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R. T. Drost
United Technologies Research Center

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OUTGASSING AND ABSORPTION RATES OF OIL/REFRIGERANT MIXTURES

Ronald T. Drost
Associate Research Engineer
United Technologies Research Center
411 Silver Lane, East Hartford, Ct 06108
(203) 727-7764
FAX (203) 727-7656

ABSTRACT

Knowledge of the solubility characteristics of an oil/refrigerant mixture is important when analyzing the lubrication system behavior in refrigerant compressors since the solubility significantly affects the viscosity of the mixture. For an operating compressor, the lubricant is a mixture of oil and refrigerant and the viscosity of the mixture is a strong function of the refrigerant concentration and of the mixture temperature. Solubility information is available for various oil/refrigerant mixtures that gives mixture viscosity as a function of temperature and pressure and can be used adequately under steady-state operating conditions. Accuracy is compromised if this information is used to analyze transient events associated with sudden changes in temperature or pressure such as may occur throughout the lubrication system. For this reason, an analytical model was developed to predict the transient outgassing/absorption rates of an oil/refrigerant mixture experiencing a sudden change in temperature or pressure. The solution method was to develop an oil/refrigerant mixture correlation by employing the conservation laws of mass and energy in a transient system.

A test rig was built to accurately measure the characteristics of an oil/refrigerant mixture experiencing a change in system temperature or pressure so that a comparison could be made between predicted and experimental results. Fast response thermocouples and pressure transducers were used to capture the system behavior during both outgassing and absorption events. A summary of the analytical model and experimental verification activities is presented.

NOMENCLATURE

a, b constants in expression for specific heat of oil
h specific enthalpy or heat transfer coefficient
m mass flow
p time
u specific internal energy
v velocity
z height

A area
Cp specific heat
H enthalpy
M mass
P pressure
Q heat flow
T temperature
U internal energy
V volume
\( \gamma \) specific volume
W work

subscripts:
cv control volume
e exit
f final
i inlet or initial
L liquid refrig.
O oil
V vapor

Greek:
\( \beta \) time constant

INTRODUCTION

The lubricant viscosity is a strong factor in the performance of the various lubrication system elements in refrigerant compressors. For an operating compressor, the lubricant is a mixture of oil and dissolved refrigerant that changes composition due to variations in operating pressure and temperature that occur inside the compressor and within the lubrication system elements themselves. Since these outgassing and absorption events can dramatically change the lubricant viscosity, accurate predictions of the instantaneous mixture composition are essential for efficient and reliable operation of the lubrication system elements.
At equilibrium conditions, the refrigerant concentration will depend on the local temperature and pressure. In general, the concentration increases with higher pressure and lower temperature. For a compressor operating at steady-state conditions, the mixture concentration approaches a constant value over time. This saturation concentration can accurately be determined using solubility information available for most oil/refrigerant mixtures. When the operating conditions are changed, however, outgassing or absorption of refrigerant will take place as the mixture tends towards the new saturation concentration. If the system pressure is increased, more refrigerant will become dissolved in the mixture; if pressure is decreased, the new saturation concentration is lower and the mixture becomes supersaturated until the excess refrigerant is outgassed from the solution. This outgassing process manifests itself in intense bubbling and foaming which results in a lubricant mixture of two Newtonian fluids that does not exhibit Newtonian behavior. This bubbly mixture can result in potentially dangerous operating conditions for the lubrication system elements. Outgassing does not readily occur until nucleation takes place. Nucleation is provided by agitation of the lubricant mixture which may be triggered by sharp corners, fluid friction, or within the oil pump. Once initiated, the outgassing process can last for several minutes or even hours and cannot be analyzed using steady-state solubility information.

An analytical model was developed to accurately predict the transient outgassing and absorption rates of an oil/refrigerant mixture experiencing a sudden change in temperature or pressure. A test rig was also built to monitor these processes and verify the predicted results. This paper summarizes the development of the analytical model and experimental test rig and compares predicted and measured test results.

**ANALYTICAL MODEL**

An analytical model employing the conservation laws of mass and energy in a transient system was developed to accurately predict the outgassing and absorption rates of an oil/refrigerant mixture experiencing a sudden change in pressure or temperature. The lubricant was assumed to be a single phase (fully-miscible liquid) oil/refrigerant mixture at uniform pressure (no stratification effects) and a uniform state, uniform flow process was assumed for the transient events. The system was modeled as two arbitrarily deformable control volumes, one each for the vapor and liquid volumes as shown in Fig. 1. Mass is allowed to enter and exit the vapor control volume while heat transfer can occur across both control volumes. Both mass and heat transfer can occur across the vapor/liquid interface. For an imposed change in vapor pressure, the solution of the system equations yields the rate of mass transfer at the interface along with vapor and liquid temperatures so that an instantaneous mixture viscosity can be obtained. The system is initially assumed to be at steady-state.

**System Equations**

The general form of the continuity equation for a control volume can be expressed as

\[ \dot{M}_{CV} = \sum \dot{m}_{in} - \sum \dot{m}_{out} \]  

(1)

For the vapor and liquid (oil + liquid refrigerant) control volumes (CVV and CVL respectively), the equation becomes

\[ \dot{M}_v = \dot{M}_{refrig} + (\dot{m}_i - \dot{m}_e) \]

\[ \dot{M}_L = - \dot{M}_{refrig} \]  

(2)

Conservation of mass for the entire system takes the form

\[ M = M_v + M_L + M_o \]

\[ M_L = \frac{(V - M_o \Phi_v - M_v \Phi_v + M_e \Phi_v)}{(\Phi_L - \Phi_v)} \]  

(3)

The first law for a uniform state, uniform flow process can be written as

\[ \dot{Q}_{CV} - \dot{W}_{CV} + \sum \dot{m}_i \left[ h_i + \frac{v_i^2}{2} + gz_i \right] - \sum \dot{m}_e \left[ h_e + \frac{v_e^2}{2} + gz_e \right] = \frac{d}{dt} \left[ m \left( u + \frac{v^2}{2} + gz \right) \right]_{CV} \]  

(4)
The following assumptions are made to simplify the solution:

1. $v_i = v_e = 0$ — no flow losses across CV's
2. $z_i = z_e = 0$ — no gravitational effect
3. $(dW/dt)_{cv} = 0$ — no work at system boundary
4. $dU_{cv} = (d(H-PV))_{cv}$ — internal energy relation
5. $(dV/dt)_{cv} = -(dV/dt)_{cvL}$ — deformable CV's
6. $H_e = H_v$ — enthalpy of exit flow
7. $H_{cvL} = H_o + H_L + H^{Ex}$ — liquid mixture enthalpy

The excess heat of mixing ($H^{Ex}$) in the liquid mixture enthalpy equation is assumed to be zero since contributions of less than 2% were estimated from this term for the expected temperature range. Chemically, this corresponds to near-ideal solution behavior.

The time derivative of internal energy for a control volume is given as:

$$\frac{d(H-PV)}{dt}_{cv} = \frac{dH}{dt} - \frac{d(PV)}{dt} = \frac{d(Mh)}{dt} - P \frac{dV}{dt} - V \frac{dP}{dt}$$  \hspace{1cm} (5)

where

$$h_v = h_v(T_v, V_v) \Rightarrow \dot{h}_v = \left( \frac{\partial h_v}{\partial T_v} \right)_V \dot{T}_v + \left( \frac{\partial h_v}{\partial V_v} \right)_T \dot{V}_v$$  \hspace{1cm} (6)

$$h_L = h_L(T_L) \Rightarrow \dot{h}_L = \frac{dh_L}{dT_L} \dot{T}_L = \frac{dh_L}{dT_L} \dot{T}_L$$  \hspace{1cm} (7)

$$h_o = C_v T_L = (a + b T_L) T_L \Rightarrow \dot{h}_o = (a + b T_L) \dot{T}_L + b T_L \dot{T}_{T_L}$$  \hspace{1cm} (8)

The partial derivatives in eqn. (6) are derived from the enthalpy of vapor equation for the particular refrigerant being used. The expression for the rate of change of vapor specific volume is found as follows:

$$P = P(T_v, V_v) \Rightarrow \dot{P} = \frac{dP}{dt} = \left( \frac{\partial P}{\partial T_v} \right)_V \dot{T}_v + \left( \frac{\partial P}{\partial V_v} \right)_T \dot{V}_v$$  \hspace{1cm} (9)

therefore,

$$\dot{Q}_v = \frac{\partial P}{\partial V_v} \dot{V}_v$$  \hspace{1cm} (10)

The partial derivatives in eqn. (10) are derived from the refrigerant vapor equation of state. Heat transfer terms are given by

$$\dot{Q}_{cv} = h(T - T_{oo})$$  \hspace{1cm} (11)

The simplified energy equation for the vapor control volume is then given by

$$\dot{Q}_{cvV} + h_i \dot{m}_i - h_e \dot{m}_e - \dot{M}_v \dot{h}_v - \dot{M}_s \dot{h}_v + P \dot{V}_v + V \dot{P} = 0$$  \hspace{1cm} (12)

and for the liquid control volume

$$\dot{Q}_{cvL} - M_o \dot{h}_o - \dot{M}_L \dot{h}_L - \dot{M}_L \dot{h}_L + P \dot{V}_L + V \dot{P} = 0$$  \hspace{1cm} (13)
Substituting for the various relations the vapor control volume energy equation takes the final form

\[
h A V (T_{L1} - T_{\infty}) + h_{V} A V_{L} (T_{V1} - T_{L1}) + (h_{i} - h_{v}) m_{i} - \dot{M}_{\text{refrig}} h_{v1} + P_{v} \dot{V}_{v} + V_{v} \dot{P} - M_{v1} \left( \frac{\partial h_{v}}{\partial T_{v}} \right)_{T} \hat{T}_{v} + \left( \frac{\partial h_{v}}{\partial V_{v}} \right)_{T} \left( \frac{\partial P}{\partial V_{v}} \right)_{T} \hat{V}_{v} = 0 \quad \text{where,} \quad \hat{T}_{v} = \frac{(T_{v2} - T_{v1})}{(t_{2} - t_{1})}
\]

and the liquid control volume energy equation becomes

\[
h_{L1}(T_{L1} - T_{\infty}) + h_{V2} A V_{L} (T_{L1} - T_{V1}) + \dot{M}_{\text{refrig}} h_{L1} - (M_{\text{C}} + 2M_{\text{p}} b T_{L1}) \hat{T}_{L} - M_{L1} \left( \frac{h_{L2} - h_{L1}}{T_{L2} - T_{L1}} \right) \hat{T}_{L} + P_{L} \hat{V}_{L} + V_{L} \hat{P} = 0 \quad \text{where,} \quad \hat{T}_{L} = \frac{(T_{L2} - T_{L1})}{(t_{2} - t_{1})}
\]

The solution can be pressure or temperature driven by using one of the following equations to describe the event

\[
P = P(t) = P_{f} + (P_{i} - P_{f}) e^{-\beta t}
\]

\[
T = T(t) = \frac{(T_{f} - T_{i})}{t_{f}} t + T_{i}
\]

**Solution Scheme**

The initial state of the mixture is assumed to be at steady-state so that the mixture solubility can be estimated from published information. A solubility subroutine was written that uses a cubic spline interpolation technique to extract the steady-state concentration data while the vapor and liquid refrigerant states are found using refrigerant property subroutines. The energy and continuity equations are solved using a Newton–Raphson–Secant iteration method to yield the system transient characteristics and rate of mass transfer at the vapor/liquid interface.

**EXPERIMENTAL APPARATUS**

A laboratory test rig was built to verify the predicted results. A schematic diagram of the outgassing/absorption experimental test apparatus is shown in Fig. 2. The rig was designed for precise real-time measurements of the system pressure, vapor and liquid temperatures, mass flow out of the vessel, and liquid oil level. The tests vessel consists of a cylindrical center shell set between a flanged lower sump and a containment plate on the top of the rig. A sight glass was located on the vessel to allow visual access of the vapor/liquid interface during the test. Fast-response, mini-hypodermic thermocouples were used to monitor the system temperatures and a high-speed miniature pressure transducer was used to record the system pressure characteristics. A Coriolis digital mass flow meter monitored the instantaneous mass flow into and out of the vessel. The liquid level history was recorded using an ultra-low range differential pressure transducer. The full scale transducer range of 0.2 psi allowed for accurate measurement of the fluid hydrostatic pressure and ultimately the fluid level. The initial state of the system was varied by changing the amount of oil and refrigerant in the system and also by changing the system temperature using a resistance heater located below the test rig. Once the system reached equilibrium, the test was initiated by opening the valve downstream of the digital mass flow meter. The secondary vessel was either pressurized or evacuated depending on the particular event being studied; in either case, the pressure in the main vessel was changed to initiate the outgassing or absorption event. The system pressure history was changed by varying the speed at which the valve was opened. High-speed data was simultaneously recorded using a multi-channel data acquisition system. Accompanying software was written to condition and plot the data.
RESULTS/CONCLUSIONS

Analytical and experimental results were obtained for a variety of outgassing and absorption events using HFC–134a and a compatible polyolester lubricant as the working fluids. Fig. 3 shows the predicted system characteristics for one test case; the corresponding experimental outgassing results are shown in Fig. 4. The measured system pressure was matched using eqn. (16) and used to drive the analytical solution. The analytical and experimental trends are in close agreement; however, a noticeable delay in the thermal response of the system can be seen. The analytical model assumes the behavior of the system is the same at every location in the system (no stratification effects) and does not account for the presence of the vessel walls. These effects are believed to significantly affect the behavior of the system. To investigate this, additional outgassing/absorption tests are planned during which the oil will be agitated in an attempt to match more closely the conditions that govern the analytical model. Figs. 5 and 6 compare the predicted and measured change in liquid level. Close agreement between these curves would ensure accurate predictions in the mixture composition since the liquid level history is directly related to the amount of refrigerant entrained in the oil. The effect of the continuum assumption is again seen in that the model predicts a faster outgassing rate than what was measured during the first few seconds of the event. The initial rise in the measured liquid level is believed to be caused by the intense bubbling at the liquid/vapor interface when the system pressure was rapidly decreasing. Fig. 7 compares predicted liquid mixture compositions from both steady-state and transient analyses for the same outgassing event. The steady-state data were generated from tabulated solubility data using the measured system pressure and liquid temperature. As can be seen, conservative steady-state predictions of quasi-static solution behavior at every point in time do not reflect the actual behavior of the system. If fact, a steady-state analysis of the event would predict that the refrigerant is completely outgassed after approximately five seconds; in reality, the system equilibrium occurs several minutes after the initial event when the transient model converges to the steady-state solution. The predicted mixture viscosities for both analyses are plotted in Fig. 8 along with the viscosity results if the mixture was assumed to be pure oil. These results clearly illustrate the importance of this investigation since accurate predictions of the lubricant viscosity is critical to the efficient and reliable operation of the lubrication system of refrigerant compressors. Steady-state solubility predictions are shown to be inadequate for analyzing most of the transient events that occur within the compressor and lubrication system elements.

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Fig. 1 Arbitrarily deformable vapor and liquid control volumes.
Fig. 2 Outgassing/absorption experimental test apparatus.

Fig. 3 Predicted system characteristics.

Fig. 4 Measured system characteristics.
Fig. 5 Predicted change in liquid level.

Fig. 6 Measured change in liquid level.

Fig. 7 Predicted mixture concentration.

Fig. 8 Predicted mixture kinematic viscosity.