaporation and condensation on the transfer of momentum between the phases is not considered, however, as shown by Wattelet (1994), the momentum exchange due to phase change is generally a small effect for refrigerants. Applying the model to both evaporation and condensation results indicates some of the differences between these two processes.

Figure 10 shows significant deviation between the model and Wattelet's (1994) data at high mass flux. The relatively thin liquid film may tend to dry out on the upper tube surface, thus reducing the heat transfer coefficient.

Figure 11 shows that R22 has similar trends to R134a evaporation data. R22, due to relatively high pressure and therefore high vapor density, has somewhat lower heat transfer coefficients from the lower vapor velocities.

Figure 11  Comparison of evaporation heat transfer coefficients from the uniform film thickness model and Wattelet's (1994) experimental results for R22. (Horizontal D = 7.75 mm L = 1.22 m T = 5° C)
Figures 12 and 13 show a comparison between uniform film model predictions and Dobson's (1994) condensation data for R134a. Heat transfer is underpredicted by the model for the 3.14 mm diameter tube shown in Figure 12. Figure 13 displays reasonably good agreement over a broad quality range for the 7.04 mm diameter tube. At qualities greater than 0.6, a trend is observed that indicates the importance of other mechanisms. Transition to a region with significant liquid entrainment in the vapor core could be responsible for thinning the liquid film layer, thus enhancing heat transfer. Another possibility may be the formation of film "streaks" due to surface tension instabilities that inhibit formation of a continuous film.

Comparison between Figures 12 and 13 also shows that uniform film model predictions do not indicate significant differences in heat transfer coefficient for the two tube sizes. Comparing experimental data, however, indicates that the smaller tube has significantly higher heat transfer. Recent refrigerant work in "microchannel" heat transfer (Huen (1995) and Zietlow (1995)) indicates enhanced heat transfer over larger diameter tubes. Increased importance of surface tension is considered to be a primary factor in structuring the flow field into a more efficient configuration. The 3 mm diameter tube results may be indicating that it is in a range in which small tube effects are important.

![Graph showing heat transfer coefficients](image)

Figure 12  Comparison of condensation heat transfer coefficients from the uniform film thickness model and Dobson's (1994) experimental results for R134a. (Horizontal D = 3.14 mm L = 0.94 m T = 45°C)
Figure 13 Comparison of condensation heat transfer coefficients from the uniform film thickness model and Dobson's (1994) experimental results for R134a. (Horizontal \( D = 7.04 \) mm \( L = 1.22 \) m \( T = 45^\circ C \))

Figure 14 Comparison of condensation heat transfer coefficients from the uniform film thickness model and Sacks' (1975) experimental results for R12. (Horizontal \( D = 9.58 \) mm \( L = 2.24 \) m \( T = 27^\circ C \))
Figure 14 compares R12 condensation data in a 9.58 mm diameter tube (Sacks (1975)) to the uniform film model. Heat transfer coefficients show similar agreement as seen in the 7 mm tube for R134a in Figure 13. A deviation between the uniform film model and experimental trends occurs for R12 in the 9 mm tube, however, the deviation occurs at a higher quality level than that observed for R134a in the 7 mm tube.

Figure 15 shows a comparison between the film model and R11 data from Sacks (1975) in the 9.58 mm diameter tube. The data is modeled reasonably well over the range of available data. Similar to the pressure drop trends, R11, with relatively high vapor velocities, may tend to be in a range with a reasonably uniform film.

Figure 16, 17, and 18 show comparisons between R22 model predictions and data from Sacks and Dobson. This is the only data set between the two studies with a common refrigerant. The uniform film model, in Figure 16, shows a tendency to underpredict Dobson's smaller tube heat transfer. This is similar to the trend seen for the small tube results for R134a. Similar trends are observed for Dobson's and Sacks' data in Figures 17 and 18 for tubes that are similar in size.

![Figure 15](image.png)

Figure 15  Comparison of condensation heat transfer coefficients from the uniform film thickness model and Sacks' (1975) experimental results for R11. (Horizontal D = 9.58 mm  L = 2.24 m  T = 30° C)
Figure 16 Comparison of condensation heat transfer coefficients from the uniform film thickness model and Dobson's (1994) experimental results for R22. (Horizontal D = 3.14 mm L = 0.94 m T = 45°C)

Figure 17 Comparison of condensation heat transfer coefficients from the uniform film thickness model and Dobson's (1994) experimental results for R22. (Horizontal D = 7.04 mm L = 1.22 m T = 45°C)
Conclusions

The uniform film model developed in this work provides a basis for understanding some of the mechanisms and trends involved in two-phase flow and heat transfer of refrigerants. Three types of refrigerants (high, mid-range, and low pressure) have been compared to the model. Generally, the low pressure refrigerant (R11) compares reasonably well in terms of pressure drop and heat transfer. Although the vapor density and transport properties are significantly different from the air-water system used for development of an interfacial shear stress model, other refrigerants in the low pressure range may also have reasonable agreement to a uniform film model over a relatively wide range of mass fluxes and qualities.

Mid-range and higher pressure refrigerants (R134a, R12, R22), which have higher vapor densities, show systematic deviations in pressure drop predictions from the uniform film model. Peak pressure drop at high qualities appears to indicate a flow structure that has other significant loss mechanisms than those modeled by a uniform film. Heat transfer is predicted reasonably well for evaporation at low mass fluxes, but the model over-predicts heat transfer at high mass fluxes. The over-prediction may be caused by a non-uniform film in which a thinner, upper tube region is prone to dryout. Condensation heat transfer data compares reasonably well with predictions over a broad quality and mass flux range, however, two areas of deviation from model predictions are
observed. First, heat transfer predictions are consistently low for results from a 3 mm diameter tube indicating that geometry may be significant as diameters reduce below that size. Second, high quality regions show increased heat transfer over the uniform model predictions, indicating other mechanisms are dominating the flow field and resulting heat transfer effects.

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