Performance Assessment of High-Efficiency Refrigerated Display Cases with Low-GWP Refrigerants

Preprint

Alexander Bulk,[1] Ramin Faramarzi,[1] Greg Shoukas,[1] Omkar Ghatpande,[1] and Steven Labarge[2]

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Performance Assessment of High-Efficiency Refrigerated Display Cases With Low-GWP Refrigerants

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ABSTRACT

As California policy makers enforce new regulations such as SB 1383 to achieve state greenhouse gas reduction goals, it will be critical to understand the energy efficiency implications of low global warming potential (GWP) refrigerants in commercial refrigeration applications. The goal of this research project, funded by Commonwealth Edison, was to assess the energy performance of two high-efficiency medium-temperature, self-contained, reach-in display cases that utilized natural refrigerant propane (R-290, GWP = 3), and hydrofluorocarbon (HFC) drop-in hydrofluoroolefin R-513A (GWP = 573). Performance of these display cases was compared to a baseline fixture using HFC (R-134A, GWP = 1301) under equal conditions in a controlled environment chamber. These display cases were selected due to their widespread use in convenience stores and small supermarkets.

The high-efficiency cases were equipped with energy-efficient lighting, improved panel insulation, high-efficiency evaporator and condenser fan motors, and high-effectiveness heat exchangers. The test method used in this project was foundationally inspired by the ANSI/ASHRAE 72-2018 standard method to evaluate each case under equivalent conditions (ASHRAE 2018). However, minor modifications to this methodology were implemented to better represent more realistic operation of the units.

While maintaining equivalent target product temperature, the energy-efficient fixtures A (containing R-290, improved insulation, heat exchangers with a higher overall heat transfer coefficient (higher UA), and more efficient lighting and fan motors) and B (containing R-513A, improved fans, and higher UA-heat exchangers, and more efficient lighting and fan motors) consumed 61.8% to 32.6% less energy, respectively, than a baseline case using R-134A refrigerant. The daily energy consumption of the R-290, R-513A, and baseline display cases at the upper limit of environmental conditions was 4.30, 7.59, and 11.26 kWh/day, respectively.

Background

In January 2020, the U.S. Environmental Protection Agency (EPA) began to enforce a production phase-down of hydrofluorocarbon refrigerants due to their high GWP, in compliance with the American Innovation and Manufacturing Act of 2020 (HR 133 2021). GWP is a ratio of the amount of heat stored by a single metric ton of a substance to the amount of heat stored in a metric ton of carbon dioxide over 100 years (EPA 2020). Thereby, a GWP of 1 corresponds to the heat absorbed by atmospheric CO_2 over 100 years.

R290 refrigerant has an extremely low GWP of 3, comparable to CO_2 and nearly $1/500^{th}$ that of R134A (Sánchez et al. 2017). R290 refrigerant is a high-purity propane (C_3H_8) charged at very low density that has had a history of use in industrial refrigeration (Choudhari and Sapali 2017). Due to the EPA phase-down of HFC refrigerants, recent refrigeration and heating,

ventilating, and air-conditioning (HVAC) technologies have begun to implement R290 refrigerant. R290 also exhibits superior thermodynamic properties. Propane has a constant pressure heat coefficient of 2.742 (liquid) / 2.036 (vapor) kJ/kgK at 25°C (Cengel 1997) and an enthalpy of vaporization of 440.1 ± 23.2 kJ/kg (NIST 2010). Because of the flammability of propane, International Electrotechnical Commission (IEC) standards have capped the allowed charge mass to 150 g in refrigeration systems (IEC 2012). The higher latent heat of vaporization of R290 allows the system to be charged with lower mass without sacrificing thermodynamic performance (Choudhari and Sapali 2017). Its properties, however, prevent R290 from being simply retrofit to existing R134A systems. Heat exchangers must be designed with smaller channels to transfer heat to and from the low-density propane (Embraco 2019). Due to flammability, plastic or aluminum non-spark fans must be used, as well as a compressor that contains overload protectors and safe-startup devices according to IEC 60079-15 (IEC 2005).

R513A, or Opteon XP10, is a hydrofluoroolefin (HFO) refrigerant blend containing 2,3,3,3-Tetrafluoropropene ($C_2H_2F_4$). R513A exhibits less than half the GWP of R134A, but still exhibits a relatively high GWP at 573 (Mota-Babiloni et al. 2017). The blend contains a small fraction of R134A that allows it to be implemented into most existing R134A systems due to its similar thermodynamic properties. This provides a cheaper replacement option than R290, which requires an entirely updated refrigeration system. Its thermodynamic properties are slightly deviated from R134A, however, to a degree that is expected to affect energy consumption. Despite this, R513A's GWP of 573 is nearly 1/3 that of R134A (Mota-Babiloni et al. 2017). The thermodynamic properties of the refrigerants evaluated in this study are shown in Table 1. The molecular structures of the three chemicals analyzed here are shown in Figure 1.

Table 1. Thermodynamic properties of refrigerants used in the evaluated technologies

Refrigerant:	Saturated Liquid Density, ρ @ 25°C (kg/m3)	Liquid Constant Pressure Heat Coefficient, cp @ 25°C (kJ/kgK)	Vapor Constant Pressure Heat Coefficient, cp @ 25°C (kJ/kgK)	Enthalpy of Vaporization (kJ/kg)
R134A	1207	1.426	0.851	234.7
R513A	492	2.742	2.036	440.1
R290	1134	1.412	0.881	194.8

(Cengel 1997; NIST 2010; Huber and McLinden 1992; Climalife 2017)

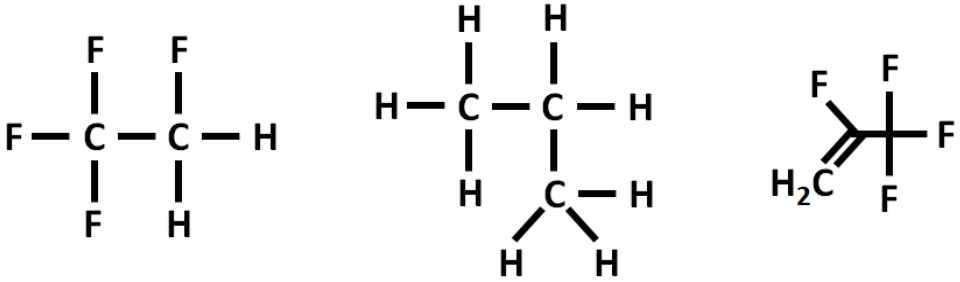

Figure 1. Molecular structure of: (left) 1,1,1,2-Tetrafluoroethane (R134A, $C_2H_2F_4$), (center) propane (R290, C_3H_8), (right) 2,3,3,3-Tetrafluoropropene (R513A, CH_2F_4)

Despite the phase-down in new stand-alone refrigerated cases, the use of R134A in existing refrigeration systems is still permitted (and in chillers until 2024). Refrigeration technologies have only recently begun adopting R513A or R290 refrigerant in more available products. Because of this, various studies have been conducted to assess the energy use of refrigerated display cases using R290 in recent years. However, the performance of self-contained medium-temperature R290 cases (as opposed to low-temperature, which is below freezing) have not been widely assessed other than to verify compliance with standards. Current studies have primarily focused on remote units (outdoor condensing units serving multiple cases) or self-contained units for low-temperature applications (SCE 2016; Mota-Babiloni et al. 2015, He et al. 2014). While remote refrigeration dominates the large supermarket sector, self-contained refrigeration is largely used in convenience stores, restaurants, pharmacies, and smaller supermarkets.

For the purpose of consistent comparison, the display cases were selected from a U.S. Department of Energy database of commercially available refrigerators that contained the closest specifications based on:

1. Equivalent overall merchandizing volume and configuration (5 deck/4 shelf, 6' length)
2. Discharge air temperatures (medium temperature)
3. Components wiring (all on 115V/3Ph/60Hz)
4. Number of glass doors (3)
5. An equal number of compressors/condensing units (1, ½ HP)
6. Close internal dimensions.

The baseline case was purchased prior to the EPA ban at the end of 2019, and a second case of the same model fitted with R513A was then purchased after the start of 2020. The R513A model was updated from its R134A model and also included upgraded energy-efficient refrigeration components. Due to the additional energy-efficient components present in the R290 case, this case is referred to as the "energy-efficient case A" or "EE case A," and the case containing R513A is referred to as "EE case B."

The refrigerator cases' energy use and product cooling performance were evaluated under indoor environmental conditions used in the ASHRAE 72-2018 method of test (ASHRAE 2018). Where applicable, the experimentation design for this work was based on this method. Modifications to the approach used in this standard were made to better address the specific questions investigated in our study and avoid altering the cases in a manner that would affect performance. Electrical instrumentation included monitoring the power of evaporator and condenser fans, the compressor, and lighting/controllers. Assessments were conducted over 24 hours (h) to evaluate the effect of compressor cycling on the cases' ability to maintain the temperature of various "product simulators" situated throughout the case interior. Door actuators were used to replicate customer traffic according to a schedule based on ASHRAE 72 (ASHRAE 2018).

Baseline Technology Using R134A

The baseline case refrigeration components consist of a fixed-speed compressor, a capillary tube, and a fin-and-tube evaporator. The case contains a top-mounted condensing unit within an aluminum enclosure, and a front grille but no ceiling or rear wall. The case has right-side hinged doors with a white-painted stainless-steel interior and exterior. The unit is

approximately 78” in length, 37” deep, and 87” tall when placed on 5” casters. The calculated volumetric capacity between shelves, product load lines, and below the evaporator is 48.29 ft^3. The left and right shelves are 22.75” wide, and middle shelves are 25” wide. All shelves are 21” between product load lines, and the bottom deck is 25.25” to the product load line. There is 10.5” of clearance between each shelf, and below the evaporator.

The case condensing unit contains a ½ HP reciprocating compressor and 11.75”-diameter aluminum condensing fan mounted to the 13.25” x 12.25” face of a 5”-thick condenser fin-and-tube heat exchanger. A 15.75” x 35.25” evaporator enclosure hanging 5.25” from the case interior’s ceiling contains the two plastic 7.5”-diameter draw-through fans directed forward at a 17.5° angle and centered 11.75” from the front of the case and 5.25” from the center. Airflow is directed into the fans, and a discharge opening of 2.25” from the rear of the evaporator enclosure delivers air to the case along the back wall.

Compressor cycling is controlled via internal cut-in and cut-out temperatures set to ±2.22°C (4°F) from a default 2.22°C (36°F) setpoint temperature that can be adjusted on a digital controller interface. Through operation, it was found that defrost cycles were time-initiated once every 24 h, and temperature-terminated when the evaporator coil outlet sensor reached 10°C (50°F). This display case, as well as both energy-efficient fixtures, leveraged off-cycle defrost. In off-cycle defrost, the condensing unit shuts off while evaporator fans stay on to introduce warm air from the sales area into the evaporator. The baseline fixture used one time-initiated/temperature-terminated off-cycle defrost per day.

Energy-Efficient Technology Using R290 (EE Case A)

This optimally rated case using R290 was selected because it shares the closest specifications to the baseline case. Unlike the other cases, this case contains a bottom-mounted condensing unit within an enclosure containing a front and rear steel-coated grille to allow airflow. The far left door also hinges from the left and not the right side, whereas the other case doors all hinge from the right. The case contains a stainless steel interior and a powder-coated, black-painted exterior around a layer of high thermal-efficiency insulation. The refrigeration components consist of a fixed-speed compressor, a capillary tube, and an evaporator enclosure containing a single 21-cm diameter plastic evaporator fan. The unit is approximately 78” in length, 30” deep, and 83” tall when mounted on 4” casters. The calculated interior volumetric capacity is 49.15 ft^3. The shelves are all 21.5” deep, 24.25” wide, and have 10.5” of clearance between them and the above shelf (or evaporator enclosure). The bottom deck is 23” deep to the product load line and has 11.5” of clearance below the lowest shelf.

This case’s condensing unit contains a ½ HP reciprocating compressor and 10.25”-diameter aluminum condensing fan mounted to the 11” x 11” face of a 5.5”-thick fin-and-tube condenser heat exchanger. Just as with the other cases, the evaporator uses a draw-through fan that delivers air into the evaporator enclosure and discharges from a grille facing the back wall. The evaporator enclosure is 16.75” x 33” and hangs 5.25” from the case interior’s ceiling and is flush with the front wall of the case above the doors. The evaporator fan is located 11.5” from the front of the case and the discharge grille located on the rear of the evaporator panel 5.75” from the back wall.

The case regulates compressor cycling via internal cut-in and cut-out temperatures set using an analog controller. The default cut-out and cut-in internal temperatures are 0.56°C (33°F) and 3.33°C (38°F), respectively. This display case has no scheduled defrost cycles, but during compressor off-cycle periods, evaporator fans run on a periodic schedule for 1 min every 6

minutes. This feature enables the display case to function similar to an off-cycle defrost system and adequately remove the frost from the coil. The condenser fan and compressor operation are interlocked and, therefore, cycle simultaneously.

Energy-Efficient Technology Using R513A (EE Case B)

When the manufacturer of the baseline case began to implement R513A into their model, they included an upgraded evaporator and condenser heat exchanger, and canopy lighting. The evaporator enclosure now contains two, more robust 8.5”-diameter aluminum evaporator fans than those of the baseline. A larger 17.5” x 36.25” evaporator enclosure hanging 5” from the case interior’s ceiling contains the two fans directed forward at an 11° angle and centered 12.5” from the front of the case and 5.5” from the center. A slightly smaller 10.5”-diameter aluminum condensing fan, however, is mounted to the 12” x 11.25” face of a 3.5”-thick condenser heat exchanger. Other than those described, all other case components and dimensions are equivalent to the baseline case. The condensing unit contains the same ½ HP reciprocating compressor model as the baseline version because it is compatible with both refrigerants.

Compressor cycling is controlled the same as the baseline, but defrost cycles occur once every 12 h as opposed to 24 h. The evaporator fan cycled off concurrently with the compressor unlike the baseline. During the two time-initiated/temperature-terminated off-cycle defrosts, the evaporator fans stayed on. Table 2 displays differences in key parameters for each of the three cases. An image of the (from left) EE case B, baseline, and EE case A set up in the lab is shown in Figure 2. Door actuators are shown on EE case A and B and the baseline case contains filler material here.

Table 2. General refrigerator display case key parameter specifications

Case:	Scheduled Defrost Freq.	Rated Btu/h	Rated Current (A)	Refrigerant GWP	Default Cut-Out/Cut-In Temperature (°F)	Evaporator Fan Cycling
Baseline (R134A)	24 h	2,600	13.8	1,300	32 / 40	Continuous
EE Case A (R290)	N/A	4,716	9.3	3	33 / 38	1 min every 6 min
EE Case B (R513A)	12 h	Not provided	Not provided	573	32 / 40	W/ compressor cycling

Figure 2. Image of the three refrigerator cases outside the environmental chamber. (Left) EE case B, (center) baseline case, and (right) EE case A. *All images/figures by NREL.*

Methodology

The goal of this project was to evaluate the energy performance of the selected display case technologies utilizing HFC, HFO, and propane refrigerants. It does not intend to replicate standard tests performed by case manufacturers for the purpose of rating. Measurement methods modified from industry-wide standards were only used to maintain consistency between the performance assessment of each display case. Power consumption was measured for the total case and for each of the case subcomponents including the compressor, condenser, evaporator fans, and lighting/controls across a 24-h evaluation at a sampling rate of 1 Hz. Power was measured through two Continental Control Systems WMC-3Y-208-MB model Wattnode power meters. Five Accu-CT ACTL-0750 model CTs were instrumented to each of the three refrigerated cases' components. Two 20 A CTs were instrumented to each case's total plug power and compressor, and three other 5 A CTs were instrumented to the condenser fan, evaporator fan, and lighting and controls. Measurement accuracy for each electrical parameter was ±0.5%. Total energy was calculated by integrating power over 24 h. Each 24-h experiment was repeated at least two times to ensure that no extraneous conditions caused results to vary between each evaluation. Case interior air temperatures, refrigerant piping temperatures, and condensate mass were monitored to provide engineers understanding of case operation, but are not provided here. Further measured data are available upon request from the author.

The cases were evaluated for 24-h cycles within a controlled-environment chamber. The interior of the chamber is 85" wide x 142.5" in length, with a 144"-high ceiling. The cases were aligned parallel to the long end of the chamber facing away from the chamber's air/steam exhaust vent. The cases were oriented exactly 12" from the back wall according to ASHRAE 72 (ASHRAE 2018), and centered in the chamber 36" from each side wall and 36" behind an ambient measurement pole for monitoring environmental conditions. There was 57" of clearance above the top of the tallest case, and no floor perforations around the perimeter of the case.

Evaluations were conducted repeatedly until approximately equivalent results were obtained between at least two tests under each environmental condition. Prior to initiating each evaluation, optional equipment such as exterior lighting not connected to internal lights, or

evaporator pans were removed from each case. Because this evaluation took place in Golden, Colorado, at 1,773 m, the lower air density reduced heat transfer through the evaporator compared to sea level. Cut-in and cut-out temperatures were therefore adjusted prior to each evaluation to reach a mean "product simulator" temperature within AHRI/FDA requirements (3.33 ± 1.11°C / 38 ± 2°F /) (AHRI 2013; FDA 2017). A controller setpoint was selected that generated mean temperatures across 18 "product simulators" closest to 3.0°C (37.4°F) with each case. Control was limited because each case's cut-in/cut-out must be adjusted concurrently in integer increments. Cases were maintained powered on until the mean temperature was maintained with less than 0.4°F (0.22°C) variation across a 12-h period prior to initiating each evaluation. Cut-in/cut-out temperatures were adjusted to 38°F / 30°F with the baseline case, 35°F/30°F with EE case A, and 39°F/31°F with EE case B to maintain average temperatures of 37.40 ± 0.04°F, 37.22 ± 0.05°F, and 37.44 ± 0.05°F, respectively.

Customer traffic and door openings were replicated by installing and programming automatic Olide 120B-model door actuators mounted above each door. The front panel of the baseline and EE case B's condensing units were raised 1.25" to install the door actuators. The case door actuators were operated based on the schedule described in ASHRAE 72-2018 (ASHRAE 2018). Approximately 3 h following the start of each assessment, the left case door was set to open for 6 s. Exactly 10 min after the left door opened (9:54 after closure), the middle door then opens for 6 s, followed by the right-side door after another 10 min. This process then repeats with the left door for a total period of 8 h. The case doors then remained closed for the latter 13 h of each assessment. Because EE case A had no scheduled defrost cycle, door actuators were not scheduled to coordinate with defrost. Evaluations were coordinated such that the baseline and EE case B defrost would not occur during the 8-h period of scheduled door openings. A webcam was set up inside the environmental chamber to monitor door openings to ensure they occurred on the prescribed schedule.

Environmental conditions were monitored and controlled from an "ambient measurement pole" mounted 36" from the front of the case (Figure 3). Measurements were collected at location T_A (5.9" above the top edge of the case doors), and location T_B (at the height of the geometric center of the case doors). Dry-bulb and dew-point temperature were controlled to T_A. ASHRAE 72 environmental conditions specify a dry-bulb at T_A of 24 ± 1°C (75.2 ± 1.8°F) and a wet-bulb of 18 ± 1°C (64.4 ± 1.8°F) (ASHRAE 2018). Because dew point was measured at T_A instead of wet-bulb temperature, the dew-point temperature was instead controlled to be maintained within the limits corresponding to the ASHRAE wet-bulb and dry-bulb limits (15.42 ± 2.19°C, 59.76 ± 3.95°F) at lab altitude (1,773 m). Only dry-bulb temperature was measured at location T_B to ensure stability within of T_A of less than 0.56°C/ft (1°F/ft). This would correspond to a range of 24 ± 2.67°C (75.2 ± 4.8°F) based on the cases' door size. Temperatures were maintained at these limits for at least 12 h prior to initiating 24 h evaluations. Omega brand 1/16" type-T thermocouple probes (model number TMQSS-062U-6) with ± 0.9°F accuracy, and an EdgeTech DewTrak II DPS3 model chilled mirror dew-point hygrometer with ± 0.2°C (± 0.4°F) accuracy were used to measure dry-bulb and dew-point temperature, respectively. Dry-bulb thermocouples were each inserted into ½"-thick brass slugs with a ¾" diameter.

Product temperatures were monitored using thermocouples placed in multiple "product simulators" situated evenly throughout the interior of the cases to ensure adequate performance, such that the average of all simulator temperatures were maintained within AHRI/FDA requirements over 24 h (AHRI 2013; FDA 2017).The product simulators are 3" x 3" (base) x 2.5" (height) plastic containers filled with grout sponges soaked in a 50/50 (±2%) mix of food-

grade propylene glycol and deionized water. Simulators were inserted with thermocouple probes (not including brass slugs) through a drillhole in the simulator lid such that the tip of each probe would rest 1.5” from the bottom. Thermocouple pre-calibration was verified using an ice bath. Eighteen simulators were placed at the left wall, right wall, and geometric center of the cases on the bottom deck, the second shelf from the bottom, and the top shelf. Each of these locations had two simulators placed at both the front and rear product load limit lines on the shelf. Figure 3 shows a schematic diagram of the location of these product simulators throughout the case interior.

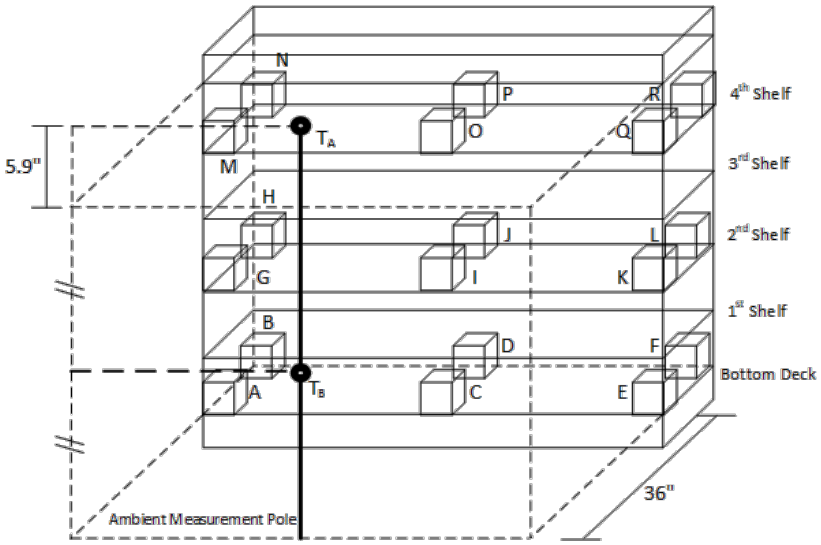

Figure 3. Schematic of product simulator temperature measurement locations (A–R), and ambient temperature locations (T_A and T_B)

The net usable interior volume of the cases was loaded with 11” tall, 1 L water bottles as filler material to simulate the thermal mass of food product. The curved-shape of the cylindrical bottles allowed air to pass between them, allowing the case to effectively cool product simulators in a manner that was not achievable using other filler. Filler bottles were organized in a 7 x 7 pattern on each shelf except those that contained product simulators. This was the maximum number of bottles that fit between the product load lines on each cases’ shelves. Shelves containing product simulators had six bottles removed to fit the simulators at the front and rear of the shelf. The baseline case and EE case B have a deeper bottom deck, and so an additional row of seven bottles was added to those shelves. The baseline and EE case B held 702 bottles (24.8 ft^3), whereas EE case A held a total of 681 bottles (24.0 ft^3). This accounted for 51.3% of the volume filled in the baseline case and EE case B, and 48.9% of the volume filled in EE case A.

Results and Discussion

Results for all experiments were obtained while controlled environmental chamber conditions were maintained equally within the required settings and temperature/humidity limits over 24 h according to ASHRAE 72 (ASHRAE 2018). The average of product temperatures was also maintained within their required limits according to AHRI and FDA requirements (AHRI 2013; FDA 2017), despite minor fluctuations during post-defrost and periodic door openings.

Only the average, and not individual simulator temperatures, are required to be maintained within the AHRI/FDA-specified range.

Energy Savings

Total mean daily energy consumption by each of the three refrigerator display case technologies is shown in Figure 4. The baseline refrigerator case consumed 11.26 ± 0.01 kWh/day over 2 repeated 24-h experiments, EE case A consumed 4.30 ± 0.00 kWh/day over three repeated experiments (61.81% savings), and EE case B consumed 7.59 ± 0.03 kWh/day over two experiments (32.59% savings). Under equivalent environmental conditions, the baseline and EE case B manufacturer reported an expected ±10% variability in daily energy consumption. The variability observed here across individual 24-h experiments was much less, despite few repeated experiments evaluated. Additional cycles may have resulted in greater variability.

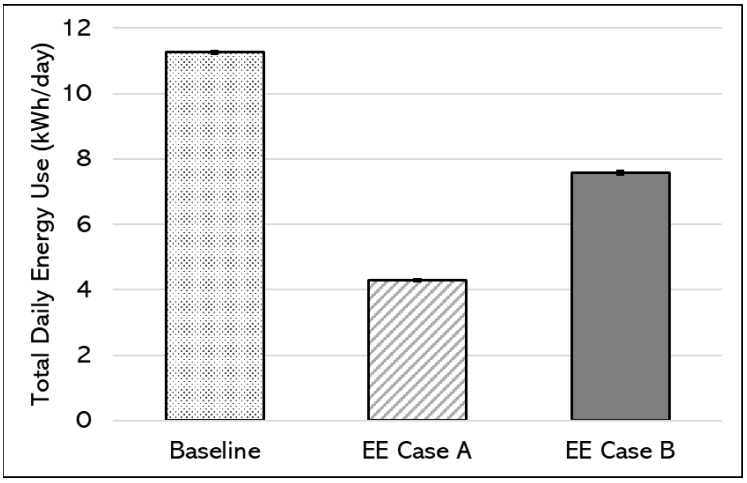

Figure 4. Total mean daily energy consumed by each of the three refrigerator case technologies. Error bars indicate standard deviation across individual 24-h experiments (negligible).

EE case A exhibited nearly two-thirds reduction in daily energy use, which can be attributed to both the improved thermodynamic properties of the R290 refrigerant as well as its various energy-efficient components. Improved insulation materials, including better-insulated door glass, meant that case interior temperatures would not increase during compressor off-cycles nearly as quickly as observed in the baseline case. This allowed the compressor to cycle on far less frequently and reduce overall energy consumption. In the baseline case, compressor cycling occurred 47 times (~ every 10 min) during the 8-h door opening period and 77 times total (~ every 30 min outside this period). The EE case A compressor, however, cycled on 6 times (~ every 80 min) during the 8-h door opening period and 17 times total (~ every 87 min outside this period).

The need for cooling was so infrequent due to the improved insulation that defrost cycles were unnecessary in EE case A. Compressor on-cycles occurred for a longer duration (20 min each in EE case A vs. 10 min each in the baseline). However, EE case A's total compressor on-time was around half the baseline's, likely due to a number of factors including the more-effective thermodynamic properties of the R290 refrigerant (Huber and McLinden 1992, NIST

2010) and improved cycle efficiency associated with higher condenser and evaporator fin-and-tube surface area (higher overall heat transfer coefficient, or "UA"). The reduced time needed to supply cooled air further reducing energy consumption. Every component of the case's vapor-compression cycle was more efficient, including the compressor energy, evaporator fan, and condenser fan, while providing equivalent cooling conditions. The lighting and controller were also much more efficient, consuming considerably less energy. The energy consumed by each of the case components can be observed in Figure 5.

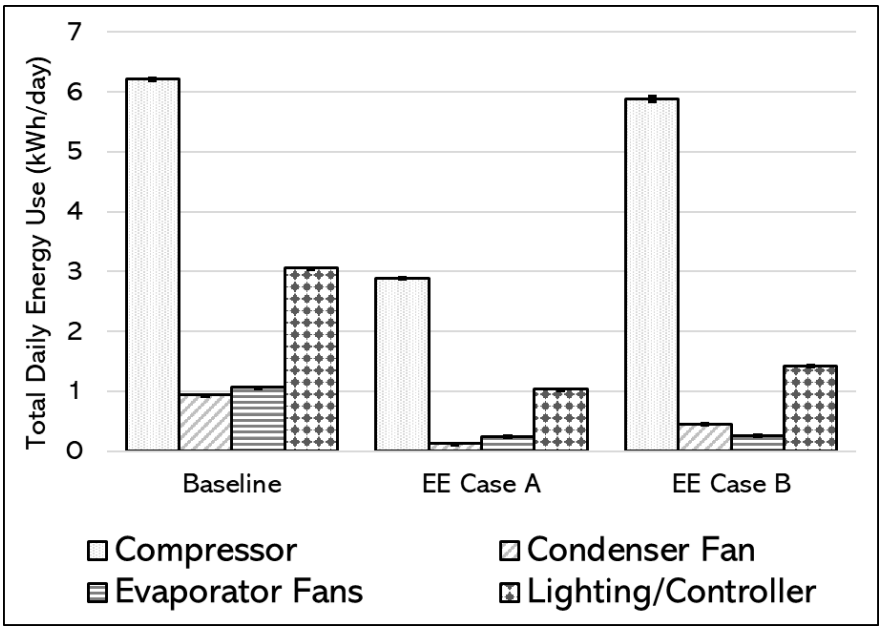

Figure 5. Mean daily energy consumed by each of the three refrigerator cases' components. Error bars indicate standard deviation across individual 24-h experiments (negligible).

Despite similar thermodynamic properties between R513A and R134A, EE case B exhibited nearly a one-third reduction in daily energy use. This can likely be attributed to the vapor-compression cycle components of the case, which include an evaporator and condenser with larger surface area and a number of coil rows contributing to a higher overall heat transfer coefficient/UA. The compressor cycled on 57 times (~ every 9 min) during the 8-h door opening period and 102 times total (~ every 19 min outside this period), which was more frequent than in the baseline. However, the approximately 8-min cycle time reduced the total compressor run time, resulting in lower daily energy use. The more-efficient lighting used by EE case B also further improved total energy savings. The fraction of the total energy consumed by each of the case components is shown in Table 3.

The different defrost cycles between cases also affected daily energy consumption. The baseline case had a single defrost cycle lasting 1:08 ± 0:08 (hours:minutes). EE case B had two defrost cycles which increased the total compressor off-time by 2:06 ± 0:09.

Table 3. Energy consumption and % contribution to total daily energy use by component

Component	Baseline (kWh/day)	% Total	EE Case A (kWh/day)	% Total	EE Case B (kWh/day)	% Total
Compressor	6.21 ± 0.02	55.08%	2.89 ± 0.01	67.41%	5.87 ± 0.04	73.30%
Condenser Fan	0.94 ± 0.00	8.31%	0.13 ± 0.00	2.95%	0.45 ± 0.00	5.67%
Evaporator Fans	1.07 ± 0.00	9.47%	0.24 ± 0.01	5.56%	0.26 ± 0.01	3.25%
Lighting/ Controller	3.06 ± 0.00	27.14%	1.03 ± 0.00	24.08%	1.42 ± 0.00	17.77%

Error indicates standard deviation. Differences between total measured energy and sum of components can be accounted for by internal losses in power meter circuits.

Power Savings

The mean power consumed by each of the three refrigerator display case technologies during compressor on-cycling is shown in Figure 6. Error bars indicate standard deviation across the total compressor on-cycle time across all evaluations. The baseline refrigerator case consumed 931 ± 372 W, EE case A consumed 559 ± 223 W (39.99% reduction in power), and EE case B consumed 892 ± 382 W (4.12% reduction in power). For both energy-efficient refrigerator case technologies, the power reduction is not nearly as significant as the total daily energy savings. This highlights the impact to the savings contributed by the reduced compressor on-cycle time. The contribution of various components to the mean power consumption can also be observed without the influence of cycle time in Figure 7.

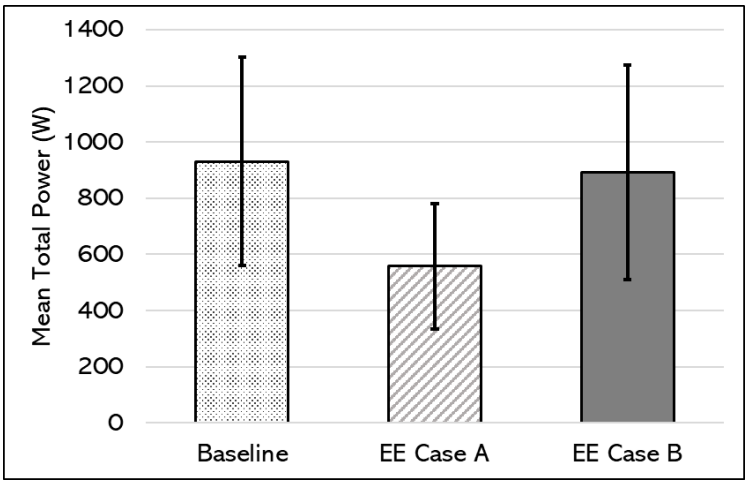

Figure 6. Mean power consumed by each of the three refrigerator cases' components during compressor on-cycling. Error bars indicate standard deviation across total compressor on-cycle time.

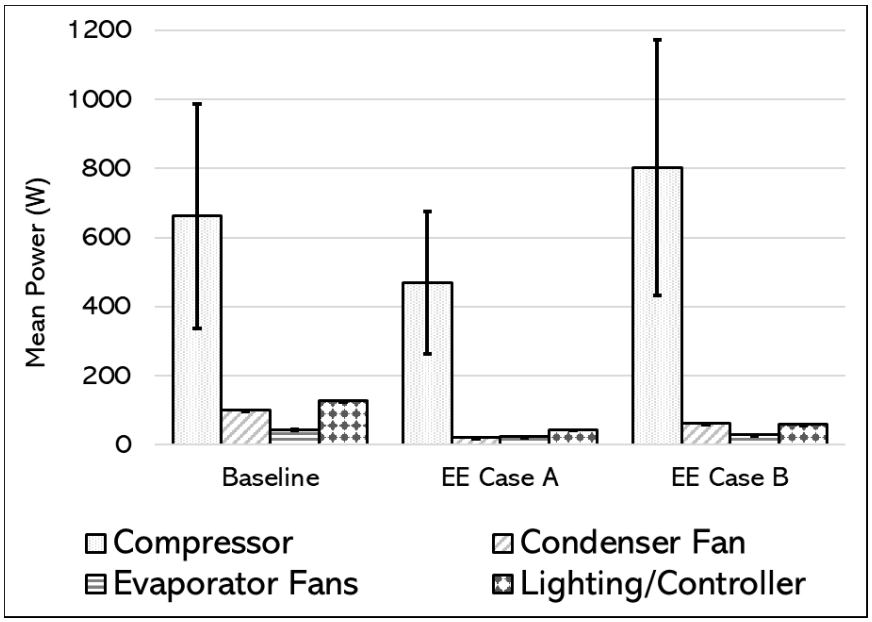

Figure 7. Mean power consumed by each of the three refrigerator cases' components during compressor on-cycling. Error bars indicate standard deviation across total compressor on-cycle time.

Table 4 shows the fraction of the total mean power consumed by each of the case components during compressor on-cycling. The mean component power consumption provides an expected scenario of the instantaneous power breakdown during compressor on-cycling. Here, we see that the components that operate continuously, including the lighting, controller, and baseline evaporator fans, contribute to a much smaller percentage of the overall power draw than they do to the daily energy consumption. The compressor consumes the large majority of the power during on-cycling. However, the power consumed by the compressor is not constant, and varies considerably during operation. For each case, there is an initial spike in power as the compressor turns on. The power draw gradually decreases at a similar rate in the baseline case and EE case B, as is evident by the variation in compressor power consumption shown in Figure 7. EE case A's compressor's power plateaus at a much faster rate than the other cases, yielding reduced variability.

Table 4. Resultant case component mean power consumption and % contribution to total mean power during compressor on-cycling

Component	Baseline (W)	% Total	EE Case A (W)	% Total	EE Case B (W)	% Total
Compressor	662 ± 325	70.92%	469 ± 205	84.22%	802 ± 370	84.27%
Condenser Fan	100 ± 49	10.66%	21 ± 9	3.73%	62 ± 29	6.51%
Evaporator Fans	44 ± 1	4.76%	24 ± 12	4.34%	28 ± 14	2.98%
Lighting/ Controller	127 ± 1	13.65%	43 ± 0.1	7.72%	59 ± 0.2	6.23%

Error indicates standard deviation across total compressor on-cycle. Differences between total measured power and sum of components can be accounted for by internal losses in power meter circuits.

Product Temperature Maintenance

The baseline case mean product temperature was 3.00°C (37.40°F), the EE case A mean temperature was 2.90°C (37.22°F), and the EE case B mean temperature was 3.02°C (37.44°F). Despite the fact that each case maintained the mean of product simulator temperatures within the required AHRI/FDA limits throughout operation (AHRI 2013; FDA 2017), the distribution of temperatures across product simulators and the variation in max and mean product temperatures were different for each case. During the 8-h period in which doors opened regularly, the compressor cycled more frequently, causing product temperatures to drop to a minimum of 2.33°C (36.19°F) in the baseline case and EE case B, but only 2.56°C (36.61°F) in EE case A. The coldest simulator dropped to -1.74°C (28.87°F) in the top shelf/center-rear simulator in the baseline, -1.15°C (29.93°F) in the top shelf/center-rear simulator in EE case A, and -0.02°C (31.96°F) in the middle shelf/center-rear simulator in EE case B during this time. Defrost cycles caused product temperatures to increase to the same maximum 3.70°C (38.66°F) in the baseline case and EE case B. EE case A only exhibited a 3.27°C (37.89°F) max mean temperature with no scheduled defrost. The warmest simulator temperature reached 6.74°C (44.13°F) in the bottom shelf/front-right simulator in the baseline, 6.17°C (43.11°F) in the bottom shelf/front-right simulator in EE case A, and 6.22°C (43.20°F) in the bottom shelf/front-left simulator in EE case B during this time. This tighter band in mean product temperature variation highlights the improved insulation materials used in the case, which limits the ability of the case's external environment to heat up product during compressor off-cycling. This tighter temperature range was also partially due to EE case A's lower cut-in temperature. However, the case's higher cut-in temperature highlights its more effective air circulation from the evaporator.

The mean temperatures of individual product simulators across the 24-h evaluation period in the baseline case are provided in Table 5. The middle shelf/center-rear simulator was the coldest average product simulator temperature, and the bottom shelf/front-right simulator was the warmest. Because door openings caused warmer air to enter the case interior, the compressor responded by cycling more rapidly during this period. This excessively cooled simulators below the evaporator discharge grille, which lowered the average product temperature. Product simulators closer to the door and sides were warmed to higher temperatures by conduction and radiative heat transfer during defrost cycles, raising the average product temperature during this period.

Table 5. Baseline refrigerator case average of individual product simulator temperatures

Top Shelf	Left	Center	Right
Rear:	2.98°C	-0.31°C	4.11°C
Front:	4.97°C	2.37°C	5.98°C
Middle Shelf	Left	Center	Right
Rear:	1.56°C	-0.38°C	2.40°C
Front:	2.97°C	1.12°C	4.07°C
Bottom Shelf	Left	Center	Right
Rear:	2.33°C	1.53°C	3.66°C
Front:	5.02°C	3.52°C	6.23°C

The individual mean product simulator temperatures in EE case A are provided in Table 6. The middle shelf/center-rear was the coldest average product simulator temperature, and the bottom shelf/front-right simulator was the warmest. The coldest product simulators below the evaporator reached lower temperatures than observed in the baseline case. The warmest simulators, however, located at the corners of the case against the doors, did not reach as extreme temperatures observed in the baseline. This improvement can be attributed to the improved thermal resistance of the case's glass doors.

Table 6. EE case A average of individual product simulator temperatures

Top Shelf	Left	Center	Right
Rear:	2.07°C	-0.29°C	1.86°C
Front:	4.27°C	1.74°C	5.02°C
Middle Shelf	Left	Center	Right
Rear:	1.54°C	-0.50°C	2.49°C
Front:	3.46°C	0.96°C	4.20°C
Bottom Shelf	Left	Center	Right
Rear:	3.96°C	1.77°C	5.13°C
Front:	5.13°C	3.93°C	5.75°C

Individual mean product simulator temperatures in EE case B are provided in Table 7. The top shelf/center-rear product simulator exhibited the coldest average temperature, and the bottom shelf/front-left product simulator exhibited the warmest temperature. The coldest and warmest simulator temperatures were not as extreme as in the baseline case. However, some simulator temperatures were further from the total mean. The more even distribution of temperatures was indicative of the improved air circulation in the case by the introduction of a more effective evaporator panel utilizing more robust evaporator fans.

Table 7. EE case B average of individual product simulator temperatures

Top Shelf	Left	Center	Right
Rear:	1.82°C	0.55°C	0.50°C
Front:	5.18°C	3.33°C	3.67°C
Middle Shelf	Left	Center	Right
Rear:	2.12°C	0.60°C	0.80°C
Front:	4.56°C	2.61°C	3.15°C
Bottom Shelf	Left	Center	Right
Rear:	3.50°C	2.69°C	3.19°C
Front:	5.69°C	4.20°C	5.58°C

Conclusions

Overall, the energy efficient units showed favorable performance compared to the baseline display case. The energy-efficient case utilizing improved insulation materials/doors, R-290 refrigerant, and higher-UA heat exchangers provided the largest energy savings. That case

consumed 61.8% less daily energy (4.30 ± 0.00 kWh/day) and 40.0% less mean power (469.05 ± 205.33 W) than the baseline case. These savings were achieved while maintaining an average product temperature of 2.92°C (37.26°F), which varied between 2.33°C (36.19°F) during door openings and a maximum 3.70°C (38.66°F). Savings may be attributed to the more-efficient heat exchangers, effective insulation, and thermodynamically efficient refrigerant that reduced the temperature lift.

The other energy-efficient display case, which utilized R-513A refrigerant, efficient lighting and fan motors, and higher-UA heat exchangers consumed 32.6% less daily energy (7.59 ± 0.03 kWh/day vs. 11.26 ± 0.01 kWh/day). Despite its higher compressor power use than the baseline, due to energy-efficient lighting and fan motors, this display case consumed 4.1% (892.38 ± 382.47 W vs 930.71 ± 371.52 W) less power. The energy and power savings were achieved while maintaining an average product temperature of 3.01°C (37.42°F), which varied between the same 2.33°C (36.19°F) during door openings and 3.70°C (38.66°F) during defrost as the baseline case. The more effective heat exchangers and fans, coupled with more efficient fan motors, lighting, and controls significantly reduced this fixture's mean power consumption and daily energy use compared to the baseline.

The mean product simulator temperatures were maintained within the AHRI/FDA allowable limits for each experimentation scenario. For all three display cases, the compressor accounted for the largest contribution to the total mean power, ranging between 71% in the baseline to 84% in both the energy-efficient cases.

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