Performance and Capacity Comparison of Two New LGWP Refrigerants Alternative to R410A in Residential Air Conditioning Applications

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ABSTRACT

Energy conservation is the core reason for the increasing interest in high performance low global warming potential (LGWP) refrigerants. Several researchers pioneered new refrigerants that have zero ozone depletion potential and GWP less than 500. This paper presents a study that breaks new ground on LGWP developmental refrigerants and focuses on heat pump systems for residential applications. An R410A 17.6 kW (5 ton) heat pump split unit commercially available off-the-shelf for US ducted HVAC application, was retrofitted with two new developmental refrigerants that have GWP ranging from 300 to 500, which is significantly lower than that of R410A. The two new refrigerants are still in the R&D stage and are referred throughout this paper as DR-4 and DR-5. The experiments were conducted in a large scale psychometric chamber at Oklahoma State University and the refrigerant cycle pressures and temperatures were measured at design and off-design conditions with outdoor temperature ranging from -8°C (17°F) to 46°C (115°F). Very high outdoor temperatures of 43°C (110°F) and 46°C (115°F) were also considered in order to assess the characteristics of the new LGWP refrigerants at extreme high temperature ambient conditions. The findings for this work showed that DR-5 had up to 4% higher capacity and up to 22% higher COP, while DR-4 showed up to 16% higher COP but 30% lower capacity in comparison with R410A. The experimental results showed that the thermal expansion valve could be further optimized for the new refrigerants. Adjustments were made to maximize the COP of the unit while preserving the cooling capacity and data showed that the COP of DR-4 could be augmented by an additional 6% with respect to drop-in tests. The experimental data discussed in this paper are part of a broader campaign on LGWP refrigerants performance in heat pump systems. The data serve to provide some guidance to the industry and regulatory agencies for the need of future research and developmental work on the next generation of high performance LGWP refrigerants.

1. INTRODUCTION

Concerns about energy security, the threat of climate change and the need to meet growing energy demand pose major challenges to energy decision makers (IEA, 2004). In 2009, the residential and commercial building sectors used 5.74×10^{12} kW-hr (19.6 quadrillion Btu) of delivered energy, that is 21 percent of total U.S. energy consumption. The residential sector for heating, ventilating, and air-conditioning (HVAC) alone accounted for 57 percent of that energy use, leaving 43 percent for the commercial sector (EIA, 2011). It is of high priority for the HVAC industry to address this critical energy challenge by improving the energy efficiencies of AC systems (Moezzi, 2000, Althof *et al.*, 2001, and EPA, 2011) and through the use of new low global warming potential (LGWP) refrigerants, thus reducing the direct and indirect greenhouse contributions for AC and heat pump systems in the short terms.

Refrigerant R410A is a near-azeotropic blend of R32 and R125, with a critical temperature of 72.8°C (163°F) and a critical pressure of 4.86 MPa (705 psi). Its ozone depletion potential is zero and it has been adopted in air conditioning and heat pump systems for residential applications (Pande *et al.*, 1996). R410A has a high volumetric cooling capacity, which means that this refrigerant can absorb significant amount of heat from the air for a unit volume of refrigerant in a direct expansion evaporator. R410A operates at higher pressures than R22 and its GWP is 2,088 (Solomon *et al.*, 2007). Several researchers investigated refrigerants that could potentially retrofit R410A in air conditioning systems. For example R32 has been proposed in mini-split systems, which are popular in China and

Japan (Pham, 2010). R32 has a GWP of 675 (Forster *et al.,* 2007) but its flammability characteristics pose some concerns in case of leakage or in case of failure of the equipment. Natural refrigerants have also been proposed as alternative refrigerants to R410A for heat pump systems. Natural refrigerants have zero ozone depletion potential and minimum global warming potential since these fluids are available in nature. However, they usually require a system designed ad-hoc for the specific application, which make retrofitting of R410A with natural refrigerants difficult, especially in existing equipment (Yin *et al.*, 1998). Few studies on refrigerants that have zero ozone depletion potential and GWP less than 500 are available in the literature. Minor and Spatz (Minor and Spatz, 2008) and McLinden (McLinden, 2011) provided overviews of the objectives of low GWP refrigerants and how these new fluids can be implemented into existing equipment. Some experimental studies for retrofitting R410A in small split systems have been published in the recent years. Developmental refrigerants from various refrigerant manufacturers were retrofitted in existing systems (Leck, 2010) and (Yana Motta *et al.*, 2010) and preliminary findings seemed to suggest that new development refrigerants were viable options. Horie *et al*. (2010) discussed the refrigerant cycle characteristics of R32 and R1234yf with respect to R410A in heat pump applications. The authors highlighted the benefits and shortcomings of these two refrigerants with respect to R410A. A companion paper in this conference described an experimental campaign in which R32 and R1234yf were retrofitted in a R410A heat pump ducted split systems (Barve and Cremaschi, 2012). This paper extends the previous studies from (Leck, 2010) and (Barve and Cremaschi, 2012) to R410A heat pump split system for ducted HVAC residential applications. Two new development refrigerants were investigated in this work and they had GWP ranging from 300 to 500, which is significantly lower than that for R410A. The two new refrigerants are still in the R&D stage and are referred throughout this paper as DR-4 and DR-5 (DR- refer to as developmental refrigerant). These refrigerants had a temperature glide of 5°C (9°F) for DR-4 and 1°C (1.8°F) for DR-5 during phase change from saturated liquid to saturated vapor. They were not toxic, compatible with POE lubricant, chemically stable, not corrosive, and had flammability characteristics of class 2L refrigerants (Leck and Yamaguchi, 2010). One of the major constituent of these two refrigerants was R1234yf.

An analysis of the refrigeration cycle for DR-4 and DR-5 was conducted based on the measured data from this work. While the details will be discussed later in the paper, the refrigeration thermodynamic cycles of DR-4 and DR-5 were drawn next to that for R410A in Figure 1. The diagrams were constructed based on the measured data for each refrigerant when charged into the heat pump unit, which run in cooling mode at A-test cooling conditions. DR-4 had a refrigeration cycle that was similar to that for R410A but shifted toward the lower pressure range. DR-5 cycle was much broader than those for R410A and DR-4. The pressure lift across the compressor was lower for both DR-4 and DR-5 compared to R410A. The pressure ratio, defined as the ratio between the discharge pressure and suction pressure, was about 2.43 and 2.60 for DR-4 and DR-5, respectively, whereas the compressor ratio of R410A was about 2.64. The discharge pressure of DR-4 was lower than R410A discharge pressure by about 570 kPa (83 psi) while DR-5 discharge pressure was 90 kPa (13 psi) lower than the corresponding discharge pressure for R410A at similar operating conditions. The superheat and sub-cooling and the pressure drops in the evaporator and condenser are shown in the *P-h* diagram. The degree of suction superheat was about 5.2°C (9.4°F) for R410A and, by adopting the same TXV, 1.9°C (3.4°F) for DR-4, and 3.5°C (6.3°F) for DR-5. The degree of subcooling at the TXV inlet location was about 5.5°C (10°F) for R410A, 3.2°C (5.7°F) for DR-4, and 6.3°C (11.3°F) for DR-5. The refrigerant flow rates were 101 g/s (810 lb_m/hr) for the unit with R410A, 81 g/s (646 lb_m/hr) for DR-4, and 82 g/s (656 lb_m/hr) for DR-5.

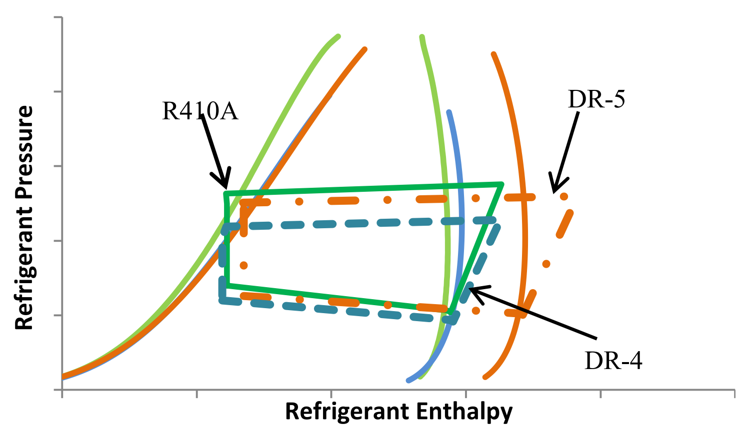

Figure 1: Schematic of the P-h diagrams for R410A, DR-4 and DR-5 during drop-in tests at AHRI A cooling conditions

2. TEST METHODOLOGY

The experiments were conducted in a large scale psychometric chamber at Oklahoma State University (OSU) and the refrigerant cycle pressures and temperatures were measured at design and off-design conditions with outdoor temperature ranging from -8.3°C (17°F) to 46°C (115°F) and in both heating and cooling modes at full load conditions. Additional tests were conducted at extreme high temperature conditions of 43°C and 46°C (110°F and 115°F) to measure the refrigerant condensation pressure and compressor discharge temperature when the unit is exposed to hot climates. These ambient conditions are extreme but often occur during the summer months in the South and Midwest regions of the United States, as well as in the Middle East areas and Southeast Asia.

Charge optimization was conducted for each refrigerant since the refrigerant charge is a key factor for the energy performance of an air conditioning system. Overcharging a system can impair the compressor run during off-design conditions and part load operations. On the other hand studies showed that, a system undercharge by 12 to 19 percent can cause an average reduction of about 13 percent in cooling capacity and about 8 percent in energy efficiency (Kim and Braun, 2010). The charge optimization was conducted at the AHRI 210 A cooling test conditions. Following the same procedures as described in (Barve and Cremaschi, 2012), once the control tolerances were satisfied and steady state conditions were achieved, data were recorded for 1 hour with a sample rate of 2 seconds. The average COPs and cooling capacities were calculated from the data and the refrigerant charge that provided the maximum COP was selected as to the optimum charge for the system. During the charge optimization process the degree of vapor superheat at the compressor suction was constrained to the above of at least 2.2°C (4°F). Then, with the optimum refrigerant charge, the system was run for a broad range of temperatures from -8.3°C (17°F) to 46°C (115°F) and in both heating and cooling modes at full load conditions. More details on the experimental test setup, test procedures, and instrumentation are available in a companion paper (Barve and Cremaschi, 2012) and details on the psychrometric facility can be found in (Worthington *et al.*, 2011). The uncertainty analysis of the measurements showed that the measured capacities and COPs had an experimental uncertainty of 3% and 4%, respectively.

The experimental campaign in this work focused on highlighting the direct drop-in replacement performance of DR-4 and DR-5. A series of experiments were carried out with the TXVs that were originally installed in the unit for R410A. These tests are referred to as drop-in tests. The cycle thermodynamic points and flow rates were measured with outdoor temperature ranging from 27.8ºC (82ºF), referred as B-test in the AHRI standards (AHRI, 2010), to 46.1ºC (115ºF). Extreme outdoor temperature of 43.3ºC (110ºF) is referred to as HT1-test and 46.1ºC (115ºF) is referred to as HT2-test throughout this paper (HT- refer to High Temperature). In heating mode of the unit, tests were conducted at three different outdoor conditions: H1-test of 8.3ºC DB/ 6.1ºC WB (47ºF /43ºF), H2-test of 1.7ºC DB/ 0.6ºC WB (35ºF /33ºF) and H3-test of -8.3ºC DB/-9.4ºC WB (17ºF/ 15ºF), with the indoor temperature at 21.1ºC DB (70ºF) for all the tests. H2 was a frost-defrost test, and the periodic cycle performances were recorded after at least 6 frost-defrost cycles of the unit. Average integrated capacity and EER were calculated from the transient data of the unit operating under frost and defrost conditions. Additional tests were conducted to investigate the potential performance of the system with the new refrigerants when minor adjustments to the thermal expansion valve were implemented. These tests are referred in this paper as tests with soft optimization of the TXV. The TXV on the indoor coil was replaced by a manual expansion valve that served to actively control the degree of superheat at the compressor suction and to set the high side and low side saturation pressures for the new refrigerants when the unit was in cooling mode. Several tests were required to optimize the refrigerant charge with the modified expansion valve. For each charge the opening of the expansion valve was varied in a parametric fashion in search of the maximum COP at similar capacities or of the maximum capacity at similar COPs. The system performance with modified expansion valve were measured for cooling mode only and with outdoor temperature ranging from 27.8ºC (82ºF) to 46.1ºC (115ºF).

3. HEAT PUMP SYSTEM SPECIFICATIONS

The heat pump used for the experiments was a R410A residential split system heat pump with a rated capacity of 17.6 kW (5 ton). The unit was commercially available off-the-shelve in the US market and it had fin-and-tube outdoor coil, an A-shape fin-and-tube indoor coil, constant speed indoor blower and constant speed outdoor fans, and constant speed fixed capacity hermetic compressor. The details of the unit and of the test set up are described in a companion paper (Barve and Cremaschi, 2012).

4. RESULTS AND DISCUSSION

The discussion of the experimental findings is organized in five sections: charge optimization, system capacity and performance, compressor discharge temperature, compressor volumetric efficiency, and compressor thermal efficiency.

4.1 Refrigerant Charge Optimization

Figure 2 shows the results of COP and degree of superheat from the tests during the charge optimization of DR-4 with the manual expansion valve installed at the indoor coil of the unit. The data are presented in normalized form, in which the COP for R410A at similar AHRI A cooling conditions was chosen as reference. The COP data are plotted versus the compressor pressure ratios, Pr, which were normalized with respect to the compressor pressure ratio for R410A. For example, for 7.0 kg (15.5 lb_m) of DR-4 charged into the unit, a point was measured and it is highlighted with an arrow in Figure 2. For this point the normalized pressure ratio was 0.935 and the normalized COP was about 1.04. This means that the pressure ratio for DR-4 was 0.935 lower than that for R410A while the COPs of the two refrigerants were similar when the unit ran at A-test cooling conditions. For 7.0 kg (15.5 lb_m) of DR-4 charged into the system there was an optimum opening of the expansion device that provided the maximum COP. Increasing the refrigerant charge of DR-4 into the system yielded to similar COPs at various pressure ratio when the expansion valve was properly adjusted for each charge. The corresponding measurements of the degree of superheat at the compressor suction during the tests for DR-4 charge optimization in the unit with manual expansion valve are plotted in Figure 3. A superheat of about 2.2ºC (4ºF) was the minimum superheat for all refrigerants acceptable for compressor safe operation. It should be noticed that the charge of DR-4 in the unit during straight drop-in tests was 6.8 kg (15 lb_m). This was the optimum charge of DR-4 when the original TXV of the unit was present and it was the initial amount of DR-4 charged in the unit during the tests for the optimization of the expansion valve.

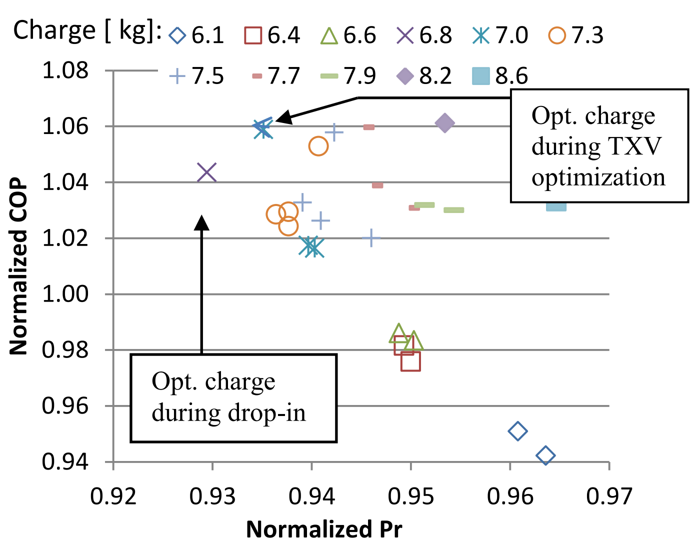

Figure 2: COP vs. Pressure ratio for DR-4 TXV optimization at AHRI A cooling conditions

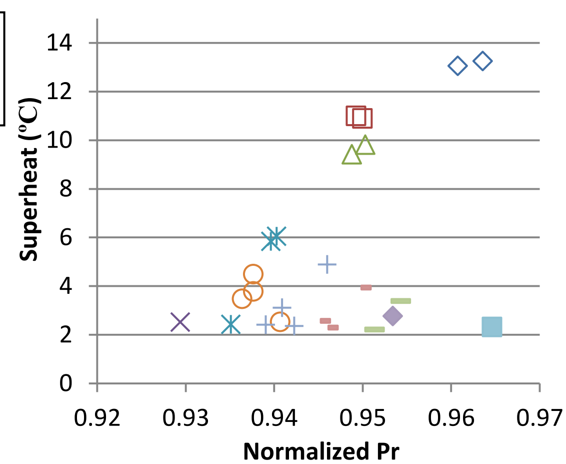

Figure 3: Superheat vs. Pressure ratio for DR-4 TXV optimization at AHRI A cooling conditions

Figure 4 shows the results of the charge optimization for DR-5, with the normalized COP data plotted against the normalized pressure ratio and with the values of R410A chosen as reference. The maximum COP is indicated in the figure with an arrow and it had a normalized pressure ratio of 0.983 and a normalized COP of 1.03 for 7.9 kg (17.5 lb_m) of DR-5 charged into the unit. Once the refrigerant charge was varied and the expansion valve was promptly adjusted, it was observed that this charge yielded to the highest COP, and thus it was chosen as the optimum charge of DR-5 in the unit with the new expansion valve. It should be noticed that the charge of DR-5 in the unit during straight drop-in tests was 8.4 kg (18.5 lb_m). This was the optimum charge of DR-5 when the original TXV of the unit was present. For DR-5, the degree of superheat was fairly sensitive to the adjustments of the expansion device. As shown in Figure 5, small variations of the needle in the expansion valve yielded to drastic change in the degree of superheat at the compressor suction.

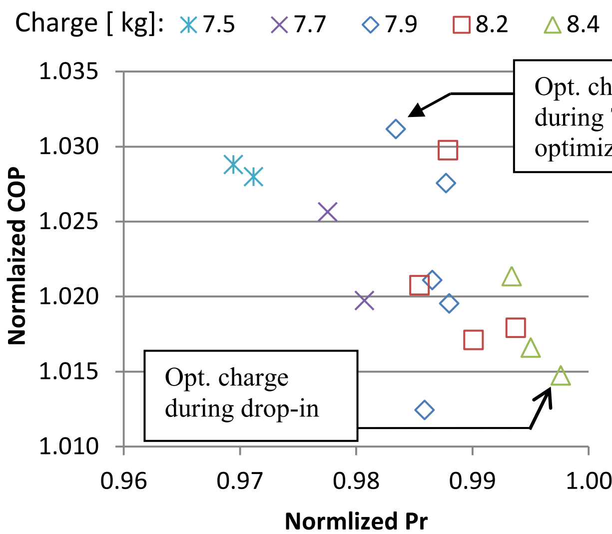

Figure 4: COP vs. Pressure ratio for DR-5 TXV optimization at AHRI A cooling conditions

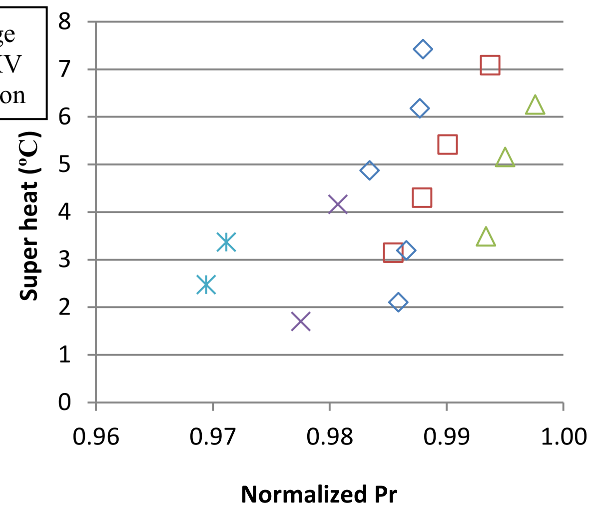

Figure 5: Superheat vs. Pressure ratio for DR-5 TXV optimization at AHRI A cooling conditions

4.2 System Capacity and Performance during straight drop-in tests

Two TXVs were present in the unit: one at the indoor coil inlet for cooling mode operation and the second one at the outdoor coil outlet for heating mode. For the straight drop-in tests, the TXVs of the R410A unit were not modified and tests were conducted with the unit run in both cooling and heating modes. Since the unit receiver was also not modified, charge management of DR-4 and DR-5 was a challenge for the broad range of outdoor temperatures investigated in this work. The original TXV in the system was designed for R410A and it controlled the evaporator capacity such that the compressor suction superheat was about 5.5ºC (10ºF) for the all outdoor temperatures. For DR-5 in cooling mode the original TXV of the system performed well and it was able to guarantee sufficient superheat at the compressor suction. For heating mode, 0.9 to 1.6 kg (2 to 3.5 pounds) of DR-5 had to be taken out from the system to guarantee enough degree of superheat at the compressor suction during the frost/defrost cycles and at very low temperature. This means that the TXV for R410A worked well for DR-5 during cooling mode but the TXV for the heating mode was too large when DR-5 was retrofitted to R410A in this unit. With the tests using DR-4, the refrigerant charge needed to be reduced for the cooling mode at extreme high temperatures and in heating mode for the very low temperatures.

During the straight drop-in tests, the capacities and COPs for DR-4 and DR-5 are shown in Figures 6 and 7, normalized with respect to that for R410A. An increase of 3 to 4% in cooling capacity during cooling mode and a decrease of 5 to 10% in heating capacity during heating mode were observed for DR-5 with respect to R410A. The COP of the unit with DR-5 was from 1 to 7% higher in cooling mode and from 1 to 22% higher in heating mode. Thus, DR-5 performed well when used for retrofitting R410A in the heat pump split system for ducted residential applications used in the present work. In heat pump mode, while the COPs were higher than R410A the capacity of the system with DR-5 was lower but it could be increased by adjusting the TXV, as it will be discussed later in regard to Figure 9. For DR-4, which has the lowest GWP among R410A and DR-5, the straight drop in tests showed that the cooling capacity decreased by about 15 to 18% in cooling mode and by as much as 30% in heating mode with respect to R410A. The COP of the unit with DR-4 was from 4 to 6% higher in cooling mode and from 11 to 16% higher in heating mode.

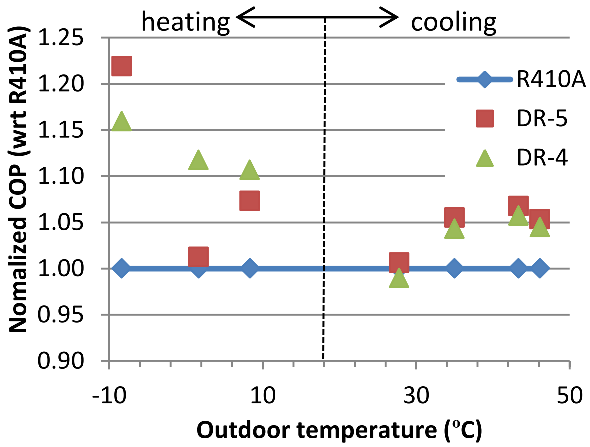

Figure 6: System performance in drop-in testing

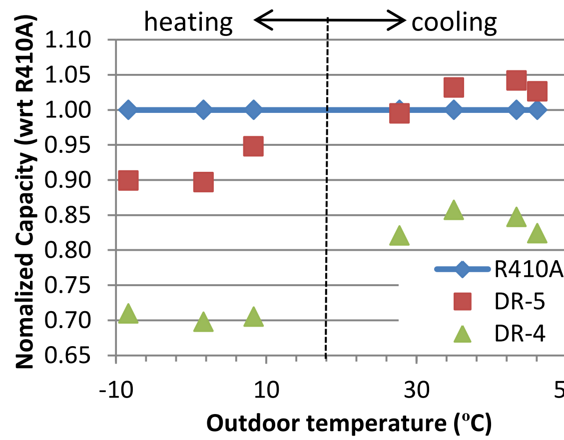

Figure 7: System capacity in drop-in testing

With a soft optimization of TXV, the capacity and the COP of DR-4 and DR-5 were further improved with respect to the ones measured during the drop-in tests. These improvements are shown in Figures 8 and 9. The data in solid bars represent the cooling capacities and COPs during drop-in tests. For example, at AHRI A cooling conditions the COP of DR-4 was about 4% higher with respect to R410A and the cooling capacity was about 15% lower. When the TXV and the corresponding refrigerant charge were further optimized, DR-4 COP increased by an additional 2% and DR-4 cooling capacity augmented by an additional 6%. This soft optimization of the unit increased the capacity from 0.85 during the drop-in test to 0.91 for the run with TXV soft optimization test. These results are shown in the bar referred to as "A" for DR-4 in Figures 8 and 9. Considering the range of outdoor temperatures in cooling mode, Figures 8 and 9 showed that optimization of the TXV overall increased the cooling capacity of DR-4 from 5 to 8% and the COP from 2 to 6% with respect to that of drop-in tests. For DR-5 it was observed that the manual expansion valve was less efficient than the TXV in the system at design cooling conditions. This could be observed from the values measured for the A-test in Figures 8 and 9 for the case of DR-5. The drop-in values were slightly higher than the values obtained with the TXV optimization leading to the conclusion that the TXV for R410A was already well suited to work with DR-5 at design cooling conditions (A-test). At B-test conditions and at very high extreme outdoor temperature (HT-2) the adjustments of the expansion valve produced additional 1 to 3% higher COPs with respect to those of drop-in tests. The capacity variations between drop-in tests and TXV optimization tests for DR-5 were practically within the experimental uncertainty of the test set up.

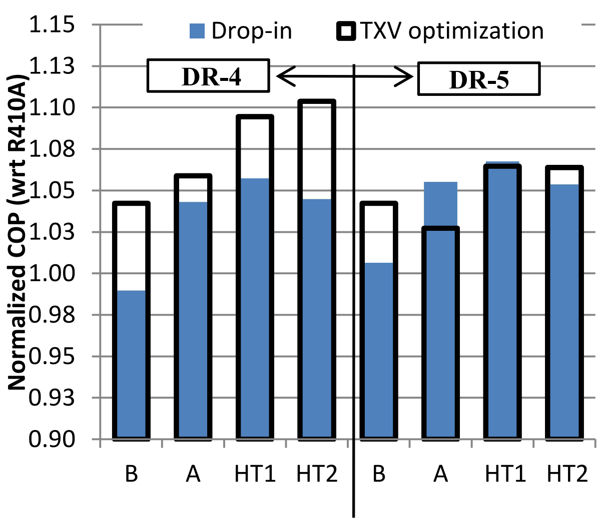

Figure 8: System performance in TXV soft optimization for DR-4 and DR-5

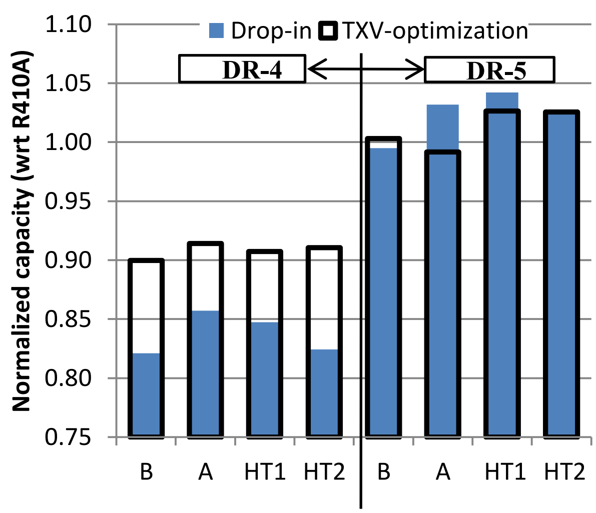

Figure 9: System capacity in TXV soft optimization for DR-4 and DR-5

4.3 Compressor Discharge Temperature

The compressor discharge temperature has a direct impact on the compressor reliability and life time since high discharge temperatures yield to metal fatigue of the valves and thermal stress of the lubricant (Leck, 2010). Figure 10 shows that during drop-in tests DR-5 exhibited a slight increase of the discharge temperature from 3 to 5ºC (5.4 to 9ºF) with respect to R410A. DR-4 had a lower discharge temperature of about 5 to 9ºC (9 to 16.2ºF) in comparison to that of R410A. Figure 11 shows the compressor discharge temperature during the soft optimization tests of the TXV. For both refrigerants an increase of COP and capacity was accompanied by an increase of the compressor discharge temperatures but the two refrigerants showed different magnitudes. DR-5 yielded to an increase of discharge temperature up to 8ºC (14.4ºF) while even if DR-4 discharge temperature increased it was always lower than that of R410A by about 2 to 6ºC (3.6 to 10.8ºF).

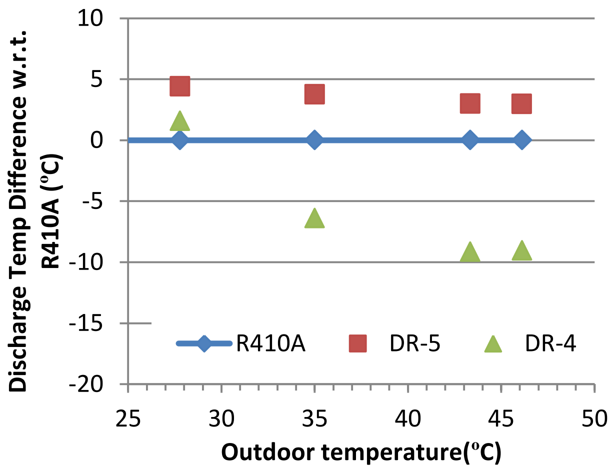

Figure 10: Discharge Temperature (Drop-in tests)

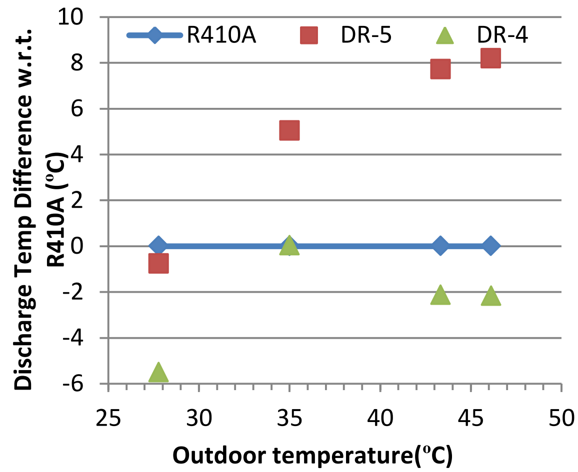

Figure 11: Discharge Temperature (TXV-optimization tests)

4.4 Compressor Volumetric Efficiency

Volumetric efficiency of the compressor was calculated as given in equation (1),

$$\eta_v = \frac{Actual\ measured\ mass\ flow\ rate}{Ideal\ mass\ flow\ rate} = \frac{Actual\ measured\ mass\ flow\ rate}{(suction\ density) \cdot \dot{V}_{comp}} \qquad (1)$$

Where $\dot{V}_{comp}$ is the compressor volumetric capacity and it was estimated from the manufacture data. The suction density was calculated from the data of pressures and temperatures measured at the suction port. Volumetric efficiency takes into account the effect due to refrigerant vapor re-expansion in the clearance volume, pressure drop across suction and discharge valves and superheating of the colder vapor being in contact with hot compressor metal surfaces. Figure 12 shows the normalized volumetric efficiency for the DR-4 and DR-5 refrigerants with respect to R410A for the drop-in cooling tests and Figure 13 represents the same quantities for the TXV soft optimization cooling tests for the entire range of outdoor temperatures. DR-5 yielded to a 2% increase in volumetric efficiency with respect to that of R410A for the drop-in tests and an increase of 1 to 3% for the TXV soft optimization tests. DR-4 volumetric efficiency was 3 to 6% lower for both the drop-in and TXV soft optimization tests.

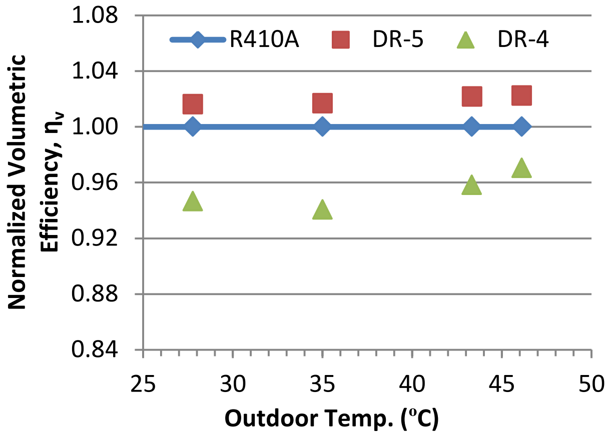

Figure 12: Volumetric Efficiency (Drop-in test)

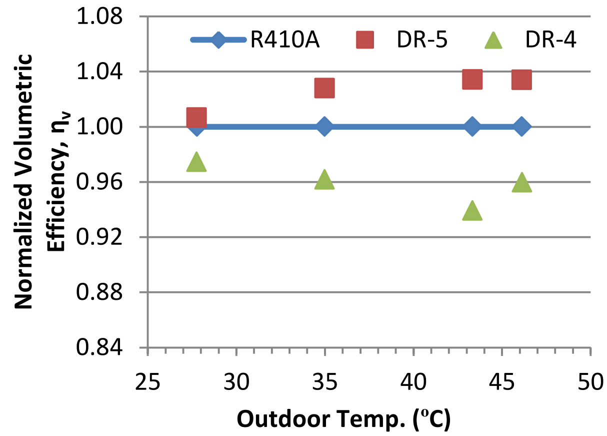

Figure 13: Volumetric Efficiency (TXV-optimization)

4.5 Thermal Efficiency of Compressor

Thermal efficiency of the compressor was defined as shown in equation (2),

$$\eta_T = \frac{Isentropic\ work\ of\ compressor}{Actual\ work\ done\ by\ compressor} \tag{2}$$

Where the isentropic work was calculated from the measurements of suction temperature and pressure and discharge pressure and the actual work was calculated based on the compressor suction and discharge temperatures and pressures. It should be noted that the discharge pressure and temperature sensors were located on the refrigerant discharge line after the 4-way. The distance between these sensors and the compressor discharge port was about 0.6 m (2 ft) of pipeline. The 4-way valve and the refrigerant pipelines were well insulated to prevent heat losses to the ambient. However, some heat exchange was expected to occur between the hot vapor in the discharge line and the cold vapor in the suction line when the refrigerant crossed the 4-way valve. Figure 14 shows that for DR-5 there was a drop of 10 to 15% in thermal efficiency compared to R410A and DR-4 had 13 to 20% lower thermal efficiencies during the drop-in tests. This could be due to the different magnitude of the heat exchanged in the 4-way valve, which could affect the values of the actual discharge temperatures read from the discharge temperature sensor. Figure 15 shows the normalized thermal efficiencies during the TXV soft optimization tests. The compressor experienced a drop of thermal efficiency from 9 to 12% for DR-5 and from 14 to 23% for DR-4. While these data are still preliminary they indicate that an optimization of the TXV yields to higher thermal efficiencies, higher volumetric efficiency and slightly higher discharge temperatures.

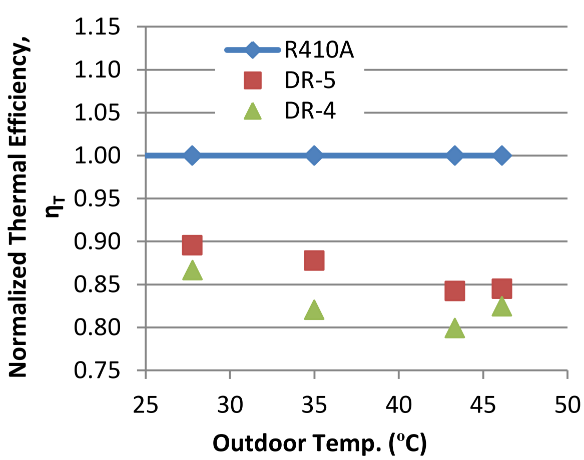

Figure 14: Thermal Efficiency (Drop-in test)

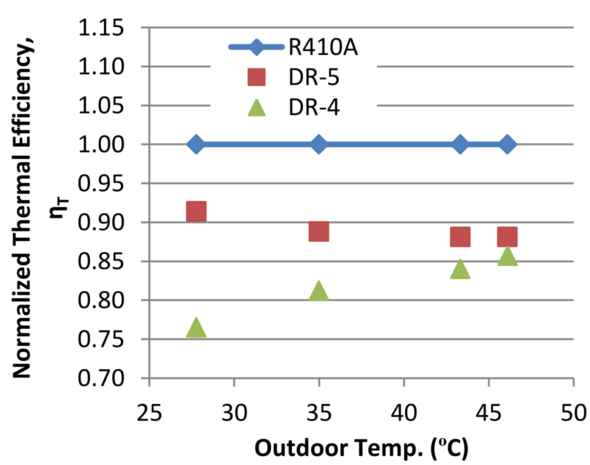

Figure 15: Thermal Efficiency (TXV-optimization)

5. CONCLUSIONS

This paper presents a study that breaks new ground on LGWP developmental refrigerants and focuses on AC systems for residential applications. A 17.6 kW (5 ton) AC ducted split unit, originally designed for R410A and commercially available off-the-shelf, was retrofitted with two new developmental refrigerants that have GWP ranging from 300 to 500, which is significantly lower than that of R410A.The following conclusions can be drawn from the discussion above:

- The new refrigerant DR-5 and DR-4 had up to 7% and up to 6% higher cooling COPs than R410A, respectively. These two refrigerants had 22% and 16% higher heating COP than R410A, respectively. The optimization of the expansion valve could improve further the COPs of these two refrigerants when the unit operates at design and extreme high temperature conditions.
- With proper charge management, DR-5 had up to 4% improvement in cooling capacity than R410A. The heating capacity was about 10% lower in comparison to R410A. DR-4 had 18% lower cooling capacity and 30% lower heating capacity when compared to R410A. By conducting an optimization of the expansion valves the drop in capacity was partially mitigated.
- The system with DR-5 had an improved compressor volumetric efficiency with an increase up to 4% with respect to R410A, while DR-4 had lower volumetric efficiency by about 6%.
- The compressor discharge temperatures and pressure of DR-5 were similar to those of R410A while DR-4 had significant lower discharge pressures and lower discharge temperatures than those for R410A. This was due to lower saturation pressures of DR-4 during condensation and evaporation processes in direct-expansion equipment.

NOMENCLATURE

CFCs	Chlorofluorocarbons
COP	Co-efficient of Performance
DR	Developmental Refrigerant
EER	Energy Efficiency Ratio
GWP	Global Warming Potential
LGWP	Low Global Warming Potential
HT	High Temperature
POE	Polyolester
Pr	Compressor pressure ratio
TXV	Thermal Expansion Valve
$\dot{V}_{comp}$	Compressor volumetric capacity
η_v	Volumetric Efficiency
η_T	Thermal Efficiency
ΔPevap	Pressure drop across evaporator

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