Compressor Performance Analyses of Refrigerants (R22 and R407C) with Various Lubricants in a Heat Pump

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ABSTRACT

System performance in heat pumps depends upon several factors: refrigerant and lubricant choice, type of compressor, type of heat exchangers, expansion devices etc. Depending upon these factors, system performance can change significantly, as much as 30% or more. An important question in analyzing system performance is determining which part of the system is responsible for changes in performance: is it due to the compressor, the system components (heat transfer, or refrigerant mass flow rate, etc.) or combined effects of both? In this study, we have examined system performance of a duct-free split heat-pump unit (2.5 kW) with a rotary compressor, designed for R-22/mineral oil and tested with R-407C and various POE oils. System performance (capacity and energy efficiency for both cooling and heating modes) showed significant variations among the different lubricants tested. As a first step to understand such behavior, we analyzed compressor performance with the various lubricants under the same ambient conditions. Based on a polytropic analysis, we concluded that the compressor performance is not responsible for the observed large variations in system performance.

1. INTRODUCTION

During testing of a duct-free split heat pump with R407C/POE, significant differences in performance were observed when the different compressor lubricants with different viscosities were tested. In this report, we show the analysis of the compressor performance with the polytropic compression process (PV^n = *constant*). When the ideal gas law ($PV = RT$) is assumed, working equations can be derived in analytically closed simple forms, and are quite convenient for the analysis of actual experimental pressure and temperature data. First, we calculated polytropic exponents (n) for R-22 and R-407C in both cooling and heating mode, using thermodynamic properties for the isentropic process. With these exponents, we analyzed compressor power requirement and the mass balance between the inlet and outlet of the compressor. If differences in the lubricant viscosity and/or solubility of refrigerant-lubricant mixtures are important for compressor work, we should observe different values in the polytropic exponent for different lubricants. Also, the exponents should be different from the ”theoretically” calculated values for the refrigerants without lubricants. However, the present analyses show that all experimental data in both cooling and heating modes have been well correlated with the same value in the exponent as the calculated value.

2. EXPERIMENTAL

2.1 Setup
In the present study, we have employed a commercial 2.5 kW ductless-split heat pump with a rotary compressor which is designed for R-22 refrigerant and 4GS mineral oil. When testing R407C, no hardware modification was made. Lubricants evaluated are listed in Table 1 with corresponding lubricant viscosity grade.

Table 1: Refrigerant Lubricant Mixtures Tested

POE Lubricant	Kinematic Viscosity (cSt) at 40^{o}C
R407C/POE 32A	32
R407C/POE 32B	34.5
R407C/POE 46A	46
R407C/POE 46B	46
R407C/POE 68	68
R22/4GS	55

The heat pump is comprised of an evaporator, a condenser, capillary-tube expansion devices for both cooling and heating modes, a suction-line accumulator, and a rotary compressor. Although no hardware modification has been made, the unit has been highly instrumented for data collection including thermocouples and pressure transducers as well as a Coriolis mass flow meter in the liquid line. For the air-side capacity measurement, all necessary instruments for the psychometric analysis were installed. Experiments were performed in an environmental chamber under specified ARI rating conditions for Cooling A (26.7°C dry bulb, 19.4°C wet bulb at indoor conditions, 35.0°C dry bulb and 23.9°C wet bulb at outdoor conditions) and Low Temp Heat (21.1°C dry bulb, 15.6°C wet bulb at indoor conditions, 31.7°C dry bulb and –9.4°C wet bulb at outdoor conditions). All data in both refrigerant-side and air-side measurements are collected in six second intervals for several hours.

2.2 Error Analysis
Before analyzing experimental data, the data quality and accompanied errors were examined. Time-series raw data such as air-side capacity, compressor power, temperatures and pressures at various locations, and Coriolis mass flow data were evaluated. Although the time-average data are typically well determined, all data show some constant variation widths (*c.a.* 2 %) around the average values.

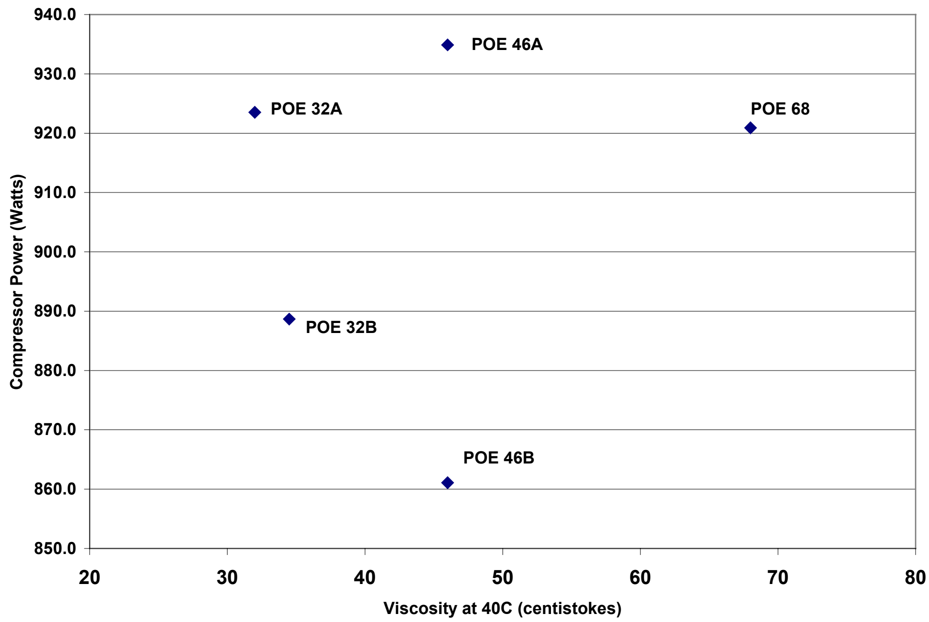
3. RESULTS AND ANALYSIS

3.1 Cooling A Test
Table 2 shows the results of the cooling air side capacity and energy efficiency ratio (EER) respectively, using different POE lubricants with R407C. An R22/4GS baseline is also shown for comparison. Surprisingly, the performance variations are significant among the different POE lubricants. It is well known that lubricant viscosity is an important factor for the compressor performance, particularly in the case of very high compression ratios (c.a. 8 – 10), which occur in refrigerator units. In the heat pump tested, the compression ratio is not very high (about 3), but viscosities of tested oils range from 32 to 68 cSt at 40^{o}C (see Table 1) which might influence compressor power consumption. Therefore, it is important to examine compressor performance.

Table 2: Cooling Capacity and EER at Cooling A Conditions

POE Lubricant	Capacity (kW)	EER
R407C/POE 32A	2.36	8.72
R407C/POE 32B	2.05	7.86
R407C/POE 46A	2.26	8.24
R407C/POE 46B	1.76	6.99
R407C/POE 68	2.49	9.22
R22/4GS	2.41	9.17

Figure 1 shows clearly that POE lubricant viscosity is not correlated with the observed compressor power consumption in the present case. Thus, the viscosity effect is not a major factor influencing power consumption.



` Figure 1: Compressor Power Versus Viscosity for R407C/POE

Then, a question is how the power is controlled, or whether there is any differential effect due to the various lubricants tested. To answer the question, compressor performance was analyzed with the commonly known "polytropic" process, where the compression and expansion processes are described by a relation PV^n = *constant*, where P is pressure, V is volume and n is a polytropic exponent. The significance of the polytropic analysis is that the compressor *work* with all different compression processes (e.g., isentropic, isothermal, isobaric processes as well as their combinations) can be modeled with this relation, and that the magnitude of polytropic exponent contains all information of the process: *i.e*., the compressor performance due to different refrigerants as well as different lubricants. One of the common misconceptions is that ΔH (enthalpy at the discharge (T, P) point minus enthalpy at the suction (T, P) point) is proportional to the compressor *work*. However, this is *only* true when the compression work is done by an isentropic (adiabatic) process. In general processes, ΔH is not simply proportional to the compressor *work*. This is because the *work* is not the thermodynamic *state* function, which is determined by the initial and final states only and does not depend on the process path, but the *work* depends upon the path, while H is a *state* function. Thus, the analysis using any *PH* (or *TS*) diagram for the *general* compressor *work* cannot be properly made.

In order to calculate the compressor work with the polytropic process, an equation of state (EOS) for the working fluid is required. The use of the ideal gas equation of state is sufficient for the present purpose. A great advantage of the use of ideal gas EOS ($PV = RT$) is that the compressor work can be analytically derived, and such an analytical expression is extremely useful for the analysis of the observed data. Then, the compressor work (W) is given in the following equation:

$$W = \frac{nV_0 P_{suc}}{n-1}\left(P_R^{1-1/n} - 1\right), \tag{1}$$

where V_0 is an *effective* fixed compressor displacement (*i.e.*, a volumetric efficiency η_v is assumed to be unity), P_{suc} is a compressor suction pressure, and P_R is a compression ratio (discharge pressure divided by suction pressure). Equation (1) is also written as:

$$\frac{W(n-1)}{nP_{suc}} = V_0\left(P_R^{1-1/n} - 1\right). \tag{2}$$

The interest here is to see whether there is any significant impact of different lubricants on compressor performance. First, it as assumed that there is no lubricant effect and the compressor work is done with the *purely* refrigerant polytropic exponent $n = n_0$. Then, a plot of the left-hand side quantity versus $P_R^{1-1/n} - 1$ will give us a straight line with a slope of V_0. If the lubricant effect is operating (or the present assumption is wrong), then such a plot would not give a straight line and scatter, because of various different n ($\neq n_0$) values. The value n_0 for R-407C has been determined for the cooling A condition in a way similar to a proposed method (Lenz, 2002), using a reliable thermodynamic EOS [Lemmon *et. al*., 2002]: n_0 (R-407C) = 1.056 ± 0.002.

With an n_0 value of 1.056, a plot for Equation (2) is shown in Figure 2. All data were on a straight line with a slope 0.0015 m^3/s (*i.e.*, V_0 = 24 cc/rev with 60 Hz), which is a physically reasonable value for the present 2.5 kW system. In addition, the *actual fixed displacement* V_a will be given by V_0 times the volumetric efficiency η_v: $V_a = V_0\eta_v$, since in Eqs. (1) and (2), we assumed η_v = 1 for V_0. The volumetric efficiency η_v is determined in a later analysis (see Figure 3): η_v = 0.858, which gives an *actual fixed displacement* V_a of 20.6 cc. In Figure 2, the straight line should possess a zero intercept according to Eq. (2), but shows a small non-zero intercept (1×10^{-5}), which can be regarded as practically zero within experimental accuracy.

As illustrated in Figure 2, all data with various oils are well correlated using a single n value of refrigerant alone. Thus, from this analysis, it is clear that various oils have no discernible contribution to changes in compressor power.

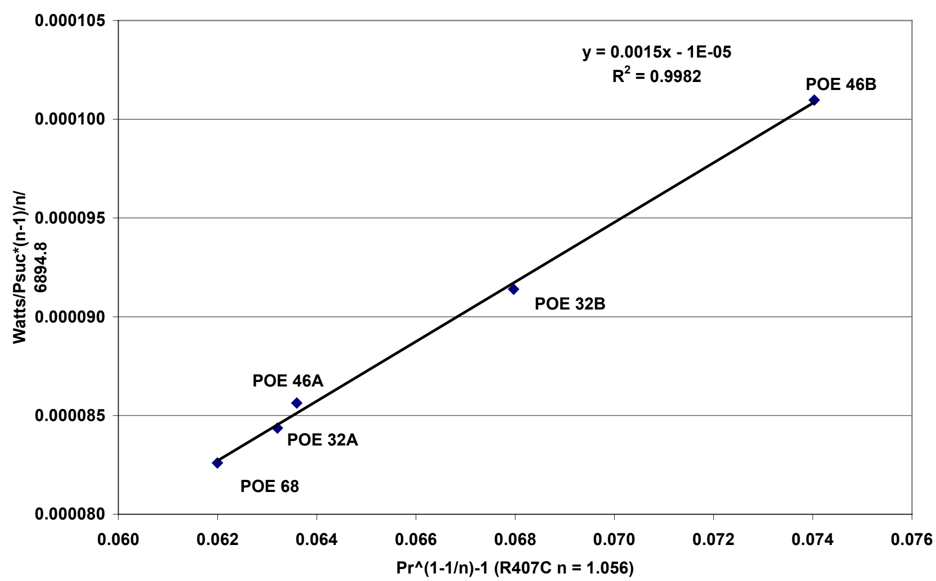


Figure 2: Compressor Work Using Polytropic Analysis – Cooling A

An additional analysis for the compressor performance can be made using a mass balance relation: *i.e.*, mass (in the compressor intake volume at suction) = mass (in the volume at the discharged point):

$$\frac{P_{suc} M_w V_0 \eta_v}{RT_{suc}} = \frac{P_{dis} M_w V_{dis}}{RT_{dis}}, \tag{3}$$

where V_{dis} (discharge gas volume) can be obtained from the polytropic equation ($PV^n = constant$: thus $P_{suc} V_0^n = P_{dis} V_{dis}^n$): M_w (molecular weight), R (gas constant), and η_v (volumetric efficiency). Then, Eq. (3) can be rearranged as:

$$\frac{P_{dis}}{T_{dis}} = \eta_v \frac{P_{suc}}{T_{suc}} P_R^{1/n}. \tag{4}$$

This relation is plotted in Figure 3 where an excellent linear correlation has been obtained using again the pure-refrigerant exponent n_0 alone. The determined slope of 0.858 corresponds to the volumetric efficiency η_v in Eq. (4). The observed non-zero intercept is merely a correction for P_{dis}/T_{dis} of the ideal gas law EOS.

Finally, from the results of both Figures 2 and 3, we conclude that the compressor performance has not been affected by the different oils studied here.

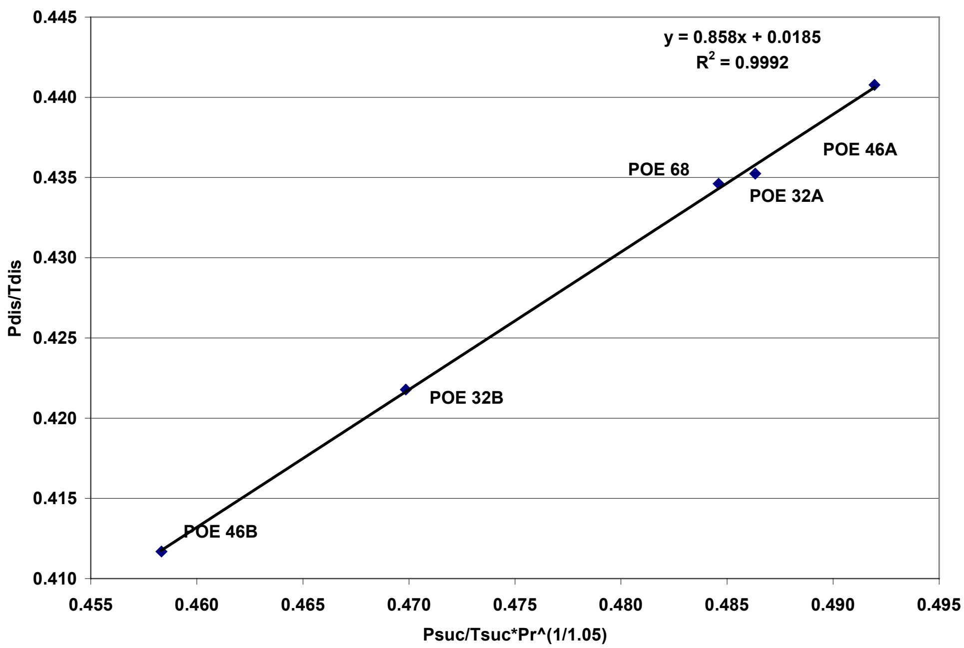


Figure 3: Pd/Td Versus Ps/Ts*Pr^(1/1.05) – Cooling A

3.2 Low Temp Heat Test

Table 3 shows the results of the low-temperature-heat capacity and EER respectively, using different POE lubricants with R407C and an R22/4GS baseline. Again performance variations are significant among the different POE lubricants.

Table 3: Cooling Capacity and EER at Low Temp Heat Conditions

POE Lubricant	Capacity (kW)	EER
R407C/POE 32	0.89	3.74
R407C/POE 34	0.86	3.11
R407C/POE 46A	1.11	4.39
R407C/POE 46B	0.76	3.20
R407C/POE 68	0.87	3.59
R22/4GS	0.97	4.16

However, it should be mentioned that all test cases with R-407C were under the evaporator-flooded condition, whereas the system was designed for R22 to operate in non-flooded condition. This is due to the fact that the capillary length was designed for R-22 which can be up to 40% longer than R407C. Although in this case, the compressor work is under the wet (two-phase) compression, we tried the same analyses as the cooling A tests (section 3.1) to determine whether the polytropic process can still be applied. Figure 4 shows the result based on Eq. (2), and again a straight line was obtained with a calculated polytropic exponent for R-407C, which was derived for the present wet compression process using the proposed method (Lenz, 2002) and a reliable EOS (Lemmon, 2002) $n = 0.9175 \pm 0.005$. The observed slope of 0.0022 m^3/s (*i.e.*, V_0 = 35 cc/rev with 60 Hz) corresponds to the effective displacement volume (see the analysis in Figure 2). The magnitude is again physically reasonable, and the actual

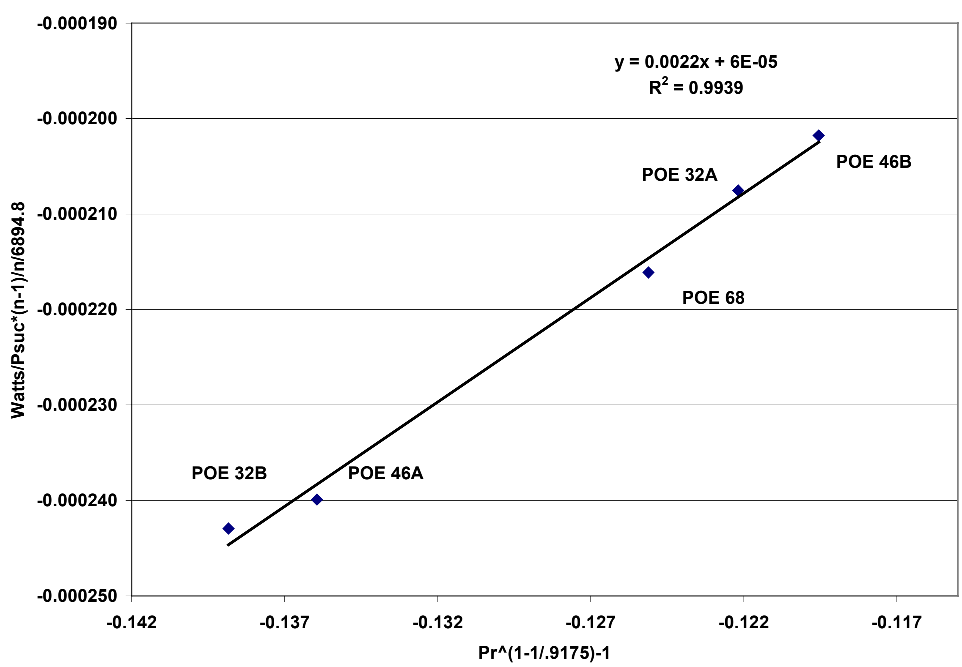


Figure 4: Compressor Work Using Polytropic Analysis – Low Temp Heat

fixed displacement volume was in this case found to be 21 cc ($V_a = V_0\eta_v$, where η_v = 0.6048, determined in the following Figure 5 analysis). This V_a value of 21 cc is in an excellent agreement with 20 cc determined in the cooling A data (see the section 3.1) According to Eq. (2), the intercept of this straight line in Fig.15 should be zero. The observed small intercept value of 6×10^{-5} can certainly be regarded to be zero within the experimental uncertainty.

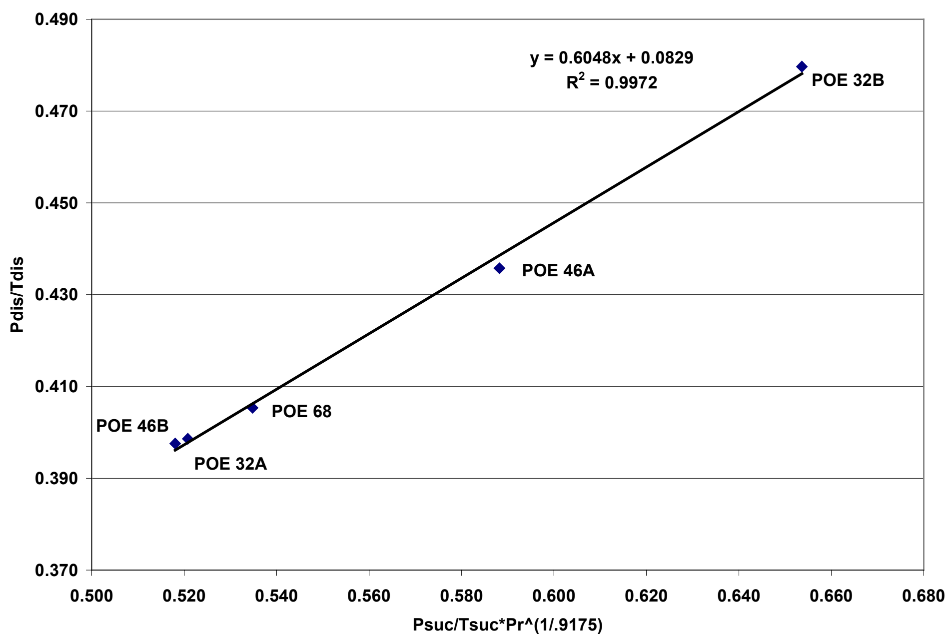


Figure 5: Pd/Td Versus Ps/Ts*Pr^(1/1.05) – Cooling A

Next we examined the mass balance, based on Eq. (4), using the same polytropic exponent of 0.9175 for pure R-407C. The result is shown in Figure 5. Again, an excellent straight line relation was obtained. The slope corresponds to the volumetric efficiency (see Eq. 4): η_v = 0.6048. This is smaller than the value of 0.8337 in Fig. 5, but this is expected because the volumetric efficiency decreases as the compression ratio increases (in the present Low Temp Heat case vs. the Cooling A case) due to the *clearance volume* effect and/or the *leakage* effect. The non-zero intercept is again simply due to the correction term for the ideal gas EOS as mentioned in Section 3.1. An important conclusion again is that the effect of different oils did not show up any unique and specific contributions to the compressor performance even for the LTH and the wet compression condition.

4. CONCLUSIONS

Polytropic analysis is a useful tool to understand compressor performance during heat pump testing using experimental measurements such as system temperatures and pressures. In this case, compressor performance of a duct-free split system heat pump was analyzed to show that different POE lubricants used with R407C were not affecting compressor behavior and that some other system effects must be responsible for variations in capacity and energy efficiency. These will be investigated in future studies.

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