Purdue University
Purdue e-Pubs

International Refrigeration and Air Conditioning Conference

School of Mechanical Engineering

2014

Extremely Low Refrigerant Charge Beverage Display Cooler Technology Using Propane

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

Stefan Elbel
stefan.elbel@creativethermalsolutions.com

Predrag S. Hrnjak
pega@creativethermalsolutions.com

Extremely Low Refrigerant Charge Beverage Display Cooler Technology Using Propane

Yadira PADILLA FUENTES[1], Stefan ELBEL[1, 2,*], 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

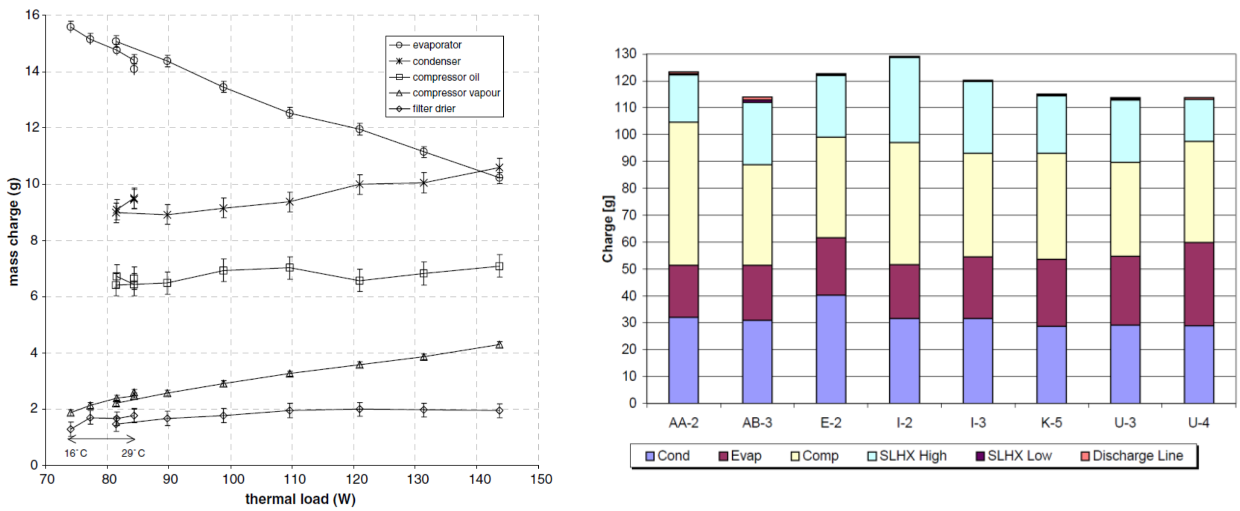
[*]Corresponding Author

ABSTRACT

The majority of beverage manufacturers prefer natural refrigerants over synthetic refrigerant options as the working fluid for their beverage display cooling equipment. The two major refrigerants considered for these light commercial applications are CO_2 (carbon dioxide) and R290 (propane). CO_2, used in a transcritical cycle, offers advantages in terms of flammability, however, reduced performance at high ambient temperatures and safety issues related to the high working pressures need to be addressed in order to design efficient and cost-competitive systems. The challenges encountered with CO_2 are typically no concern for R290; however, the major risk associated with propane is the flammability of the refrigerant. One way to mitigate the risk is rigorous reduction of refrigerant charge to levels that are substantially below legal limits. This task has been accomplished by employing experimentally validated simulation models that can be used to reliably predict the refrigerant charge in each component at different ambient conditions. Design and optimization efforts have to be focused on the compressor oil type and charge amount which dissolves large quantities of propane that does not contribute to generating the desired cooling effect. The other component that bears significant potential to reduce refrigerant charge is the condenser; low internal volumes are desired that can provide an optimal balance between required refrigerant charge, heat transfer, and pressure drop. A serpentine style microchannel heat exchanger design featuring low fin density on the air side was developed and implemented. The reduced number of fins on the air side reduces maintenance requirements and allows the system to be used in dusty environments. The improvements along with further optimizations were implemented into a beverage cooler holding 700 cans with a volume of 355 ml per can. The achieved cooling capacity was on the order of 1 kW featuring a propane refrigerant charge of 50 g. The performance of the redesigned beverage display cooler was experimentally validated at an ambient temperature of 32.2 ^{o}C and it was found that all of the manufacturers' pull-down and energy consumption test requirements were successfully achieved.

1. INTRODUCTION

Propane is an A3 fluid (ASHRAE Standard 34, 2004), therefore it is a flammable fluid which poses safety risks. To reduce this risk for R290 in bottle coolers sold in the United States, the cooler must comply with UL standards as well. The charge requirement for a flammable fluid is 150 g or lower in a single unit (UL Standard 471, 2010). For light commercial applications such as beverage display cooling equipment the most challenging part of system design is the reduction of refrigerant charge to this low limit of 150g. System design must also take into consideration cost of components and their manufacturability. Figure 1 shows refrigerant charge distribution data from Björk and Palm, (2006) and Hoehne and Hrnjak, (2004). The data confirm that as with other refrigerants, the majority of the R290 refrigerant charge is located on the high pressure side of the system due to the high densities that are encountered.



(a) Björk and Palm, 2006 (b) Hoehne and Hrnjak, 2004

Figure 1: Experimental refrigerant charge data of refrigerant hold-up in various system components

The components containing most of the charge are the condenser, compressor, and liquid line for a conventional vapor compression system. There are two ways to lessen charge. One way is through the reduction of system volume. The other option is the optimization of mass flux (Niño *et at.*, 2002) encountered by the heat exchanger tubes. Of these two methods, the more viable option for manufacturing is volume reduction since the options for cost-competing R290 compressors are limited. The first task was to theoretically model total charge in the compressor to understand the limits of charge in the total system. Then multiple condenser designs were studied and a final microchannel condenser design was developed to find a condenser with the lowest charge possible. The modeled microchannel design was then manufactured in-house. The improvements along with further optimizations were implemented into a beverage cooler holding 700 cans with a volume of 355 mL per can. The final results are then compared with model results.

2. MODELING CHARGE IN SYSTEM COMPONENTS

2.1 Compressor charge modeling

The compressor used for the R290 system had a total oil charge of 350 ml of Emkarate RL22H oil, which is of the POE type. The majority of refrigerant charge in the compressor is dissolved in the oil as discussed by Björk and Palm (2006). Using density and miscibility data, the total refrigerant charge absorbed in the oil can be calculated fairly accurately. Based on available VLE data for the mixture shown in Figure 2, an approximate refrigerant charge amount was calculated in the compressor. Since the oil is located on the low pressure side of the compressor, suction pressure was used to calculate the desired parameters. From baseline data, the shell of the compressor was measured and was used for charge prediction in the compressor. The average compressor shell temperature used for charge prediction was 60 ^{o}C while the suction pressure was assumed to be 400 kPa. Figure 2 shows that at these conditions the mixture consists of approximately 92% oil and 8% refrigerant. Using Table 1 the density of the compressor oil at 60 ^{o}C was estimated to be 0.972 g/cm^{3}. Therefore, an oil volume of 350 ml in the compressor equates to 340 g of oil at the given conditions. Using Equation 1 and the determined refrigerant concentration of 8% in the oil refrigerant mixture, the total mass of refrigerant absorbed in the oil was calculated to be 30 g. This corresponds to 20% of the allowed refrigerant charge of the 150 g, realizing that the refrigerant dissolved in the compressor lubricant does not contribute to generating the desired cooling effect. Furthermore, the free internal volume was estimated and assumed to be filled with vapor refrigerant at 60 ^{o}C. This led to charge of 14 g of R290 in the vapor phase in the free internal volume of the compressor. Table 2 shows the total charge prediction of R290 in the compressor, distinguishing between refrigerant that is absorbed in the oil and refrigerant that is stored in the free volume of the compressor, neither of which contributes to generating desired cooling effect.

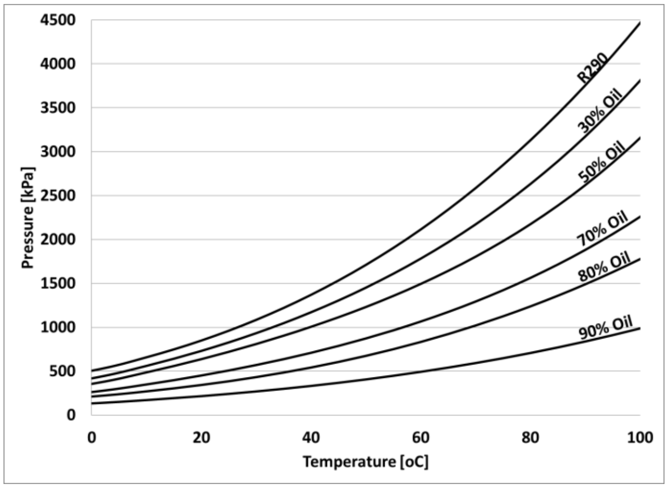


Figure 2: VLE data for Emkarate RL22H oil and R290 (Lubrizol, 2010)

Table 1: Density data for Emkarate RL22H oil (Lubrizol, 2010)

Temperature	**Viscosity RL22H**		**Density RL22H**
°C	**Absolute cP**	**Kinematic cSt**	**g/cm^3**
20	*	*	0.993
30	20.61	20.84	0.989
40	15.55	15.84	0.983
50	10.40	10.65	0.977
60	7.73	7.95	0.972
70	5.85	6.05	0.966
80	4.62	4.81	0.960
90	3.70	3.89	0.952
100	1.92	3.12	0.945
110	2.69	2.87	0.939
120	2.72	2.81	0.935
130	2.51	2.70	0.930

$$w = \frac{m_{ref}}{m_{ref} + m_{oil}} \tag{1}$$

Table 2: Theoretical distribution of refrigerant charge in compressor volume and oil

	Baseline compressor oil	Reduced compressor oil
Free internal volume (estimate) [ml]	2115	2115
Oil charge [ml]	350	200
Mass of refrigerant in free volume [ml]	14	14
Mass of refrigerant dissolved in oil [g]	30	21
Total mass of refrigerant in compressor [g]	44	35

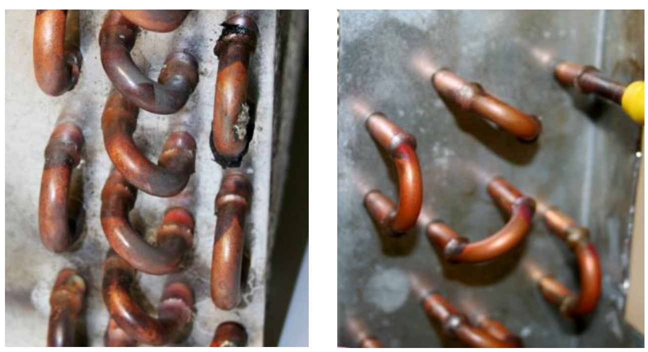
Due to this high amount of refrigerant absorbed in the oil, reduction of the total oil amount in the compressor was necessary. A collaboration with the compressor manufacturer led to a reduction of 100 ml of oil to be used in the compressor. This led to a total oil mass of 243 g in the compressor. Using the same process as before, 21 g of refrigerant charge absorbed in the oil was calculated with the new lower oil amount, representing a reduction of the dissolved refrigerant amount of 30%.

The major goal of this study was to demonstrate that the total propane charge could be reduced much further, and the aim was to achieve satisfactory cooling performance and efficiency with only 50 g of propane for a cooler that can hold 700 cans with a volume of 355 ml per can. The proposed propane charge of 50 g represents 33% of the allowable limit of 150 g. The authors believe that demonstrating the potential of extremely low-charged propane systems will further facilitate the introduction of R290 systems to the market, because of a substantial reduction of the risk associated with flammability.

The compressor oil charge calculation in Table 2 showed that 15g of R290 was available to the rest of the system when using a decreased oil amount. Since the condenser holds the highest amount of refrigerant charge in the system, the aim of this study made it necessary to entirely redesign the condenser in order to implement a design that would require the least amount of refrigerant possible while still being able to reject enough heat to pass all performance requirements of the cooler

2.2 Condenser design selection

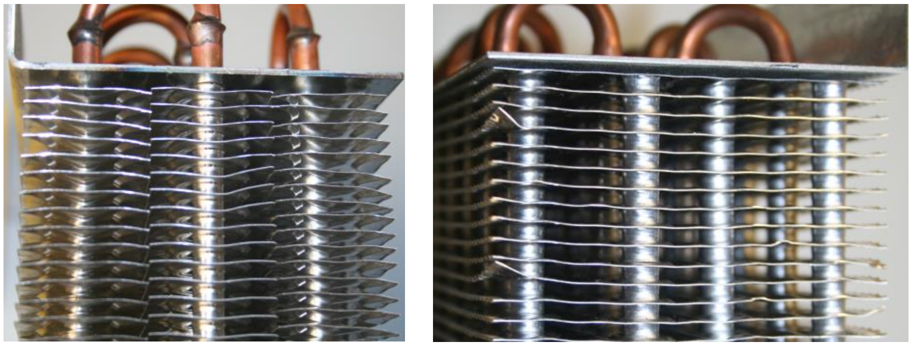
Multiple condenser designs were considered. The typical solution is to use an off the shelf condenser that is used in current bottle cooler systems. These are condensers with high fin density, typically copper, and large internal tube diameters (typically 7.94 mm internal diameter as shown in Figure 3a). Although the high fin density is an advantage in increasing the heat transfer area, it leads to large pressure drop on the air side and increased fouling on the condenser. Furthermore, the volume created by the large internal diameter leads to large refrigerant charge and makes these designs undesirable. The next logical step is to reduce the internal diameter of the round tube. Current cost effective manufacturing can bring the internal diameter to sizes as small as 5 mm for a RTPF design as shown in Figure 3b.



(a) (b)

Figure 3: Round-tube-plate-fin condensers used for bottle coolers

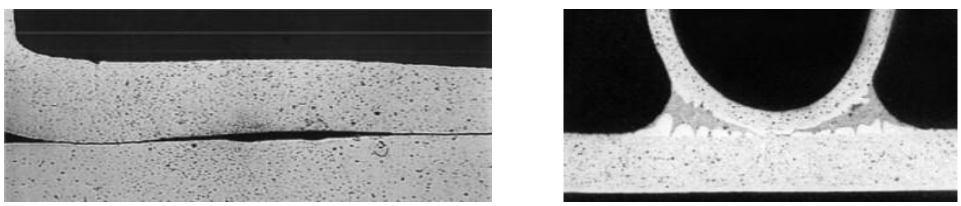
However, the reduction in internal diameter in the round tube decreases heat transfer area on the air side. Another negative impact on overall heat transfer has been observed due to the conduction of heat in the direction of opposite of the air flow in cross-counterflow designs. The hot refrigerant on the air outlet side conducts heat through the fin material in the direction opposite to the air flow, where it heats the refrigerant exiting the condenser on the side where it meets the cold air inlet. To accommodate this decrease in heat transfer the new condenser was designed with split fins as shown in Figure 4a. Typical RTPF condensers have fins that are connected from one tube row to the next as shown in Figure 4b. The cut fin design which ends the fins at each tube row not only reduces unwanted conduction opposite to the air flow, but also increases the local heat transfer coefficient on the air side by increasing turbulence and disrupting the boundary layer formed on the fin. Conversely, this also increases the pressure drop on the air side of the condenser. The split fin round tube, small internal diameter condenser is a viable option for a system to achieve charge less than 150 g of R290. Experiments showed that the system passed test requirements with a charge of 70 g of the allowed 150 g with the cut fin and smaller internal diameter RTPF condenser design. However, to further decrease charge by reduction of internal volume leads to the need to use a microchannel design resulting in even smaller refrigerant-side internal volume.



(a) (b)

Figure 4: Heat transfer augmentation in RTPF designs

Due to small internal volume, the microchannel design was considered next. There are several advantages of a microchannel tube. Air side pressure drop is reduced due to the flat shape of the tube. The fin density can also be decreased in comparison to RTPF counterparts since there is better contact of the fin to the microchannel plate as shown in Figure 5 (brazed instead of expanded) and thus better heat transfer. Internally, the small volume of the microchannel enables very low refrigerant charge. Furthermore the small diameter of the microchannel ports increase the local heat transfer coefficient in the tube which allows for better heat transfer and reduction of the coil size (Niño *et al.*, 2002). Finally, microchannel tubes are made from aluminum which is a low-cost and light material which can reduce expenses in production and shipping.



RTPF: Small gaps between fins and tubes when tube is expanded onto fins

Microchannel: No gaps in brazed fin connection

Figure 5: Fin contact between tube and fins in RTPF and microchannel coils

2.3 Microchannel condenser charge modeling

A microchannel condenser was designed using a simulation model implemented with Engineering Equation Solver (2011) assuming steady state conditions. The microchannel tube has the advantage of low internal volume. However addition of headers can drastically increase the total refrigerant charge in the coil. Therefore to reduce header size and total charge, a split serpentine design with 2 small headers was designed shown in Figure 6.

Figure 6: Designed microchannel condenser with internal volume of 80 ml

Although microchannel tubes have the advantage of small charge due to their small volume this small space also accounts for an increase in pressure drop across the condenser (Niño *et al.*, 2002). Therefore, care was taken in the design of the condenser to keep the pressure drop in the condenser within a satisfactory range (less than 100 kPa). To model the two-phase refrigerant pressure drop across the condenser the correlation by Cavallini *et al.* (2009) was used. The condenser was divided into two serpentine components which helped reduce the total pressure drop across the condenser as well as reduce charge by avoiding using large headers with high internal volumes.

To calculate the two-phase heat transfer within the two-phase region on the refrigerant side, the correlation by Cavallini *et al.* (2006) was used. The heat transfer coefficient correlation by Kondou and Hrnjak (2011) was used in the region between the superheated and the two-phase zone. The effects of oil on the refrigerant heat transfer coefficients and pressure drops was neglected.

In the single phase regions for superheated and subcooled zones the heat transfer coefficient by Dobson and Chato (1998) and friction factor by Churchill and Usagi (1972) to calculate single phase pressure drop were used.

For the heat transfer coefficient on the air side, the correlation by Park and Jacobi (2009) for louvered fins on microchannel tubes was utilized. Studies by Nino *et al.* (2002) and Padilla and Hrnjak (2012) showed that mass flux has a significant impact on the total charge in the condenser. Therefore, the correlation by Graham *et al.* (1999) which takes into account the effects of mass flux in void fraction was used.

The condenser was designed with lower fin density (315 fins per meter) to reduce fouling of the condenser. Using the above correlations a coil with a modeled internal volume of 80 ml and a refrigerant charge amount of 20 g was designed and is shown in Figure 6. The condenser was manufactured in-house and was installed in a bottle cooler system with 700 cans, each can containing 355 ml carbonated soft drink.

Similarly, the RTPF condensers were modeled using EvapCond V.3 by Domanski and Yashar (2010). The final RTPF condenser for charge reduction was modeled with 5 mm internal diameter. Table 3 shows the modeled results comparing the microchannel condenser and the RTPF condenser. It shows that the microchannel design has similar charge to the RTPF. However the microchannel design achieves this similar charge amount with lower fin density and higher cooling capacity. This makes the microchannel design the best option for charge reduction.

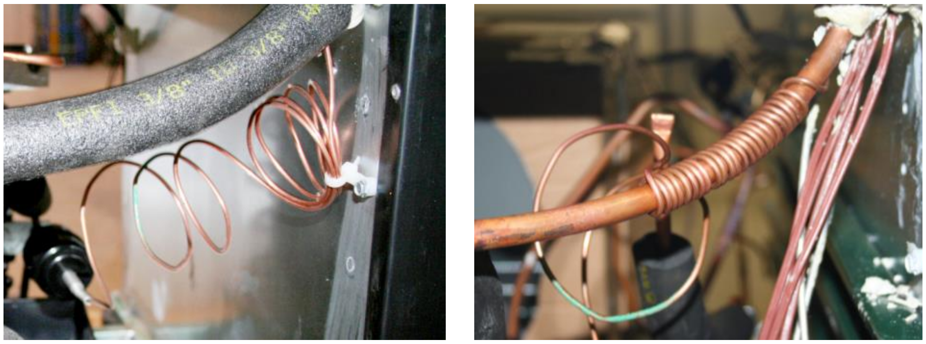
Table 3: Modeled condenser results

	RTPF Condenser	Microchannel Condenser
Fin pitch [m^{-1}]	630	315
Capacity [W]	690	740
Refrigerant Pressure Drop [kPa]	10	50
Internal Volume [ml]	225	80
Charge [g]	23	20

3. EXPERIMENTAL SETUP AND RESULTS

3.1 Other component improvements

In order to reduce charge further, the internal diameter and total length of the liquid line was reduced. Furthermore vestigial components such as the accumulator and filter drier were removed from the baseline system. A filter drier may not be needed if sufficient care is taken during the assembly process of the cooler. This reduced the system internal volume by 125 ml. Finally, an internal heat exchanger was made using the capillary tube and the return line as shown in Figure 7. The capillary tube was wrapped around the suction line and the contact was made with solder. For CO_2 bottle coolers, using an internal heat exchanger has large benefits in reducing total system energy consumption. For R290 this effect is smaller, however due to the simplicity of the component it was added to the system. Adding the internal heat exchanger also adds to the total liquid volume in the system, however the benefit on energy consumption outweighs the small addition of charge.



Before (baseline) After

Figure 7: Installation of internal heat exchanger using capillary tube and evaporator discharge line

3.2 Experimental setup

The improvements along with further optimizations were implemented into a beverage cooler with a UA value of 4.5 W/ ^{o}C. The cabinet was loaded with 700 cans with a volume of 355 ml per can. Each can was filled with sugar carbonated soft drink. For every shelf, the top most cans closest to the front of the door and were aligned either to the left, center, or right of the cooler were filled with 33% propylene glycol and 67% water solution. The top most cans on the back of the cooler located on the left and right edge closest to the cooler walls were also instrumented with 33% propylene glycol and 67% water solution. All cans filled with 33% propylene glycol and 67% water solution were instrumented with Type-T thermocouples and served as the temperature representing the product temperature distribution in the cooler. The performance of the redesigned beverage display cooler was experimentally validated at an ambient temperature of 32.2 ^{o}C and 65% relative humidity. The allowed time to bring all the products from ambient conditions to the desired temperature range between 0 ^{o}C and 7 ^{o}C was 19 hours. Subsequently, energy consumption of the cooler is run for a 24 hour period. During energy consumption period the cooler door is closed and the cooler cycles from on and off periods for a single speed compressor in order to maintain the product temperature between 0 ^{o}C and 7 ^{o}C.

3.3 Experimental results

The theoretical models showed that the compressor and the condenser total refrigerant charge was 55 g. However to accommodate for other components in the refrigeration system an initial charge of 65g was used. Using 65 g of R290 as working fluid with the serpentine microchannel condenser, a reduction by 100 ml of oil in the compressor, and the system improvements discussed earlier, the system attained a product pull down time of 16 hours and 30 minutes. The data showed that at 65 g of R290 the system was slightly overcharged with refrigerant and had 2 hour and 30 minutes lower pull down time than permitted. Therefore to reduce charge further, a propane charge of 50 g was used for the total system charge instead. Figure 8 shows the pull down curve of the system using 50 g of R290. The cooler passes the pull down time with 1 hour and 30 minutes to spare. For energy consumption tests, the cooler had approximately 4 kW/hr for a 24 hour period which was well within requirements. For R290 energy consumption is usually not an issue due to excellent thermophysical properties of propane, requiring less compressor power. However using a compressor that is too large or other components with high power requirements such as lighting or fans will increase the systems energy consumption. Hence energy consumption tests and improvements are still necessary for R290 even though energy consumption of the compressor for R290 is generally much lower than for comparable systems using CO_2.

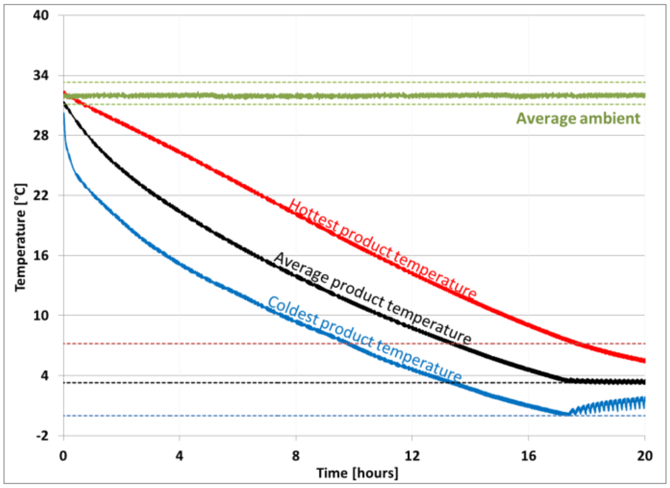


Figure 8: Pull down results of products for 50 g propane system

Previous studies have shown that charge predictions often under predict total charge in multiple microchannel condenser designs, for example Litch and Hrnjak (1999), Graham *et al.*(1999), Niño *et al.* (2002), Padilla and Hrnjak (2012) and others. This was seen once again in the modeling of the components of the system in the current study. Nevertheless, the prediction of refrigerant charge in the compressor and the condenser provided an accurate starting charge prediction to use for the system, so the usefulness of the modeling tool has been confirmed.

Moreover, the refrigerant charge amount used provided the necessary capacity for the system in order to pass the imposed time limit to bring the product temperatures from ambient conditions to required temperature conditions. The microchannel design for the condenser and the reduction in total oil amount in the compressor were essential system improvements necessary to reduce the total refrigerant charge.

4. CONCLUSIONS

The modeled propane charge in the compressor of 38 g showed that a significant amount of refrigerant was absorbed in the oil. Therefore to reduce the system charge, the amount of oil in the compressor was decreased. This led to an estimated 21 g of refrigerant charge absorbed in the oil in the compressor. Based on the compressor estimation, multiple condenser designs were studied. A microchannel condenser design was selected due to its small volume that minimizes the charge in the component. The condenser was designed to split into two parallel sections on the refrigerant side to reduce pressure drop. A low fin density was used to reduce possibility of fouling. The condenser was then manufactured in-house and was installed in the system. Other system improvements were incorporated on the cooler with a total capacity of 700 cans with 355 ml of carbonated soft drink each. These improvements included reducing the tube internal diameter in the liquid line, removing the accumulator and filter drier, and installing an internal heat exchanger by using the capillary tube wrapped around the evaporator return line. The cooler was validated at 32.2 ^{o}C and 65% relative humidity. The allowed pull down time to reach product temperatures between 0 ^{o}C and 7 ^{o}C was 19 hours. The system achieved a pull down time of only 17 hours and 30 minutes. Therefore the system passes this requirement in the standard. The results show that for a system with cooling capacity of approximately 1 kW, a highly efficient bottle cooler can be designed to have total R290 refrigerant charge of 50 g or less.

NOMENCLATURE

Symbols

m	mass	(g)
w	local oil mass fraction	(-)
RTPF	round-tube-plate-fin	

Subscripts

ref	refrigerant
oil	oil in compressor

REFERENCES

American Society of Heating, Refrigerating, and Air-conditioning Engineers (ASHRAE), 2004, Designation and safety classification of refrigerants, Standard 34-2004, Atlanta, GA, USA

Björk, E., Palm, B., 2006, Refrigerant mass charge distribution in a domestic refrigerator. Part II: steady state conditions, *Appl. Therm. Eng.*, vol. 26, no. 8-9: p. 866-871

Cavallini, A., Doretti, L., Matkovic, M., Rossetto, L., 2006, Update on condensation heat transfer and pressure drop inside minichannels, *Heat Transfer Eng.*, vol. 27, no. 4: p. 74-87

Cavallini, A., Del Col, D., Matkovic, M., Rossetto, L., 2009, Pressure drop during two-phase flow of R134a and R32 in a single minichannel, *J. Heat Transf.*, vol. 131, no. 3: p. 033107-1-033107-8

Churchill, S.W., Usagi, R., 1972, A general expression for the correlation rates of transfer and other phenomena, *AICHE J.*, vol. 18, no. 6: p. 1121-1128.

Dobson, M.K., Chato, J.C., 1998, Condensation in smooth horizontal tubes, *J. Heat Transf.*, vol. 120, no. 1: p. 193-213

Domanski, P.A., Yashar, D., 2010, EVAP-COND - Simulation Models for Finned-Tube Heat Exchangers with Circuitry Optimization, Version 3, National Institute of Standards and Technology; Gaithersburg, MD, USA

Engineering Equation Solver (EES) - Academic Professional V8.874, 2011, F-Chart Software, Middleton, WI, USA

Graham, D.M., Kopke, H.R., Wilson, M.J., Yashar, D.A., Chato, J.C., Newell T.A., 1999, An investigation of void fraction in the stratified/annular flow regions in smooth, horizontal tubes, Air Conditioning and Refrigeration Center, University of Illinois, Technical Report 144

Hoehne, M., Hrnjak, P.S., 2004, Charge minimization in systems and components using hydrocarbons as a refrigerant, Air Conditioning and Refrigeration Center, University of Illinois, Technical Report 224

Kondou, C., Hrnjak, P.S., 2011, De-superheating and condensation from R744 and R410A in smooth and microfin tubes near the critical point, Air Conditioning and Refrigeration Center, University of Illinois, Technical Report 264

Litch, A.D., Hrnjak, P.S., 1999, Condensation of ammonia in microchannel heat exchangers, Air Conditioning and Refrigeration Center, University of Illinois, Contract Report 22

Lubrizol, 2010, Low temperature miscibility & vapour liquid equilibrium Data Emkarate RL 22H with R-290

Niño, V.G., Hrnjak, P.S., Newell, T.A., 2002, Characterization of two-phase flow in microchannels, Air Conditioning and Refrigeration Center, University of Illinois, Technical Report 202

Padilla, Y., Hrnjak, P.S., 2012, Experimentally validated models of refrigerant distribution in microchannel heat exchangers used to evaluate charge reduction of various working fluids, Air Conditioning and Refrigeration Center, University of Illinois, Technical Report 290

Park, Y.G., Jacobi, A.M, 2009, Air-side heat transfer and friction correlation for flat-tube louver-fin heat exchangers, *J. Heat Transf.*, vol. 131, no. 2: p. 021801-1-021801-12

Underwriters Laboratories (UL), 2010, UL 471: Standard for commercial refrigerators and freezers