

TAKING INTO ACCOUNT OF OIL-REFRIGERANT MIXTURES INFLUENCE DURING EXPERIMENTAL TESTING OF REFRIGERATION COMPRESSORS.

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ABSTRACT. *This paper presents an investigation on the performance of a reciprocating hermetic refrigeration compressor, using a superheated vapor phase cycle. As a result from experiments, the uncertainty of enthalpy calculation, mainly at compressor exhaust, due to the high oil solubility at high pressures, indicates that big errors in energy balance, leads to big differences in refrigerant mass flow rate when compared with catalogs information. Using an equation of state for these mixtures and thermodynamic relationships, as well as taking into consideration low oil partial pressure it was possible to calculate the effects of enthalpy on mass flow rate calculations, considering mixture effect. Results obtained allow, comparison with experimental results, and shows that these effects should be taken into account in refrigeration testing methodologies proposed in literature.*

Key-words: *Compressor, Lubricant, Refrigerant, Oil, Mixture.*

1.0 INTRODUCTION

Normally, refrigerating compressors are tested using conventional cycles where both condensation and evaporation occurs (ASHRAE, 1993 and ISO 1989) and mass flow rate may be determined by direct (flowmeters) or indirect (energy balances) methods. When applying an energy balance for the evaporator for example, we must consider that a considerable oil separation occurs because at low pressure, the lubricant solubility in the refrigerant is low. The composition of both vapor and liquid phases (when not pure) influences the equilibrium pressure. With refrigeration lubricants, the type, boiling range, and viscosity are also factors influencing vapor pressure; (naphthenic oils of a specific viscosity grade generally show higher vapor pressures than paraffinic oils). Above the critical solution temperature (CST), many refrigerant/lubricant mixtures in this class are completely miscible, and their behavior is identical to that just described. Below the critical solution temperature, however, the liquid in the mixture may separate into two phases.

Usually, the lubricant effects are not taken into account in the thermal balances. Besides, no matter how little lubricant the entering refrigerant carries, the liquid phase, as it progresses through the evaporator, passes through the critical composition which usually lies between 15% and 20% lubricant in the total liquid phase.

The available standards imposes that the mass oil flow must always lies below 3% of the total flow, in order to allow neglecting its effects.

Due to common accuracy problems (Pimenta, 1997), such as thermal inertia effect on thermal properties, inaccuracy in determination of both evaporation pressures and superheating degree, etc, some studies were started, aiming to decrease such problems using a

totally vapor phase cycle (Dirlea *et al.*, 1997, Duarte *et al.*, 1998). In this work, a state equation for mixtures proposed by Yokozeki (1994) based on the Soave-Redlich-Kwong (SRK) state equation, is considered with determination of the critical properties of lubricant by a methodology developed by the authors, together with the supposition accepted by Yokozeki and other authors, that considerate the lubricant partial pressure as negligible.

Lubricant/refrigerant (naphthenic oil/R22) mixture enthalpy may be calculated, which shows in quality and quantity that the effects of the mixture should be taken into account in such analysis.

2.0 STATE EQUATION AND ITS CONSTANTS

The SRK state equation such as described by Yokozeki (1994) is used in this paper in the following form:

$$p = \frac{RT}{v-b} - \frac{a}{v(v+b)} \quad (1)$$

where, a and b are known constants (Van Wylen, 1973) for real gas mixtures, given by:

$$a = (x_1 a_1^{0.5} + x_2 a_2^{0.5})^2 \quad (2)$$

$$b = x_1 b_1 + x_2 b_2 \quad (3)$$

considering that the constants values a_1 , a_2 , b_1 and b_2 are resulting from application of mathematic condition for critical point, where:

$$\left(\frac{\partial p}{\partial v} \right)_{T_{ci}} = 0 \quad (4)$$

$$\left(\frac{\partial^2 p}{\partial v^2} \right)_{T_{ci}} = 0 \quad (5)$$

and also considering that each component in the lubricant/refrigerant mixture behaves as an isolated components, that obey Eqs. (4) and (5), leads to Eqs. (6) and (7) which give four constants (two for each component) to be used in the Eqs. (2) and (3):

$$a_i = \frac{R_i T_{ci} v_{ci}^2 (v_{ci} + b_i)^2}{(2v_{ci} + b_i)(v_{ci} - b_i)^2} \quad (6)$$

$$3v_{ci} b_i (v_{ci} + b_i) = v_{ci}^3 - b_i^3 \quad (7)$$

3.0 CRITICAL CONSTANTS DETERMINATION

The R22 critical constants were obtained from the pure property tables for R22 ($T_c=368,9$ K; $P_c=0,0497$ Pa). Since the vapor pressure of the lubricant is negligible compared with refrigerant, the vapor phase is essentially pure refrigerant, and only the composition of liquid phase needs to be considered. It was needed an approximated methodology based on the Clapeyron equation in order to calculate the lubricant critical constants. According to some

authors (Yokozecky, 1994 and Parise *et al.*,1998), a rough estimation for T_c and P_c of the lubricant may be sufficient, as the critical temperature of oil is very high than any refrigerant and its vapor pressure is essentially zero under the present case. So, from the Clapeyron equation:

$$h_{fg_i} \cong (v_{g_i} - v_{l_i}) T_{sat} \left(\frac{dp}{dT} \right)_{sat} \quad (8)$$

where,

h_{fg_i}	oil vaporization latent heat	[J/kg]
v_{g_i}	oil saturated vapor specific volume	[m ³ /kg]
T_{sat}	saturation temperature	[K]

$$p_{sat_i} = 3513 \cdot 10^{-20} T_{sat}^{8.526} \quad (9)$$

$$v_{g_i} = Z_i R_i \frac{T_{sat}}{P_{sat}} \quad (10)$$

where v_{l_i} represents the specific volume of the liquid lubricant at 20°C ($v_{l_i}=0,001182 \text{ m}^3/\text{kg}$), taken as approximately constant until the critical point. The calculation procedure using Eq. (8) was done until reaching $h_{fg_i} \cong 0$. A previous estimation of the critical constants values T_{ci} and P_{ci} , was made aiming to reach the initial Z_i value. After some interactions, $T_{ci}=818 \text{ [K]}$ and $P_{ci}=75,2 \cdot 10^5 \text{ [Pa]}$ could be obtained for the lubricant oil.

4.0 MIXTURE ENTHALPY CALCULATION

From enthalpy definition ($h = u + pv$), Maxwell correlations, and from the Eq. (1) the following expression can be obtained for the mixture enthalpy change, between two states:

$$\Delta h \cong C_v(t)(T - T_0) + \left[\frac{a}{bM_{mix}} \ln \left(\frac{v+b}{v} \right) \right]_{v_0}^v + pv - p_0v_0 \quad (11)$$

where:

$$M_{mix} = X_1M_1 + X_2M_2$$

M_{mix}	molecular mass of oil/refrigerant mixture	[kg/kmol]
$C_v(t)$	specific heat at constant volume of pure R22	[J/kg-K]
M_1 e M_2	molecular mass of oil and refrigerant, respectively.	[kg/kmol]
X_1 e X_2	volume composition of oil and refrigerant, respectively.	

5.0 MIXTURE COMPOSITION

The total vapor cycle for the refrigeration compressor testing presented here, was performed in a test bench as shown in the Fig. 1. Measurements of pressure (P) and temperature (T) were made in the indicated points, as well as the water mass flow and the electric power consumed by the compressor. The refrigerant mass flow in the cycle is indirectly determined by means of thermal balance in the compressor and in the desuperheater, as well as directly using venturi flowmeters (Duarte, 1998a, Oliveira, *et al.*,

1998). Temperature was measured with type T thermocouples which have uncertainty around $0.5\text{ }^{\circ}\text{C}$. Pressure measurements were made by means of transducers with uncertainty around 0.25% F.S. and the compressor power consumption was obtained using an electric power meter device.

The evaporation and condensation pressures presented here, are called (P_{ev}) and (P_{cd}) respectively. Analogously, (T_{ev}) and (T_{cd}) are their respective evaporation and condensation temperatures. The Pressure-Enthalpy graphic, on the other hand, reproduces the complete cycle shown in the first schematic figure. The total amount of refrigerant contained in the system, can be controlled by operating two valves assembled on its refrigerant tank device.

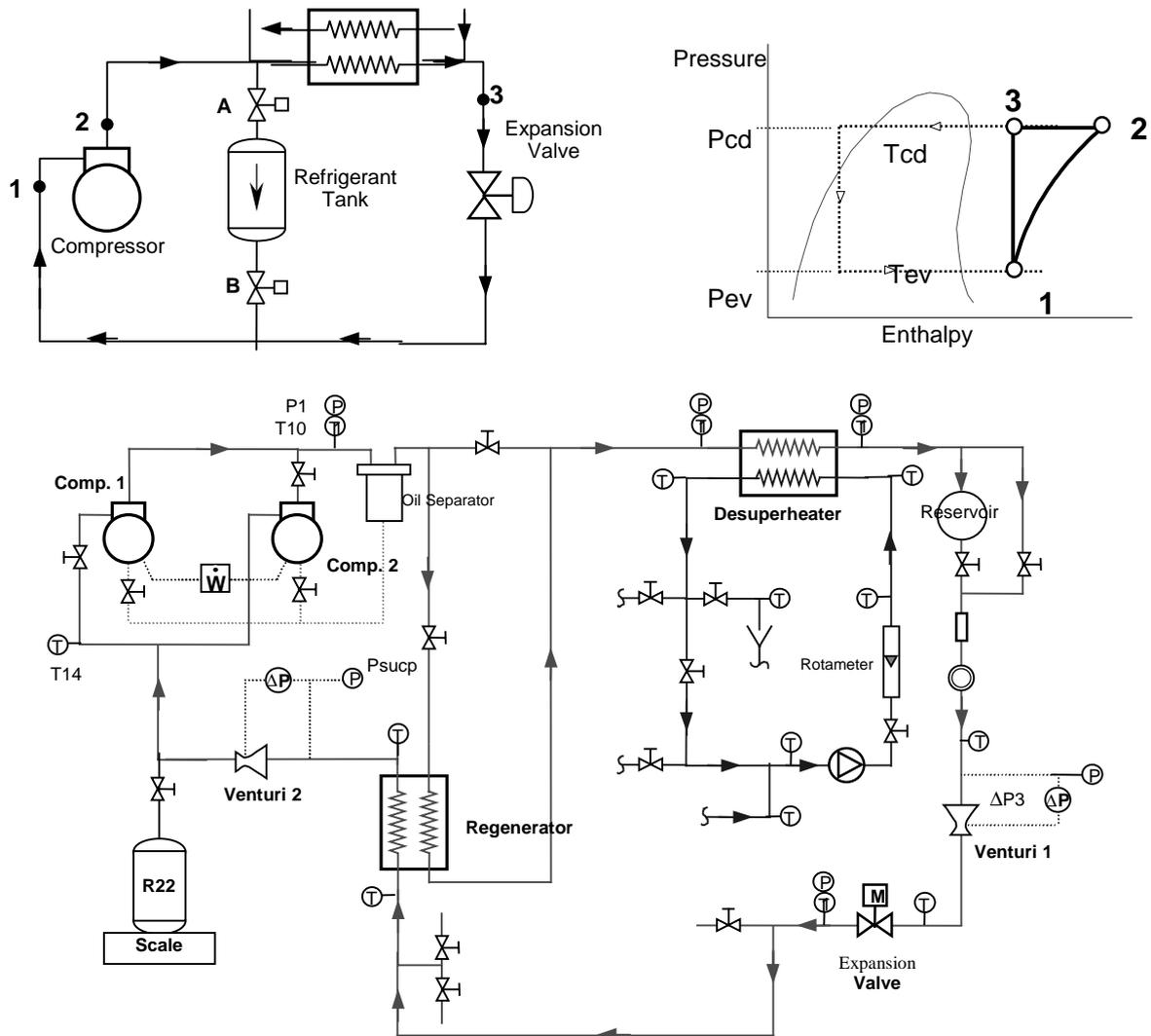


Figure 1 – Schematic view of the experimental test bench.

The pressure and temperature experimental data at compressor supply (P_{sucp} , T_{14}) and exhaust (P_1 , T_{10}), were used to determinate the mixture composition, and Eq. (1) was applied to the mixture. The composition of the mixture is determined considering that pure lubricant in vapor phase obeys Eq. (1) having a null pressure at the mixture temperature. In this terms, the oil volumetric percentage (X_{v1}) in the mixture can be obtained by making:

$$X_{v1} = \frac{v_1}{v_m} \frac{M_{mis}}{M_1} \quad (12)$$

where,

v_l	oil specific volume at null pressure.	[m ³ /kg]
v_m	mixture specific volume obtained from the Eq. (1).	[m ³ /kg]
M_l	oil molecular mass.	[kg/kmol]

As a result, the oil mass composition (X_{m1}) in the mixture, is given by:

$$X_{m1} = X_{v1} \frac{M_l}{M_{mis}} \quad (13)$$

6.0 EXPERIMENTAL RESULTS

All gases are soluble to some extent in lubricants, and many refrigerant gases are highly soluble. The amount dissolved depends on the pressure of the gas, the temperature of the lubricant, the nature of the gas, and the nature of the lubricant.

From the realized simulations, considering ten (10) experimental points, we can establish that: The oil percentage in the mixture lies below 10 % in mass for most of the results, observing that the oil percentage calculated as indicated, is strongly dependent on pressure (see Table 2).

TEST Number	Psucp [kPa]	T14 [K]	P1 [kPa]	T10 [K]	\dot{W} [kW]
1	620	316	2130	418	2.30
2	560	309	2200	421	2.20
3	480	301	2220	426	2.10
4	360	290	1990	428	1.70
5	490	307	2030	422	2.00
6	660	304	2060	386	2.32
7	600	303	2150	399	2.38
8	560	298	2210	408	2.21
9	590	319	2440	428	2.37
10	570	315	2330	425	2.28

Table 1: Experimental operating conditions.

TEST Number	X_{v1} %	X_{m1} %	\dot{W}_{amb_cp} [watts]	$\Delta h_{12\ pure}$ [J/kg]	$\Delta h_{12\ mix}$ [J/kg]	$\dot{m}_{r, pure}$ [kg/s]	$\dot{m}_{r, mix}$ [kg/s]	$\dot{m}_{r, fab}$ [kg/s]	$\dot{m}_{r, vent}$ [kg/s]
1	2.59	9.73	94	67971	67514	0.032	0.022	0.032	0.030
2	2.68	10.04	89	73096	72488	0.029	0.021	0.028	0.027
3	2.68	10.04	84	81964	80883	0.025	0.019	0.022	0.022
4	2.33	8.82	74	90703	89155	0.018	0.012	0.015	0.016
5	2.43	9.17	88	75645	74793	0.025	0.017	0.023	0.023
6	2.71	10.12	49	49914	51924	0.046	0.035	0.035	0.036
7	2.75	10.28	59	60200	61181	0.039	0.030	0.034	0.034
8	2.78	10.39	54	70062	70362	0.031	0.024	0.027	0.028
9	3.00	11.12	94	71183	70392	0.032	0.026	0.028	0.027
10	2.85	10.61	87	72094	71341	0.030	0.023	0.027	0.027

Table 2: Main calculation results.

where,

$\dot{m}_{r, fab}$	manufacturer refrigerant mass flow rate	[kg/s]
$\dot{m}_{r, pure}$	pure refrigerant mass flow rate	[kg/s]

$\dot{m}_{r,mix}$	oil-refrigerant mixture mass flow rate	[kg/s]
$\dot{m}_{r,vent}$	venturi meter refrigerant mass flow rate	[kg/s]
$\dot{W}_{amb,cp}$	compressor heat losses to surroundings	[watts]

The calculated mass flow of refrigerant was estimated considering an energy balance that includes the compressor heat losses to surroundings:

$$\dot{m}_{r,mix} = \frac{\dot{W} - \dot{W}_{amb,cp}}{h_2 - h_1} \quad (14)$$

Enthalpy difference in Eq. (14) is obtained using Eq. (11); Equation (11) was corrected accepting that this equation is also valid for pure refrigerant, and by comparison with enthalpy value obtained from thermodynamic tables. Then, the obtained difference was considered as due to reference level and introduced in Eq. (11).

This methodology shows that, when the mixture flow rate is calculated considering only pure refrigerant, elevated values are obtained. On the other hand, when the mass flow rate is calculated and the presented methodology is applied, it was observed that the mass flow rate lies on the lowest values.

In order to analyze oil effect on this mixture, the relative difference between mass flow rate (*Dif1*) was obtained considering that oil is not present ($\dot{m}_{r,pure}$), and the mass flow ($\dot{m}_{r,mix}$) calculated by considering oil effect. Analogously, the relative difference compositions (*Dif2*) and (*Dif3*) were obtained by replacing ($\dot{m}_{r,mix}$) for ($\dot{m}_{r,fab}$) and ($\dot{m}_{r,vent}$) respectively.

$$Dif1 = \frac{\dot{m}_{r,pure} - \dot{m}_{r,mix}}{\dot{m}_{r,pure}} * 100 ; Dif2 = \frac{\dot{m}_{r,pure} - \dot{m}_{r,fab}}{\dot{m}_{r,pure}} * 100 ; Dif3 = \frac{\dot{m}_{r,pure} - \dot{m}_{r,vent}}{\dot{m}_{r,pure}} * 100 \quad (15)$$

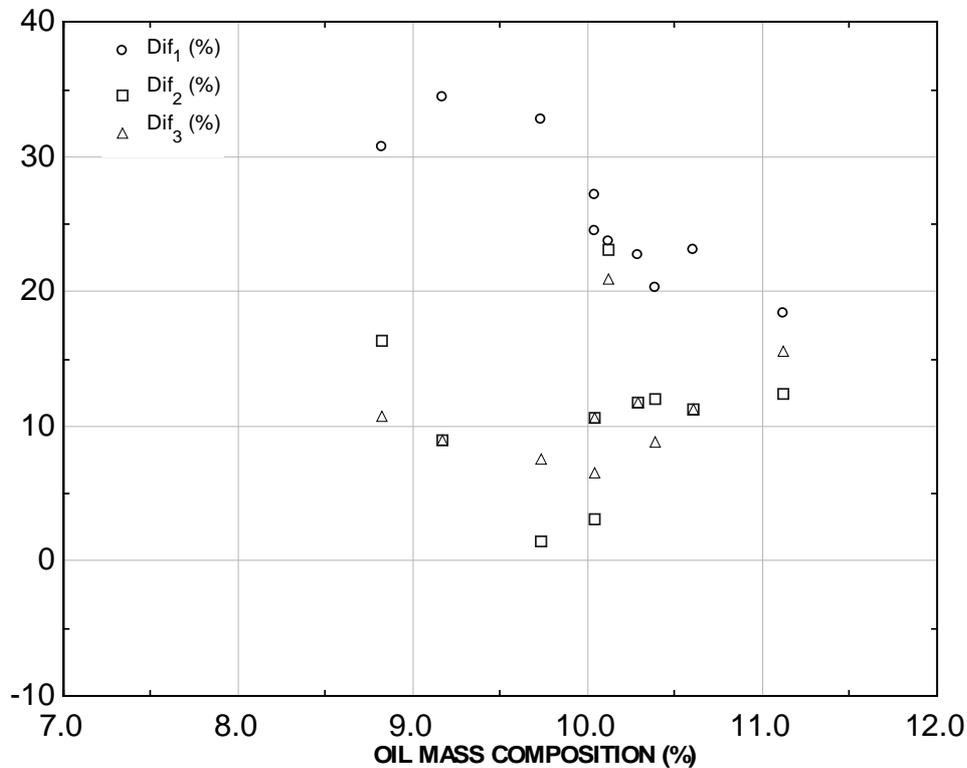


Figure 2: Mass flow comparison.

7.0 CONCLUSIONS AND RECOMMENDATIONS

The effect of the lubricant in mixture with refrigerant causes a decreasing in the refrigerant mass flow. As a result, the coefficient of performance (COP) and others cycle characteristics are effectively affected by the oil presence.

In other words, taking into account the presence of lubricant mixed with refrigerant leads, as a result, to lower refrigerant flows than expected, which are more accurate. This, even considering all theoretical assumptions, represents an encouraging result.

The thermodynamic modelling of mixture properties relies on the fugacity and activity coefficients. Some simplified hypothesis are essential when using an equation of state with mixing rules, because the application of the state equation method requires detailed oil property information. This is more correct than using the thermodynamic properties of mixture as if it was pure refrigerant only.

The next steps in this work:

- Take the $C_v(T)$ calculation as a more accurate form for real mixture.
- Introduce the fugacity concept for the partial oil pressure.
- Measurement of oil mass composition in our test bench, comparing with the theoretical results obtained in this paper.

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