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R32 And HFOs As Low-GWP Refrigerants For Air Conditioning

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ABSTRACT

This paper will review the status worldwide on technical and policy search for next-generation refrigerants with both low Global Warming Potential (LGWP) and low Life Cycle Climate Performance (LCCP) with particular focus on R410A replacement for unitary A/C & H/P. R32 and the HFO blends offer potential solutions but all involve trade-offs among Global Warming Potential (GWP), efficiency, safety, and cost. With the U.S. mandating new higher regional efficiency standard taking effect January 2015, there is even more pressure in finding a Low-GWP refrigerant solution that is affordable and can sustain efficiency and reduce charge requirement. This paper focuses more on R32 as the available data for HFO blends is limited. Theoretical analyses and test results from various drop-in system tests with different heat exchangers and scroll compressors will be presented comparing R32 and HFO blends versus R410A. LCCP, system economics, compressor discharge temperature envelopes for typical US applications, as well as A2L flammability safety aspects will be discussed relative to the HFOs.

Key words: R32, HFO, A2L, GWP, LCCP, drop in, discharge temperature

1. INTRODUCTION

Since the announcement of the Kyoto Conference in 1997 and the Europe F-Gas regulation, further regulations have been proposed globally to amend the Montreal Protocol to gradually phase-down the production of HFCs in the next 20 years due to their high GWP. Australia has imposed a carbon tax of $50/kg for R410A. Figure 1 shows the North American Proposal (NAP) for both developed and developing countries that was suggested by the United States, Canada and Mexico. However, there has been no agreement from developing countries such as China and India where HCFC-22 is still the predominant refrigerant which is already subject to phase-out in the next few years under the Montreal Protocol for Ozone Depletion Potential (ODP). In the residential air conditioning application, global research efforts have been performed to search for substitutes for HFC-410A which is under pressure due to its GWP of 2,088. Adding to the global environmental pressure is also the simultaneous push globally for higher minimum efficiency standards to minimize the indirect carbon emission from power plants.

Figure 1: North American Proposal (NAP)

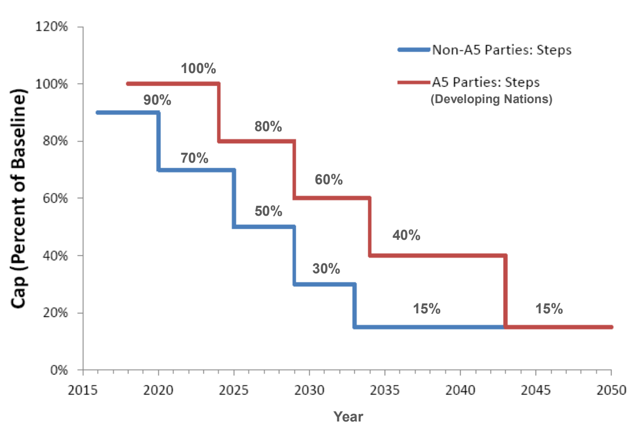
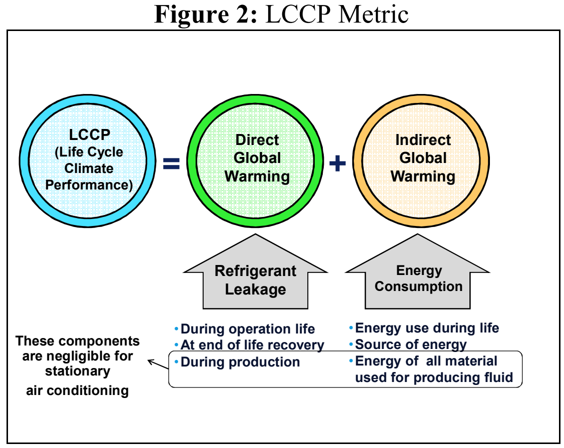


Figure 2: LCCP Metric



China has been investigating HFC-32 which has a GWP of 675 (68% reduction over R410A) due mainly to its cost being lower than R410A while the U.S. and others have also been investigating blends of HFC with HFO-1234yf or HFO-1234ze. HFC-32 is not a new refrigerant - it was studied during the 1990s in the search for a zero-ODP solution but was not adopted due to concerns about its flammability characteristics. But now, the requirement for Low-GWP likely requires accepting some flammability constraints for the refrigerants. HFC-32 and HFOs are both expected to be mildly flammable with an ASHRAE A2L flammability safety rating and involve significant tradeoffs among efficiency, GWP, and cost. The Air Conditioning, Heating and Refrigeration Institute (AHRI) has launched since July 2011 a global Low-GWP Alternative Refrigerants Evaluation Program (LGWP AREP) to pool industry resources for system and compressor test evaluations involving over forty LGWP candidates for various applications. Most of these candidates are either pure HFC-32 or blends with HFC-32, HFO-1234yf or HFO-1234ze.

Although regulation efforts are focused on the GWP metric due to the simplicity of relating to the direct emission impact of the refrigerant, it is actually more important to recognize the indirect emission from the efficiency impact of the refrigerant which can be more overwhelming than its direct emission, particularly for applications such as stationary A/C where the direct emission is typically no more than 5% of the total emission. As illustrated in Figure 2, the concept of using Life Cycle Climate Performance (LCCP) has been widely recognized globally as a better metric for weighing both the direct and indirect effects. This paper will review further the merit of R32 versus other candidates in the context of both GWP and LCCP, ability to reduce charge and to sustain efficiency and affordability.

2. PROPERTIES COMPARISON - R32 vs. R410A

The relative merits of R32 can be summarized based on a comparison of theoretical properties as shown in Table 1 :

- considerably lower refrigerant cost than R410A and potentially better affordability
- available now in high volumes globally since it is 50% of R410A composition
- 8% higher critical temperature, better performance at higher ambient conditions
- similar pressure and pressure ratio, a close drop-in replacement for R410A without major system redesign
- 9% lower liquid density, lower system charge requirement
- 28% lower vapor density and lower system mass flow rate, about 50% lower pressure drop expected
- higher volumetric capacity despite the 28% lower mass flow due to 43-50% higher latent heat
- 41% higher liquid thermal conductivity, higher heat transfer coefficient at same mass flux
- No glide and potential to optimize heat exchanger with smaller tube volume for further charge reduction

The disadvantages are cited below:

- A2L mild flammability rating (difficult to find a Low-GWP A1 non-flammable fluid)
- higher compressor discharge temperature from higher vapor specific heat
- New oil likely required since existing polyolester (POE) oil is not miscible with R32

Overall, R32 seems to offer more advantages than disadvantages. Its lower cost provides incentive for investing development time for mitigating its disadvantages through compressor and system design optimization.

Table 1: Properties of R32 vs. R410A at 110°F condensing and 50°F evaporating conditions (Refprop8)

Parameter	**Unit**	**R410A**		**R32**	
Critical Temperature	ºF	158.9		172.6	
Temperature	ºF	<u>110</u>	<u>50</u>	<u>110</u>	<u>50</u>
Saturated vapor pressure	psia	380.0	157.3	389.6	160.5
Liquid density	lbm/ft^3	59.5	70.4	54.7	63.6
Vapor density	lbm/ft^3	7.14	2.617	5.043	1.887
Latent heat	Btu/lbm	65.3	89.7	98.3	128.6
Liquid specific heat	Btu/(lbm·R)	0.483	0.377	0.535	0.431
Vapor specific heat	Btu/(lbm·R)	0.475	0.295	0.509	0.327
Liquid thermal conductivity	Btu/(hr.ft·F)	0.0457	0.0563	0.0648	0.0794
Liquid viscosity	lbm/ft-s	5.12E-05	9.59E-05	6.1E-05	0.9E-05
Vapor viscosity	lbm/ft-s	1.02E-05	0.8E-05	0.9E-05	0.8E-05

Figures 3 and 4 show respectively the relative comparisons of theoretical capacity and energy efficiency ratio (EER) of R32 over R410A at various evaporating (Te) and condensing (Tc) conditions based on standard 20ºF superheat and 15ºF liquid subcooling per AHRI 540 standard assuming equal volumetric and overall isentropic efficiency. As can be seen, R32 has theoretically 3-14% higher capacity in the A/C operating range (45-55°F Te) and 7-16% higher capacity in the H/P operating range (-10° - 30°F Te). Correspondingly, R32 has theoretically -1% to +5% higher EER in the A/C cooling operating range and 0% to 7% higher EER in the H/P heating operating range, thus little more favorable for H/P than A/C operation. The relative gains increase with lower Te and higher Tc.

Figure 3: Relative theoretical capacity R32 vs. R410

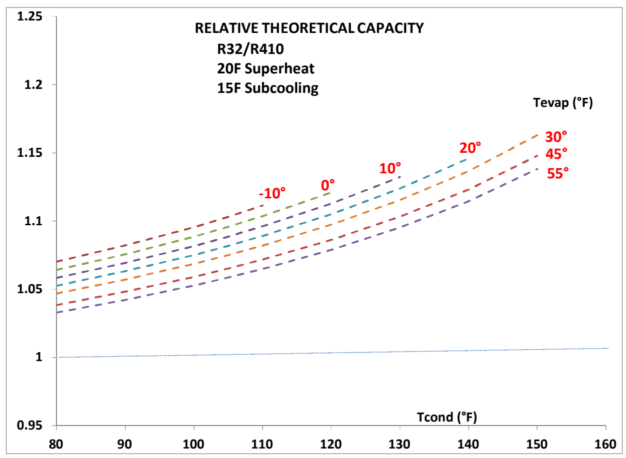
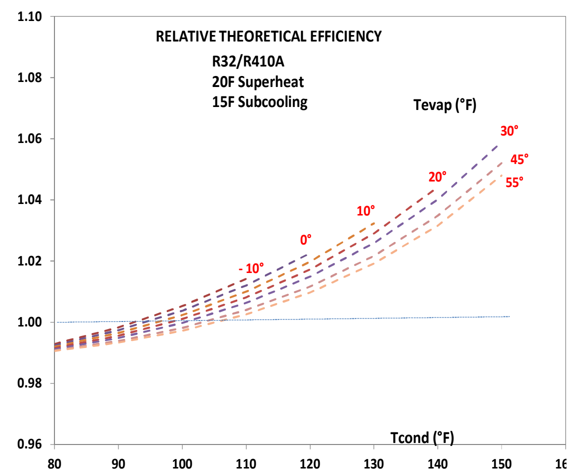


Figure 4: Relative theoretical EER R32 vs. R410



3. COMPRESSOR TEST COMPARISON - R32 vs. R410A

The relative comparison above is based on theoretical properties only. Practical compressor performance depends on actual volumetric and isentropic efficiencies. Figures 5 and 6 show actual relative capacity and EER respectively for a 3-ton scroll compressor at three A/C and two H/P Te/Tc conditions typically encountered in system performance rating tests compared to theoretical at same 20 ºF superheat and 15 ºF liquid subcooling.

Figure 5: Actual vs. theoretical compressor capacity

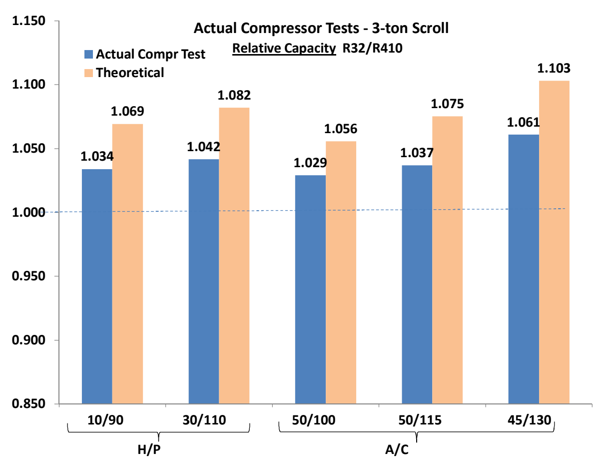
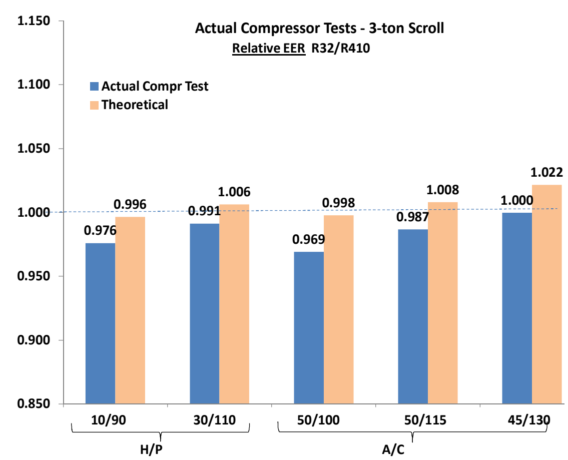


Figure 6: Actual vs. theoretical compressor EER



As can be seen, the actual relative compressor capacity is on average about 3-4% lower than theoretical due to volumetric efficiency reduction from the higher heat of compression due to its higher vapor specific heat as well as its 28% lower mass flow which culminate in its higher discharge temperatures as shown further in Figures 7. Correspondingly, the actual relative compressor EER is on average about 2-3% lower than theoretical due to lower overall isentropic efficiency. It is expected to get worse at higher compression ratios conditions. It should be pointed out that these tests were conducted using an R410A optimized compressor and not a R32 optimized product.

Figure 7: Discharge temperature R32/R410A

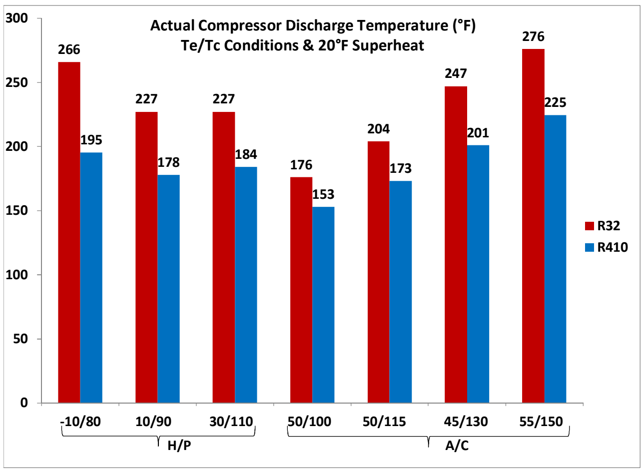


Figure 8: Discharge temperature correlation

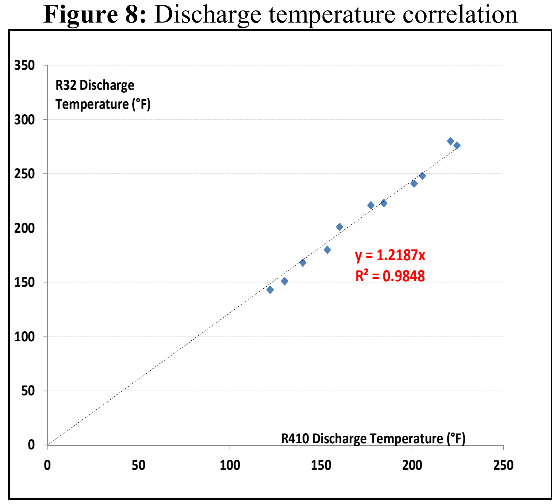
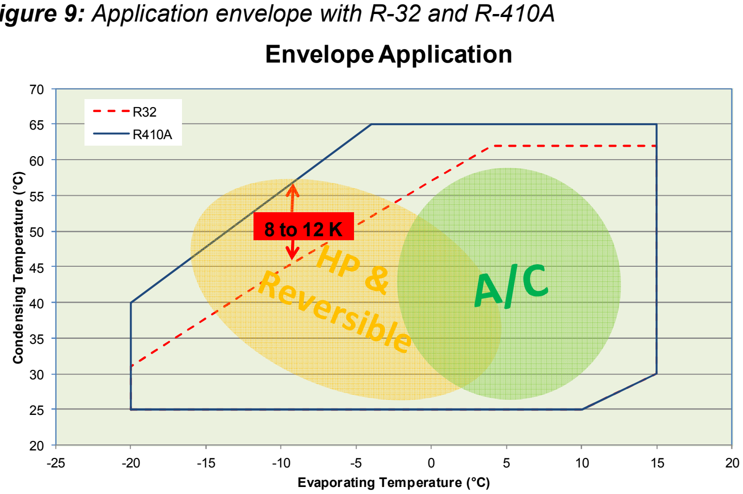


Figure 8 shows the compressor discharge line temperature (DLT) which is generally about 22% higher with R32 than R410A. This implies that the Te/Tc envelope limits for R32 would be less than R410A for the same maximum allowable discharge temperature limit which is currently about 300°F. Options for reducing the impact of compression heat include 1) reducing the entering suction superheat through better system flow control such as Electronic Expansion Valve (EXV), 2) optimizing compressor internal design, 3) employing compressor Vapor Injection (VI) or Liquid Injection (LI) cycle, or 4) improve the oil to enable higher maximum allowable discharge temperature. The saving from R32 lower refrigerant cost enables deploying these options to still provide cost effective system solutions. Which options to use depend on the particular Te/Tc envelope requirement for the applications as shown in Figure 9. For US residential 14+SEER air-to-air A/C and H/P systems as required by DOE 2015 regional standards, the envelope is less challenging and options 1) and 2) are likely adequate. For air-to-water reversible hydronic H/P such as those in Europe where much higher Tc is required for heating water, the addition of option 3) is likely needed.

Figure 9: *Application envelope with R-32 and R-410A*



4. SYSTEM DROP-IN TEST COMPARISON

4.1 Test Conditions

The system test conditions followed AHRI 210/240 Standard as shown in Table 2. Two tests, A (95°F ambient) and B (82°F ambient) were conducted for cooling mode, and two tests High Temp2 (47°F ambient) and Low Temp (17F ambient) were conducted for heating mode. The purpose of this experimental study is to investigate the drop-in system performance difference with only adjusting the thermostatic expansion valve (TXV) or optimizing system charge. The test system is a 3-ton heat pump with a scroll compressor designed for R410A with TXV for cooling and heating mode, and 3/8" tubing for the outdoor and indoor heat exchangers. Both R410A and R32 systems were charged to about 6ºF subcooling at test A condition which results in about 15% lower charge with R32.

Table 2: Test conditions

Test	Indoor			Outdoor				Operation
	DB	WB	RH	DB	WB	RH	DP	
Extended condition	26.7°C (80°F)	19.4°C (67°F)	50.66%	46.1°C (115°F)	NA	NA	NA	Steady State Cooling
A				35.0°C (95°F)				Steady State Cooling
B				27.8°C (82°F)				Steady State Cooling
C		≤13.9°C (≤ 57°F)	≤21.41%					Steady State Cooling, dry coil
D								Cyclic Cooling, dry coil
High Temp2	21.1°C (70°F)	≤15.6°C (≤ 60°F)	≤56.42%	8.3°C (47°F)	6.1°C (43°F)	72.9%	3.7°C	Steady State Heating
High Temp1				16.7°C (62°F)	14.7°C (58.5°F)	81.1%	13.4°C	Steady State Heating
Low Temp				-8.3°C (17°F)	-9.4°C (15°F)	69.8%	-12.3°C	Steady State Heating
High Temp Cyclic				8.3°C (47°F)	6.1°C (43°F)	72.9%	3.7°C	Cyclic Heating
Frost Acc.				1.7°C (35°F)	0.6°C (33°F)	82.0%	-0.9°C	Steady State Defrost
Extended condition				-17.8°C (0°F)	NA	NA	NA	Steady State Heating

4.2 Drop-In System Performance Comparisons - R32 vs. R410A

Figure 10 shows the changes in system capacity and EER at the two A/C and two H/P ambient conditions tested. The 3-4% capacity gain and 1-1.5% EER reduction from drop-in system comparison seem to correlate reasonably close to the compressor performance data as shown previously in Figure 5 and 6. The system EER can be about parity with R410A if downsizing the compressor displacement about 5% to equalize system capacity. The heating performance is slightly more favorable than cooling as expected from theoretical analysis. Figure 11 shows the changes in Te/Tc ratio and suction superheat from the same four tests. Despite getting higher capacity, R32 Te is still 0.3-1.6°F higher than R410A due to its higher evaporating heat transfer and lower pressure drop. Correspondingly, the Tc increases slightly due to the higher capacity and higher DLT heat loading on the condenser. However, the suction line superheat is noted to be 2-5°F higher with R32 due to its 28% lower mass flow which would penalize A/C system performance slightly.

Figure 10: Drop-In R32/R410A system performance

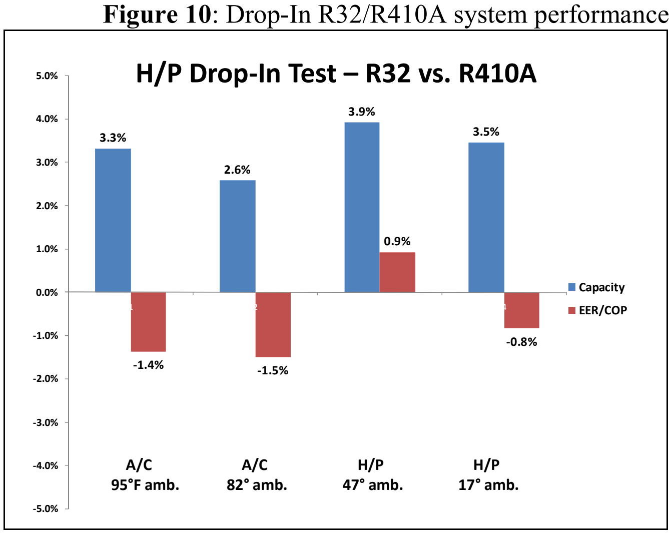
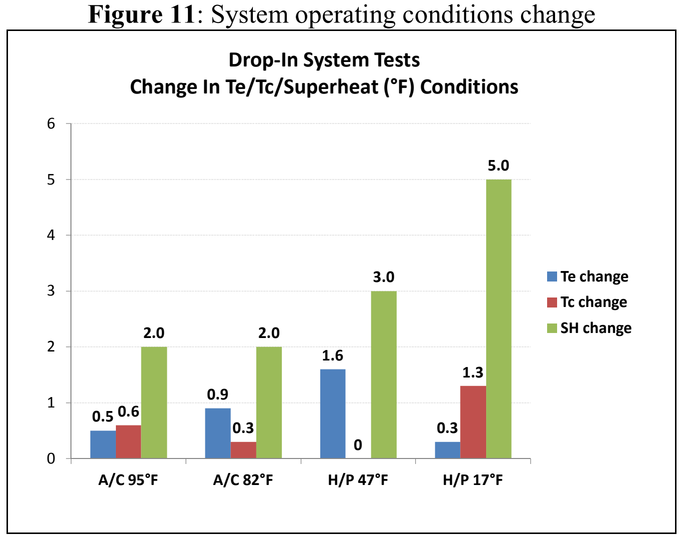


Figure 11: System operating conditions change



4.3 Drop-In System Performance Comparison - R32 vs. HFO Blends

As mentioned in the introduction section, several blends of HFO-1234yf and HFO-1234ze are also Low-GWP candidates as the R1234yf or R1234ze component of these blends has ultra low GWP of 4 and 6 respectively. Previous literature [3] has reported that pure R1234yf or R1234ze is targeted as an R134a alternative, thus not efficient as an R410A alternative from a system performance standpoint without substantial redesign with larger compressor, piping and heat exchangers and added cost to compensate for the high pressure drop. Blending R1234yf or R1234ze with HFCs like R32 offers new alternatives as reported in recent literature [5,6]. However, a main concern with these HFOs is that they are considerably more expensive than R410A [10] and their long-term cost position is yet to be defined.. These HFO blends are still proprietary in development stage and their properties are not yet publicly available.

Figure 12 shows a drop-in system performance comparison of R32 versus three HFO blends with GWP around 500 in the same system described above at those same four test conditions with only TXV and charge adjustments allowed. Plotting the relative capacity and EER of the alternatives versus R410A on the x- and y- axes respectively enables a quick visual comparison as typically a +5% capacity can be traded away for about +2% higher EER by downsizing compressor displacement for same capacity as shown by the diagonal line. The upper right quadrant is the first desired zone (both higher capacity and higher EER) and the upper left quadrant is a close second. As can be seen, R32 performs reasonably close on a drop-in basis without much compressor and system optimization. The three HFO blends are more spread to the upper left quadrant and having more significant drop in H/P performance compared to R32 except for HFO #1 which could be a close drop-in for R410A. These HFO blends are subject to more optimization changes in the future and the results could change.

Figure 13 shows further system performance comparison from additional testing at the two A/C A and B cooling conditions using another 3-ton condensing unit with compact microchannel heat exchanger (MCHX) which enables significant 30-40% charge reduction, an important system design consideration, particularly for reducing the high cost of HFO refrigerant as well as further mitigating the risk with A2L mild flammability. However, it is interesting to note that R32 shifts further to the upper right quadrant while the other two HFO#1 and HFO#3 shift further to the upper left quadrant which suggests R32 is more optimized for smaller tubes (MCHX, 7mm, 5mm) while the HFO blends are likely better with larger 3/8" tubes due to their higher pressure drop. It was also noted that the system charge for these HFO blends is about same as R410A while R32 is 15% lower. This brings the question of how to integrate efficiency, charge, GWP into one comparison and the LCCP is the industry-recognized overall metric for that purpose.

Figure 12: System performance R32 vs. HFO blends

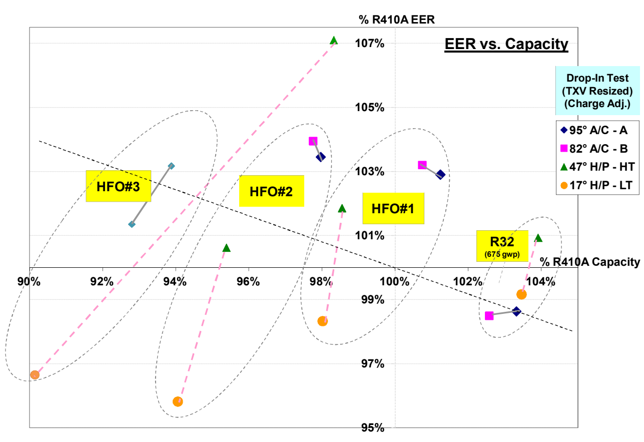
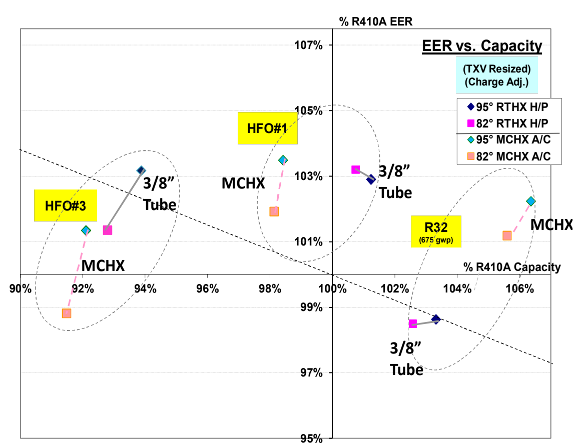


Figure 13: Impact of low-charge condenser



7. LCCP AND OVERALL ASSESSMENT

Figure 14 shows an LCCP analysis for a R410A 3-ton H/P with 13.0 SEER and 7.7 HSPF meeting current DOE regulation in comparison to R32 and the other three HFO blends tested as well as pure R1234yf for comparative reference. As can be seen, the direct component with R410A is only about 5% of the total emission even assuming a conservative 4% annual leak rate and 30% end-of-life loss and 15 year life. The indirect component is based on calculating the changes in SEER and HSPF based on the four tests and using 0.65 kg CO_2/kwh representative of the US average power plant emission rate. Consequently, the overall LCCP index as shown at the bottom of the chart is reduced less than 5% indicating it is better to reduce the indirect content through efficiency. The direct content would increase about 3x to 15% for a trophy H/P like 20+SEER/12.0 HSPF where the charge could be 2x higher and the indirect content is 35% less, but this trophy segment is less than 2% of the market. It is interesting to note that there is diminishing return pursuing only the GWP since the direct content is < 1.5% with 675 GWP of R32. Another good example is R1234yf has same LCCP as R410A baseline despite having a GWP of 4 due to its higher indirect trading off the lower direct content.

Figure 14: LCCP R32 vs. HFO blends

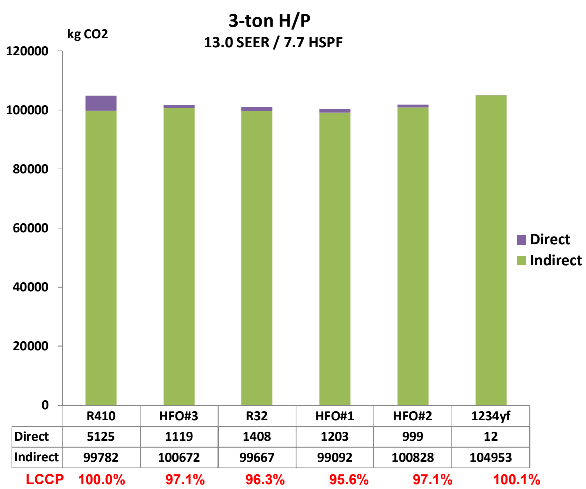


Figure 15: Low-GWP options assessment

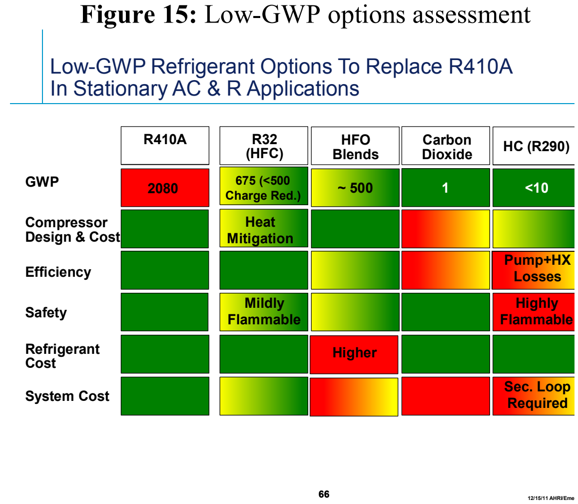


Figure 15 shows a qualitative high-level comparison summary for R32, HFOs as well as the so-called natural refrigerants such as CO_2 and hydrocarbon (HC) R290. R32 seems closer to R410A on several metrics compared to the other alternatives. The main disadvantage of HFO blends is the higher refrigerant cost from R1234yf or R1234ze. For CO_2, it is both efficiency and cost from ultra high-pressure design and transcritical operation. For HC greater than 150g charge like in Unitary A/C applications, its A3 high flammability will likely require a secondary loop and will impact cost as well as efficiency due to added pump and secondary HX losses.

8. A2L STANDARD DEVELOPMENT

Refrigerants are classified by ASHRAE under latest standard 34 as either class 1, 2, 2L or 3 for their flammability characteristics, and class A or B for non-toxic or toxic characteristic respectively. The specific classification is based on their respective flammability variables such as flame velocity, Lower Flammability Limit (LFL) and heat of combustion. Flame velocity is the variable used to classify refrigerants between class 2 and class 2L. Figure 16 below shows the flame velocity of the various refrigerants (including R32 and the HFOs) relative to each other. All refrigerants with a flame velocity below 10 cm/s were recently classified as "2L" by ASHRAE 34, thus introducing two new categories of refrigerants, A2L and B2L (Figure 17).

Figure 16: Flame velocity of different refrigerants

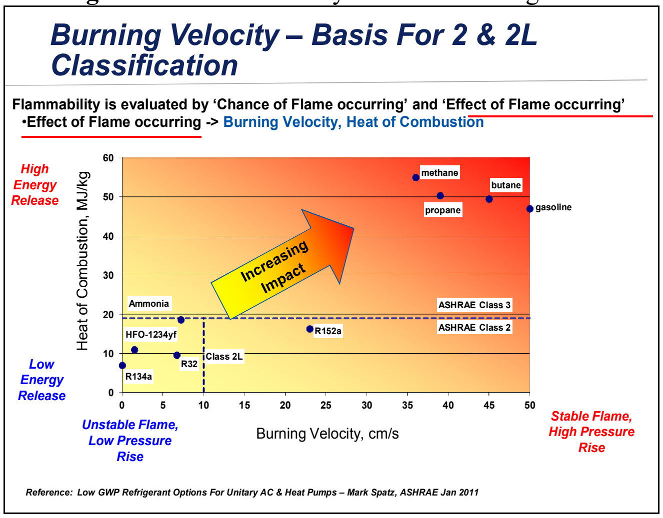
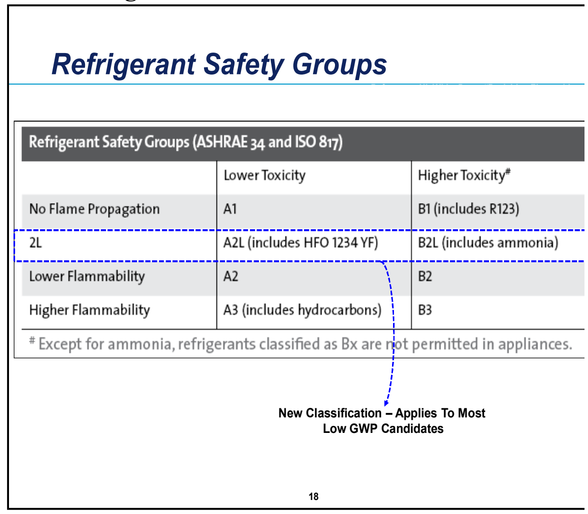


Figure 17: The new 2L classification



As mentioned above, a key challenge with R32 as well as the HFOs is the lack of standards and codes for the new A2L mild flammability rating which does not enable their quick commercialization. Several committee efforts are currently underway globally in the US, Europe and China to accelerate the development for these A2L standards (ASHRAE standards 15 and 34, UL 471 and 1995, IEC 60335-2, ISO 5149, pr EN 378-2) for both ductless and ducted A/C applications. Most of these standards are based on defining the maximum allowable charge based on a leak not exceeding 25% of the lower flammability limit (LFL) of the A2L refrigerant and whether the space is ventilated or unventilated. Tight installation spaces (closet) may require some application restrictions and/or modifications, etc. R32, R1234yf and R1234ze have comparable LFL at about 0.3 kg/m3.

Three levels of charge, $m1$, $m2$, $m3$, are proposed as follows based on the equations for A2 class of refrigerants for ventilated spaces :

$$m1 = 1.5 * (4\ m^3) * LFL \qquad (eqn.1)$$

$$m2 = 1.5 * (26\ m^3) * LFL \qquad (eqn.\ 2)$$

$$m3 = 1.5 * (130\ m^3) * LFL \qquad \text{(eqn. 3)}$$

where, LFL is the lower flammable limit in kg/m^3 for the refrigerant used. If the charge amount in the system (M) is below $m1$ then the minimum requirements for flammable refrigerants have to be met. When the charge level (M) is greater than $m1$ and less than $m2$, then in along with the minimum requirements, additional restrictions apply to the allowable room size. If the charge amount (M) is greater than $m2$ and less than $m3$ then in additional ventilation requirements are included and finally, if the charge amount (M) is greater than $m3$ then national standards apply.

The 1.5 multiplication factor in the above calculations was specifically proposed for A2L refrigerants to account for the lower flammability characteristics of the A2L refrigerants compared to the A2 class. This proposed 1.5 factor is being discussed in the working groups and may be revised upwards to 2.0 or even 3.0 depending on new data being made available on A2L refrigerants as well as the risk analyses on the use conditions of these refrigerants being conducted by various organizations worldwide.

The equation for the maximum charge level permitted for unventilated spaces if the charge (M) falls between $m1$ and $m2$, is as follows and is also under evaluation:

$$m_{max} = 2.5 \times (LFL)^{(5/4)} \times h0 \times (A)^{1/2} \qquad \text{(eqn. 4)}$$

where,

A = room area in m^2;
LFL = Lower Flammable Limit (LFL) in kg/m^3;
$h0$ = installation height of the appliance in m:
0.6 m for floor location;
1.8 m for wall mounted;
1.0 m for window mounted;
2.2 m for ceiling mounted.

None of the above four equations are final yet since much of this work was done for non ducted systems – working groups in Europe and the United States are actively pursuing finalization of these documents including consideration of ducted systems etc.; more is expected by the end of 2012. However, without these A2L standards available sooner, the industry will be at risk in meeting any future potential phase down scenarios since delaying early implementation of Lower-GWP solutions in new equipment will cause more R410A build up in the installed aftermarket where A2L likely cannot be a retrofit solution.

9. CONCLUSIONS

On the basis of efficiency and cost, R32 offers an attractive Low-GWP/Low-LCCP solution for mainstream A/C and H/P applications with performance comparable to R410A. This is based on drop-in evaluation with R410A optimized components and could be potentially improved by further optimizing the compressor and system towards achieving its theoretical potential as well as mitigating its higher compressor discharge temperature and finding new compatible oils. Its GWP of 675 can be equivalent to 500 GWP when factoring in its 15% lower system charge. Its heat transfer and pressure drop characteristics are synergistic with the direction of lower-charge, lower-leak compact heat exchangers for further mitigating the GWP phase down as well as the A2L flammability. R32 can serve as the initial candidate for new equipment to meet any potential HFC phase down proposal for at least until 2020+. It is still uncertain what GWP will ultimately be needed to meet the suggested 15% cap in the NAP proposal by 2030+ due to uncertain future R410A build up in the aftermarket service sector.

The HFO blends need to offer more compelling advantages over R32 due to their currently higher cost. They have same A2L flammability as R32. To sustain efficiency, their GWP is constrained to around 500, not much lower than R32 when charge is factored in. No visible advantages over R32 have been seen in these early drop-in system test results. More system data from the AHRI LGWP AREP program are needed to see further trends. On an LCCP basis, the HFO blends are comparable to R32, but their use may be limited initially until their cost position becomes clearer. R1234ze is reportedly a fraction of the cost of R1234yf, so R1234ze blends may offer potential to be closer to R410A cost and merit further investigation.

Natural refrigerants (CO_2 and R290) with ultra-low GWP so far have not shown to be cost-effective solutions for mainstream air conditioning due to their low efficiency and/or A3 flammability safety reasons. R290 is limited to much smaller units such as <1-ton mini splits where the charge can be limited closer to 150g that has become accepted as a standard without requiring secondary loop.

NOMENCLATURE

A2L	:	ASHRAE refrigerant mild flammability rating index
A3	:	ASHRAE refrigerant high flammability index
A/C	:	Air Conditioner
AREP	:	Alternative Refrigerants Evaluation Program
DOE	:	Department of Energy (US)
DLT	:	Discharge Line Temperature
EER	:	Energy Efficiency Ratio (Btu/wh)
GWP	:	Global Warming potential
HC	:	Hydrocarbons
HFO	;	HydroFluoro Olefins
H/P	:	Heat Pump
HSPF	:	Heating Seasonal Performance Factor (Btu/wh)
HX	:	Heat exchanger
LCCP	:	Life Cycle Climate Performance
LGWP	;	Low Global Warming Potential (Low GWP)
ODP	:	Ozone Depletion Potential
SEER	:	Seasonal Energy Efficiency Ratio (Btu/wh)
Te	:	Evaporating Temperature
Tc	:	Condensing Temperature

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