

Purdue University Purdue e-Pubs

International Refrigeration and Air Conditioning
Conference

School of Mechanical Engineering

2016

Conversion of cold beverage dispenser's refrigeration system to R744 refrigerant

Matej Visek

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

Stefan Elbel

stefan.elbel@creativethermalsolutions.com

Pega Hrnjak

pega.hrnjak@creativethermalsolutions.com

Follow this and additional works at: <http://docs.lib.purdue.edu/iracc>

Visek, Matej; Elbel, Stefan; and Hrnjak, Pega, "Conversion of cold beverage dispenser's refrigeration system to R744 refrigerant" (2016). *International Refrigeration and Air Conditioning Conference*. Paper 1794.
<http://docs.lib.purdue.edu/iracc/1794>

This document has been made available through Purdue e-Pubs, a service of the Purdue University Libraries. Please contact epubs@purdue.edu for additional information.

Complete proceedings may be acquired in print and on CD-ROM directly from the Ray W. Herrick Laboratories at <https://engineering.purdue.edu/Herrick/Events/orderlit.html>

Conversion of cold beverage dispenser's refrigeration system to R744 refrigerant

Matej VISEK^(1,*), Stefan ELBEL^(1,2), Pega HRNJAK^(1,2)

¹Creative Thermal Solutions, Inc.
2209 North Willow Road, Urbana, IL 61802, USA

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

* Corresponding Author Email: matej.visek@creativethermalsolutions.com

ABSTRACT

Cold beverage dispensers belong to the group of light commercial refrigeration machines that includes beverage coolers, ice machines and storage units for food items. Beverage dispensers use a vapor compression system to charge a thermal storage, usually an ice bank, which is used to indirectly cool tap water and beverage concentrate. For these machines, R290 (propane) and R744 (carbon dioxide) are among the favorite candidates to completely replace high GWP HFC refrigerants in the future. Because of flammability and charge restrictions for R290, carbon dioxide is in some cases preferred even if some of its thermodynamic and transport properties are less favorable than those of R290, especially at high ambient temperature.

This study evaluated the performance of a baseline R134a beverage dispenser machine by experimentally measuring initial pull down time and energy consumption during compressor cycling operation at 24°C and 38°C ambient temperatures, respectively. Another important performance criteria for dispensers is the burst capacity, which is the number of 350 ml drinks dispensed at the rate of 3 drinks per minute while maintaining a beverage temperature below 5°C. This test simulates the typical load profile of the machine during peak serving time in hotels and motels. Burst capacity is closely related to thermal storage capacity and therefore the ice amount was also closely monitored. The machine's refrigeration system was afterwards converted to R744 refrigerant, using a transcritical cycle. Special attention was given to optimization of capillary tube size, refrigerant charge amount and gas cooler design where the strongest impact on the performance at different ambient temperatures was expected. Appropriate instrumentation and theoretical analysis helped to identify optimum selection of capillary tube and R744 refrigerant charge. The optimized R744 system used 385g of charge and pulled down in approximately half the time and dispensed at least 25% more drinks while keeping the energy consumption comparable with its baseline R134a system. Slight increase in gas cooler cost was well overcome by the benefits of larger burst capacity and significantly shorter pull down time allowing: the redesigned beverage dispenser can be kept off longer and needs to be turned on shortly before the serving time.

1. INTRODUCTION

Cold beverage dispensers belong to the group of light commercial refrigeration machines that includes beverage coolers, ice machines and storage units for food items. The light commercial refrigeration is gradually transforming to natural refrigerants all over the world. There are two typical groups of natural refrigerants which can replace currently used HFCs in light commercial refrigeration units. The first group consists of flammable refrigerants such as propane and isobutane and second group includes carbon dioxide (R744). Flammable refrigerant have disadvantage of flammability risks which can be mitigated but not completely removed. The HC refrigerants are also limited in amount of charge to 150g by government standards. Certainly there is a limit to the refrigeration capacity reachable with 150g of charge. As it was shown by Fuentes et al. (2014) on the two door commercial beverage cooler, it is possible to push the cooling capacity limit very high by careful design of system components for low charge.

On the other side R744 is non-flammable and non-toxic with no limit of charge. Though its working pressures are considerably higher at usual operating ambient conditions of light commercial unit. The system components have to be redesigned from mechanical point of view to withstand higher burst pressure. Carbon dioxide has low critical

temperature and transcritical operation of the system is necessary at most operating conditions. CO₂ fluid efficiency degrades rapidly above the critical point. Many researches such as DeAngelis and Hrnjak (2005), Kim et al. (2004), Boccardi et al. (2013) and Fuentes et al. (2015) have shown that it's possible to build energy efficient commercial equipment operating with R744 at comparable cost with equivalent R134a unit. One of the largest beverage company in US committed to the use of R744 in its beverage coolers and already successfully introduce thousands of functional and efficient coolers to the market (TCCC, 2014).

This study demonstrates that in addition to regular beverage coolers also beverage juice dispensers can be successfully converted to R744 refrigerant with relatively low cost changes of the refrigeration system and reach on par energy efficiency at usual operating ambient temperatures.

2. PRINCIPLE OF OPERATION AND PERFORMANCE CRITERIA

2.1 Beverage dispenser principle of operation

Beverage dispenser is designed to provide cold drinks by mixing cooled tap water with juice concentrate which is stored in the machine's refrigerated cabinet. The tap water is flowing through separate heat exchanger immersed in the center of the thermal storage tank. In addition the tank includes also evaporator coil and water circulation pump as shown in the Figure 1. Water is used as a thermal storage medium in the tank. When refrigeration system operates it's freezing the storage water on the evaporator coil creating approximately 3cm thick wall of ice on the perimeter of the tank visible in the Figure 1. The thickness of the ice layer is controlled with immersion temperature probe keeping frost free the central part of the tank with tap water coil and circulation pump. It's important to keep the tap water heat exchanger coil frost free. Freezing of the tap water in the coil could block the dispensing flow and cause failure of the machine. The evaporator is bare copper tube with several passes bended to fit the tank. The circulation pump is mixing the storage bath the keep its temperature as uniform as possible and enhances heat transfer between cooled tap water and cold storage water. In addition the circulation pump flow is also diverted to a small fin and tube heat exchanger located in the concentrate storage cabinet. Small DC voltage fan circulates cabinet air across this water cooled heat exchanger and keeps the temperature of the concentrate constant. The storage cabinet cooling system is not designed to pull down temperature of the concentrate. Concentrate reservoirs are moved to the dispensing machine from separate refrigerator already at the desired temperature around 5°C. The condenser and compressor are located at the back of the machine. Round tube and fin condenser is air cooled with fan. The air flow from the condenser is cooling also the compressor shell. Capillary tube serves as simple expansion device. The suction line accumulator is positioned before the compressor to protect it from potential liquid refrigerant coming from the evaporator. The baseline system JDF-4D was charged with 283g of R134a refrigerant.

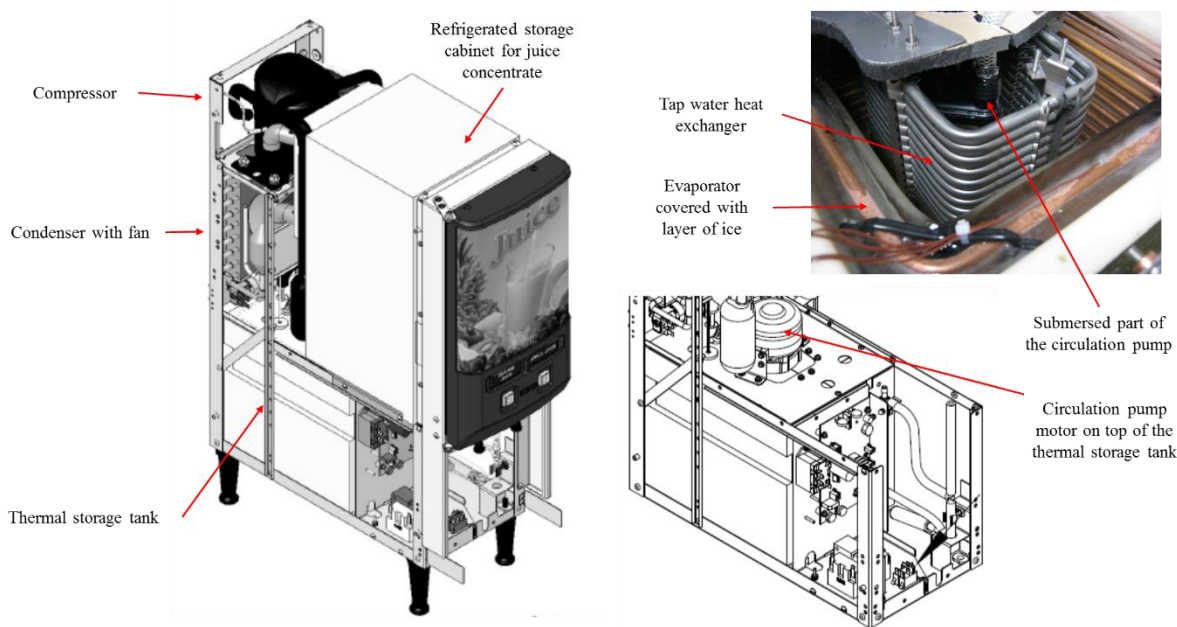


Figure 1: Description of beverage dispenser machine

2.2 Installation, performance requirements and challenges of conversion to transcritical R744

The beverage dispenser must be connected to a cold water system with operating pressure between 138kPa and 690kPa. This water source must be capable of producing a minimum flow rate of 89ml per second. Inlet water is required to have temperature less than 24°C. The dispenser requires a minimum of 100mm of air clearance at the rear and 200mm above the dispenser for proper air flow across the condenser. The ambient temperature should not exceed 38°C. Before initial use, the thermal storage tank needs to be filled up according to the procedure described in the installation manual (BUNN-O-MATIC Corp. 2012). The dispensing flow rate and mixing ratio with concentrate have to be tuned according to the Table 1. These requirements are essential in order to achieve correct functionality and sufficient capacity of the dispenser during testing.

Table 1: Dispensing flow rates and mixing ratios depending on the product (BUNN-O-MATIC Corp. 2012).

* Maximum flow rate may be less depending on the water pressure supply at each location.

Mix Ratio (water + concentrate)	Adjust water flow rate to:	Product	Brix %
2+1	*3.0 fl. oz (89 ml) Per 3-second test	Prune Juice	16.0%
		Other	*
4+1	*3.0 fl. oz (89 ml) Per 3-second test	Orange Juice	11.8%
		Pineapple Juice	12.8%
		Cranberry Fruit Juice	14.0%
5+1	*3.0 fl. oz (89 ml) Per 3-second test	Lemonade	
		Apple Juice	12.0%
		Fruit Punch	
		Grape Juice	13.0%
3+1 through 7+1		Other	*
Above 7+1	*4.0 fl. oz (118 ml) Per 3-second test	Other	*
High Viscosity Juice	*3.0 fl. oz (89 ml) Per 3-second test	As Required	-

The usual drinks dispensing rate is 3 drinks per minute with volume 355ml each. The beverage juice dispenser capacity (also called burst capacity) is then determined by number of drinks that can be dispensed with temperature less than 5°C with inlet tap water temperature 24°C. The baseline R134a system had specified burst capacity of 145 drinks at 24°C ambient temperature. Obviously at higher ambient temperature the burst capacity is expected to be lower.

Other performance criteria are pull down time and energy consumption which have to be watched closely during conversion of dispensing machine to R744. The pull down time is directly linked with the refrigeration system capacity and energy consumption reflects the energy efficiency of the cooling cycle (compressor ON and OFF). Achieving same or higher cooling capacity with carbon dioxide system is not expected to be a problem as the volumetric capacity of R744 is known to be approximately 8 time higher compared to R134a refrigerant. However, achieving comparable efficiency especially at high ambient temperature is much more demanding. While R744 can be comparably efficient refrigerant at low temperatures the COP degrades quickly at higher ambient temperatures due to transcritical heat rejection in the gas cooler. In contrast to R134a cycle where the discharge pressure is directly linked with the condensing temperature, the discharge pressure in transcritical R744 system is not determined by ambient temperature and can be almost freely adjusted by expansion device. For each discharge pressure there is gas cooler outlet temperature which maximizes the cycle COP. Both the gas cooler outlet temperature and discharge pressure have significant effect on the performance of the CO₂ cycle as it is shown in the Figure 2. In transcritical operation optimum discharge pressure dramatically changes with the gas cooler outlet temperature thus the ambient temperature. Therefore some tradeoff will be necessary when designing and optimizing CO₂ system with single capillary tube for wide ambient temperature range. Gas cooler, capillary tube and charge are of utmost importance during redesign. Gas cooler must provide as low outlet temperature as possible to achieve comparable performance of CO₂ system with R134a system. The refrigerant charge together with capillary tube have to be adjusted to maximize COP at given gas cooler outlet temperature.

The R744 system operates at much higher pressures compared to R134a system, thus all the components and refrigeration tubing need to withstand significantly higher burst pressures. Usual approach taken in redesigning the

components is to reduce internal tube diameter and increase wall thickness. High volumetric capacity of CO₂ reduces required refrigerant volume flow and thus also velocity in the pipes. Because of higher absolute pressures the potentially higher pressure drop in the small diameter tubes impact CO₂ system performance less than equivalent pressure drop in R134a system.

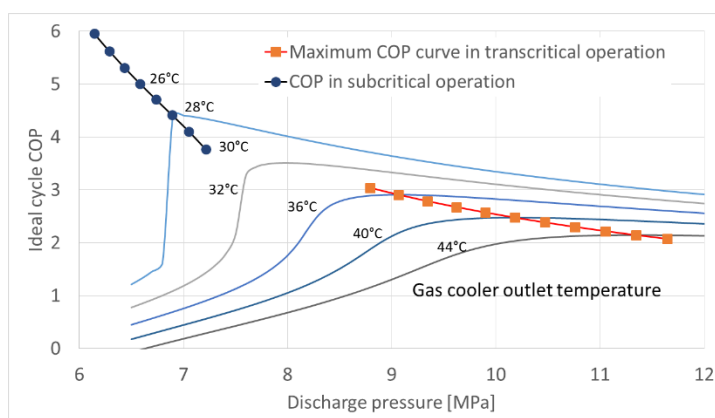


Figure 2: COP of ideal R744 cycle as a function of gas cooler outlet temperature close to critical point

3. INSTRUMENTATION, TEST PROCEDURES AND BASELINE PERFORMANCE

Baseline dispenser was instrumented with surface mounted type T thermocouples at the locations indicated in the Figure 3. Sensotec pressure transducers were connected to the compressor suction and discharge to have more accurate information about operating conditions. The Ohio Semitronics power transducer measured power consumption of the system. A set of immersion thermocouples as depicted in Figure 4 was used to detect thickness of the ice layer in the thermal storage tank. Thermocouples soldered to the surface of the evaporator provided information about amount of liquid refrigerant within the evaporator during compressor ON while charging the thermal storage.

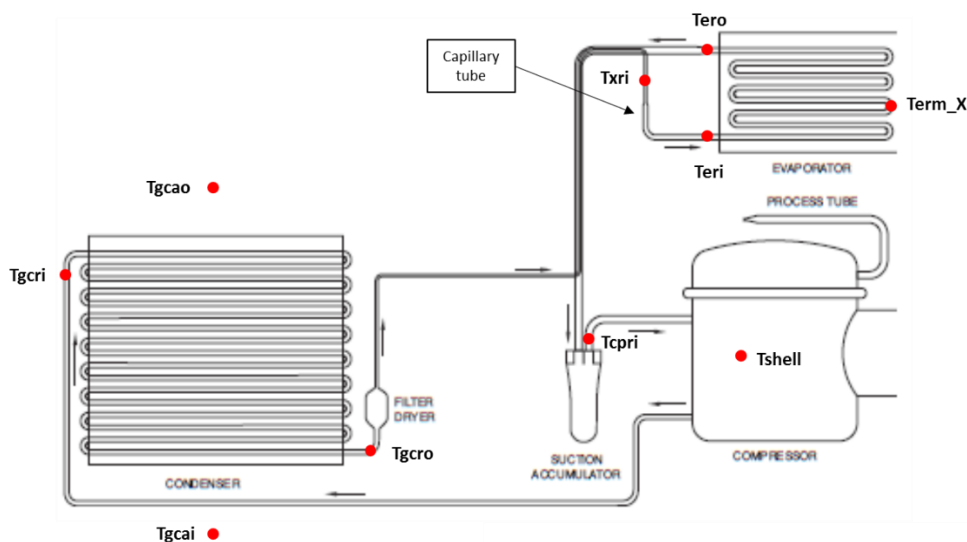


Figure 3: Baseline R134a dispenser refrigeration system instrumentation

Such instrumented dispenser was placed in the environmental chamber equipped with tap water supply and drain. The tap water was preconditioned by the heater showed in the Figure 5 to keep constant 24°C during dispensing test. The dispensing of drinks was fully automated with control panel located outside of the chamber for testing convenience and chamber temperature stability.

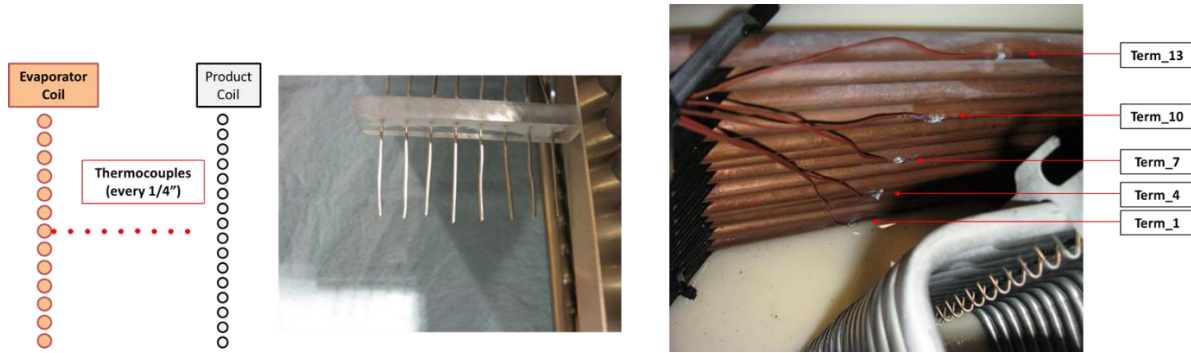


Figure 4: Measurement instrumentation for ice thickness and evaporator flooding

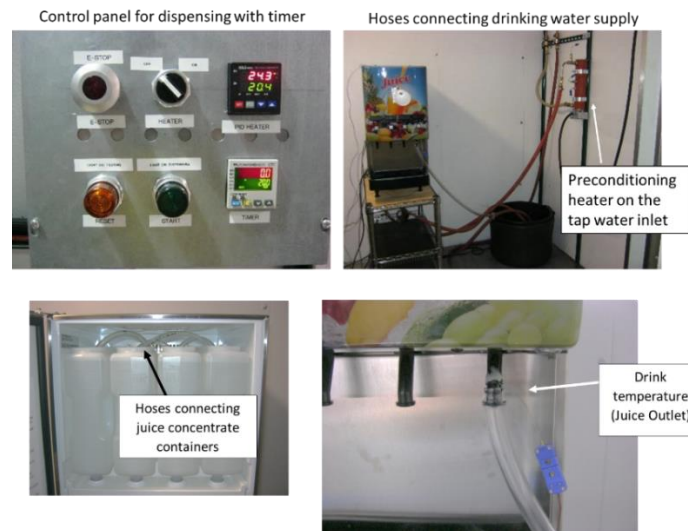


Figure 5: Chamber equipment and dispenser instrumentation for fully automated dispensing (burst capacity) test

Three types of tests were performed with the baseline dispenser:

1. Pull down
2. Energy consumption
3. Dispensing at ambient temperatures 24°C, 32°C and 38°C.

Before the pull down test the chamber and machine was stabilized at the desired ambient temperature. The machine's thermal storage bath was filled with water at chamber ambient temperature. The refrigerated cabinet was empty without concentrate containers. The machine was turned ON. The time until the compressor OFF was measured and is reported as a pull down time. Typical pull down chart is shown in the Figure 6.

During the energy consumption test the system was left to cycle without dispensing the drinks and with the refrigerated cabinet empty. Once stable cycling was reached the energy consumption was calculated from 2 full compressor ON and OFF cycles. Typical compressor ON and OFF cycle of the baseline dispenser took approximately 10 hours with compressor run time of around 20% at 24°C ambient temperature as shown in the Figure 6. 2 full ON and OFF cycles are covering almost full day (24 hours) of testing. The power was recorder every 5 seconds and energy consumption was calculated from the power by integrating it over the time.

For dispensing test the refrigerated cabinet was loaded with 4 concentrate containers filled with 5°C water and interconnected with hoses in the way that all the containers supply one dispensing nozzle. The nozzle was equipped with drink measuring thermocouple, for details see Figure 5. The machine was hooked up to the water supply with controlled temperature at approximately 24°C for all ambient conditions. After the tanks were inserted, the machine was left to cycle at least one complete compressor ON and OFF cycle to stabilize water temperature in the concentrate tanks. The dispensing test always started 5 minutes after compressor turned OFF. The timer was set based on the water volume flow in the way to dispense 355ml every 20s. Data acquisition speed was 2Hz and the drink temperature was determined as an average of juice outlet temperature measurements during actual dispensing time. Burst capacity was assessed as maximum number of drinks dispensed with drink temperature below 5°C.

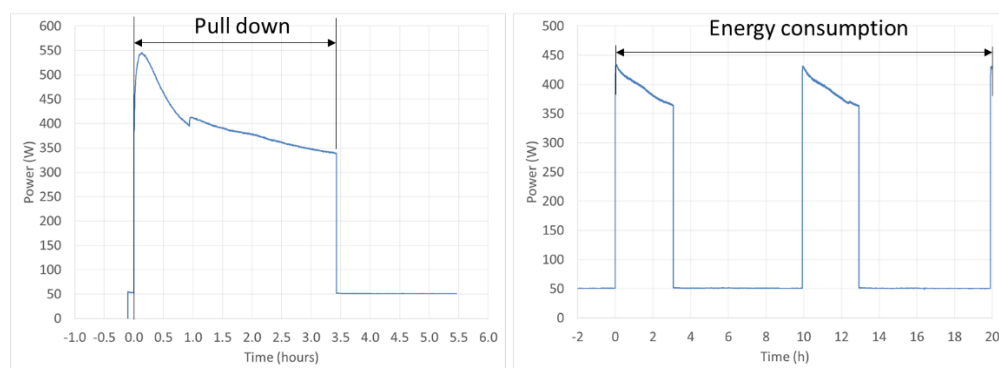


Figure 6: Typical results for pull down and energy consumption of baseline at 24°C ambient temperature

The baseline power measurement for pull down and energy consumption at 24°C ambient is shown in the Figure 6 and performance of baseline is summarized in the **Error! Reference source not found.** (page 8) together with the results of modified R744 system. The pull down of baseline took 3 hours 25 minutes. Two full compressor ON and OFF cycles taken for energy consumption calculation lasted almost 20 hours. An average drink temperature during baseline burst capacity testing is reported in the Figure 7. The burst capacity met the machine's specification of 145 drinks at 24°C ambient and it is gradually decreasing at higher ambient temperatures as it was expected.

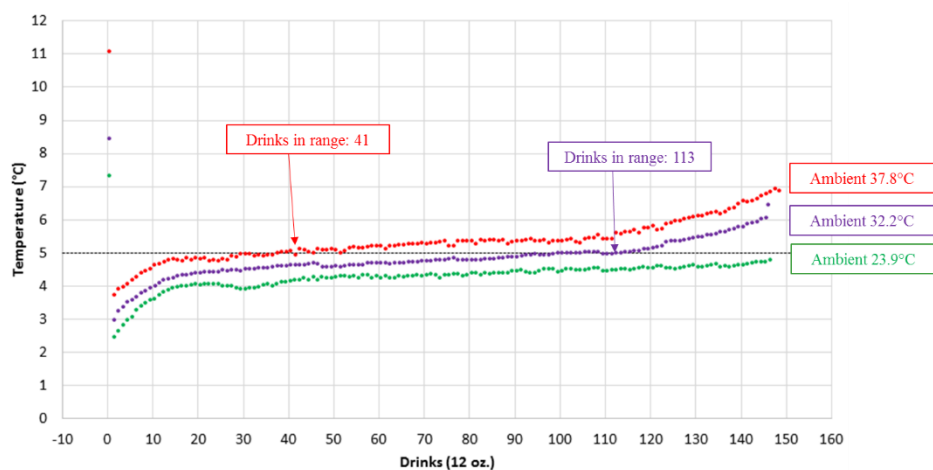


Figure 7: Burst capacity results of baseline R134 dispenser at 24°C, 32°C and 38°C ambient temperatures

4. BEVERAGE DISPENSER CONVERSION TO R744

The baseline dispenser's R134a compressor GL90PE had cooling capacity of 573W at -10°C evaporating, 40°C condensing, 8K sub-cooling and 32°C suction temperature. There are only limited number of transcritical CO₂ compressors available on the market in this capacity range. The cooling capacity steps between the CO₂ models are also significantly larger compared to R134a compressors. The Table 2 compares the baseline R134a compressor with two carbon dioxide compressors with closest higher and lower refrigeration capacity. The CO₂ compressor performance is evaluated at the -10°C evaporating and 32°C suction temperature. The gas cooler outlet temperature was chosen 32°C to correspond with liquid temperature at the outlet of the R134a condenser. The discharge pressure (around 8MPa) maximizes the COP at the given gas cooler outlet temperature. The larger SRcADB CO₂ compressor model was used further in the conversion process of the dispenser. It has 30% higher cooling capacity compared to baseline R134a model and was expected to shorten the pull down time by 20%. Slightly improved burst capacity of the system was also anticipated because of cooling capacity increase during the dispensing test when compressor is running. The COP of SRcADB compressor is 20% lower compared to its R134a counterpart and this can have negative effect on the system energy consumption.

Table 2: Performance comparison of baseline R134a compressor with two transcritical CO₂ compressors at corresponding operating condition (R134a: -10°C evaporating, 40°C condensing, 32°C suction and 8K sub-cooling, R744: -10°C evaporating, 32°C suction, 32°C gas cooler outlet and 8MPa discharge pressure)

Compressor model	GL90PE	SRcADB	SRFDB
Refrigerant	R134a	R744	R744
Displacement (ccm)	8.85	1.75	1.1
Cooling capacity (W)	573	730	462
Power (W)	295	483	320
Compressor COP	1.94	1.51	1.44

The gas cooler was redesigned in order to account for higher pressure of CO₂ refrigerant and larger capacity of the replacement compressor. Same technology of aluminum fins and copper tube heat exchanger was utilized for the R744 system as well. It should be noted that size of the gas cooler in the optimized R744 system slightly increased compared to R134a system. In the course of the supercritical heat rejection process in the gas cooler refrigerant experiences large temperature glide. Therefore, transcritical carbon dioxide gas cooler should always be designed in counter flow arrangement. Depending on the circuitry of the gas cooler hot inlet refrigerant might conduct heat through the fins towards the cold outlet. Very effective way of avoiding this unwanted conduction in the fins is cutting the fins between different tube rows as showed in the Figure 8. Alternative attractive technology for gas cooler to keep it small is aluminum microchannel heat exchanger which is found in many other applications. Steel is also a competitive material in design of CO₂ gas coolers because of its higher mechanical strength and low cost.



Figure 8: Cut fins to reduce heat conduction from warm to cold tube rows



Figure 9: Needle valve to speed up optimization of capillary tube and charge

It is clear that variable expansion valve could improve energy efficiency of the dispenser across wide range of ambient conditions. However because of low-cost requirements, reliability and robustness of the capillary tube was a preferred solution even though some energy efficiency had to be sacrificed. Typical capillary tube vary between 1.5 to 5m in length and its internal diameter is usually in the range between 0.5 to 1.5mm. Higher vapor density of R744 drives the selection to smaller diameters or longer tubes. Smaller diameters can lead to clogging issues thus it is very common to have much longer capillary tubes in the CO₂ systems. The capillary tube can easily form internal or external heat exchanger with suction tube and improve the efficiency of the CO₂ system. During capillary tube optimization in R744 system it is important to closely observe discharge pressure, flooding of the evaporator and gas cooler outlet temperature. The capillary tube and refrigerant charge have to be optimized with any component change in the system. Therefore during energy consumption optimization process manual needle valve as seen in Figure 9 was used to rapidly adjust discharge pressure maximizing the COP at corresponding gas cooler outlet temperature without needing to replace the capillary tube and system recharging. The Figure 10 shows experimental measurement of discharge pressure and gas cooler temperatures with calculated optimum discharge pressure to maximize the COP at measured gas cooler outlet temperature. The calculated optimum discharge pressure is slightly lower compared to actual measured pressure especially during supercritical operation at the beginning of the pull down. Adjustable electronic expansion device could minimize the difference and provide even higher efficiency of the system. The refrigerant charge level was adjusted in the way to flood the evaporator during the pull down but still keep some degree of superheat at the compressor inlet at the end of the pull down. It can be seen in the Figure 11 that during the pull down the evaporator is most of the time superheated resulting in non-optimal use of the heat exchanger even though the system energy consumption was optimized. This is a typical behavior of capillary tube system caused by varying load of evaporator during the pull down, starting with high load when the coil is frost free. As the layer of ice builds up the load decreases leading to flooding of the evaporator. Thermostatic or electronic expansion valve would

be able to optimize evaporator surface usage even during varying evaporator load and further improve energy consumption.

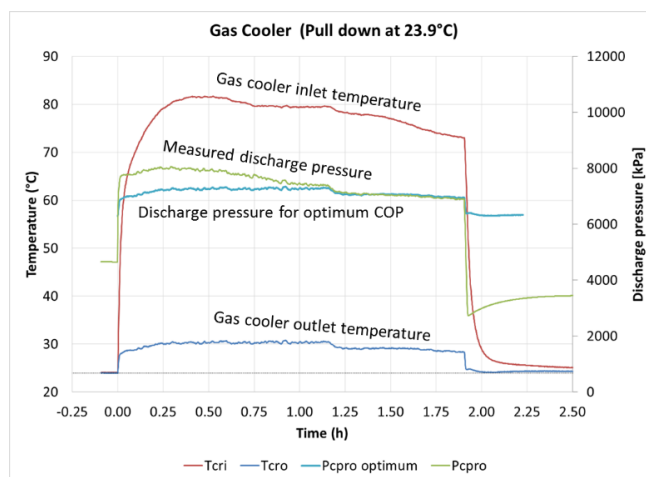


Figure 10: Gas cooler temperatures and discharge pressure for optimized CO₂ system

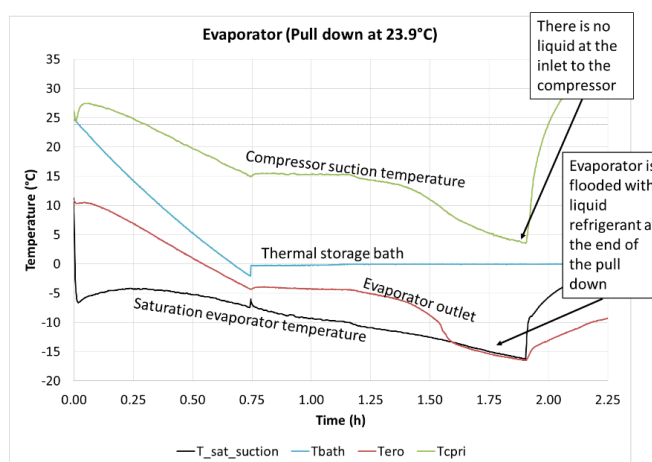


Figure 11: Evaporator temperatures during pull down of optimized CO₂ system

5. EXPERIMENTAL PERFORMANCE RESULTS OF R744 JUICE DISPENSER

The baseline R134a juice dispenser was converted to R744 refrigerant by implementation of the system modifications outlined in the paragraph 4. The mechanical system conversion was followed by rigorous refrigerant charge and capillary tube optimization to tune the performance of the modified R744 unit across range of ambient temperatures. The results of pull down, energy consumption and burst capacity are summarized in the Table 3 next to the baseline R134a system.

Table 3: Comparison of experimental performance results of baseline R134a and modified R744 beverage dispenser

Ambient	Refrigerant	R134a	R744
	Refrigerant amount	283g	385g
24°C	Pull down time (hr:mm)	3:25	1:54 (-45%)
	Energy consumption (kWh/day)	2.66	2.60 (-2%)
	Burst capacity (drinks)	>150	>>200
32°C	Energy consumption (kWh/day)	3.75	3.99
	Burst capacity (drinks)	113	-
38°C	Energy consumption (kWh/day)	4.33	5.86
	Burst capacity (drinks)	41	142

As a result of larger compressor cooling capacity the modified R744 beverage dispenser has significantly shorter pull down time only 1 hour 54 minutes compared to baseline system with 3 hours 25 minutes at 24°C ambient temperature. The energy consumption of the system was again calculated from two full compressor ON and OFF cycles. Power measurement of the modified system during pull down and energy consumption tests is shown in the Figure 12. Absolute value of the power at the pull down peak is approximately 100W higher for the modified R744 system but the compressor run time is shorter and thus the energy consumption can still be comparable to the R134a baseline system at 24°C and 32°C ambient conditions. The energy efficiency degrades rapidly at 38°C as seen in the Table 3. In addition to lower ideal COP of the carbon dioxide the capillary tube was tuned to perform better at low ambient temperatures and it's not enough restrictive at high ambient temperature. The preference of low ambient conditions is reasonable because most of the juice dispensing machines operate in the air-conditioned spaces.

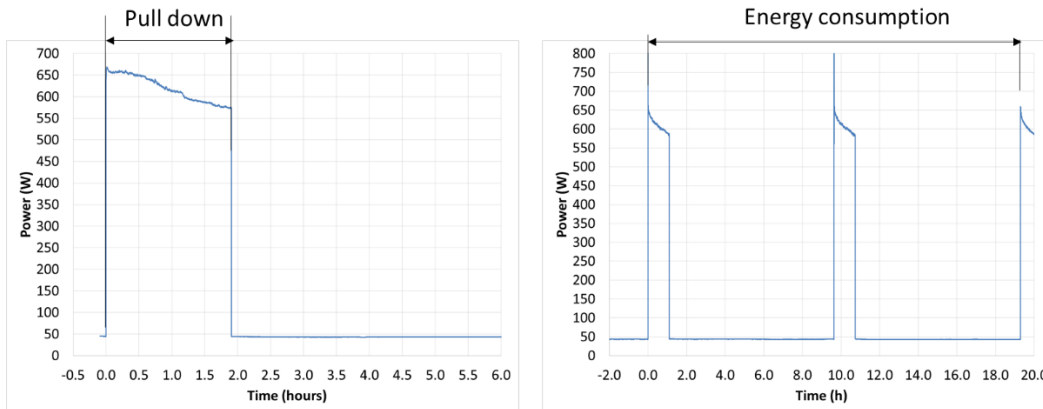


Figure 12: Power measurement during pull down and energy consumption tests of R744 juice dispenser

For modified R744 system the burst capacity results in the Figure 13 shows increase at both lowest and highest ambient conditions. This a result of larger capacity CO₂ compressor. At 24°C ambient almost unlimited dispensing is possible with CO₂ compressor running. At high ambient condition 38°C the CO₂ system is able to dispense 141 drinks which is very close to the baseline specification at 24°C condition.

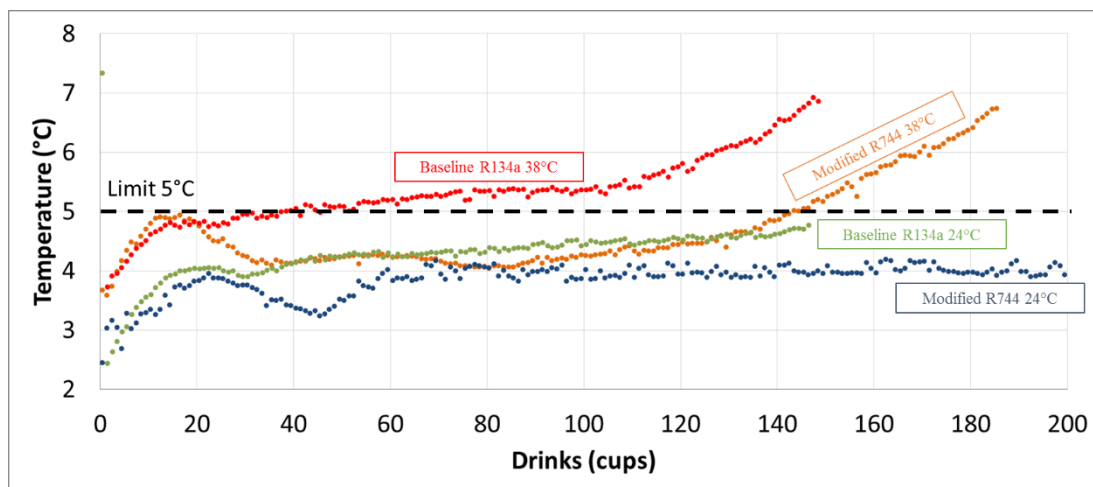


Figure 13: Burst capacity of optimized CO₂ system versus baseline R134a

5. CONCLUSIONS

During the conversion of juice dispenser's refrigeration system from HFC to transcritical R744 refrigerant it was necessary to modify following components of the system:

1. Compressor
2. Condenser – gas cooler including shroud and fan
3. Capillary tube
4. Refrigerant charge amount
5. Suction line accumulator

Because of high volumetric capacity of carbon dioxide, the compressor displacement needs to be small to provide same refrigeration capacity. Design of refrigeration lines and heat exchangers have to be adjusted to withstand considerably higher pressures. Capillary tube together with refrigerant charge amount have to be very well tuned to the required ambient conditions in order to reach comparable performance results with R134a. If the modifications are done thoughtfully it is possible to convert the beverage juice dispenser successfully to R744 with shorter pull down time, higher burst capacity and comparable energy efficiency with R134a system at lower ambient conditions (24°C and 32°C) with only a small increase in the system cost. At higher ambient temperature (38°C) the energy consumption quickly degrades for the CO₂ system but the burst capacity is still higher compared to baseline R134a dispenser.

NOMENCLATURE

CO ₂	carbon dioxide
COP	coefficient of performance (-)
DC	direct current
GWP	global warming potential
HC	hydrocarbon
HFC	hydrofluorocarbon
HC	hydrocarbon

REFERENCES

- Boccardi G., Calabrese N., Celata G.P., Masturullo R., Mauro A.W, Perrone A., Trichieri R., (2013). Experimental evaluation for a carbon dioxide light commercial cooling application under transcritical and subcritical conditions. *Applied Thermal Engineering*, 528-535.
- BUNN-O-MATIC Corporation (2012). Installation and operating guide (44744.0002). Retrieved from <http://www.bunn.com/product-manuals> as of April 27, 2016.
- DeAngelis, J.M., Hrnjak, P.S., (2005). Experimental study of system performance improvements in transcritical R744 systems with applications to bottle coolers. University of Illinois at Urbana-Champaign, Air Conditioning and Refrigeration Center (ACRC), Contract Report 57.
- Fuentes, Y.P., Elbel, S., Hrnjak, P., (2014). Proceedings from 15th International Refrigeration and Air Conditioning Conference at Purdue, July 14-17, 2014: Extremely Low Refrigerant Charge Beverage Display Cooler Technology Using Propane. West Lafayette IN USA.
- Fuentes P.Y., Hrnjak P., Elbel S., (2015). High-efficiency, low-cost glass door merchandisers using transcritical carbon dioxide. 24th IIR International Congress of Refrigeration, Yokohama, Japan, August 16-22, Paper 305.
- Kim, M.-H., Pettersen, J., Bullard, C.W., (2004). Fundamental process and system design issues in CO₂ vapor compression systems. *Progress in Energy and Combustion Science*, vol. 30, no. 2: p. 119-174.
- The Coca-Cola Company, (2016). 2014/2015 Sustainability Report. Retrieved from: <http://www.coca-colacompany.com/content/dam/journey/us/en/private/fileassets/pdf/2015/09/2014-2015-sustainability-report.pdf>.