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ABSTRACT

Compressor suction and discharge lines in refrigeration systems must be designed to move compressor lubricants along the pipeline. This paper presents experimental and analytical results that describe the characteristics of two-phase, viscous liquid flows. Smooth-walled, flat plate test sections have been designed to allow flow visualization studies of the oil flow in different flow orientations. Experimental pressure drop and liquid film thickness results obtained with air and 300-SUS alkybenzene oil will be presented. A variety of interfacial shear stress models obtained from the general two-phase flow literature have been applied to predict liquid film thickness. Comparisons between measured and predicted film thickness indicate that the measured film thickness is thinner, as expected, from a smooth surface approximation. Predictions of amount of oil holdup in large system pipes will also be presented.

NOMENCLATURE

D	Tube diameter (m)
f	Friction factor
f_i	Interfacial friction factor
f_s	Smooth wall friction factor
h	Mean film thickness (m)
h_L	Liquid film thickness (m)
h_L^+	Dimensionless liquid film thickness
r	Tube radius (m)
Re	Reynolds number
Re_{D_h}	Reynolds number on hydraulic diameter
Re_{LF}	Liquid film Reynolds number
Re_v	Vapor Reynolds number
u	Velocity (m/s)
u_i	Interfacial Velocity (m/s)
u_v	Vapor Velocity (m/s)
u_L^*	Liquid friction velocity (m/s)
u_s^*	Smooth wall friction velocity (m/s)
u_v^*	Vapor friction velocity (m/s)
y	Radial distance (m)
y^+	Dimensionless radial distance

Greeks

Γ_m	Mass flow rate of liquid film per unit width of wall surface (kg/(m·s))
ε	Wall-roughness height (m)
μ	Dynamic viscosity (Kg/(m·s))
ν	Kinematic viscosity (m^2/s)
ρ	Density (kg/m^3)
δ	Surface tension (N/m)
τ	Shear stress

Subscripts

i	Interfacial
L	Liquid
v	Vapor

INTRODUCTION

During the operation of a refrigeration system, the lubricating oil in the compressor invariably leaks through the seals and mixes with the refrigerant. Generally, the lubricant flows as a thin film along the walls of the pipe, driven by the turbulent flow of refrigerant vapor. Oils accumulated in the compressor discharge and suction lines must be circulated through the system by the vapor drag on the oil. If the vapor drag is insufficient, oil holdup increases. This can decrease the oil level in the compressor crankcase and lead to poor compressor lubrication and compressor failure.

Literature for Oil Return

Information in the literature for oil return in refrigeration systems is limited to a few rule-of-thumb correlations and experimental observations. Sharpe [1967] studied a glass P shaped trap between a horizontal and vertical length of suction line in a small refrigeration unit. For vapor velocities in the order of 3 m/s, oil transport by a slow moving oil film was observed.

Jacobs et al [1976] studied oil transport with refrigerant vapor by watching the liquid accumulating in the lower unit of the test section. Tests were conducted for combinations of R-12, R-22 and 150 (~32 mm^2/s) and 300 SUS (~66 mm^2/s) napthenic oil. A criterion for minimum transport of oil in vertical tubes was correlated with buoyancy forces (i.e., the density difference between the liquid and the vapor, and the momentum flux of the vapor) in terms of minimum tonnage of the refrigeration unit.

Hwang et al. [2000] investigated the flow characteristics of R134a and three kinds of oils (mineral ISO 10, alkylbenzene ISO 8, and alkylbenzene ISO 10) in the vertical upward suction line of the refrigeration system. Churn flow and annular flow were observed in their experiments. Mean oil film thickness was estimated from the amount of oil stored in the tube. They found that the oil with poor miscibility and high viscosity caused a larger oil volume stored in the tube. But with the increase of refrigerant mass flow rate, the influence of oil type and viscosity becomes less dominant.

Fukuta and Yanagisawa [2000] studied the flow characteristics of air/oil mixtures in vertical upward pipes (inner diameter 8 mm and 10 mm). Two different mineral oils, VG56 (260 mm^2/s) and VG20 (73 mm^2/s), were used. They found that the air velocity was dominant over the transition of the flow pattern. Average oil film thickness was obtained by measuring the capacitance between two thin electrodes. The oil film thickness was found to increase with the increase of the oil flow rate, however, the fractional thickness increase was smaller than that of flow rate.

Literature for Two-Phase Flow

In addition to a few investigations related to the oil return problem, there are a large number of publications focused on the adiabatic, two-phase flow, which are mostly concerned with air-water or water-steam mixtures. Currently, very little information is available in the literature describing the characteristics of a viscous liquid film driven by a turbulent vapor.

Friction Factor

Friction factor is a very important parameter to characterize the system pressure drop. For single-phase flow, Churchill [1977] developed a clever correlation that combines the friction factor for the laminar, transition and turbulent flow regimes in circular pipes as below:

$$f = 2\left[\left(\frac{8}{\mathrm{Re}}\right)^{12} + \frac{1}{(A+B)^{3/2}}\right]^{1/12} \tag{1}$$

where $A = \left\{2.457\ln\left[\left(\frac{7}{\mathrm{Re}}\right)^{0.9} + 0.27\left(\frac{\varepsilon}{D}\right)\right]\right\}^{16}$ and $B = \left(\frac{37530}{\mathrm{Re}}\right)^{16}$

For two-phase flow, the friction factor is typically defined as the ratio of the interfacial shear stress τ_i to the kinetic energy of the vapor phase.

$$f_i = \frac{2\tau_i}{\rho_v u_v^2} \tag{2}$$

The interfacial shear stresses between liquid and vapor phases are very complicated. There is no universal theory to predict this parameter. The liquid film behaves as a roughened boundary which causes an increase in the drag. For a fully rough regime, the friction factor would depend on the ratio of the mean film height h to the pipe diameter. Wallis [1969] suggested a correlation based on this assumption. He correlated his experimental annular flow data by using a friction factor of the form:

$$f_i = 0.005\left(1 + 300\frac{h}{D}\right) \quad (3)$$

For situations in which the film is thin enough that disturbance waves are not present, wave heights appear to be too small for the assumption of a completely roughened surface to hold. For these cases, Asali et al. [1985] suggested that when the flow is upward

$$\frac{f_i}{f_s} = 1 + 0.45\,\mathrm{Re}_v^{-0.2}\left(\mathrm{Re}_v\sqrt{\frac{f_i}{2}}\frac{h}{D-2h} - 4\right) \quad (4a)$$

and when the flow is downward

$$\frac{f_i}{f_s} = 1 + 0.45\,\mathrm{Re}_v^{-0.2}\left(\mathrm{Re}_v\sqrt{\frac{f_i}{2}}\frac{h}{D-2h} - 5.9\right) \quad (4b)$$

Fukano et al [1998] investigated the effects of liquid viscosity on the mean film thickness and interfacial shear stresses. Air and four different water and glycerol solutions were used as the test fluids. The viscosities of liquid solutions range from 0.85×10^{-6} to 8.5×10^{-6} m^2/s. They proposed a correlation which accounted for the effects of the liquid viscosity by trial and error as below:

$$f_i = 0.425\left(12 + \nu_L/\nu_W\right)^{-1.33}\left(1 + 12\frac{h}{D}\right)^8 \quad (5)$$

where υ_W is the kinematic viscosity of water at 20°C.

Film Structure

One of the most important parameters for the liquid layer structure is the mean film thickness. This is very clearly demonstrated by the series of friction factor correlations developed from the Wallis correlation as above. Beginning in the early 1960's, investigators began developing methods for measuring liquid film thickness. Ambrosini [1991] recommended the following correlation proposed by Asali et al [1985]:

$$h_L^+ = \left[\left(0.34\,\mathrm{Re}_{LF}^{0.6}\right)^{2.5} + \left(0.037\,\mathrm{Re}_{LF}^{0.9}\right)^{2.5}\right]^{0.4} \quad (7)$$

for $\mathrm{Re}_{LF} < 1000$, and the correlation proposed by Kosky [1971]

$$h_L^+ = 0.0512\,\mathrm{Re}_{LF}^{0.875} \quad (8)$$

for $\mathrm{Re}_{LF} > 1000$. The dimensionless film thickness h_L^+ and the liquid film Reynolds number Re_{LF} are defined as:

$$h_L^+ = \frac{h_L u_L^*}{\nu_L},\ u_L^* = \sqrt{\frac{\tau_i}{\rho_L}} \quad (9)$$

$$\mathrm{Re}_{LF} = \frac{4\Gamma_m}{\mu_L} \quad (10)$$

Shearer and Nedderman [1965] investigated the equivalent sand-roughness of the liquid film with liquid viscosities between 1.15 and 12.4×10^{-3} kg/(m·s) in the small ripple regime and proposed the following equation:

$$h_L^+ = K_1 D^2\sigma + K_2\frac{u_v - u_i}{\mu_L}\sqrt{\rho_v}\frac{\varepsilon}{D} \quad (11)$$

where $K_1 = 5.98\times10^5\ \frac{\sec^2}{kg\cdot m^2}$ and $K_2 = 0.1196\left(\frac{kg}{m}\right)^{0.5}$ are empirical constants.

A better understanding of the mechanism of oil circulation by the refrigerant vapor would permit more reliable design of the refrigeration system. A basic study of a viscous liquid film in contact with a faster moving gas is described in the following sections. This investigation extends the basic knowledge of two-phase annular flow with high viscosity liquids.

EXPERIMENTAL APPARATUS

Figure 1 shows a schematic of the adiabatic, near-ambient pressure, air/oil flow visualization loop for this study. A liquid pump is used to circulate the oil. A blower is used to circulate air through the test section. The air and oil mix together before entering the test section. The two-phase mixture from the test section is separated and returns to the oil reservoir and air blower, respectively. Oil mass flow rate is measured by a mass flow meter. Air mass flow is determined by measuring pressure drop through a straight tube section that is before the injection of the oil. The air/oil mass flow rate can be controlled by adjusting by-pass valves and shut-off valves on the air/oil lines. Three different orientations: horizontal, vertical up, vertical down flow in the loop can be investigated.

Flat plate test sections have been made from clear PVC sheets. The thickness of the sheets is 0.635 cm. The length of the test sections including the inlet and outlet ports is 50.8 cm and the width is 12.7 cm. Three smooth test sections with plate spacing of 1.27 cm, 0.64 cm, and 0.32 cm were built. Two pressure taps are attached at the centerline of the test section with a distance of 35.6 cm to measure the pressure drop.

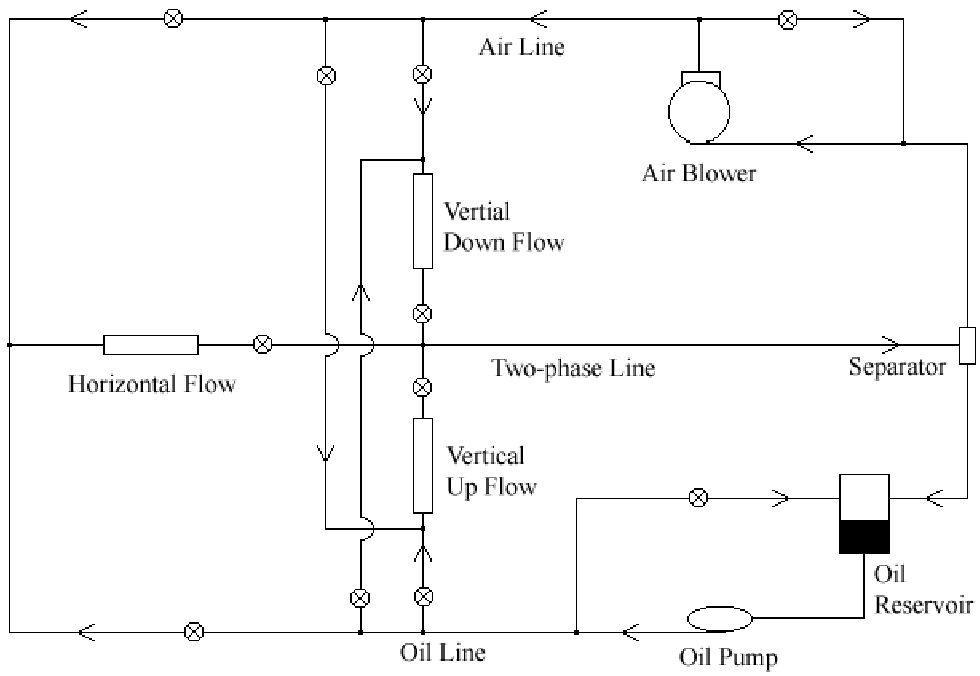

Figure 1: Schematics of Flat Plate Experimental Apparatus

RESULTS AND DISCUSSION

Tests were conducted with air and alkylbenzene oils as working fluids at ambient pressure and temperature. Alkylbenzene oil is a common refrigeration and air conditioning compressor lubricant. The oils used here have a density of 875 kg/m^3 at 20°C and a kinematic viscosity of 66×10^{-6} m^2/sec for 300 SUS oils at 40°C, which is almost same as the viscosity of the VG20 mineral oils used by Fukuta and Yanagisawa [2000] and more than ten times of the maximum viscosity investigated by Fukano et al [1998].

Friction Factor

Pressure drop data were first collected in three smooth test sections with dry air over a range of flow conditions. Figure 2 compares the experimental single-phase friction factors to Churchill's correlation. As shown in Figure 3, all the experimental data are within ±15% difference to Churchill's correlation. The air Reynolds numbers are calculated with the following equation: $\mathrm{Re}_{D_h} = \dfrac{4Q}{P_W \upsilon_{air}}$, where Q is the air volume flow rate and P_W is the wetted perimeter. Figure 4 shows the experimental air/oil flow friction factors and Wallis's, Asali's, Fukano's and Shearer's friction factors to the dimensionless mean liquid film thickness, *h*/D. The predictions by Asali's and Shearer's correlations are closest to our experimental friction factors. Other correlation predictions are much larger than the experimental results. There is one special note behind this chart. Wallis's and Fukano's friction factor correlations all directly depend on h/D. As we have not collected the film thickness data correspondent to the friction factor data in the chart, Asali's model was used to predict the value of h/D. Figure 5 compares the difference between our experimental friction factors and Asali's predictions. Even the difference between these two is very clear.

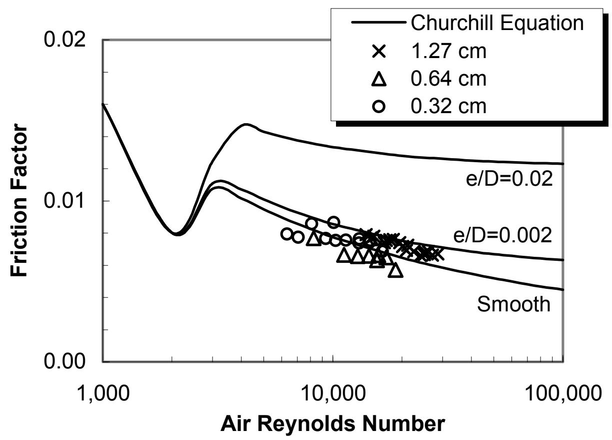

Figure 2. Friction Factor Vs. Air Reynolds Number (Air Flow)

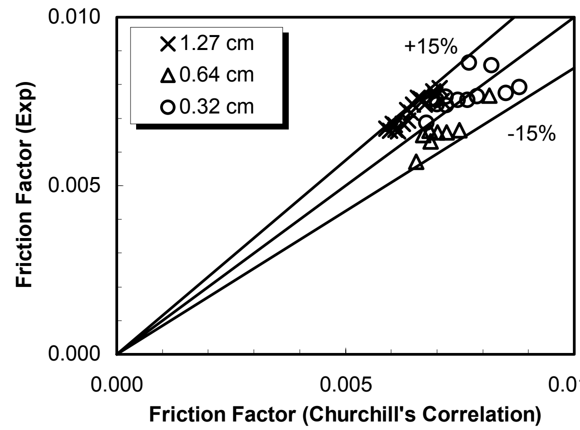

Figure 3. Experimental Vs. Churchill's Friction Factors (Air Flow)

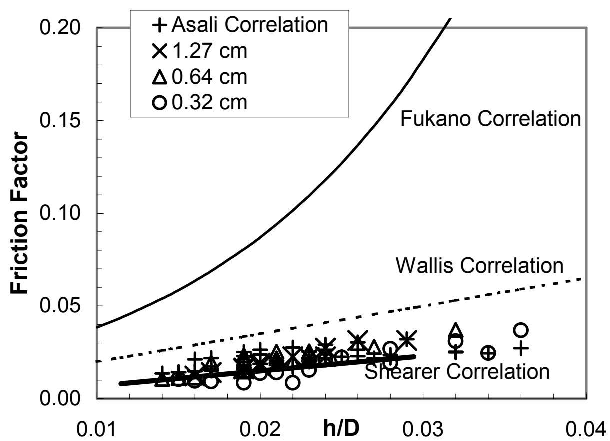

Figure 4. Friction Factor Vs. Dimensionless Mean Film Thickness h/D (Air/Oil Flow)

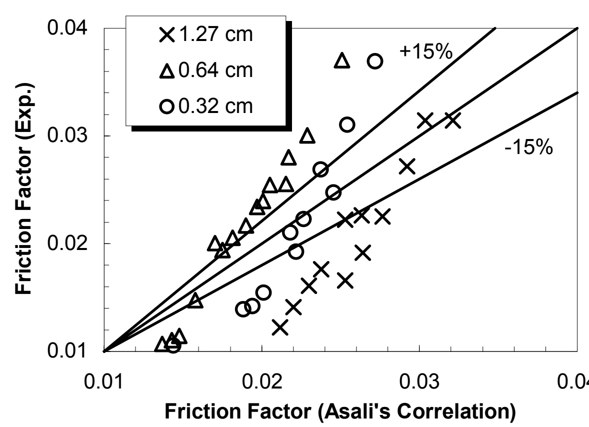

Figure 5. Experimental Vs. Asali's Friction Factors (Air/Oil Flow)

Liquid Film Thickness

Local liquid film thicknesses along smooth plates were obtained using an optical liquid film thickness measurement method developed by Shedd and Newell [1998] in the 0.64 cm vertical-up test section. Meanwhile, modeling of the liquid film thickness in smooth plates is developed to compare with the average film thicknesses obtained by experiments. In our experiment, the vapor core is driven by the pressure gradient in the plates. The liquid film is driven not primarily by the pressure gradient, but by momentum transferred from the vapor. This momentum exchange is very large resulting in high shear stress in the liquid film. Based on the momentum exchange between liquid and vapor, modeling of the liquid distribution in smooth plates is under development.

Laminar Liquid Layer and Turbulent Vapor Core Model

In this model, we modeled the high viscous liquid layer as laminar flow. The liquid layer has linear velocity profile. The vapor core is turbulent flow and can be modeled using von Karman's form of the Law of the Wall velocity profile [1939]. The following are the description equations for the model:

Laminar Liquid Layer:

$$u_L^+ = y^+ \tag{12}$$

$$\frac{u_{L,avg}}{u_L^*} = 0.5h_L^+ \tag{13}$$

where $u_L^+ = \frac{u_L}{u_L^*}$, $y^+ = \frac{yu_L^*}{\nu_L}$, $u_L^* = \sqrt{\frac{\tau_i}{\rho_L}}$, τ_i is the interfacial shear stress.

Turbulent Vapor Core

Viscous Sublayer: $$u_V^+ = y^+ \qquad y^+ < 5 \tag{14}$$

Buffer Region: $$u_V^+ = 5\ln(y^+) - 3.05 \qquad 5 < y^+ < 30 \tag{15}$$

Log Region: $$u_V^+ = \frac{1}{\kappa}\ln(y^+) + B \qquad 30 < y^+ < h_V^+ \tag{16}$$

where $u_V^+ = \frac{u_V - u_i}{u_V^*}$, $y^+ = \frac{yu_V^*}{\nu_V}$, $u_V^* = \sqrt{\frac{\tau_i}{\rho_V}}$, B=5.5, $\kappa = 0.4$.

Shear Stress Model

In this model, both the liquid layer and the vapor core are modeled using von Karman's form of the Law of the Wall velocity (Equations 14~16). The liquid layer velocity profile is decided by the value of h_L^+. The interfacial shear stress is predicted using a correlation developed by Asali (Equation 4a). Integrating the vapor velocity profile from 0 to r-h_L gives the following equation for the average vapor velocity:

$$\frac{u_{V,avg} - u_i}{u_s^*} = \frac{1}{\kappa}\ln\left[\frac{(r-h_L)u_s^*}{\nu_V}\right] + B - \frac{3}{2\kappa} \tag{17}$$

where $u_s^* = \sqrt{\frac{\tau_s}{\rho_g}}$, τ_s is the smooth wall shear stress.

The equation set can be solved simultaneously at a specified condition to predict the liquid film thickness. Figure 6 compares the liquid film thickness predicted by the Shearer correlation, the laminar and the Asali shear stress models, and the experiment liquid film thickness data in the 0.64cm spacing vertical up smooth plate when the air mass flow rate $\dot{m}_{air} = 15g/s$, oil mass flow rate $\dot{m}_{oil} = 1.1 \sim 13.4g/s$, temperature T=40°C. Although Shearer's correlation can predict the fiction factor quite well, its liquid film thickness predictions are about 3 times larger than that obtained experimentally. An important hint from the two model predictions is that h_L^+ values are always less than 5 in the whole range of oil mass flow rate, which means the base liquid layer should be very close to laminar flow. The shear stress model based on Asali's interfacial shear stress correlation is closest to our experimental data. But the shear stress model also overpredicts the liquid film thickness. From the flow patterns observed in our experiments, there is a wavy layer on the top of the base viscous layer. Due to the nature of the measurement, the experimental results exclude the large liquid waves from the calculated average thickness, so the thicknesses reported reflect only the base film that exists between the large, liquid waves. This is one of the reasons that both models over-predicted the liquid film thickness compared to the experimental data. The shear stress model predictions are better than those are from the laminar model. This is because the shear stress model can reflect the momentum exchange between the vapor and liquid better compared to the laminar model. Asali's correlation was developed for situations in which the film is thin enough that disturbance waves are not present. But very little of his data occurs in the low h_L^+ region. It seems that Asali's correlation still under-predicts the momentum exchange caused by those wave structures.

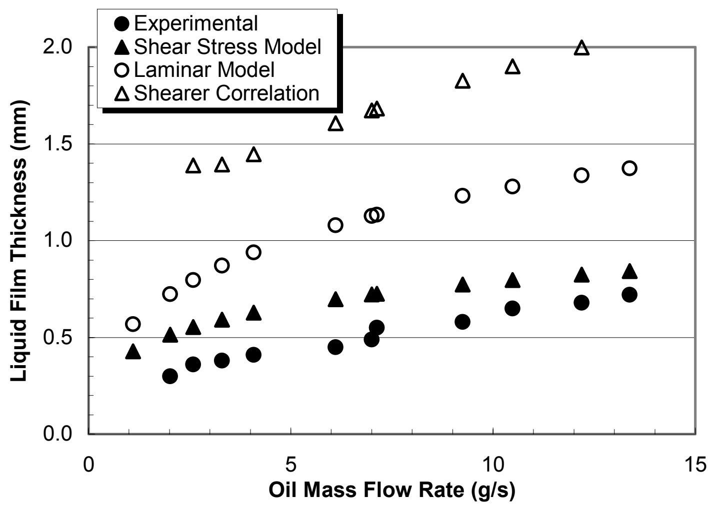

Figure 6. Comparison of Oil Film Thickness between Different Models and Experimental Data

Prediction of Oil Holdup

With the shear stress model developed above, we can predict the oil film thickness along the pipes of a refrigeration system. Also the amount of oil holdup can be estimated. The following is an example of the application. A 50~100 tons refrigeration system for the ice rink at the University of Illinois uses R22 as the working fluid and operates between 0.4 and 1.0 MPa. Alkylbenzene SUS 300 oil is used as the lubricating oil. Figures 7 shows the prediction of oil film thickness along the pipes in different conditions by the shear stress model. The high temperature curves are for the compressor discharge line and the low temperature curves are for the suction line. Figure 8 is the prediction of the oil accumulation along the compressor discharge line when the oil concentration is 0.5%.

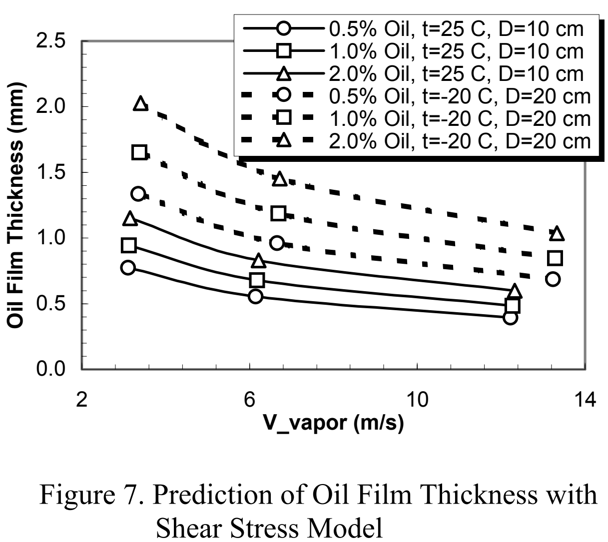

Figure 7. Prediction of Oil Film Thickness with Shear Stress Model

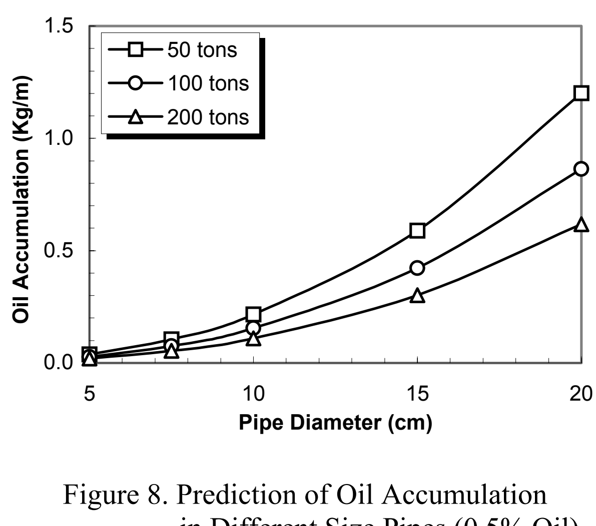

Figure 8. Prediction of Oil Accumulation in Different Size Pipes (0.5% Oil)

CONCLUSIONS

Initial experimental and analytical work has been carried out to investigate the characteristics of two-phase, viscous liquid flows. The results from current study indicate that the shear stress plays a key role in determining the pressure drop and liquid film thickness. The measured film thickness is thinner, as expected, from a smooth surface approximation. Shear stress correlation based on air-water data also overpredicts the liquid film thickness. Further experimentation is needed for better understanding of the mechanism of a viscous liquid film driven by a faster moving gas.

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