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HVAC Systems with Low Global Warming Potential Refrigerants: A Case Study

Vijay Bahel
Emerson, United States of America, Vijay.Bahel@Emerson.com

Rajan Rajendran

Brian Butler

Drew Welch

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Vijay BAHEL, Rajan RAJENDRAN, Brian BUTLER and Drew WELCH*

Emerson, Dayton, Ohio, USA

Ph: 937-726-5246, Email: Vijay.Bahel@Emerson.com

Ph: 937-726-0620, Email: Rajan.Rajendran@Emerson.com

Ph: 937-726-7634, Email: Brian.Butler@Emerson.com

Ph: 937-407-1940, Email: Drew.Welch@Emerson.com

* Corresponding Author

ABSTRACT

The objective of this case study is three-fold: (1) identify promising alternative refrigerants with lower global warming potential (GWP); (2) among those, select refrigerant(s) that could be "drop-in" replacements for R-410A and would not require significant system redesign or compressor changes — with the exception of minimal changes such as lubricating fluid and expansion valves; (3) evaluate the impact of another easy-to-implement option: replacing lower-efficiency, permanently split capacitor (PSC) condenser fans and evaporator blower motors with electronically commutated motors (ECM) for additional system efficiency improvements. This study leverages the steady-state heat pump design model (HPDM) developed by the Department of Energy (DOE) and the Oak Ridge National Laboratory (ORNL) to demonstrate three key findings: (1) two popular refrigerant replacement candidates with a GWP less than 750, R-32 and R-454B; both have system performance equal to or better than R-410A; (2) the lower-GWP refrigerant options with a GWP below 300 all underperform compared to R-410A; however, heat exchanger optimization may improve system performance; (3) using an ECM instead of a PSC evaporator blower motor increased system seasonal energy efficiency ratio (SEER) performance ~8% for all refrigerants evaluated.

1. INTRODUCTION

California is among a group of states seeking to limit the global warming potential (GWP) of refrigerants in new residential HVAC systems. To reduce GWP, the California Air Resources Board (CARB) has proposed restricting the use of refrigerants greater than or equal to 750 GWP in new stationary air conditioning equipment beginning Jan. 1, 2023 (ACHR, 2020). R-32 and R-454B (Shen *et al*, 2017) are two leading alternative refrigerants capable of achieving this target. Thus, this report evaluates and simulates the performance these alternatives compared to R-410A. We're also working under the assumption that future regulations will reduce refrigerant GWP limits even further, so we have also simulated the impacts on system performance using several emerging low-GWP refrigerant options less than 300 and 150 GWP. The primary objective of this case study is to evaluate a system designed for R-410A and leverage a theoretical model to estimate the impact on performance using these lower-GWP refrigerant alternatives.

2. LOW-GWP REFRIGERANTS

While progress toward wider adoption of lower-GWP refrigerants continues, the HVAC industry still has very limited information regarding the performance of the most commonly proposed alternatives. For our analysis, we've identified five alternative refrigerants, all of which are classified as A2L (non-toxic, mildly flammable, low burning velocity) by the American Society of Heating and Refrigeration Equipment (ASHRAE). We selected these refrigerants based on the following criteria: commercial availability or relative proximity to it; R-32 and R-454B have been considered as future candidates by residential original equipment manufacturers (OEMs) and other organizations (Shen *et al*, 2017); R-444B (< 300 GWP) appears to be an emerging alternative; R-454C and R-455A have reasonable capacity among options less than 150 GWP.

2.1 Refrigerant glide

Most of the refrigerants that have historically been used in split systems have been pure fluids, azeotropic mixtures, or nearly azeotropic mixtures. Refrigerant blends with capacity near R-22 or R-410A that have a GWP of less than 300 would require changes to system design and service practices to account for large differences between the saturated temperatures of liquid and vapor at typical operating conditions — or what's commonly known as glide. One common problem caused by glide is fractionation due to preferred phase change, both in heat exchanger and as leaks occur. In addition, systems may experience a performance penalty caused by higher condensing and lower evaporating temperatures. This is because the temperature change of refrigerant in the heat exchangers would not be matched with the ambient temperature change of the air, and because current typical residential split systems are designed with crossflow heat exchange. The modeling described in our study is based on an existing 2½-ton system that employs crossflow heat exchangers.

3. METHODOLOGY

3.1 Selected system

For our analysis, we selected a 2½-ton, commercially available 13.5 SEER R-410A split system, as test data and system information were readily available. System modeling required preparing detailed inputs of the system for the model, which were obtained from the OEM's product literature. These inputs are listed in Table 1. Note that the indoor blower and outdoor fan were equipped with PSC motors. Test points A and B define conditions prescribed in (ARI Std. 210/240, 2008) which N. American OEMs use to determine Performance Rating of their Unitary Air-Conditioning & Air-Source Heat Pump Equipment.

Table 1: System information of R-410A, 2½-ton split heat pump

Description	**Units**	**Indoor Unit**	**Outdoor Unit**
Heat Exchanger		Finned-Tube	Finned-Tube
Finned Height / Length	(in)	17.5 / 47	44.5 / 44.4
Horizonal / Vertical Tube Spacing	(in)	0.866 / 1.000	0.866 / 1.000
Number of Tubes / Rows	---	96 / 2	40 / 1
Number of Circuits	---	6	5
Fin Pattern	---	Slit-Lanced	Slit-Lanced
Fin Density	---	16	20
Fin Material	---	Aluminum	Aluminum
Tube Material	---	Copper	Copper
Outside Diameter	(in)	⅜	⅜
Tube	---	Rifled	Rifled
Air Flow Rate	CFM	988	2,800
Blower / Fan Motor Power	(W)	308	257
Condenser Exit Subcooling	(F)	---	10
Evaporator Superheat	(F)	10	---
Liquid Line Length	(ft)	---	25
Outside Diameter	(in)		⅜
Suction Line Length	(ft)	---	25
Outside Diameter	(in)		¾
Discharge Line Length	(ft)	---	4
Outside Diameter	(in)		½
Compressor	---	---	ZP25K5E-PFV

3.2 Analytical model

For the analysis, we used DOE/ORNL's steady-state heat pump design model (HPDM) (Shen and Rice, 2016), which is a well-known, public-domain HVAC equipment modeling and design tool. As this is a hardware-based model, user can specify the inputs of each component, i.e. compressor, heat exchanger, fan, pump, etc. It uses published manufacturer provided AHRI 10 coefficients (AHRI Standard 540, 2015) of compressor to improve accuracy of the system simulation by calculating mass flow rate and power consumption of compressor at system conditions. Knowing refrigerant power and mass flow rate enable calculations of refrigerant-side heat transferred at condenser and evaporator using their respective inlet/outlet enthalpies. The model also considers the actual suction state to correct the map mass flow prediction using the method described in published industry data (Dabiri and Rice, 1981).

HPDM uses the compressor's published AHRI coefficients provided by compressor manufacturer (to calculate mass flow rate and power consumption for the baseline system using refrigerant R-410A. For modeling performance systems with low-GWP refrigerants, our model assumes that the compressor has the same volumetric and isentropic efficiencies for the alternative refrigerants operating at the same suction and discharge pressure conditions. Essentially, the efficiencies of alternative refrigerants are derived from the baseline R-410A compressor map as a function of the suction and discharge pressures. In the absence of actual compressor performance maps for alternative refrigerants, studies (Shen *et al*, 2017) have shown that this approach can be used with reasonable accuracy to represent performance. The HPDM can connect directly with the Reference Fluid Thermodynamic and compresspr4Transport Properties Database (REFPROP) (Lemmon and Huber, 2010) to reference refrigerant properties per model; it also can use property tables for faster execution, which is the approach used in this study.

4. VALIDATION OF MODEL

We prepared the input file using the system information described in Table 1 and compressor maps to simulate cooling mode performance under the conditions described in (ARI Std. 210/240, 2008) — 95 ºF (A test) and 82 ºF (B test) — for R-410A and compared it to the test data published in (Stoben *et al*, 2015). Comparative results for test point A are summarized in Table 2.

Table 2: Test data vs. simulated performance for R-410A for A point

Description	**Test Data**	**Simulated**	**Difference (%)**
Capacity (Btu/hr / kW)	27,879/8.17	29,825/8.74	+ 6.5%
Indoor Blower (W)	308	308	0%
Outdoor Fan (W)	257	257	0%
Total Power Input (W)	2,363	2,603	+ 9.2%
EER (Btu/Wh)/COP	11.8/3.46	11.46/3.36	- 2.7%
Subcooling (F)/(K)	10.0/5.56	10.0/5,56	0%
Compressor Superheat (F)/(K)	13.0/7.2	13.0/7.2	0%

During the simulation exercise, we did not adjust any of the several scaling factors available in the model — such as: heat transfer or pressure drop — to improve the correlation between the test data and simulated results. Since resulting capacity and power exceeded normal ± 5 accuracy tolerance permitted by AHRI for compressor performance, we decided to identify the sources that may have contributed to the additional errors. See section 4.1 for a detailed breakdown of the sources of errors.

4.1 Error analysis and breakdown of simulation and test data

The consideration of new refrigerants is best evaluated by combining both testing and simulation to understand how actual performance differs from the model. In addition, model-based approaches may vary from the simple cycle performance of refrigerants to a complete system model, as described in this paper. Our study utilizes both testing and modeling by characterizing individual differences between refrigerants to arrive at an expected actual performance.

While we are primarily interested in the difference in the capacity and efficiency of refrigerants, the system performance can be divided into many smaller characteristics. For the example below, R-410A was the baseline refrigerant; this baseline was used by the evaluation model to compare the changes per each specific characteristic with each refrigerant. As the examples below describe, our analysis was performed by isolating characteristics using combinations of baseline conditions, alternate refrigerant conditions, baseline refrigerant properties and alternate refrigerant properties. The cycle performance of the refrigerant by itself is described by Equation 1.

Equation 1

$$\%\ of\ Baseline = \frac{\left(\dfrac{h_{alt\ ref,suc,at\ b/l\ Tsat\ suc\ and\ Super\ heat} - h_{alt\ ref,liq,at\ b/l\ Tliq}}{h_{alt\ ref,dis,at\ b/l\ Tsat\ dis\ and\ entropy\ of\ b/l\ suc\ conditions} - h_{alt\ ref,liq,at\ b/l\ Tliq}}\right)}{\left(\dfrac{h_{b/l\ ref,suc,at\ b/l\ Tsat\ suc\ and\ Super\ heat} - h_{b/l\ ref,liq,at\ b/l\ Tliq}}{h_{b/l\ ref,dis,at\ b/l\ Tsat\ dis\ and\ entropy\ of\ b/l\ suc\ conditions} - h_{b/l\ ref,liq,at\ b/l\ Tliq}}\right)}$$

Where:

$h_{alt\ ref,suc,at\ b/l\ Tsat\ suc\ and\ super\ heat}$ is enthalpy of the alternate refrigerant at the saturated suction temperature and superheat of the baseline refrigerant test or simulation

$h_{alt\ ref,liq,at\ b/l\ Tliq}$ is enthalpy of the alternate refrigerant at the liquid temperature of the baseline refrigerant test or simulation

$h_{alt\ ref,dis,at\ b/l\ Tsat\ dis\ and\ entropy\ of\ b/l\ suc\ conditions}$ is enthalpy of the alternate refrigerant at the saturated discharge temperature suction entropy of the baseline refrigerant test or simulation

$h_{b/l\ ref,suc,at\ b/l\ Tsat\ suc\ and\ super\ heat}$ is enthalpy of the baseline refrigerant at the saturated suction temperature and superheat of the baseline refrigerant teste or simulation

$h_{b/l\ ref,liq,at\ b/l\ Tliq}$ is enthalpy of the baseline refrigerant at the liquid temperature of the baseline refrigerant test or simulation

$h_{b/l\ ref,dis,at\ b/l\ Tsat\ dis\ and\ entropy\ of\ b/l\ suc\ conditions}$ is enthalpy of the baseline refrigerant at the saturated discharge temperature suction entropy of the baseline refrigerant test or simulation

The temperature split across the air-side impact on performance is calculated by Equation 2.

Equation 2

$$\%\ of\ Baseline = \frac{\dfrac{\left(\dfrac{h_{alt\ ref,suc,at\ Tsat\ suc\ of\ alt\ supply\ air\ and\ std\ s/h} - h_{alt\ ref,liq,at\ Tamb}}{h_{alt\ ref,dis,at\ Tdis\ sat\ of\ alt\ exhaust,isen} - h_{alt\ ref,liq,at\ Tamb}}\right)}{\left(\dfrac{h_{b/l\ ref,suc,at\ Tsat\ suc\ of\ b/l\ supply\ air\ and\ std\ s/h} - h_{b/l\ ref,liq,at\ Tamb}}{h_{b/l\ ref,dis,at\ Tdis\ sat\ of\ b/l\ exhaust,isen} - h_{b/l\ ref,liq,at\ Tamb}}\right)}}{\%\ of\ Baseline_{Refrigeran\ t\ Cycle}}$$

Where:

$h_{alt\ ref,suc,at\ Tsat\ suc\ of\ alt\ supply\ air\ and\ std\ s/h}$ is enthalpy of the alternate refrigerant at a saturated suction temperature equal to the supply air temperature of the alternate refrigerant test or simulation with a standard superheat

$h_{alt\ ref,liq,at\ Tamb}$ is enthalpy of the alternate refrigerant at the liquid temperature equal to the ambient temperature of the baseline refrigerant test or simulation

$h_{alt\ ref,dis,at\ b/l\ Tdis\ sat\ of\ alt\ exhaust,\ isen}$ is enthalpy of the alternate refrigerant at a saturated discharge temperature equal to the exhaust air temperature of the alternate refrigerant test or simulation with the entropy at a saturated suction temperature of supply air temperature of the alternate test or simulation with a standard superheat

$h_{b/l\ ref,suc,at\ Tsat\ suc\ of\ alt\ supply\ air\ and\ std\ s/h}$ is enthalpy of the baseline refrigerant at a saturated suction temperature equal to the supply air temperature of the baseline refrigerant test or simulation with a standard superheat

$h_{b/l\ ref,liq,at\ Tamb}$ is enthalpy of the baseline refrigerant at the liquid temperature equal to the ambient temperature of the baseline refrigerant test or simulation

$h_{b/l\ ref,dis,at\ b/l\ Tdis\ sat\ of\ alt\ exhaust,\ isen}$ is enthalpy of the baseline refrigerant at a saturated discharge temperature equal to the exhaust air temperature of the baseline refrigerant test or simulation with the entropy at a saturated suction temperature of supply air temperature of the baseline test or simulation with a standard superheat

$\%\ of\ Baseline_{Refrigerant\ Cycle}$ is the cycle performance of the refrigerant by itself as described by Equation 1

After accounting for the advantage of the cycle, observed differences in subcooling and superheat, changes to air temperature splits, pressure loss changes, compressor performance map differences, and heat exchange impact differences, the impact changed from the observed to the expected for SEER. Condition and SEER values are shown in Table 3, where SEER was calculated with a constant coefficient of degradation of 0.10.

Table 3: Adjusted test data vs. simulated performance for R-410A, R-454B and R-32

	Tested			**Modeled**		
	R-410A Baseline	R-454B	R-32	R-410A Baseline	R-454B	R-32
SEER A Direct	100% (b/l)	102.8%	99.9%	100% (b/l)	105.0%	103.7%
SEER A Adjusted	100% (b/l)	101.3%	99.9%	100% (b/l)	102.3%	103.0%
	Tested			**Modeled**		
	R-410A Baseline	R-454B	R-32	R-410A Baseline	R-454B	R-32
SEER B Direct	100% (b/l)	100.0%	98.8%	100% (b/l)	104.1%	103.2%
SEER B Adjusted	100% (b/l)	101.7%	98.6%	100% (b/l)	100.5%	101.9%

The simulation consistently favored both alternate refrigerants compared to the R-410A baseline. Further, upon characterizing performance aspects to adjust the output, our analysis showed that there may be a performance improvement available with R-454B beyond what can be achieved with R-32.

5. PERFORMANCE WITH ALTERNATIVE REFRIGERANTS

5.1 Performance with constant compressor displacement

Our performance analysis began by simulating the system level performance of each alternative refrigerant under constant compressor displacement. For this step, we kept the baseline R-410A compressor's displacement constant at 1.39 in^3/revolution. Performance for test points A and B are listed in Table 4. We did not adjust any of the scaling factors available in the model. In addition, we kept the model's existing indoor and outdoor fan airflow rates and power consumption, heat exchanger sizes, circuiting and fan performances, including the condenser subcooling and compressor superheat.

Our findings show that R-32 has a better overall heat transfer in both the condenser and evaporator, followed by R-454B. All the other alternatives had a slightly poorer heat transfer performance, thereby yielding lower system capacity. This indicates that a larger compressor displacement is needed to match the point A capacity of refrigerant R-410A. For reference, see Table 3 for data on the system performance of the alternative refrigerants compared to the baseline refrigerant R-410A.

Table 4: Performance with alternative refrigerants at constant compressor displacement of 1.39 in^3/revolution

Test Point	Refrigerant	Compressor Displacement Scaling Factor	Capacity [Btu/hr]	System EER [Btu/Wh]	System SEER [Btu/Wh] @ Cd = 0.10
A	R-410A	1.000	29,825	11.46	---
A	R-32	1.000	32,222	11.88	---
A	R-454B	1.000	29,392	12.03	---
A	R-444B	1.000	22,726	12.09	---
A	R-455A	1.000	22,781	11.44	---
A	R-454C	1.000	21,490	11.80	---
B	R-410A	1.000	31,838	13.90	13.21
B	R-32	1.000	33,913	14.35	13.63
B	R-454B	1.000	31,136	14.47	13.74
B	R-444B	1.000	24,036	14.30	13.59
B	R-455A	1.000	24,492	13.71	13.02
B	R-454C	1.000	23,079	14.10	13.39

5.2 Performance with modified compressor displacement

Next, we scaled the baseline compressor's displacement of 1.39 in^3/revolution to match performance of other refrigerants with the test point A capacity R-410A. Results are summarized in Table 5. Refrigerants R-444B, R-454C and R-455A required significantly larger compressor displacement versus R-410A. R-32 and R-454B proved to be the most promising replacement candidates for R-410A. R-32 had the best SEER values at 13.85 Btu/Wh versus. 13.21 Btu/Wh for R-410A. In addition, the R-32 system required 10% smaller compressor, whereas R-454B had the closest compressor displacement with equally good SEER of 13.67 Btu/Wh. Furthermore, R-32's similar thermophysical properties and operating pressures allow it to operate with compressors designed for R-410A duty. If selecting a refrigerant below 500 GWP is a primary criterion, R-454B would be the preferred option over R-32.

It's important to note because our testing uses heat exchangers that are designed for R-410A duty, the drop-in characteristic of our investigation does not provide an adequate comparison of refrigerants. Alternative refrigerants have different thermophysical properties, which causes the existing heat exchangers to be improperly sized, compared to the heat exchangers designed specifically for R-410A application. In addition, some of the alternative refrigerants examined have large temperature glides, indicating the need for optimal heat exchanger configurations for those refrigerants. The efficiency and capacity of systems with alternative refrigerants would be expected to improve through design modifications that manufacturers generally perform before introducing the new product to market.

Table 5: Performance with modified compressor displacement to match A point of baseline unit with R-410A

Test Point	Refrigerant	Compressor Displacement Scaling Factor	Capacity [Btu/hr]	System EER [Btu/Wh]	System SEER [Btu/Wh] @ Cd = 0.10
A	R-410A	1.000	29,825	11.46	---
A	R-32	0.903	29,789	12.11	---
A	R-454B	1.020	29,774	11.95	---
A	R-444B	1.440	29,927	11.67	---
A	R-455A	1.460	29,900	10.82	---
A	R-454C	1.575	29,824	11.12	---
B	R-410A	1.000	31,838	13.90	13.21
B	R-32	0.903	31,351	14.58	13.85
B	R-454B	1.020	31,545	14.39	13.67
B	R-444B	1.440	31,690	14.00	13.30
B	R-455A	1.460	32,204	13.16	12.50
B	R-454C	1.575	32,084	13.51	12.84

6. PERFORMANCE WITH ECM MOTORS

6.1 Efficiencies of ECM and PSC motors

Another easy-to-implement strategy we evaluated to further enhance system efficiency was the option to replace a system's existing lower-efficiency PSC condenser fan and evaporator blower motors with ECM motors. We started by examining the efficiency of existing PSC and ECM blower and fan motors on the system, as shown in Tables 6 and 7.

Table 6: PSC and ECM efficiencies of indoor unit's blower motor

Indoor Blower Motor					
Type	**Horsepower**	**Efficiency (%)**	**Shaft Power (W)**	**Power Input (W)**	**Air Flow Rate (CFM)**
PSC	0.23	56	171	308	988
ECM	0.23	82	171	209	988

Table 7: PSC and ECM efficiencies of outdoor unit's fan motor

Outdoor Fan Motor					
Type	**Horsepower**	**Efficiency (%)**	**Shaft Power (W)**	**Power Input (W)**	**Air Flow Rate (CFM)**
PSC	0.2	58	149	257	2,800
ECM	0.2	80	149	186	2,800

6.2 System gain with ECM motors

Next, we re-calculated system efficiency and SEER with ECM motors per each refrigerant and summarized the results (see Figure 1). Our analysis evaluated SEER of all six refrigerants with PSC and ECM motors and demonstrated their respective efficiency gains. ECM motors in outdoor and indoor units provided an average gain of 8.4% — with R-32 delivering the greatest improvement of nearly 9%.

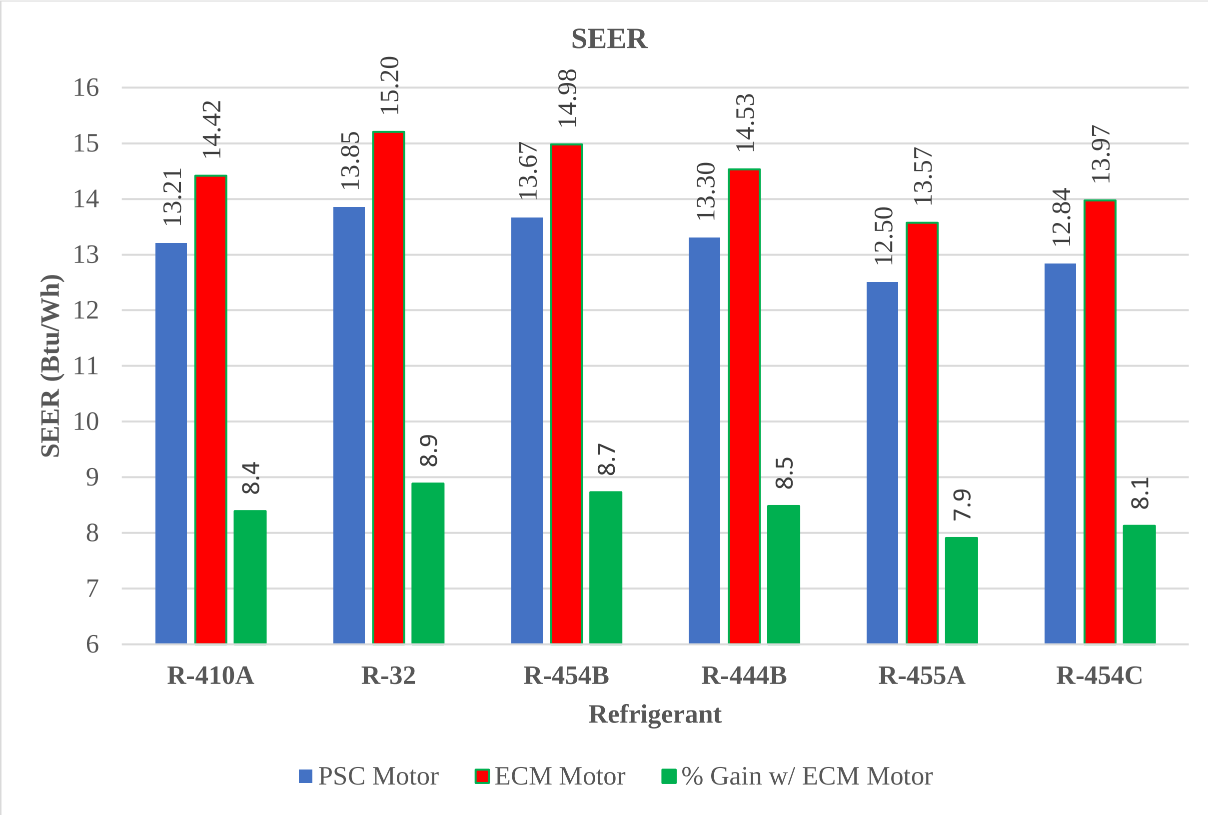

Figure 1: Simulated SEER gain with ECM vs. PSC evaporator blower motors

7. CONCLUSIONS

From the simulations/modeling performed in this report, we have three key takeaways: (1) we can conclude that R-32 and R-454B, two popular R-410A replacement candidates with a GWP below 750, both have system performance equal to or better than R-410A; (2) refrigerant options with a GWP below 300 all underperform versus R-410A; however, heat exchanger optimization for these refrigerants may lead to system performance improvements, and (3) the use of an ECM versus PSC evaporator blower motor increased theoretical system SEER performance ~8% for all refrigerants evaluated.

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