A fair comparison of CO2 and propane used in light commercial applications featuring natural refrigerants

Stefan Elbel
University of Illinois at Urbana-Champaign, United States of America / Creative Thermal Solutions, United States of America, elbel@illinois.edu

Matej Visek
Creative Thermal Solutions, United States of America, matej.visek@creativethermalsolutions.com

Pega Hrnjak
pega@illinois.edu

A Fair Comparison of CO_2 and Propane Used in Light Commercial Applications Featuring Natural Refrigerants

Stefan ELBEL[a,b,*], Matej VISEK[b], Pega HRNJAK[a,b]

[a] Air Conditioning and Refrigeration Center
Department of Mechanical Science and Engineering
University of Illinois at Urbana-Champaign
1206 West Green Street, Urbana, IL 61801, USA

[b] Creative Thermal Solutions, Inc.,
2209 North Willow Road, Urbana, IL 61802, USA

[*] Corresponding Author
Email: elbel@illinois.edu

ABSTRACT

Glass door merchandisers are used in grocery and convenience stores to display chilled beverages or refrigerated foods. Among the natural refrigerants available, only propane (R290) and carbon dioxide (CO2, R744) are realistic candidates to replace currently used HFC refrigerants. While many of the thermodynamic and transport properties of R290 are favorable over R744, propane is listed as an ASHRAE A3 refrigerant and therefore bears an inherent flammability risk even when the refrigerant charge is limited to 150 g or less as required by applicable safety standards. While it seems possible to design and implement safe, low-charge hydrocarbon refrigeration systems, some manufacturers prefer solutions that completely eliminate the flammability risk and therefore focus on R744, which is a non-flammable, non-toxic, natural refrigerant listed as ASHRAE A1. The challenges encountered with R744 are mostly due to lower performance at elevated ambient temperatures requiring a transcritical cycle, which often makes R744 systems more expensive for designs that are aimed at providing comparable cooling capacity and energy efficiency. Therefore, the component and system design challenges encountered with the two fluids are very different, which drives design solutions in very different directions. This study elaborates on the different fluid-specific challenges that are inherent to the each of the two refrigerants and demonstrates the resulting consequences in terms of system and component design. Since the cooling target is the same in both cases, the pros and cons of R290 and R744 can be fairly compared and meaningful conclusions can be drawn.

1. INTRODUCTION

The aim to introduce environmentally friendly refrigerants is among the most dominant technology drivers in numerous sectors of the HVAC&R industry. It appears that some of the major companies that purchase light commercial refrigeration systems for beverage display cooling are driving this part of the industry towards the use of natural rather than synthetic refrigerants. One of the major players in the soft drink industry recently announced the goal to implement hydrofluorocarbon (HFC) free technology for all new cold-drink equipment by 2015. The same company announced that as of now it has installed more than 1.4 million HFC-free coolers using natural refrigerant in the marketplace (TCCC, 2016). Among the natural refrigerants available, only propane (R290) and carbon dioxide (CO_2, R744) are realistic candidates to replace currently used HFC refrigerants. While many of the thermodynamic and transport properties of R290 are favorable over R744, propane is listed as an A3 (ASHRAE Standard 34, 2013) refrigerant and therefore bears an inherent flammability risk even when the refrigerant charge is limited to 150 g or less as required by applicable standards. While it seems possible to design and implement safe, low-charge hydrocarbon refrigeration systems, some manufacturers prefer solutions that completely eliminate the flammability risk and therefore focus on R744, which is a non-flammable, non-toxic, natural refrigerant listed as ASHRAE A1

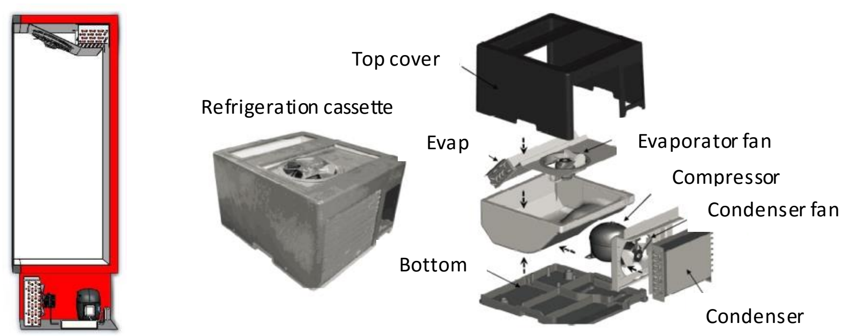
(ASHRAE Standard 34, 2013). While R290 system designs are relatively similar to currently used R134a coolers, the situation with R744 is quite different. It has been shown that R744 can deliver acceptable cooling performance in a small-size package (DeAngelis and Hrnjak, 2005; Cecchinato and Corradi, 2011). The major drawback with R744 is the reduced cooling performance at elevated outdoor temperatures and the high working pressures (Kim *et al.*, 2004). Glass door merchandisers are often used at ambient conditions between 25 ^{o}C and 40 ^{o}C, at which R744 operates in a transcritical cycle, characterized through subcritical heat absorption in the evaporator and supercritical heat rejection in the gas cooler. Slightly adjusted component designs can be implemented to manage the higher working pressures, which typically range from 9 MPa and 12 MPa depending on the ambient temperature. However, the reduction of energy efficiency at elevated ambient temperatures is much more difficult to overcome, especially when overall system cost should not increase to ensure competiveness with currently used HFC systems.

This paper summarizes the most relevant R290 and R744-specific design issues and differences, and also compares them to conventional bottle coolers that typically use HFC R134a. Experimental data for different systems using these two natural refrigerants are presented and compared. In order to make the comparison as fair as possible, similar levels of technology were used for both systems, for example fixed speed compressors, round-tube-plate-fin (RTPF) heat exchangers, and fixed geometry expansion devices. The results show that it is possible to achieve similar performance in terms of capacity and energy efficiency for both refrigerant options even when using the same, relatively low-cost technology; while these results may be somewhat counter-intuitive they demonstrate that well engineered systems can overcome even large technical hurdles.

2. QUALITATIVE COMPARISON

2.1 Design Constraints

The global glass door merchandiser market comprises several million units per year. The coolers are available in a wide range of sizes. The smallest table-top units are designed for approximately 20 soft drink cans with a beverage volume of 330 ml or 355 ml per can. The largest available glass door merchandisers have multiple access doors and can hold up to approximately 2000 cans. The majority of the available systems are split-type, as shown in Figure 1a. In that system type the evaporator is mounted on the interior ceiling of the cabinet and the evaporator is connected to the condensing unit components mounted underneath the cabinet via extended refrigerant tubing. Another system type is shown in Figure 1b. In this cassette-style system all refrigeration components, including the evaporator, are mounted in a removable cassette that can easily be connected to the insulated beverage display cabinet.



a) split system b) cassette type system (Waltrich *et al.*, 2010)

Figure 1: Available glass door merchandiser system types

The cassette discharges the cooled air with its own blower into the cabinet in the upwards direction where it reverses its flow direction before it is entrained back into the cassette as return air. While the cassette-type has advantages in terms of system assembly and maintenance, uniformity of air distribution and the resulting product temperature uniformity are often not as good as in split-type systems. As an immediate consequence of the different air flow management, cassette-type systems often require slightly stronger evaporator fans than comparable split systems. While for R744 the system style should not matter much, the implications are much stronger for R290 designs. More compact cassette systems allow for lower refrigerant charges, especially when the internal volume of the high-pressure side can be reduced where higher refrigerant densities are encountered.

2.2 Performance Requirements

A large variety of different test standards exist, some of which are available in the open literature (e.g. ANSI/AHRI 1201, 2013), while others are proprietary to the equipment suppliers in the beverage industry. Depending on the application, testing requirements can be very different for different products. While cabinet temperature set points for carbonated soft drinks are above 0 ºC, certain types of beer coolers need to pass tests at product temperatures below 0 ºC. Furthermore, a glass door merchandiser of a given size is often rated at different ambient conditions (often 3 different ambient conditions: low, medium, and high) with corresponding humidity levels. That means that the same size cabinet may require larger refrigeration components in order to be able to provide higher capacities that are required at higher ambient conditions.

The cooling capacity of most coolers is checked through the time required to cool a full load of warm cans to the desired set point. Depending on the temperature class and specific cooler application, pull down time targets vary, but are often on the order of 18 to 24 hours. Another requirement at the end of the pull down is that the products stay within certain maximum, average, and minimum temperature bands that again vary slightly between the different cooler applications. Some coolers are also investigated in tests that are designed to simulate realistic cooler operation. For example, a cooler that is half filled with cans that have been cooled to the desired set point is opened, and an additional half load of warm cans is added in a specified amount of time. The additional sensible and latent load has to be removed in a given amount of time and at the end of the pull down period, all products have to be again within the specified range of temperatures.

Energy efficiency is typically determined when all products have reached the desired set point and the system is running continuously for a specified amount of time. Fixed speed compressor systems cycle on and off many times during this test to yield an average daily energy consumption. Systems with variable speed compressors can be controlled in such a way that the compressor runs continuously and cycling of the system is not necessary. These systems often have lower energy consumption than their fixed speed counterparts due to the use of more efficient compressor motors and the fact that continuous compressor operation eliminates losses that occur every time the compressor cycles on and off.

In addition to the test procedures outlined above, many additional tests are carried out that take into account realistic operating scenarios and intelligent cooler controls. It should also be noted that in some regions of the world government specific energy consumption requirements exist that may be more stringent than those specified by beverage companies and equipment suppliers.

2.3 Comparison of Design Challenges for Transcritical R744 and R290 Systems

There are a number of design challenges that need to be addressed when designing transcritical R744 components for light commercial applications, such as glass door merchandisers. R744 is known to have an approximately eight times higher volumetric capacity than R134a, which means that for identical component sizes, the capacity of an R744 system can be considerably larger than for an R134a system. The comparison between R744 and R290 shows that the volumetric capacity of R744 is still about 6 times larger than that of R290; that means that the volumetric capacity of R290 is approximately 1.3 times larger than that of R134a. This is interesting, because both the vapor density and the liquid density of R290 are much lower than for R134a, but it is the higher latent heat of vaporization of R290 that overall results in the higher volumetric capacity. This is an important trend, because it allows reducing the internal volume of an R290 system in comparison to an R134a system. That is because the required refrigerant mass flow rates for the R290 system are less than for R134a (despite R290's lower vapor density). If the goal is to design for comparable pressure drop, the lower R290 mass flow rate allows to use smaller diameter tubing which helps to achieve low refrigerant charge.

Because of the undisputable flammability of R290 it is necessary to reduce refrigerant charge to low levels. Global differences exist for specific system types that use A3 flammable refrigerants, but the majority of systems are designed to use no more than 150 g when used in commercial applications. This charge is further reduced to approximately 50 g for household refrigerators. While realizing domestic refrigerator designs with 50 g of charge (mostly isobutane, R600a) is typically not too big of a challenge for appliances sold in Europe, the much larger internal volumes of typical refrigerators used in homes in the United States require larger cooling capacities and component sizes, making it much more challenging to achieve all performance targets for the larger units. For light commercial applications the design intent is to maximize the available cooling capacity that can be achieved with 150 g, although it appears that the industry pushes for approval of higher charge amounts. The capacity envelope can be pushed with low-internal volume heat exchangers, for example microchannel designs, that make the biggest difference when used on the high pressure side where high refrigerant (liquid) densities are encountered. For the same reason, it is important to use heat exchanger (especially condenser) designs that do not require excessive amounts of liquid to fill header tubes. For R290 systems, such low charge designs can be achieved by using serpentine microchannel tube designs (as shown in Figure 2a) that eliminate headers altogether. Often, higher refrigerant-side pressure drop results from the use of serpentine condenser designs in comparison to microchannel designs that use headers to allow for many parallel microchannel tubes. However, the lower refrigerant-side mass flow rate in comparison to R134a, in combination with higher thermodynamic cycle efficiency of R290 over R134a, caused by smaller throttling loss and lower desuperheating losses at the condenser inlet, help mitigate this problem, i.e. the COP of an R290 system is still higher than for a comparable R134a system even if the pressure drop across heat exchangers is slightly increased.

Achieving high energy efficiency with R744 is much more demanding, especially at elevated ambient temperatures at which many bottle coolers have to operate. While R744 can be a very efficient refrigerant at lower temperatures, even in a transcritical cycle, the cycle COPs degrade rapidly at higher temperatures due to the supercritical heat rejection process in the gas cooler. The crossover temperature at which the performance of the transcritical R744 cycle becomes inferior to a comparable R134a or R290 systems occurs often at around 25 ^{o}C to 30 ^{o}C, of course depending on the specific application and performance characteristics of the baseline system. While energy consumption of the majority of R744 bottle coolers will be rated at temperatures higher than 30 ^{o}C, it is possible to optimize the cooler performance for the specific rating condition.

It should be noted that the evaporator design can be of conventional round-tube-plate-fin design, because for typical vapor mass fractions observed at the evaporator inlet (approximately 20%), the high void fraction causes the refrigerant tubes to be relatively empty, causing the evaporator to contribute relatively little to the overall refrigerant charge of the R290 system, as shown in Figure 2b. By being able to use round-tube-plate-fin designs (often all-aluminum) for the evaporator, it is possible to use low-cost designs that are also more forgiving to frost formation and condensate drainage than microchannel designs. Interestingly, the same evaporator design has been found to deliver very good system performance in the transcritical R744 cycle. Aluminum microchannel designs are still seen as more prone to corrosion problems, even in comparison to aluminum round-tube designs, which could be linked to the reduced wall thickness of microchannels, and different alloys that are used for the aluminum extrusions.

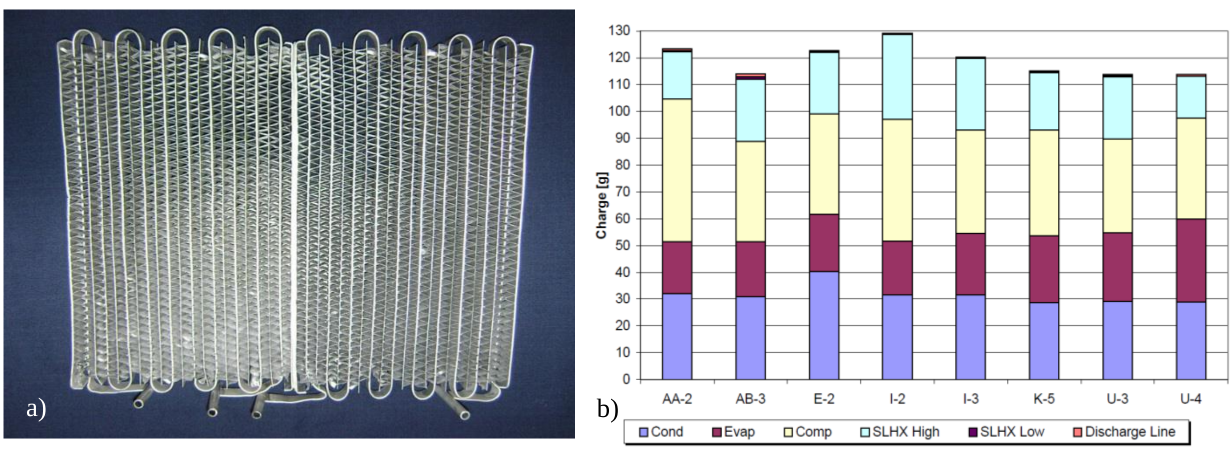


Figure 2: a) Headerless microchannel heat exchanger design; b) Typical refrigerant charge distribution of a light commercial R290 system at different test conditions AA-2, AB-3, E-2, I-2, I-3, K-5, U-3, U-4 (Hoehne and Hrnjak, 2004)

When it comes to internal heat exchangers, or liquid-to-suction-line heat exchangers, the situation is quite different for the different refrigerant choices. While it is common for R134a system to bring the capillary tube in contact with the suction line, R744 requires higher internal heat exchanger effectiveness in comparison to R134a. Padilla Fuentes *et al.* (2015) have demonstrated how to increase IHX effectiveness by extending the larger diameter tube coming from the gas cooler in a transcritical R744 system and by bringing that tube extension in contact with the suction line. This helps because of the larger available temperature differences in this particular IHX design (the temperature of the high pressure refrigerant starts to drop immediately when sent through the capillary tube). A more effective IHX would also improve the COP of R134a, but by a much smaller absolute amount. Therefore, it is the transcritical R744 system that needs the more effective IHX to achieve good overall performance. It should be noted, however, that an IHX effectiveness beyond 80 to 90% may result in substantial capacity loss because the lower suction density at the compressor inlet reduces the mass flow rate the compressor can deliver. At the same time, too high compressor inlet temperatures may cause excessive temperatures at the compressor exit. Nevertheless, it has been observed in numerous experiments that conventional capillary tube to suction line contact may not yield high enough IHX effectiveness in case of R744. The IHX situation is quite different in case of R290. While theoretically the IHX can help to improve R290 cycle efficiency it can, depending on the specific design, easily lead to an increase of the required refrigerant charge, especially if more effective IHX designs, such as the one described above for R744 are used. Since the thermodynamic benefit of using an IHX for R290 is only on the order of a few percent (even slightly less than for R134a), and the more important goal is to realize low refrigerant charges, typical R290 designs should still use the conventional capillary tube to suction line IHX design. This could become a bigger design challenge should this system type require adjustable expansion devices to further optimize the system performance when used with variable speed compressors or when maximum possible energy efficiency is needed at different ambient temperature conditions. In that case, the capillary tube would be replaced with a valve, which could possibly extend the liquid line length of split-type glass door merchandisers. Switching to an expansion valve could have dramatic impact on overall refrigerant charge inventory of an R290 system, since the capillary tube provides expansion at very low internal volume (in comparison to an orifice that has to be connected with larger diameter tubing to be close to the top-mounted evaporator).

Another issue that is directly related to refrigerant charge is the tolerance to small losses of refrigerant over time. The system pressures of R744 are much higher than those of R134a and R290, so the tendency for R744 to leak is higher for a leak of same geometric dimensions. However, since refrigerant charge is much less of a concern in R744 systems, additional charge can be added through a charge accumulation volume (typically located at the evaporator exit in a transcritical R744 system) to compensate for possible leaks. This is much more challenging for R290 systems where the allowable refrigerant charge is limited due to legal requirements. The challenge is that a system that has been rigorously designed for extremely low refrigerant charge amounts will be much more sensitive to even smallest refrigerant losses, because missing refrigerant liquid either at the evaporator inlet or at the condenser exit will immediately result in lower performance levels. The addition of extra charge to compensate for leaks (low-side accumulator or high-side receiver) is not easily achieved, because the high density liquid quickly drives refrigerant charges beyond the allowed limit. That means that a low-refrigerant charge system requires leak-free (or at least extremely low-leak) technology or otherwise it is difficult to guarantee unchanged system performance over the life of the appliance.

It has been already mentioned that the evaporator design of an R744 or an R290 system can be very similar to standard R134a evaporator designs. However, when it comes to the heat rejection heat exchanger (gas cooler in R744 or condenser in R290), the use of low-internal volume heat exchangers is the preferred option. It is interesting that both R744 and R290 benefit from microchannel (or small diameter round tube) designs, but for very different reasons: the microchannel in R744 reduces the required wall thickness needed to withstand high pressures and makes component design much more manageable. It also enhances the air-side of the gas cooler which is essential for good performance. For R290 on the other hand, the microchannel condenser allows for charge reduction in comparison to conventional round-tube-plate-fin designs.

A key difference exists when it comes to designing the heat rejection heat exchanger: while R290 condenser designs can be very similar to R134a designs, an R744 gas cooler greatly benefits from counterflow design to further reduce the refrigerant exit temperature. Because of the continuous temperature glide that single-phase, supercritical R744 undergoes during the gas cooling process the temperature differences between air and refrigerant vary substantially between refrigerant inlet and outlet. The row closest to the air inlet is essential to achieve lowest possible approach temperature differences. In comparison to R290 or R134a an R744 heat rejection heat exchanger achieves much better performance when the face area can be increased rather than the depth of the heat exchanger. That results in large surface areas in the region of the heat exchanger where the available temperature difference is the smallest (refrigerant exit row) and reduced driving force (temperature difference) has to be compensated by larger surface area. It should be noted that increasing the face area of the heat exchanger requires a fan with sufficient air flow rates to ensure sufficient air face velocities, often on the order of 2 m/s. It should be mentioned, however, that neither R744 nor R290 systems require microchannel technology on the heat rejection side; it just greatly facilitates and improves the design. The systems presented below used round-tube designs while still meeting all performance criteria. Especially for R744 the use of small diameter steel tubes proves to be beneficial because of the low cost and the high mechanical strength of the material.

When it comes to compressors, similar technology is available, for example single piston, reciprocating designs. Compressor efficiencies for the available designs are comparable, although at this point a much wider selection exists for R290 compressors, including variable speed options that are not yet available (in mass production) for small-capacity R744 compressors. With additional R744 compressor suppliers most likely entering the market in the near future, it remains to be seen if the additional competition will help to further reduce the cost. Currently available compressors for R744 sell for approximately 30% more than comparable R134a or R290 designs. One could argue that an R744 compressor will always be more expensive because of the higher pressure and motor torque requirements, but it remains to be seen if this theoretical difference will translate into substantial cost differences when high pressure compressors are produced in similar numbers. When it comes to R290 compressors the amount and type of compressor lubricant becomes very important, as hydrocarbons such as R290 readily dissolve into lubricants in much larger quantities than R744 and R134a. For R290 it seems that AB and POE oils are preferred by many compressor suppliers.

In addition to these differences there are a number of other factors that need to be considered: because of R290's lower saturation pressures and high cycle efficiency R290 cycle designs are much more forgiving than R744 designs, where even small design mistakes such as optimum high side pressure, IHX effectiveness, refrigerant charge, or gas cooler circuitry have much larger effects on system performance. That means that R744 requires more experienced designers who understand that strictly applying R134a design methods to R744 systems will result in sub-optimal results. On the other hand, a well-designed system R744 results in very good performance and energy efficiency, especially when used at moderate outdoor temperatures or in conditioned indoor locations. The reduced risk of using a non-flammable refrigerant compensates for the larger design effort to overcome performance drawbacks at high ambient temperatures. While R290 system design is more similar to R134a, R290 (and also R744) require well trained assembly workers and maintenance technicians, although for different reasons: the higher pressure of R744 does pose a higher risk during maintenance work and proper system charging is very important for R744 (it is charged by mass and not by saturation pressure, which makes it easy to overcharge R744 systems). On the other hand, flammability of R290 is of big concern during assembly and maintenance, especially when charging equipment is present that contains much larger R290 amounts than what is found in a single cooler.

3. EXPERIMENTAL RESULTS

After successful implementation of the design aspects discussed in the foregoing sections, the cooling performances of a transcritical R744 glass door merchandiser and a comparable beverage cooler using R290 were determined experimentally. The test systems had an internal cabinet volume of approximately 600 l and a capacity of 600 soft drink cans, each having a volume of 355 ml. The systems were tested at an ambient temperature of 32 ºC and a relative humidity of 65%. The overall heat loss rate of the insulated cooling cabinet was measured in a reverse-heat-leak test. The UA value of the cabinet was determined to be approximately 4.5 W/K. For every shelf, the top most cans in the corners closest to the front of cooler were filled with a 33% propylene glycol and 67% water solution. The top most cans of each shelf on the back of the cooler located on the left and right edges closest to the cooler walls were also filled with a 33% propylene glycol and 67% water solution. The mixture containing cans were instrumented with Type-T thermocouples and served as the temperature representing the product temperature distribution in the cooler. The allowed time to bring all products from ambient conditions to the desired temperature range between 0 ºC and 7 ºC was 19 hours. Subsequently, energy consumption tests of the coolers were run for a 24 hour period. During the energy consumption test period the cooler door is closed and the cooler cycles between on and off periods for a single speed compressor in order to maintain the product temperatures between 0 ºC and 7 ºC, as specified by the manufacturer of the cooler.

The performance results in terms of pull down time and energy consumption, both shown as relative values in comparison to an identical cooler using an optimized production R134a refrigeration system which serves as a baseline in this case, are shown in
Figure **3**.

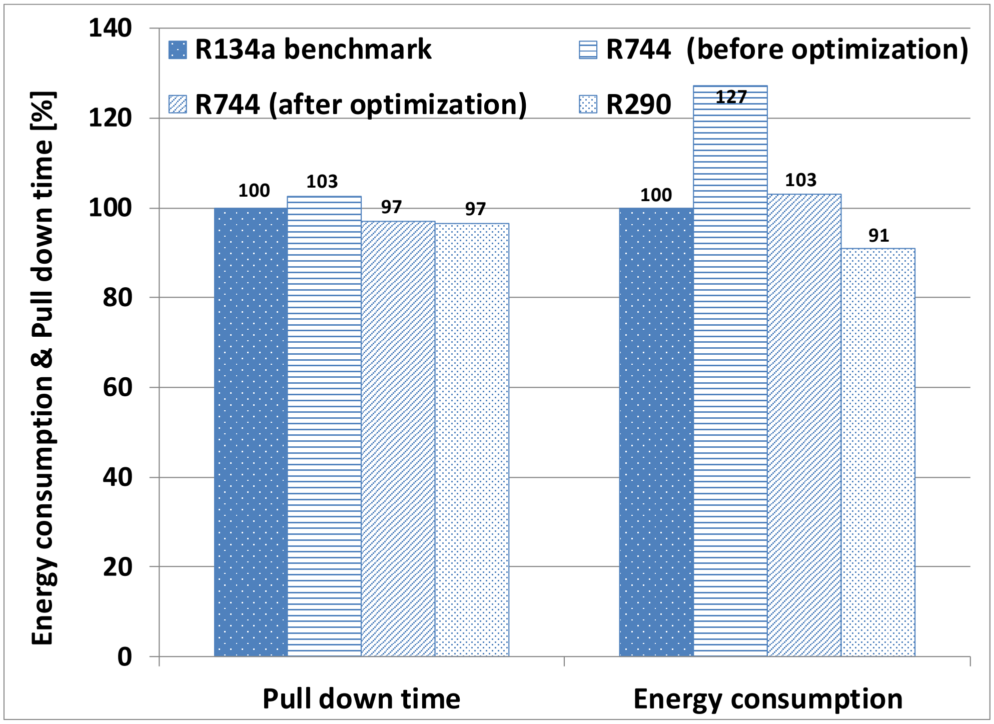


Figure 3: Comparison of pull down times (cooling capacity) and energy consumption (COP) of production R134a (benchmark) and prototype transcritical R744 and R290 glass door merchandiser systems

For the transcritical R744 system, the performance of an originally non-optimized system is also shown for reference. It can be seen that the improved R744 system which had an improved round-tube gas cooler, IHX, and fully optimized refrigerant charge and capillary tube, outperformed the benchmark R134a system in pull-down time by a small margin. More important is the comparison of energy consumption: while the original R744 system showed a 27% higher energy consumption than the baseline, the improved transcritical R744 system was on par by achieving essentially the same energy consumption as the baseline R134a cooler (103% vs. 100%). These results are very impressive, considering that the improved R744 system did not make use of any high cost components, but used low-cost technology comparable to the baseline R134a system. In fact, the overall system cost of the R744 system was just a few percent higher than for the R134a baseline.

The pull down time of the R290 system was measured to be identical to that of the optimized R744 system. The energy consumption of the R290 system is approximately 9% less; this result is not too surprising considering the excellent thermodynamic properties and theoretical cycle efficiency of the hydrocarbon system. Figure 4 shows detailed pull down data for the R290 system. This data can be compared to the pull down performance of the optimized R744 system presented in Figure 5. It should be noted that the pull down time is determined from how long it takes to bring all product temperatures to within the specified temperature band of 0 °C to 7 °C. For example, the product temperature of the coldest product shown in Figure 5 is slightly below 0 °C at which point the compressor is turned off for the first time and its controller begins to cycle the compressor on and off. This causes the products to warm up. Pull down time continues to be counted until the coldest can has warmed up sufficiently to satisfy the temperature requirement (while all other products stay within the specified temperature band as well).

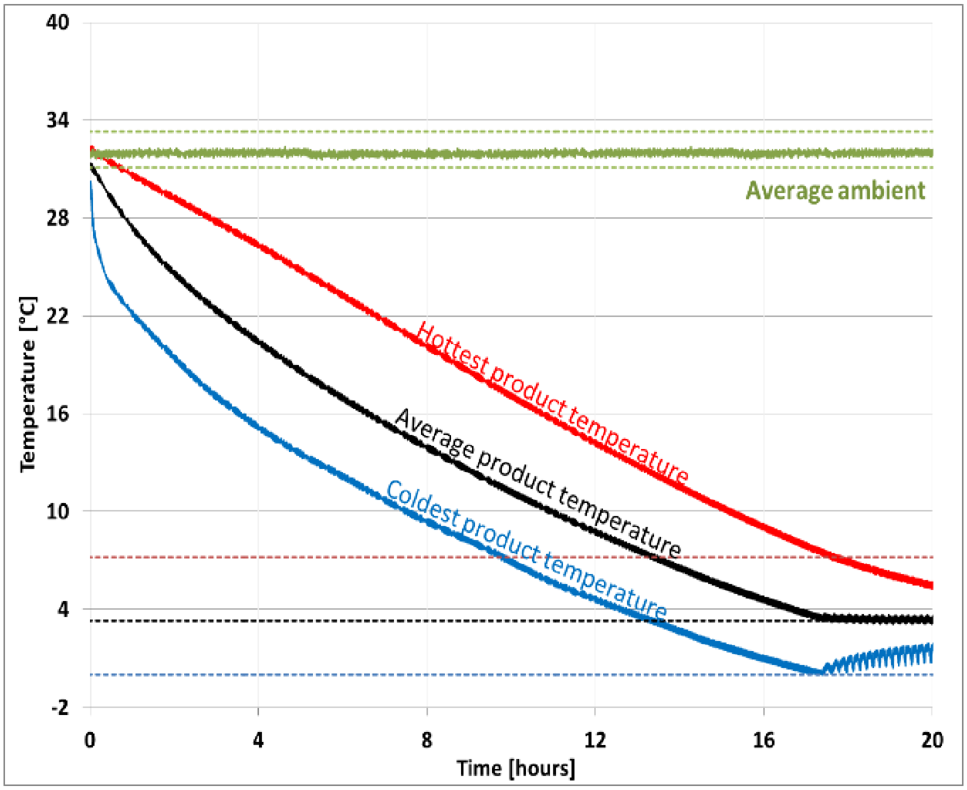


Figure 4: Experimental pull down times for R290 system

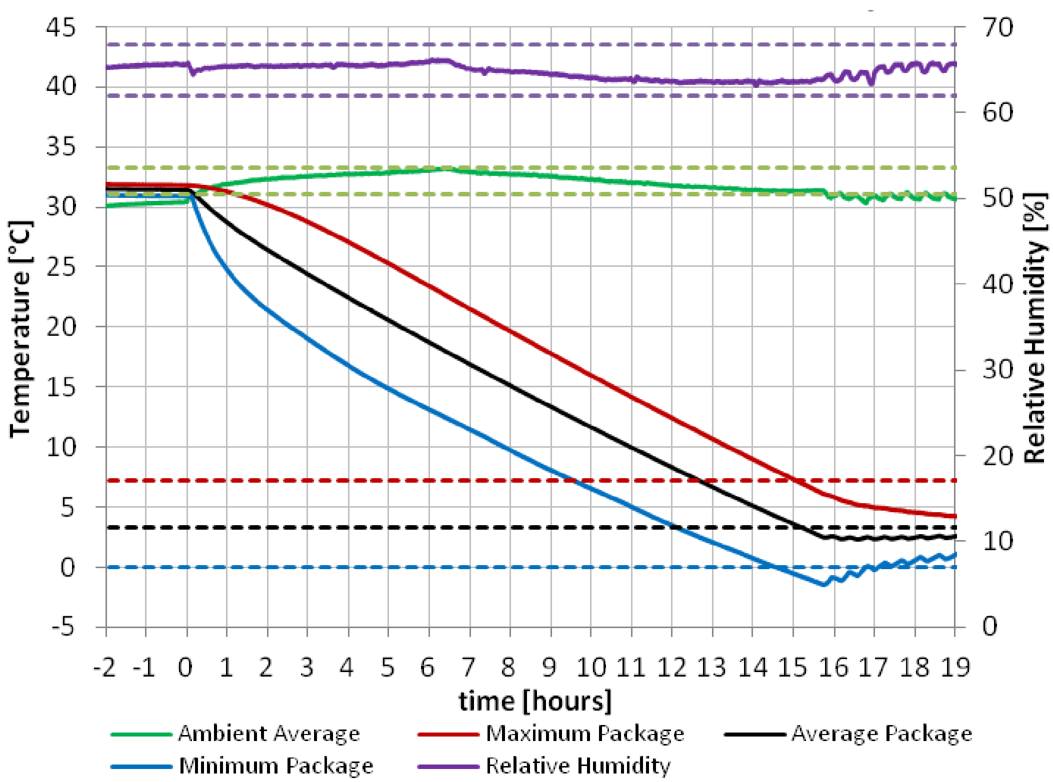


Figure 5: Experimental pull down times for optimized R744 system

4. CONCLUSIONS

This paper demonstrated that it is possible to design high-performance transcritical R744 and R290 glass door merchandiser systems that deliver cooling capacities and energy efficiencies that are comparable to results obtained with an R134a benchmark system. While the energy efficiency of R290 is undoubtedly higher than for R744, the results show that R744 can deliver satisfactory results even at elevated ambient temperatures. The discussion of the fluid specific differences shows that the more conventional design and the higher energy efficiency of R290 are offset by the higher risk due to flammability. R744 on the other hand is far less forgiving in terms of design mistakes and shows lower energy efficiency at higher ambient temperatures, yet has the advantage of being non-flammable. Proper training of maintenance and assembly technicians is required in both cases, although for slightly different reasons. Probably most surprising is the fact that system cost is not too different at this point, and that the slightly higher prices for R744 systems are expected to be lower in the future when additional component manufacturers become available.

NOMENCLATURE

Symbols and Abbreviations

COP	coefficient of performance (-)	IHX	internal heat exchanger
Cond	condenser	Low	low pressure side
Comp	compressor	RTPF	round-tube-plate-fin
Evap	evaporator	SLHX	suction line heat exchanger
HFC	hydrofluorocarbon	UA	overall heat loss rate (W/K)
High	high pressure side		

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