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Stefan Elbel
Creative Thermal Solutions, USA / University of Illinois at Urbana-Champaign, USA,
stefan.elbel@creativethermalsolutions.com

Yadira Padilla Fuentes
Creative Thermal Solutions, USA, yadira.padilla@creativethermalsolutions.com

Chad D. Bowers
Creative Thermal Solutions, USA, bowers.cd@gmail.com

Predrag S. Hrnjak
pega@creativethermalsolutions.com

Successful Design, Implementation, and Validation of Transcritical R744 Technology for Beverage Display Coolers

Stefan ELBEL[1,2,*], Yadira PADILLA FUENTES[1], Chad BOWERS[1], Pega HRNJAK[1,2]

[1]Creative Thermal Solutions, Inc.
2209 North Willow Road, Urbana, IL 61802, USA
Phone: (217) 344-7663, Fax: (217) 344-7552, Email: stefan.elbel@creativethermalsolutions.com

[2]University of Illinois at Urbana-Champaign
Department of Mechanical Science and Engineering
1206 West Green Street, Urbana, IL 61801, USA

[*]Corresponding Author

ABSTRACT

Glass door merchandisers are used in grocery and convenience stores to display chilled beverages or refrigerated foods. Among other possible choices for this application, carbon dioxide (R744, CO_2) is seen as a promising low global warming potential (GWP) refrigerant alternative that is non-flammable and non-toxic. While R744 itself is less costly than some synthetic alternatives, successful implementation of high-performance, low-cost transcritical R744 technology is challenging. This paper summarizes important R744-specific design issues and differences in comparison to conventional R134a bottle coolers. Due to cost reasons, it is highly desired to use relatively conventional components, including round-tube-plate-fin heat exchanger designs, fixed geometry capillary tubes (instead of variable geometry expansion valves), and single-speed compressors. While conventional round-tube-plate-fin evaporator designs deliver acceptable results, transcritical R744 systems require substantially different heat exchanger designs in order to deliver suitable performance when used as gas cooler. Internal heat exchange, which in conventional R134a systems is often achieved by wrapping the capillary tube around the compressor suction line, plays a much more important role in transcritical R744 systems, and shows large performance improvement potential. Experiments also show that proper capillary tube sizing and refrigerant charge optimization have much bigger impact on transcritical R744 systems in terms of cooling performance and energy efficiency in comparison to R134a systems. Presented is an example of a successful R744 bottle cooler design that is on par with a comparable R134a system in terms of performance and cost. This low-cost, high performance design has been implemented and experimentally validated. The improved transcritical R744 system achieved pull down times and energy efficiencies that were comparable with the R134a baseline system results: 97% and 103%, respectively.

1. INTRODUCTION

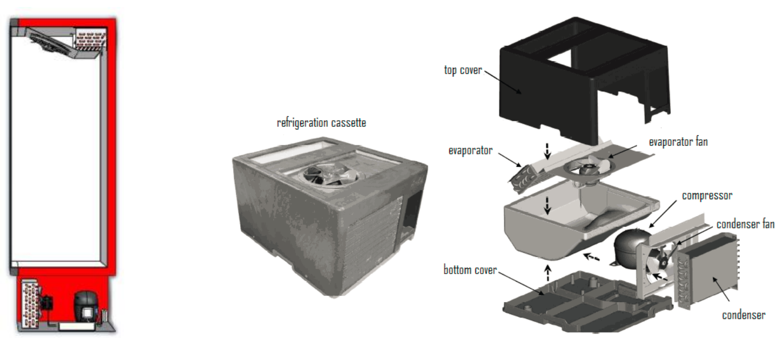
The aim to introduce environmentally friendly refrigerants is among the most dominant technology drivers in numerous sectors of the HVAC&R industry. The situation for light commercial refrigeration products can be seen as different from many other cooling applications that use vapor compression systems. 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 its 1 millionth HFC-free cooler using natural refrigerant in the marketplace, preventing the emission of 5.25 million metric tons of CO_2 over 10 years (TCCC, 2014). 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, 2004) 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. Mitigation of the flammability risk is possible by rigorous refrigerant charge reduction which can be achieved through the use of system components with very low internal volume and compressor lubricants with reduced hydrocarbon solubility characteristics. However, when using hydrocarbon refrigerants the flammability risk can only be mitigated, but it is not possible to completely eliminate it. Therefore, in cases where a non-flammable, non-toxic, natural refrigerant has to be used, R744 appears to be the only suitable option that 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 ambient 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 costs should not increase to ensure competiveness with currently used HFC systems.

This paper summarizes important R744-specific design issues and differences in comparison to conventional bottle coolers that typically use HFC R134a. In order to minimize additional costs when implementing transcritical R744 it is highly desirable to utilize relatively conventional components, such as round-tube-plate-fin heat exchanger designs, fixed geometry capillary tubes (instead of variable geometry expansion valves), and single-speed compressors. Presented is an example of a successful R744 bottle cooler design that is on par with a comparable R134a system in terms of performance and cost. This low-cost, high performance design has been implemented and experimentally validated.

2. DESIGN CHALLENGES, CONSTRAINTS, AND PERFORMANCE REQUIREMENTS

2.1 Design constraints

The global glass door merchandiser market comprises several million units per year. The coolers are available with 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. For transcritical R744, both cassette and split-type designs are possible.

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 ^{o}C, certain types of beer coolers need to pass tests at product temperatures below 0 ^{o}C. 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 aimed at simulating 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 cycling losses that otherwise occur every time the compressor cycles on or 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 Design challenges for transcritical R744

There are a number of design challenges that need to be addressed when designing transcritical R744 components for 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. However, achieving comparable energy efficiency 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 ambient 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 system 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. Cooling capacity tests are conducted in form of transient pull down tests as described above. However, because of the higher volumetric capacity of R744, meeting capacity targets is rarely the problem so that design efforts can be focused on improving energy efficiency.

Another major design challenge exists due to the limited availability of low-cost system components. A well-developed component supplier base with a large number of different manufacturers exists for R134a. The situation is currently very different for R744 components. While the situation will most likely improve over time when additional suppliers will enter the market, it is commonly believed that at least some of the R744 components could remain more expensive than their R134a counterparts due to the increased mechanical strength requirements. The increased requirements are a consequence of the much higher operating pressures of R744 with typical heat absorption pressures of 3 MPa to 3.5 MPa and heat rejection pressures of approximately 9 MPa to 12 MPa. Considering the current cost situation it is very difficult for manufacturers to justify additional components or complexity, such as electronic expansion valves, since low-cost light commercial systems designed for R134a almost always use capillary tubes. This adds additional challenge to the task of designing high performance, low-cost R744 systems, and the only options are to rigorously optimize gas coolers, refrigerant charge and capillary tube dimensions to achieve optimum heat rejection pressures. Another component that plays a more significant role for R744 than for R134a is the internal heat exchanger (IHX). The design specifics of these elements are presented next.

3. R744 COMPONENT DESIGN AND SELECTION

In principle, evaporators for transcritical R744 bottle coolers can be of the same design as for R134a. Most commonly used are round-tube-plate-fin designs with copper tubes and aluminum fins. All aluminum designs become more popular due to cost considerations, although corrosion can lead to problems. There are no major changes on the air side, where most of the heat transfer resistance is located. Typical fin pitches are 6 to 10 FPI, and the tube wall thickness is adjusted to withstand the higher R744 evaporation pressures. Typical outer tube diameters are between 6 and 8 mm. Much more critical is the design of the R744 gas cooler. Round-tube-plate-fin is the preferred design for many manufacturers, but lately, their interest has shifted towards low-maintenance designs. In many designs, low-maintenance directly translates into reduced air-side fin area (reduced fin pitch as shown in Figure 2a), since this is where fouling and corrosion problems occur. Other designs include wire-and-tube or jelly-roll (Figure 2b), but the aim to reduce air-side area is the same.

a) round-tube-plate-fin design with reduced fin pitch

b) jelly-roll design

c) severe fouling/corrosion problems

Figure 2: Low-maintenance heat rejection coils

From a low-maintenance standpoint it makes sense to reduce the air-side area to mitigate fouling and corrosion problems (Figure 2c). Nevertheless, transcritical R744 systems are much more sensitive to reduced heat rejection than R134a systems, because higher refrigerant outlet temperatures at the gas cooler increase the so-called approach temperature difference between the refrigerant outlet and the air inlet temperature. Slight increases in the approach temperature difference directly impact COP and capacity of a transcritical R744 system, because of the S-shaped isotherms in the region above the critical point where heat rejection takes place. Therefore, reducing air-side area of the gas cooler penalizes the performance of a transcritical R744 system much more than a comparable area reduction in an R134a condenser. Many low-maintenance designs therefore increase tube length to compensate for the reduced air-side fin area. However, tubes are more expensive than fins, and adding tube increases refrigerant charge and pressure drop in the heat exchanger. It should be noted that aluminum microchannel heat exchangers, which are a very attractive design option for condensers and gas coolers in many other applications have not been introduced yet to the light commercial sector, at least not on a large scale. Steel seems to be an attractive option for some of the gas cooler designs due to low cost and high mechanical strength.

Unlike for R134a, currently there are a very few compressor manufacturers that offer commercially available transcritical R744 compressors for bottle coolers. Several different displacements are available, but due to the limited number of different models and displacements some cooler designs result in oversized compressors, which further limit the achievable energy efficiencies due to increased cycling losses. Thermal storage technologies could be applied to mitigate the capacity-size mismatch for now; however, it is expected that the situation will improve as soon as additional compressor sizes or variable speed options become available.

It is clear that variable geometry expansion devices could improve energy efficiency, especially if the cooler has to operate in various ambient conditions. Nevertheless, due to the low-cost requirements and the fact that the vast majority of R134a glass door merchandisers use a capillary tube, it seems unlikely that the additional component cost is justifiable for transcritical R744. However, since energy consumption is typically only rated at a single ambient condition, it is possible to optimize the equipment with proper capillary tube and refrigerant charge selection. Typical capillary tube lengths vary between 1.5 and 5 m and diameters between 0. 5 and 1.5 mm are common. The higher vapor density of R744 in comparison to R134a drives the designs towards smaller diameters which can lead to increased clogging issues. It is therefore not uncommon to choose similar diameters as for R134a and increase the length of the R744 capillary tube. For a typical soda drink cooler that is designed to work at an ambient of approximately 30 ^{o}C, capillary tube and refrigerant charge are adjusted to achieve evaporation and gas cooling pressures of approximately 2.9 MPa and 8.8 MPa, respectively. A positive aspect of using a capillary tube is the fact that it can easily be brought in contact with the tube connecting the evaporator exit and the compressor suction (especially in split-type designs) to form an internal heat exchanger that is known to provide substantial performance gains in transcritical R744 systems.

4. LOW-COST IMPROVEMENT POTENTIALS FOR R744 SYSTEMS

Among the refrigeration system components identified above several offer promising improvement potentials that can be implemented at low-cost: gas cooler design with low heat conduction in the opposite direction of air flow, internal heat exchanger with improved effectiveness, simultaneous optimization of refrigerant charge and capillary tube dimensions. This is not to say that the available compressors do not offer any improvement potential; however, the items listed above can more easily be addressed and optimized by the designer of the refrigeration system. Also, it is worth mentioning that the evaporator bears the smallest improvement potential when trying to implement typical R134a designs into transcritical R744 systems.

4.1 Gas cooler

The design of a well performing gas cooler differs substantially from a heat exchanger that is designed for condensation. Aside from the desuperheating zone at the entrance of the condenser, the constant temperature condensation process reduces internal temperature differences in comparison to a supercritical heat rejection process in a gas cooler where the refrigerant experiences a continuous temperature glide of approximately 60 K to 80 K between the inlet and the outlet of the refrigerant. Consequently, the capacity distribution in a gas cooler is less uniform than in a condenser, as shown in the simulated refrigerant temperature profile in Figure 3. Therefore, R744 gas coolers should always be designed as counterflow. The largest temperature difference between the air and the refrigerant occurs at the inlet side of the refrigerant. As the refrigerant is cooled and travels towards the exit, the available temperature difference between the air and the refrigerant further diminishes. Two important design implications can be derived: 1) Adding heat exchanger depth does not help as much in increasing the overall rate of heat transfer as adding face area; low approach temperature differences that are required for high COP can be better realized with increased face area than with increased heat exchanger depth; 2) Depending on the refrigerant circuitry of the gas cooler, it is possible that the hot refrigerant tubes at the refrigerant inlet conduct heat through the fins against the direction of air flow and ultimately heat up the refrigerant at the exit of the gas cooler. This problem has been seen to be most severe if both inlet and outlet of the refrigerant are located either both at the bottom or both at the top, which generally is the case with an even number of passes in heat exchanger depth. A low-cost, yet very effective way of mitigating this unwanted heat conduction is by cutting the fins between the different tube rows as shown in the example in Figure 4. More details on this approach can be found in Yin *et al.*, 2001.

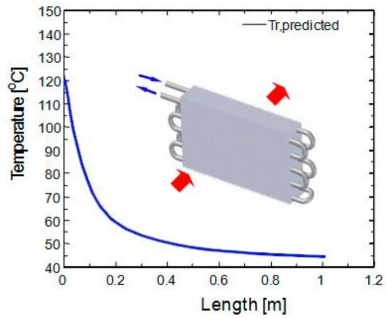


Figure 3: Uneven capacity distribution in a gas cooler due to large temperature gradients

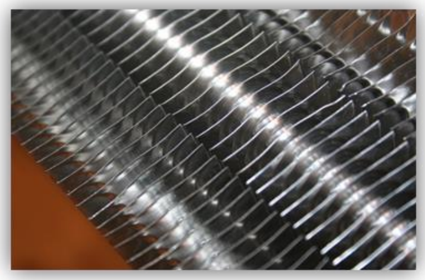


Figure 4: Cut fins as a way to mitigate detrimental heat conduction in the opposite direction of air flow

Another important aspect of gas cooler design is the required air flow rate due to its strong impact on approach temperature difference. A heat exchanger simulation tool has been developed to study the effect on COP. From Figure 5 it can be seen that system COP decreases by approximately 20% when the approach temperature difference increases from 0 K to 5 K. It was found that variable speed fans can be very beneficial in terms of system performance. The largest heat exchanger capacity is needed at the beginning of a pull down test, which is where the highest air flow rate is required. After all products have been cooled, reducing the fan speed is an opportunity to reduce the overall energy consumption of the cooler. This is possible, because heat exchanger capacities drastically decrease after the cooler has been pulled down and the products have reached their target temperature.

In Figure 6 it can be seen that increasing the fan speed and the resulting air flow rate does not return significantly higher gas cooler capacities beyond a certain point. In that case, it is better to stay to the left of the dashed line and reduce fan speed even further once the heat exchanger capacity decreases.

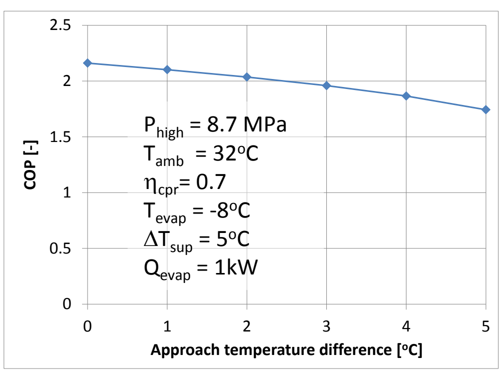


Figure 5: Effect of gas cooler approach temperature difference on COP

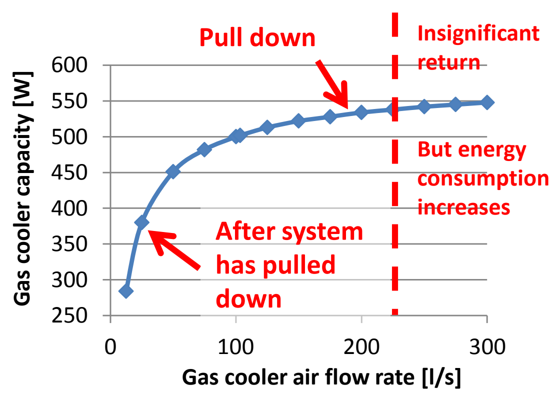


Figure 6: Effect of gas cooler air flow rate on gas cooler capacity

4.2 Internal heat exchanger

It has already been mentioned that internal heat exchange is much more important for R744 than for R134a. In conventional vapor compression systems, the use of an IHX increases the cooling capacity, provided the reduction of compressor mass flow rate caused by lower suction densities is less than the additional phase change enthalpy difference caused by lower inlet vapor qualities to the evaporator. Nevertheless, COP does not necessarily increase when an IHX is used, as this depends on the fluid properties of the refrigerant used. COP increases in case the additional compressor work caused by the larger superheat of the suction gas is less than the extra cooling capacity obtained in the evaporator. The governing equations are shown in Equations (1) - (3). Figure 7 illustrates the trade-off between additional compressor work and extra cooling capacity that determines whether COP increases or decreases with the use of an IHX. As in the case of R134a, but to an even larger extent, the use of an IHX increases both cooling capacity and COP of glass door merchandisers that use transcritical R744.

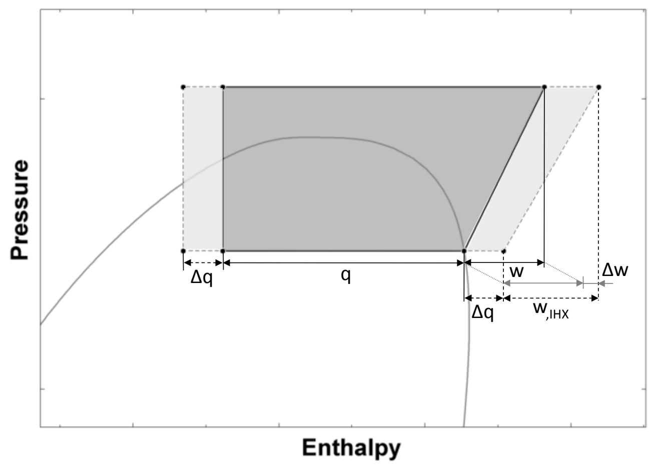
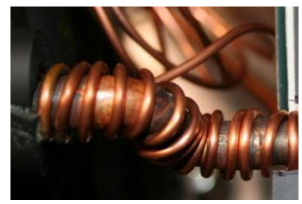


Figure 7: IHX trade-offs between cooling capacity and compressor work

a) capillary tube inside suction line



b) capillary tube externally attached

Figure 8: Internal heat exchanger designs

$$COP = \frac{q}{w} \tag{1}$$

$$COP_{IHX} = \frac{q + \Delta q}{w + \Delta w} \tag{2}$$

$$COP_{IHX} \approx COP \cdot \left(1 + \frac{\Delta q}{q} - \frac{\Delta w}{w}\right) \tag{3}$$

In systems that use a capillary tube as expansion device the simplest IHX design is implemented by bringing the capillary tube in contact with the suction line. However, this leaves the question which of the two designs shown in Figure 8 yields higher IHX effectiveness: capillary tube inside the suction line or capillary tube in external contact with the suction line (both are insulated to prevent external heat transfer from the environment). An analysis of the heat exchanger effectiveness shows that the case where the capillary tube is externally connected to the suction line results in increased heat transfer provided the contact resistance between the externally attached capillary tube and the suction line is sufficiently small. This result is somewhat counterintuitive, but in the case of the externally attached capillary tube, the larger diameter suction line acts as a fin, thereby increasing the available heat transfer area.

However, there are additional design considerations that need to be taken into account when designing an effective IHX. Big potential for improvement is seen in adding a piece of tube extending the larger diameter tube of the gas cooler exit before a transition is made to the small diameter capillary tube. This simple modification can substantially improve the overall capacity of the IHX and consequently the performance of the entire cycle. The reason is that the fluid, once it enters the capillary tube, immediately starts expanding from the high side pressure to the low side pressure. While the temperature change associated with the pressure drop is small for a sufficiently compressed liquid, the temperature starts dropping rapidly once the fluid gets closer to and finally enters the two-phase region. As a consequence, the available temperature difference in an IHX that only has contact between the suction line and the capillary tube is much smaller than in an improved IHX where the contact between the suction line and the high pressure side already starts at the larger diameter tube extension following the gas cooler, which is illustrated by the lengths L1 and L2+L3 in Figure 9.

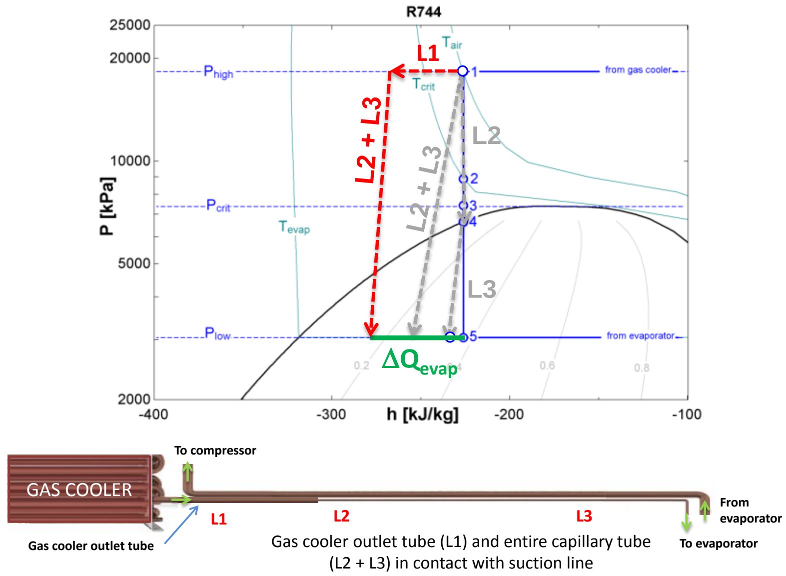


Figure 9: Increase of available temperature difference for heat transfer in the IHX by adding large diameter tube extension at gas cooler exit prior to entering capillary tube

4.3 Refrigerant charge and capillary tube sizing

The effect of changes in refrigerant charge and capillary tube dimensions have been found to have much greater effect on the performance of transcritical R744 than on R134a systems, because both parameters affect the heat rejection performance in the gas cooler. In addition, changes in high side pressure immediately affect the system performance, because for each ambient temperature a performance maximizing high side pressure exists. Figure 10 shows the effect on evaporator performance.

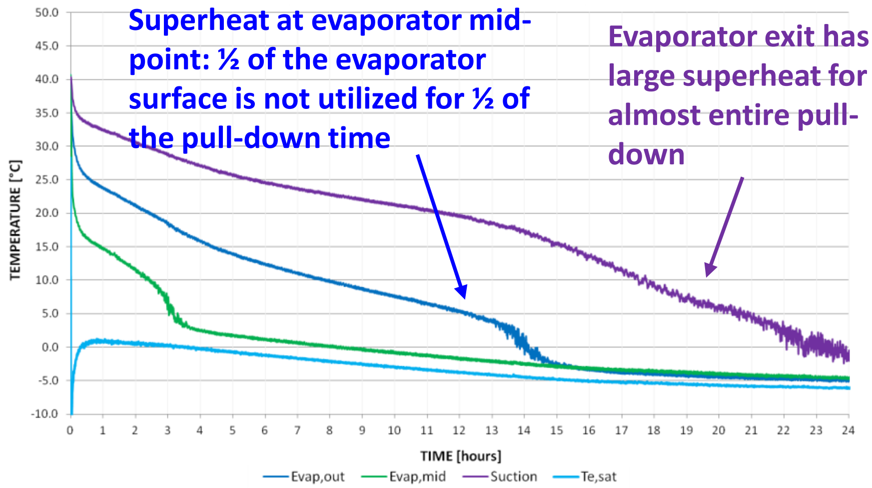


Figure 10: Effect of refrigerant charge and capillary tube sizing on evaporator performance

It can be seen that for during the entire pull down, more than 50% of the evaporator surface are superheated, resulting in suboptimal evaporator performance. Additional refrigerant charge or capillary tube adjustments could certainly improve the utilization of the heat exchanger, but the resulting shift in high side pressure may been seen to have a more severe impact on system performance (energy efficiency) than the loss of evaporator area due to inefficient surface utilization. Moreover, increasing refrigerant charge or reducing the capillary tube restriction will affect the refrigerant condition at the evaporator exit, which means that the system has to be designed so that at least a small amount of superheat is available at the compressor inlet after the products have been cooled, because this is when the lowest heat load is removed by the evaporator. It is speculated that a small refrigerant accumulator at the exit of the evaporator could mitigate the problem and would yield further performance enhancements. This modification is subject to further experimentation.

5. EXPERIMENTAL EVALUATION

After successful implementation of the improvements discussed in the foregoing sections, the performance of an improved transcritical R744 glass door merchandiser with an internal cabinet volume of approximately 500 l and a capacity of 550 soft drink cans, each having a volume of 355 ml was tested at an ambient temperature of 32 ^{o}C and a relative humidity of 65%. The performance results in terms of pull down time and energy consumption, both shown as relative values in comparison to the R134a baseline cooler (optimized production system), are shown in Figure 11.

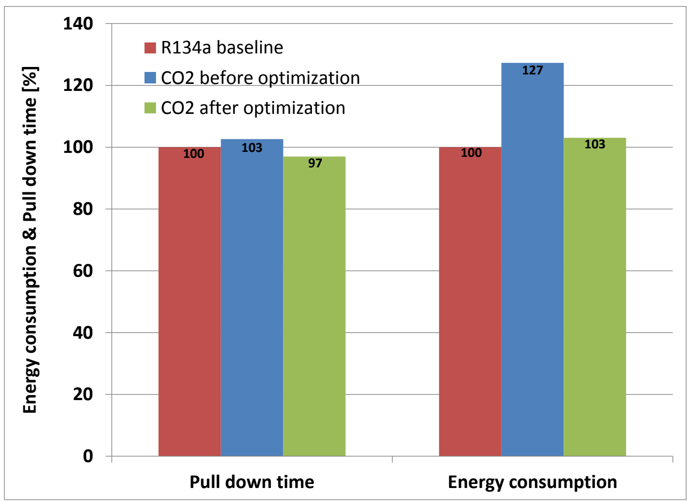


Figure 11: Comparison of pull down times (cooling capacity) and energy consumption (COP) of baseline R134a and transcritical R744 glass door merchandiser systems

For the transcritical R744 system, the performance of the original R744 system is also shown as a reference. It can be seen that the improved R744 system outperformed the baseline R134a system in pull-down time by a small margin. More important is the comparison of energy consumption: while the original R744 system showed 127% 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%). 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.

6. CONCLUSIONS

This study demonstrates that it is possible to design high-performance transcritical R744 glass door merchandiser systems that deliver cooling capacities and energy efficiencies that are comparable to results obtained with an R134a baseline system. In the present case, pull down time of the improved R744 was slightly lower (97%), while energy efficiency was almost equal (103%) for tests conducted at an elevated ambient temperature of 32 ^{o}C and 65% relative humidity. The most important observation is that these promising system results were obtained with low-cost technology, which was very comparable to what was used in the R134a system: round-tube-plate-fin gas cooler and evaporator, capillary tube, and a fixed speed compressor. The improvements were realized by optimization of gas cooler circuiting, an improved IHX, and a rigorous optimization of refrigerant charge and capillary tube geometry.

NOMENCLATURE

COP	coefficient of performance	(-)
GWP	global warming potential	
HFC	hydrofluorocarbon	
IHX	internal heat exchanger	
P	pressure	(MPa)
q, Q	heat transfer rate	(kW)
T	temperature	(^{o}C)
w	work rate	(kW)
Δ	difference	
η	efficiency	(-)

Subscript

amb	ambient
cpr	compressor
evap	evaporator
high	high pressure side
IHX	internal heat exchanger
sup	superheat

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