2014

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ABSTRACT

Polyol ester (POE) lubricants of different viscosity ISO grades (32-80) and possessing distinctly different compatibilities (miscible vs. immiscible) were tested with R-410A, R-32, and L-41b. For each refrigerant-lubricant pair tested, the cooling coefficient of performance (COP), heating performance factor (HPF), and oil circulation ratio (OCR) were determined while operating at AHRI Standard 210/240 conditions A, B, C, H1 & H2. The results were correlated to the properties of the working fluids. Due to its higher density, yet comparable specific heat, R-32 showed increased cooling capacity compared to R-410A. However, the COPs of these refrigerants were similar because the capacity increase was offset by increased compressor power consumption. L-41b required the least compressor power, but also had the lowest cooling capacity and COP of the three refrigerants. Lubricant choice had minimal impact on cooling capacity. However, immiscible lubricants lowered cooling capacity by about 4% for R-32, condition B. A larger effect was observed in the compressor, where lubricants specifically designed for R-32 lowered discharge temperatures by 6 °C and reduced power consumption by up to 10%. For R-32-lubricant pairs tested under AHRI cooling condition B, the highest and lowest COPs measured were 4.19 (optimized ISO 68 POE) and 3.72 (commercial ISO 32 POE) – a 12% improvement by replacing the standard R-410A lubricant.

1. INTRODUCTION

The transition to lower global warming potential (GWP) refrigerants is critical to the realization of environmentally sustainable and more energy efficient refrigeration technologies (Ritter, 2013). Leading candidates to replace R-22 and R-410A in air conditioning and heat pump applications include R-32 (difluoromethane) and many HFC/hydrofluoroolefin blends with GWPs in the range of 400-650. Leading candidates in the latter category include blends containing 70% or more R-32 (e.g., L-41a & b, DR-5) (Kedzierski *et al.*, 2014).

Considerable data have been generated comparing R-410A with various low-GWP alternative refrigerants in full system tests. Most notable is the work sponsored by AHRI under the Alternative Refrigerants Evaluation Program (AREP) (Wang *et al.*, 2013). These studies have been either refrigerant "drop-in" tests to commercial R-410A systems or "soft-optimized" tests, where minor component modifications were made to adapt a system to the properties of the new refrigerant. In all cases, the lubricants used for these studies were commercial polyol esters (POEs) designed for use with R-410A. However, lubricants optimized for one refrigerant may not be the best choice for others. In this study, we show that the choice of lubricant is another parameter that should be considered when switching from R-410A to lower-GWP alternatives.

Although lubricant is added to the system for the sole purpose of lubricating the moving parts of the compressor, it also plays a thermo-fluidic role in the system, impacting capacity and efficiency. For example, lubricants can influence capacity by altering refrigerant-side heat transfer coefficients, lowering pressures necessary to reach operating temperatures, and increasing pressure drops. Lubricants also affect efficiency by changing the isentropic efficiency of the compressor, which would raise or lower the discharge temperature for a given discharge pressure.

Systems which are convertible between air conditioning and heat pump modes are gaining market share in the residential marketplace. In consideration of this trend, we tested a 3-ton convertible system with the next generation refrigerants R-410A, R-32, L-41b and traditional and advanced POE lubricants. Since POEs intended for R-410A (i.e., traditional POEs or TPOE in Figure 1) are not as miscible with R-32 and HFC/HFO blends, their use leads to concerns over inadequate lubrication, poor oil return, and excessive lubricant hold-up in the system, especially when switching between cooling and heating modes. Although some of these problems were not observable in the short-term energy efficiency tests reported here, there is also interest in understanding if optimized lubricants can improve the overall performance of low-GWP-based systems.

Type	ID#	ISO	R-410A		R-32		L-41b	
TPOE	1	32	X	+	X	-		+
TPOE	2	68	X	-	X	-	X	+
APOE	3	32		+	X	+	X	+
APOE	4	46		+	X	+	X	+
APOE	5	68	X	+	X	+		+
APOE	6	80		+	X	+	X	+

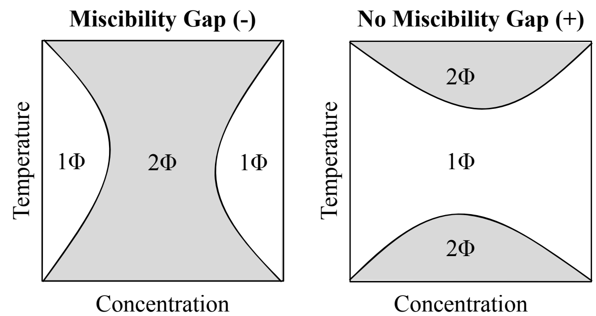

Figure 1: Left: Refrigerant-lubricant pairs tested (indicated by X's) and their sealed-tube miscibility behavior (indicated by - or +). Right: Refrigerant-lubricant mixtures can be classified by whether they have a miscibility gap. 1Φ = one liquid phase; 2Φ = two liquid phases.

To answer this question, we designed, synthesized, and tested a series of advanced polyol ester lubricants (APOE, Figure 1) having improved compatibility with R-32 and low-GWP blends. The results of this study and related work illustrate that these new lubricants have the potential to increase performance while providing improved lubricity and wear protection (Hessell *et al*., 2014; Urrego *et al*., 2014). The refrigerant-lubricant pairs tested are listed on the left side of Figure 1. The six lubricants are differentiated by their viscosity grade and miscibility with R-32. TPOEs (lubricants 1 & 2) are commercially available and commonly used with R-410A. The APOEs (lubricants 3-6) possess optimized molecular structures that decouple lubricant viscosity and miscibility such that even high-viscosity lubricants with good load-carrying properties are miscible with R-32. Similar APOEs have been demonstrated to maintain superior working viscosities in R-744 (Hessell *et al*., 2010).

2. EXPERIMENTAL

The test facility (Figure 2) allowed detailed measurements of the major components in each of two psychrometric test chambers. Added instrumentation measured refrigerant flow rates, temperatures, and pressures at key points in the cycle. Air temperature, humidity, and flow rates past each heat exchanger were controlled to match AHRI Standard 210/240 conditions. Upstream of each heat exchanger, an inlet thermocouple grid ensured uniform air temperatures while an outlet thermocouple grid was used as an indicator of refrigerant distribution. The air pressure drop across the heat exchangers was measured and a nozzle assembly allowed for the determination of air flow rate in the indoor unit. The indoor wind chamber was also equipped with a helper blower to match the air flow rate across the evaporator.

Two independent energy balances were utilized for calculating capacity: (1) air-side energy balance and (2) refrigerant-side energy balance. For the air-side energy balance, the mass flow rate of the air was calculated from differential static pressure measurements across air flow nozzles with known diameters according to equation (1).

$$\dot{m}_{air} = C_D A_{nozzle} \sqrt{2\Delta P \rho_{air}} \tag{1}$$

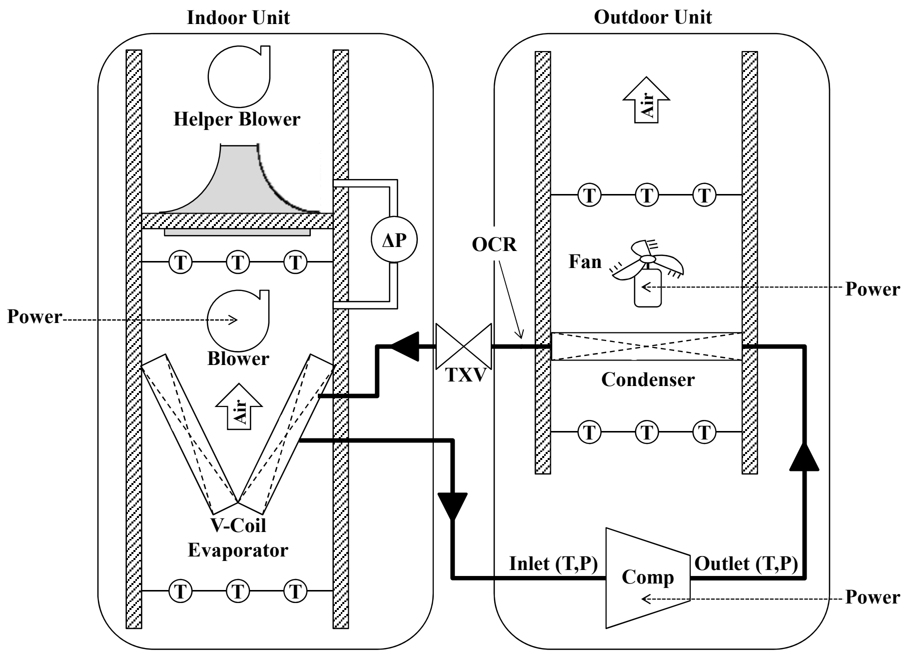

Figure 2: Diagram of the test facility, which conformed to ASHRAE Standard 37-2005. OCR represents the removable test section used for gravimetric analysis of the oil circulation ratio.

Thermocouple grids allowed for the calculation specific enthalpy changes of dry air as it was heated or cooled by the heat exchangers. Since water condensation may occur on heat exchangers that absorb heat from the air, chilled mirror dew point sensors were located before and after the evaporator so that the latent load could be calculated. Combining measurements of air flow rate, temperature, and humidity change allowed the calculation of capacity exclusively from air measurements according to Equation 2.

$$\dot{Q} = \dot{m}_{air} c_{p,air} \left(T_{in} - T_{out}\right) \tag{2}$$

The refrigerant-side energy balance used flow rate data collected by a Coriolis-type mass flow meter placed in the liquid line. The change in specific enthalpy from component inlet to outlet was determined from temperature and pressure measurements. Multiplying the refrigerant mass flow rate by the enthalpy difference across the heat exchanger yielded the refrigerant side energy balance, Equation 3, where the heat of mixing (Q_{mixing}) is assumed to be negligible [e.g., $\dot{m}_{oil}Q_{mixing}$ = (0.1 kg/s)(2% OCR)(0.5 kJ/kg) = 1 W].

$$\dot{Q} = \dot{m}_{refrigerant} \left(h_{out} - h_{in}\right) + \dot{m}_{oil} c_{p,oil} \left(T_{out} - T_{in}\right) + \dot{m}_{oil} Q_{mixing} \tag{3}$$

COP measures system efficiency by relating cooling capacity to the power requirements of the system. Power to the outdoor unit was consumed by the compressor and a blower. Power to the indoor unit was used by a fan and 24 Volt transformer. Each unit's power requirements were measured separately by calibrated transducers. COP was calculated via Equation 4. An analogous equation describes the HPF for heating mode operation.

$$COP = \dot{Q}_{evap} / \dot{W}_{in} \tag{4}$$

The OCR was measured gravimetrically in a test section of the liquid line according to ASHRAE standards and Equation 5. As OCR is strongly dependent on operating conditions and loading, this was measured at the end of testing of each cooling condition (A, B, C). The reported results are an average of the three measurements.

$$OCR = \frac{\dot{m}_{oil}}{\dot{m}_{oil} + \dot{m}_{refrigerant}} \tag{5}$$

POE lubricants were paired with each refrigerant, as listed in Figure 1. The lubricant and refrigerant were charged to the system per industry standard procedures. Specifically, refrigerant was added until the subcooling in the liquid line matched that specified by the manufacturer. The system was run for several hours while the proper test conditions were achieved. Following 30 minutes at steady state, data were collected for 30 minutes per test condition and the equilibration-testing cycle was repeated for each AHRI test condition. After completing data collection for a refrigerant-lubricant pair, the system was thoroughly flushed with R-134a to remove any residual working fluid from all components. Then it was evacuated to 75 mTorr and the next lubricant-refrigerant pair was charged. The flushing, charging, and data collection procedure was repeated for each pair marked with an "X" in Figure 1.

3. RESULTS AND DISCUSSION

3.1 Cooling Mode Performance

Neglecting lubricant effects and looking at trends in the data when grouped by refrigerant (Table 1), we observed that the discharge temperature correlated with the amount of R-32 in the refrigerant (R-32 > L-41b > R-410A). Cooling capacity, discharge pressure, and the power consumed by the compressor (i.e., the main component of the outdoor unit) followed the trend R-32 > R-410A > L-41b. Figure 3, where the data is grouped by test condition (A, B, C) and refrigerant, contains more detail regarding COP. The average and its 95% CI are included for each group, neglecting lubricant effects. These results show that, under test conditions A and B, the COP of L-41b was significantly lower than that of the other two refrigerants, while R-32 was comparable to R-410A. Under test condition C, all three refrigerants performed similarly, with only marginal improvement going from L-41b to R-32 to R-410A.

Table 1: Average values for key performance characteristics during cooling mode operation

AHRI Test Condition:		A			B			C	
Refrigerant:	L-41b	R-32	R-410A	L-41b	R-32	R-410A	L-41b	R-32	R-410A
Discharge T (°C)	94.5	97.9	79.6	79.9	84.2	68.3	82.2	89.4	71.6
Discharge P (bar)	26.4	30.3	29.1	22.0	25.5	24.7	21.4	24.8	24.1
Outdoor Unit Power (kW)	2.59	3.00	2.79	2.19	2.52	2.37	2.14	2.47	2.35
Cooling Capacity (kW)	8.90	11.05	10.43	9.53	11.66	11.21	8.25	9.59	9.48
COP	2.99	3.27	3.28	3.69	4.01	4.06	3.26	3.35	3.47

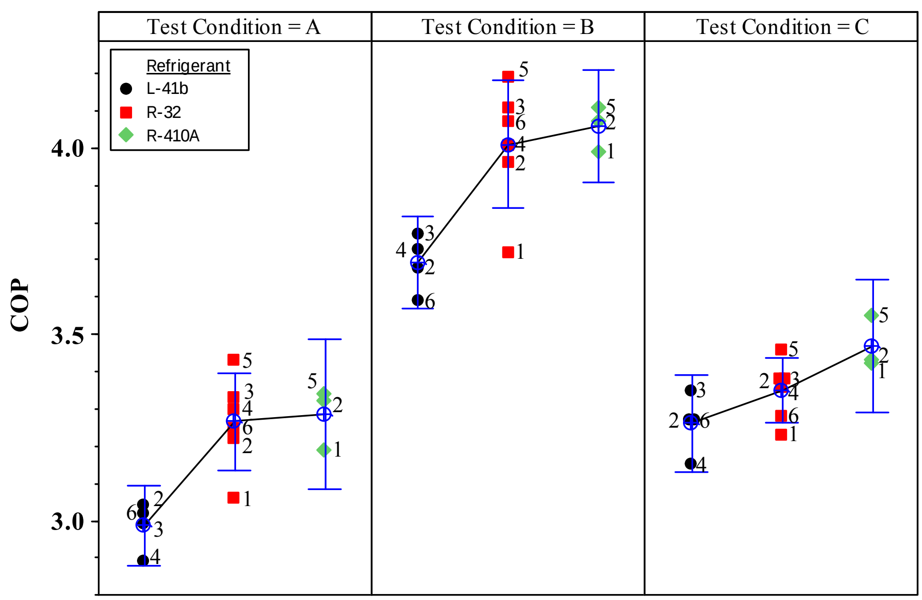

Figure 3: Interval plot of COP grouped by AHRI test condition. Labels indicate lubricant.

However, the data labels in Figure 3 provide additional information and illustrate the effect lubricant optimization has on efficiency. All lubricants were tested with R-32, the refrigerant for which APOE miscibility was optimized, while only a subset was tested in R-410A and L-41b. So the differences between lubricants and their effects on performance were highlighted in the R-32 data. In R-32, under test conditions A and B, all APOEs (3-6) surpassed the TPOEs (1,2) in both R-32 and R-410A. The results in L-41b were less differentiated, presumably because all of the lubricants tested were miscible in this blended refrigerant. Considering the R-32 and R-410A data, it is clearly seen that miscible lubricants outperformed those with a miscibility gap (Figure 1). Lubricant 5, the miscible ISO 68 APOE, provided the best COP in all cooling conditions tested. Lubricant 1, the commercial ISO 32 TPOE, was lowest wherever tested. The improvement ranges from 7% to 12% for R-32.

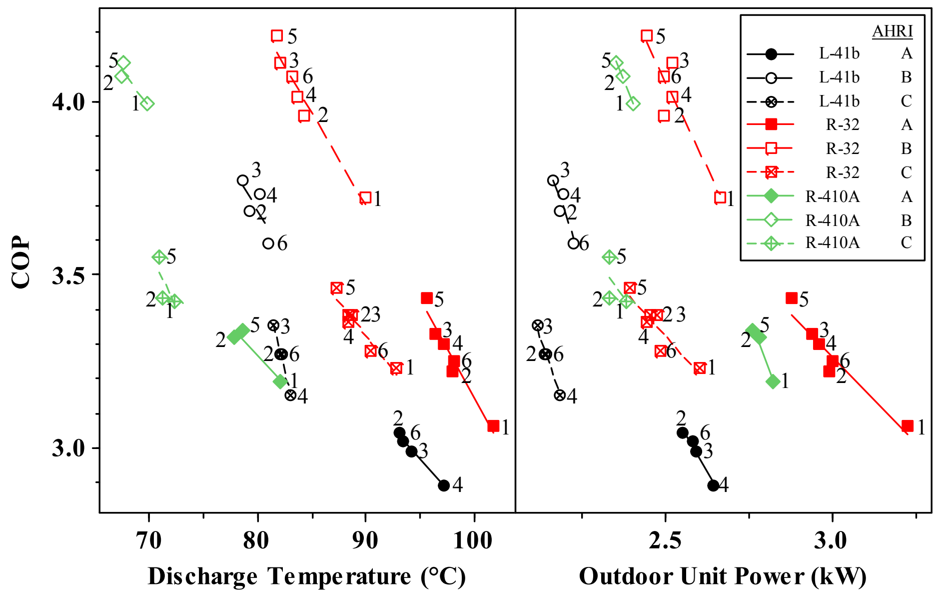

Figure 4: COP vs. compressor outlet temperature grouped by refrigerant and AHRI test condition

The chart above depicts the relationship between COP, compressor discharge temperature, and power consumed by the outdoor unit. As expected, R-32 produced the highest discharge temperatures, on average, for each test condition, followed closely by L-41b. Within each group, discharge temperatures and compressor power were inversely correlated to COP and, interestingly, the lubricant appeared to play a role. Again focusing on just the R-32 data, the effects of optimized lubricants were observable: under each condition, lubricants 1 and 5 produced the highest and lowest discharge temperatures, respectively.

3.2 Heating Mode Performance

Metrics related to heating conditions are listed in Table 2. The discharge temperatures and pressures observed were generally lower for heating than cooling. Again, of the three refrigerants, R-32 gave the highest average temperature and pressure in both conditions. However, the trend in HPF varied by test condition: for H1, R-32 ≥ L-41b > R-410A; for H2, L-41b > R-32 ≥ R-410A. This appeared to be caused by changes in heat pump capacity when switching from H1 to H2. The capacity measured for R-32 under condition H1 was much greater than that for L-41b. The capacities of L-41b and R-32 were nearly equal under condition H2.

Table 2: Average values for key performance characteristics during heating mode operation

AHRI Test Condition:	H1			H2		
Refrigerant:	L-41b	R-32	R-410A	L-41b	R-32	R-410A
Discharge T (°C)	62.6	69.7	63.1	61.7	68.2	60.4
Discharge P (bar)	21.4	25.1	23.9	20.6	22.8	22.2
Outdoor Unit Power (kW)	2.15	2.51	2.35	2.09	2.31	2.23
Heat Pump Capacity (kW)	8.53	9.85	8.95	7.76	7.86	7.57
HPF	11.48	11.65	11.17	10.70	9.96	9.90

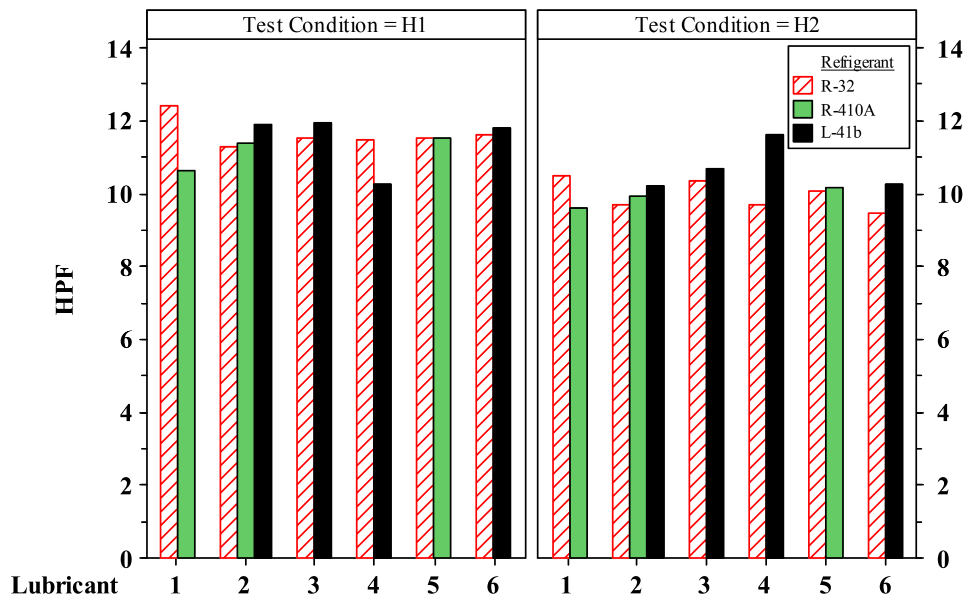

Figure 5: Comparison of HPF for each lubricant-refrigerant pair tested

Heating efficiency (HPF) as a function of lubricant and refrigerant is illustrated in Figure 5. The pair R-32/lubricant 1 (TPOE ISO 32) was the best performer at condition H1; the pair L-41b/lubricant 4 (APOE ISO 46) was the best performer at condition H2. Aside from these two stand-outs, trends were less apparent. It is noteworthy that HPF did not decrease with higher ISO grade lubricants. This suggested that other factors aside from efficiency, such as compressor reliability or working viscosity under extreme conditions, would determine the best lubricant for these cases. Nevertheless, lubricant 5 (APOE ISO 68) remained a strong candidate for heat pump conditions and performed equally well in R-32 and R-410A.

3.3 Lubricant Effects

As mentioned above, two main classes of oils were tested in this study: traditional POEs which tend to have miscibility gaps in R-32 and R-410A; advanced POEs which do not have gaps. Immiscibility could have negative consequences on system performance due to phase separation in the heat exchangers. In extreme cases, immiscibility in the sump could lead to lubrication problems and thus compressor reliability concerns. So good miscibility over a wide range of temperatures and compositions is desirable.

Miscibility is often used as an indicator of the solubility of a refrigerant in the lubricant. It is generally accepted that a refrigerant can be too soluble in a lubricant, such that the viscosity of the working fluid becomes too low to provide adequate lubrication of the compressor. Other problems may also arise due to excessive solubility, such as bubble formation when the fluid's temperature and pressure suddenly change (i.e., refrigerant flashes off).

Optimizing the chemical structure of the lubricant mitigates these problems. The APOEs do this by allowing for controlled miscibility (lower limits near -10 °C) over a wide range of ISO grades. This allows for the use of higher viscosity grade lubricants because the solubility of the refrigerant in the oil decreases the viscosity of the mixture to values similar to those of TPOE/R-410A combinations. These statements are illustrated in Figure 6, which contains data for R-32 at AHRI test condition A. Each panel depicts the viscosity of the working fluid at measured operating conditions. The lubricants are classified according to their miscibility with R-32. TPOEs are less miscible than APOEs. The top panel (suction conditions) shows that TPOEs possess a higher viscosity in the suction line, which could lead to oil return concerns. APOEs, on the other hand, have a reduced viscosity that matches the baseline cases (6.23-7.16 cSt for TPOE/R-410A) near the ISO 68 viscosity grade. The bottom panels show that both classes of oils have similar viscosities at the elevated temperatures inside the compressor. All APOEs provide similar or better working viscosities than TPOEs. However, at the discharge temperature, the APOE working viscosities have a slight maximum at the ISO 68 grade. Therefore, we conclude this ISO grade represents the best trade-off between miscibility, solubility, and working fluid viscosity (i.e., lubrication) for R-32.

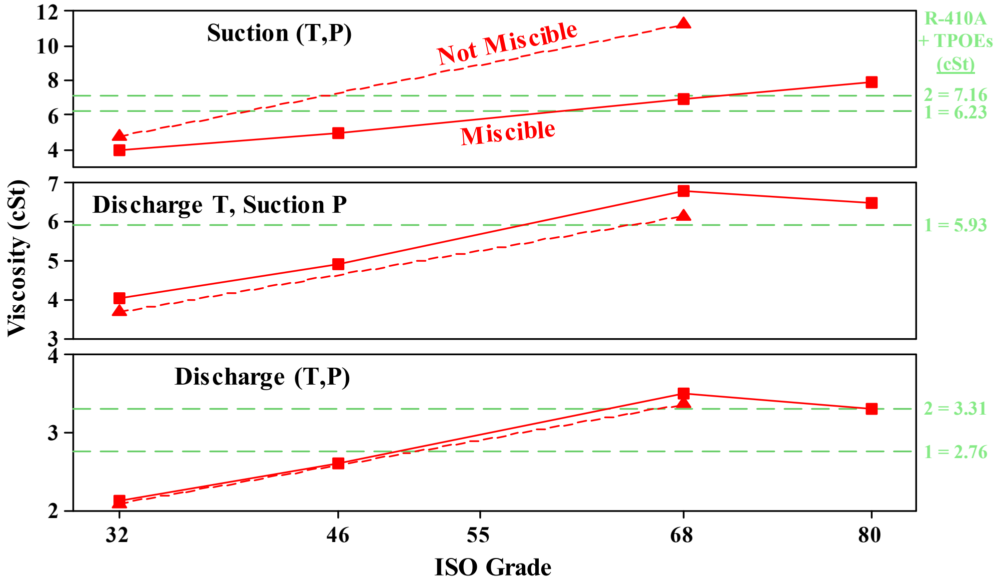

Figure 6: R-32, Condition A: Working fluid viscosity vs. ISO grade as a function of compressor inlet and outlet conditions. Horizontal reference lines illustrate the viscosities attained with R-410A + TPOE Lubricants 1 & 2 (ISO Grades 32 & 68) under identical operating conditions.

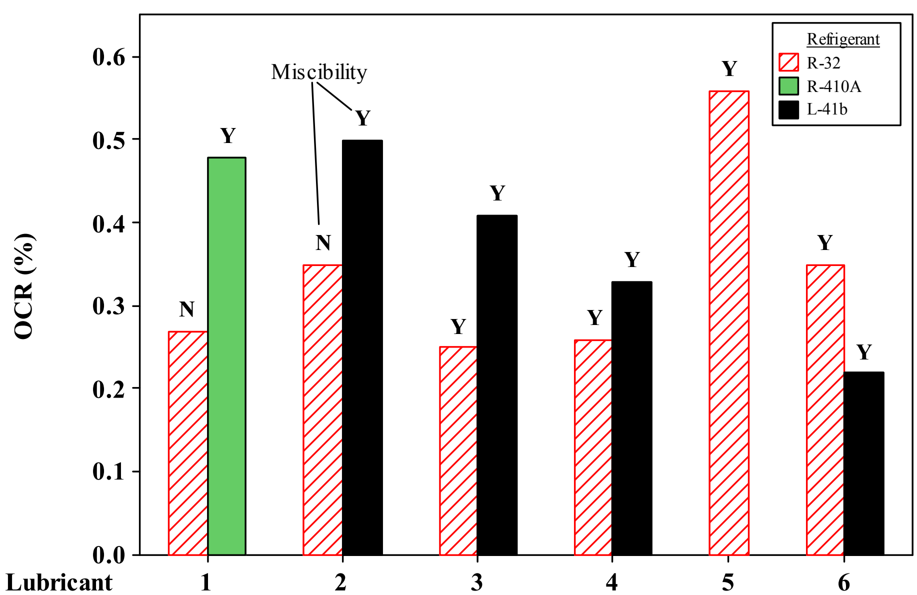

Figure 7: Oil circulation ratios for each lubricant-refrigerant pair. Y = miscible oil (+ in Figure 1); N = miscibility gap (- in Figure 1). Measurements for the following 2 pairs were unreliable and are not shown: R-410A/Lubricant 2 and R-410A/Lubricant 5.

The oil circulation ratio measurements are summarized in Figure 7. All measurements were well below 1% and more soluble pairs tended to have higher OCRs (e.g., L-41b > R-32). Lubricant 5 in R-32 had an OCR very near that of the TPOE-1/R-410A pair. The study of lubricant and refrigerant distribution throughout the system components is the topic of our related work (Wujek *et al*., 2014).

4. CONCLUSIONS

- The miscibility of polyol esters can be decoupled from their ISO grade. Higher viscosity oils can maintain good miscibility with low-GWP refrigerants such as R-32 and L-41b.
- Lubricants have a measurable effect on system performance and efficiency.
- Optimized lubricants with improved R-32 miscibility produce higher COPs than commercial POEs under all conditions tested.
- ISO 68 advanced esters perform best with R-32. They produce the lowest discharge temperatures, require the least compressor power, and yield the highest COPs.

NOMENCLATURE

A_{nozzle}	area of nozzle	(m^2)
$c_{p,i}$	specific heat at constant pressure of fluid i	(kJ/(kg-K))
C_D	discharge coefficient	(-)
h	enthalpy	(kJ/kg)
$\dot{m}_i$	mass flow rate of fluid i	(kg/s)
P	pressure	(Pa)
$\dot{Q}$	heat flow rate	(kW)
T	temperature	(K)
$\dot{W}$	rate of work, power	(kW)
ΔP	pressure drop	(Pa)
ρ_i	density of fluid i	(kg/m^3)

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ACKNOWLEDGEMENT

The authors thank Dr. Dale Carr, Mr. Jeffrey Hutter, and Mr. John Bayusik whose dedication and hard work made the preparation of this paper possible. Components were provided by Carrier Corporation and Emerson Climate Technologies. Financial support was provided by the Department of Energy (DOE-EE0003986).