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Refrigerant Replacement from HFC to HC in a Medical Ultra-low Temperature Appliance: Experimental and Simulation Approach

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ABSTRACT

This work explores the migration of single speed HFC compressor to variable speed HC compressor in a cascade refrigeration circuit for Ultra Low temperature freezer application. In this way, the paper first describes a case study to replace HFC refrigerant pair R404A/R508B by low GWP alternative HC R290/R170. In order to define the best average working condition for the HC pair with variable speed compressors, a semi-empirical model is applied. The results showed that some circuit modifications are important to optimize system performance and keep the HC refrigerant charge below 150g. Experimental data indicated that to take the most advantage of the VCC technology it is paramount to optimize the compressor working speed for given appliance. As a final result the a reduction of 41.5% in energy consumption was obtained and allowed the system to reach 8.6% lower energy consumption than Energy Star target.

1. INTRODUCTION

With the Kigali Amendment to the Montreal Protocol, the global community made another important step towards the reduction of CO2 emissions. The global phase-down of HFCs in the refrigeration sector is reinforced by different regulations, as EU F-gas, California CARB (2018), and Canada HFC regulations (2017), which limit high global warming potential (GWP) refrigerants in different applications. Appliance and compressor manufacturers are being pushed to perform tests and research to find the most successful product portfolio. McLinden (2015) performed a broad theoretical evaluation of several fluids and concluded there are few potential Low-GWP refrigerants for volumetric capacity above 3 MJ, but they range from slightly flammable to highly flammable. One of the solutions is R290, which has 3.85 MJ/m^3 and a relatively high COP for its respective volumetric capacity. For the refrigerant studied so far, from compressor point the HC solution show the highest efficiency and lowest thermal profile, allowing broader operating envelope and increased reliability (Zgliczynski and Sedliak, 2018).
Even thou ULT appliances were excluded from those regulations there is a clear trend for the refrigeration industry (market) towards low GWP refrigerants (EU Regulation 517/2016). The attention to Ultra-Low Temperature (ULT) freezing application has increased much since the outbreak of the COVID-19 pandemic, especially due to mRNA vaccine preservation temperature requirement (Pfizer 2022, Australian Department of Health, 2022, Ontario Ministry of Health, 2022). The ULT freezers usually have an internal temperature range of -50°C to -90°C, which require evaporation temperature to down to -100°C. The conventional single-stage vapor compression circuits are not suitable for ULT applications due to high pressure-ratio to which the compressor is exposed, leading to low efficiency and high discharge temperatures. Although there are different technologies to reach such low temperatures, such as stirling cycle, auto-cascade and two-stage cascade, the vast majority of the Medical ULT appliances use the two-stage cascade refrigeration circuit. In this type of systems, traditional non-HC refrigerants used by the industry for the high temperature stage are R22, R404A, R410, R134a and for the low stage R23, R508B, R41 and R744a (Sun et al., 2019 and Rodriguez-Criado, 2022).

Table 1: Thermo physical properties of refrigerants

Refrigerant	R404A	R508B	R290	R170
Type	HFC blend	HFC	HC	HC
GWP	3922	13396	3	6
Safety Class	A1	A1	A3	A3
Boiling Temp @ 1atm (°C)	-46.6	-87,2	-42.1	-88.6
Critical Temp (°C)	72.2	11.2	96.7	32,2
Critical pressure (MPa)	3.73	3.77	4.25	4.87
Bubble-Dew @1atm (K)	0.85	0.45	0	0
Molar mass (kg/kmol)	97.6	95.4	44.1	30.0
LFL (kg /m3)	-	-	0.038	0.038
Max Charge IEC 60335-2-89:2019 (kg)	No limit	No limit	0.494	0.494
Lubrication oil	POE	POE	POE / AB	POE

In recent years a special interest has been shown in the replacement of R404A/R508B pair, for which one of the most important candidates is R290/170 (Table 1). The high GWP values of R404A and R508B refrigerants are included in the referred regulation phase-out list, although R404A is still largely used in a broad range of light commercial applications. Difficulties related to direct replacement with HC refrigerant are mainly related to the charge limitation of 150g, since the original cascade high temperature stage circuit may run with as much as 1kg R404A. The solution is usually related to tubes and heat exchanger redesign to reduce its internal volume. Industry trend has been to use 5mm mini-channel finned tube for larger appliances and micro-channel solutions for lower capacity appliances.
This paper does not have the intention to detail the component design procedures and methods or to provide deep theoretical explanation for two-stage cascade refrigeration operation and control particularities. The aim of the current study is to show that it is feasible to reduce component internal volume to attain the flammable refrigerant charge limit for each circuit and still reach the Energy Star certification target.

2. REFERENCE APPLIANCE AND MODIFICATIONS

This section describes the original ULT application circuit and the modifications required for refrigerant migration.

2.1 Original appliance

The ULT medical freezer tested in this study has ambient temperature working range from 15°C (59°F) to 32°C (89.6°F) with relative humidity 80% at 25°C (77°F). The storage compartment holds up to 480 standard 51mm boxes, with a volume of 680L (24 ft^3) and control range from -50°C (-58°F) to -86°C (-122.8°F). The original two stage cascade circuit applies the same 220V/60Hz hermetic reciprocating on-off compressor model with displacement 27.8cm^3 in both circuits. Original compressor has an ASHRAE LBP rating capacity of 1673W (5710 BTU/h) and COP 1.42 W/W (4.85 BTU/h/W) in R404A, with liquid and return temperature 32.2°C (90°F). The high stage working condition is -38.5°C (-37.3°F) evaporation, 30.0°C (86°F) condensation, 9.3°C (48.7°F) return temperature and sub-cooling 2.2°C (4.0°F), with estimated capacity is 868W (2960 BTU/h) and COP 1.18 W/W (4.0 BTU/Wh) in R404A. For the low stage R508B circuit the average working condition for -80°C (-112°F) set point was -96.7°C (-124°F) evaporation, 33.8°C (92.8°F) condensation and -14.7°C (5.5°F) return temperature.
The cascade circuit is represented with its main components in figure (1) scheme. The R404A high stage comprises a finned tube condenser, TXV expansion device, cascade plate evaporator, liquid line frame heater and tube-in-tube suction line heat exchanger. The low stage R508B component types are a finned tube pre-condenser, cascade plate condenser (CHX), capillary tube, suction line tube-in-tube heat exchanger (iHX), tube-in-tube sub-cooling heat exchanger (iHX.SUB) and auxiliary oil separator and expansion tank.

2.2 Refrigerant circuit modifications

Some original components were replaced allow the circuit to work at proper condition with R290/R170 refrigerant pair. As the modification was performed directly at the original appliance, it was not possible to replace all the desired components due to technical restrains. The decision was to not modify what could damage the cabinet insulation. As a result, even thou the LS skin evaporator and HS frame heater should be replaced by smaller

diameter tubes, these components were maintained as original. So current prototype is jeopardized regarding the required refrigerant charge and its effect on the heat exchangers performance.
In order to reach the energy consumption for Energy Star, the target was to select the compressor with highest available COP. It was possible to select a considerable smaller displacement VCC model because the original on-off compressor RTR was close to 65% to reach -75°C (-103°F) compartment temperature. The selected model has a displacement of 10.85 cm^3, COP 1.98 W/W (6.75 BTU/Wh) at 3000rpm and capacity 958W (3277 BTU/h) at max speed for ASHRAE LBP condition.
The expansion devices were replaced to adjust the required mass flow for the desired working condition at each circuit. For the HS the TXV solution was maintained and Danfoss model TD-1 (068N2020) was selected. In the same way, for the LS the capillary tube solution was maintained. The geometry was calculated according to procedure showed by (Melo *et al.*, 1999). Compared to R508B, the R170 capillary tube should have 30% lower equivalent nitrogen flow. For -80°C (112°F) internal compartment temperature, a condition of -85°C (-121°F) evaporation and 35°C (95°F) condensation was considered to selected capillary tube geometry 0.77mm internal diameter and 4.0 length. Because original design was with tube-in-tube suction line heat exchangers, the modified capillary tubes were maintained adiabatic.
The original HS condenser tube size of 9.5mm (3/8'") result in an internal volume of 0,830 L and would require a high R290 refrigerant charge, so the selected prototype condenser used 7.94 mm (5/16") tubes with internal volume 0,353 L. Following the condenser size reduction the original condenser fan with 305mm (12") blade was reduced to a 254mm (10"). Only slight power consumption reduction was obtained by the fan replacement, with original model consumption at appliance 40W and selected model 35W. After testing experimentally the modified circuit design, it was noticed that air flow distribution over the new HS condenser and the compressors was not satisfactory. As a result the HS condensing temperature increased, as well as compressor shell and compressor niche air temperature. These penalties led to lower cycle efficiency and higher thermal load for the bottom compartment.
To reduce further the internal volume the LS expansion tank was removed from the circuit. During the study it was also noticed that the HS frame heater needed to be removed. Due to its large tube diameter 9.5mm (3/8'") and 1.3m vertical line, the R290 vapor velocity reached values below 1.0 m/s and liquid refrigerant was not dragged to feed the TXV with sub-cooled refrigerant. The removal of the frame heater reduced the thermal load and contributed to lower energy consumption. The impact of this change was not quantified.

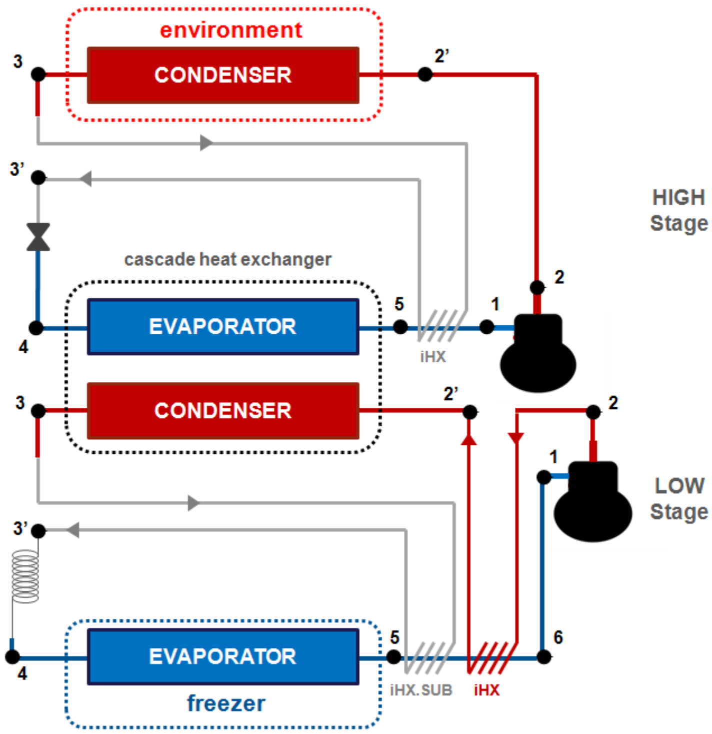

Figure 1: Cascade circuit scheme

3. CASCADE CIRCUIT SIMULATION FOR VCC OPERATION

A simulation was performed for the two stage cascade circuit to estimate the working condition which minimizes the energy consumption of the appliance. Current work is not focused on a detailed simulation and description of the method. The estimated simulation results are presented to guide the experimental tests and most suitable VCC working speeds to achieve the target internal compartment temperature of -75°C.

3.1 Simulation Model

The simulation was performed with a semi-empirical mathematical model, following the equations and procedures detailed in the work of Gonçalves *et al.* (2018) using the EES software (Klein, 2021). In summary it considers the mass and energy balances in each circuit component to converge the compressor working condition and total energy consumption. The energy balance equations are repeated for each component, following similar structure. Examples are shown at equation (1) and (2) for HS condenser, equation (3) and (4) for the HS suction line heat exchanger and equation (5), (6) and (7) for the cascade heat exchanger which connects the LS circuit to HS circuit equations. The equations subscripts indicate the figure (1) points, refer to the nomenclature section for details.

$$\dot{Q}_{cd} = \dot{m}_{HS}(h_3 - h_{2'})_{HS} \tag{1}$$

$$\dot{Q}_{cd} = UA_{cd}(T_{cd.HS} - T_{amb}) \tag{2}$$

$$(h_1 - h_5)_{HS} = (h_3 - h_4)_{HS} \tag{3}$$

$$\varepsilon_{iHX.HS} = (T_1 - T_5)_{HS}/(T_3 - T_5)_{HS} \tag{4}$$

$$\dot{Q}_{CHX} = \dot{m}_{HS}(h_5 - h_4)_{HS} \tag{5}$$

$$\dot{Q}_{CHX} = UA_{CHX}(T_{cd.LS} - T_{ev.HS}) \tag{6}$$

$$\dot{Q}_{CHX} = \dot{m}_{LS}(h_{2'} - h_3)_{LS} \tag{7}$$

In order to reach proper simulation assertiveness the model is put forward considering the specific circuit arrangement and control. The reference circuit operation with On-Off compressors was used to calibrate the empirical overall heat transfer coefficient multiplied by the area (UA parameter) for the heat exchangers and the suction line heat exchangers effectiveness ε.
Current model uses a simplified approach, where the expansion device mass flow model is not incorporated to the equation system. For both HS and LS, an isenthalpic expansion is considered from condenser pressure and sub-cooling degree to evaporator pressure. In the same way, current simulation model does not consider the refrigerant mass distribution over heat exchanger geometry or transient behavior in the components. To avoid high pressure peaks at the LS circuit, the original control start first the HS circuit and LS circuit is started only when the cascade temperature reach a given temperature. The simulation model has no capability to attain for this transient behavior, therefore it is expected that the simulation RTR values are slightly lower than the experimental ones. In this way the simulation results should be considered as estimations to narrow the experimental test investigation.

3.2 Simulation analysis

The simulation tool was used to estimate the (i) optimum cascade temperature (HS evaporation and LS condensation) and (ii) VCC speed ratio between HS and LS. It is know that by replacing some components, as described in section 2.2, the absolute energy consumption value and optimum working condition may shift. However, it is still considered a suitable way to fasten the experimental work. The results are shown in figure (2), for internal compartment -75°C (-103°F), original circuit components, refrigerant pair R290/R170 and VCC model.
The figure (2a) show the simulated RTR value as a function of the LS speed (N.LS) and HS speed (N.HS), where higher values are shown in green background and lower values in red background. In order to get a full picture, the simulation model was not limited to RTR values below 1.00 (100%). So the region where RTR is above 1.00 must be considered as not feasible working conditions and is marked with a gray background. It includes the point LS 2500rpm / HS 2000rpm and LS 2000rpm for any HS speed.
Energy consumption deviation from optimum point (ΔEC) is shown at the figure (2b), where dark green represents lower energy consumption and red higher value. Taking into accountant only the feasible working conditions, a relatively broad region is shown where ΔEC is below 2.5%. This region encompasses conditions with RTR 85% to 100%, HS evaporation -32.0°C (-25.6°F) to -41.2°C (-42.2°F) and HS speed 2500rpm to 4500rpm. Another more

restrict region, with ΔEC less than 1.0%, may be defined. In this case the LS compressor speed is 2500rpm, HS speed may range from 2500rpm to 4000rpm and RTR from 0.91 to 0.97. In this narrow region, a plateau is reached where the benefit in one stage is balanced by the penalty in the other stage. As an outcome of simulation analysis it is highlighted that energy consumption is more dependent to the LS compressor speed than to the HS for the analyzed circuit and compressor pair. Preliminary experimental analyses were conducted and confirmed simulation trend for RTR and energy consumption.
It is important to point out that not only the capacity or speed ration between HS and LS is important as well as the absolute capacities. Conditions with similar speed ration may differ up to 4% in energy consumption and conditions with same HS evaporation (intermediary pressure) may differ up to 9% in energy consumption.

(a)

Running Time Ratio [-]

N.HS \ N.LS	2000	2500	3000	3500	4000	4500
2000	1.14	1.01	0.91	0.85	0.80	0.76
2500	1.10	0.97	0.88	0.81	0.76	0.72
3000	1.08	0.94	0.85	0.79	0.74	0.70
3500	1.05	0.92	0.83	0.77	0.72	0.68
4000	1.04	0.91	0.82	0.75	0.71	0.67
4500		0.89	0.81	0.74	0.69	0.66

(b)

ΔEC [%]

N.HS \ N.LS	2000	2500	3000	3500	4000	4500
2000	-0.3	1.9	4.8	8.4	12.7	17.6
2500	-1.2	0.5	3.1	6.3	10.0	14.3
3000	-1.4	0.0	2.3	5.2	8.6	12.5
3500	-1.2	0.1	2.2	4.9	8.0	11.7
4000	-0.6	0.5	2.5	5.0	8.0	11.5
4500		1.3	3.2	5.6	8.5	11.8

(c)

T.ev.HS [°C]

N.HS \ N.LS	2000	2500	3000	3500	4000	4500
2000	-31.4	-28.1	-25.3	-23.0	-21.0	-19.1
2500	-35.2	-32.0	-29.4	-27.1	-25.2	-23.4
3000	-38.2	-35.1	-32.5	-30.4	-28.5	-26.8
3500	-40.5	-37.5	-35.1	-33.0	-31.1	-29.5
4000	-42.5	-39.5	-37.1	-35.1	-33.3	-31.7
4500		-41.2	-38.8	-36.8	-35.1	-33.5

Figure 2: Simulation result for original circuit configuration with R290/R170 VCC model.

Following Gosney (1982) indication for optimum intermediary pressure in equation (8), it was expected that the HS compressor speed would lead to higher energy consumption deviation. Higher HS compressor speed would lead to direct change in HS condensation, HS evaporation and LS condensation, thus deviate the intermediary pressure from optimum point. As discussed, figure (2c) show that the HS evaporation may change up to 7.5K (13.5°R) with energy consumption deviation below 1.0%.

$$P_{int} = \sqrt{P_{HS.cd}\, P_{LS.ev}} \tag{8}$$

It is important to point out that the low energy consumption sensitiveness to evaporation and condensation temperatures is mainly due to large original heat exchangers, where HS condensation and LS evaporation change few with the compressor speed. In the review performed by Jiang (2016), there is indication that energy consumption deviation is small when working close to optimum theoretical intermediate pressure. It is important to notice this plateau-like behavior for the selected VCC model efficiency curve and heat exchanger components converge to specific working pressures. Presented results and trends should not be considered as a reference for different circuits, since different optimum speeds, speed relation and intermediary cascade temperature may occur for other components and compressor models.

4. EXPERIMENTAL RESULTS

Current study targeted the energy consumption criteria for Energy Star Program (2014), where ambient dry-bulb temperature should be 24±1°C (75.2±1.8°F), relative humidity circa 55% and interpolated compartment temperature -75°C (-103°F). The program certification determines the maximum daily volume normalized energy consumption at -75°C of 19.42 kW/day/m^3 (0.55 kW/day/ft^3). For current application the target energy consumption is 13.2 kW/day. The experimental procedure followed two appliance modification steps to be described below.

		(a) On-Off		(b) VCC (150g)		(c) VCC.cd (150g)	
		Setpoint -70°C	Setpoint -80°C	Setpoint -70°C	Setpoint -80°C	Setpoint -70°C	Setpoint -80°C
High stage	Speed	3600 RPM	3600 RPM	3000 RPM	4500 RPM	3000 RPM	4500 RPM
	Cd.in	88,0°C	84,4°C	50,7°C	56,9°C	61,0°C	71,9°C
	Cd	30,2°C	29,2°C	25,9°C	26,0°C	29,1°C	30,8°C
	TXV.in	29,1°C	27,9°C	23,5°C	23,2°C	26,3°C	27,5°C
	SUB	1,1K	1,3K	2,4K	2,8K	2,8K	3,3K
	Ev.in	-37,9°C	-41,1°C	-27,3°C	-34,0°C	-26,0°C	-31,5°C
	Ev	-	-	-28,2°C	-35,2°C	-26,4°C	-32,1°C
	Ev.out	-24,0°C	-34,9°C	-10,3°C	-15,7°C	-12,3°C	-16,3°C
	SUP	13,9K	6,2K	17,9K	19,5K	14,1K	15,8K
	Return	9,5°C	9,5°C	-	-	18,5°C	16,9°C
Low stage	Speed	3600 RPM	3600 RPM	2000 RPM	3600 RPM	2000 RPM	3450 RPM
	Cd.in	4,6°C	-1,7°C	1,7°C	-1,7°C	3,3°C	1,2°C
	Cd	-	-	-24,9°C	-31,0°C	-24,7°C	-30,1°C
	Cap.in	-39,3°C	-43,4°C	-28,1°C	-35,6°C	-26,2°C	-32,1°C
	SUB	-	-	3,2K	4,6K	1,5K	2,0K
	Ev	-86,0°C	-87,2°C	-79,5°C	-88,3°C	-78,5°C	-87,4°C
	Ev.out	-52,0°C	-64,8°C	-28,7°C	-37,8°C	-25,3°C	-34,0°C
	SUP	34K	22K	50,8K	50,5K	53,2K	53,4K
	Return	-10,7°C	-16,6°C	12,2°C	10,1°C	12,3°C	9,4°C
T.cab.avg		-67,6°C	-75,3°C	-73,5°C	-82,5°C	-69,6°C	-77,7°C
Energy Consumption (RTR)		16.55 kWh/day (55.5%)	20.79 kWh/day (73.4%)	10.97 kWh/day (100%)	13.66 kWh/day (100%)	10.34 kWh/day (87.2%)	12.94 kWh/day (79.4%)
EC@-75°C		**20.62 kWh/day**		**11.42 kWh/day**		**12.07 kWh/day**	
Cabinet 1 (top)		-63,1°C	-68,7°C	-68,9°C	-76,8°C	-66,3°C	-72,8°C
Cabinet 2		-68,6°C	-76,1°C	-74,7°C	-83,3°C	-70,4°C	-78,5°C
Cabinet 3		-69,8°C	-78,1°C	-75,9°C	-85,0°C	-71,7°C	-80,1°C
Cabinet 4		-69,1°C	-78,2°C	-74,3°C	-84,9°C	-69,9°C	-79,5°C

Figure 3: Experimental results for original circuit heat exchangers.

4.1 Circuit modification step 1

The original appliance circuit arrangement with on-off compressor was tested. This is considered the reference condition and results are showed on figure (3a). A first circuit modification, comprising VCC models and expansion device modifications, was tested with the allowed refrigerant limit of 150g per circuit and maintaining compressor 100% on. Comparing the results in figure (3a) and (3b) show that by replacing compressor model and expansion device it was possible to reduce the interpolated energy consumption from 20.62 kWh/day to 11.42 kWh/day. As figure (3) shows that by maintaining the SUB values close to 2K (3.6R), the evaporator SUP increased in both circuits due to lack of refrigerant charge and the heat exchanger area was not being completely at the evaporation temperature. The VCC speed control allowed reaching an operating condition close to the simulation optimum region, which together with the compressor continuous operation (RTR 100%) reduced the cut-in/cut-off cycle penalty and allowed to reach the expressive energy consumption reduction of 44.6%.

With latest updated of the IEC 60335-2-89 the question about results with refrigerant charge above 150g could arise. It is expected that the HS and LS evaporator outlet temperature would reduce, leading to higher cooling capacity and lower cabinet temperatures for fixed 100% compressor RTR. The VCC model datasheet states a maximum allowed refrigerant charge of 150g, so no tests with more than this limit are presented in current study.

4.2 Circuit modification step 2

A second test arranged aimed to reduce the optimum refrigerant charge to attain low SUP value and continued the component modifications by replacing the HS condenser, HS condenser fan and LS pre-condenser. The LS receiver and LS expansion tank were removed from the circuit in this step. The VCC control was also modified to follow the original on-off thermostat cut-in/cut-off signal. Only one test was performed at this circuit arrangement. The VCC at same fixed speed as former VCC tests from figure (3b), exception to the LS at -80°C set-point condition, where the VCC working speed was 3450rpm instead of the planned 3600rpm.

Comparing results from figure (3b) to figure (3c), the LS operate at similar condition, but the HS has some differences. Even thou the HS evaporator SUP is reduced, the condenser inlet and condensing temperatures increased as a result of the smaller condenser area and deteriorated air flow. The step 2 modification energy consumption of 12.07 kWh/day was 5.7% higher than step 1 modification, however it is important to remark that besides circuit component modification, the least test also includes inefficiencies related to equalization and LS cooling delay. The step 2 control mode rough penalty is estimated to be 6% due to LS cooling delay, as 2,5min out of average 40min on cycle duration, with an additional penalty due pressure equalization. So it is considered that circuit modification step 2 showed slight advantage over earlier results.

The step 2 result, shown in figure (2c), was considered the final configuration for ULT prototype. An energy consumption of 12.07 kWh/day was reached, which represents a reduction of 41.5% over the original on-off configuration and 8.6% below the Energy Star certification threshold. As a secondary output, the proposed design step 2 also allowed component size reduction.

5. CONCLUSIONS

This paper presented a sequence of experimental two stage cascade circuit modifications for migration from single speed HFC compressor arrangement to low GWP HC refrigerant pair R290/R170, working with VCC in both HS and LS circuit. The experimental evaluation was supported by a semi-empirical steady state simulation approach to define a favorable cascade heat exchanger temperature working region and HS/LS VCC speed to attain the condition. It was possible to reduce much the original On-Off compressor displacement, which allowed to select a VCC model with 40% higher COP. As a final result the a reduction of 41.5% in energy consumption was obtained and allowed the system to reach 8.6% lower energy consumption than Energy Star target. Current study shows that components modifications are required when replacing the refrigerant pair, mainly to reach proper working conditions and refrigerant charge reduction. Even thou it is not a drop-in solution, it is shown that it is feasible to perform the migration towards low GWP refrigerants and still improve the appliance efficiency. With the preliminary evaluation of modification on an existing cabinet, it is showed that, the study could continue by implementing the design changes in a new factory made cabinet. Suggestions to further components redesigned to be considered are the overall tube diameter reduction (e.g. HS frame heater and LS evaporator), LS oil separator size, smaller tube-in-tube iHX, internal natural air flow evaluation to reduce top shelf temperature and cabinet bottom insulation. This study maintained fixed VCC speed for each compartment set-point, for future work it is also suggested to explore the VCC speed control during the on cycle.

NOMENCLATURE

ε	heat exchanger effectiveness	(°C/°C)
ΔEC	energy consumption deviation from optimum point	(%)
U	overall heat transfer coefficient	(W/K)
A	heat exchanger area	(m^2)
N	compressor speed	(RPM)
LBP	low back pressure	(–)
HS	high stage	(–)

LS	low stage	(–)
COP	coefficient of performance	(W/W)
RTR	running-time-ratio	(min/min)
VCC	variable capacity compressor	(–)
TXV	thermostatic expansion valve	(–)
GWP	global warming potential 100 years	(–)
HC	hydrocarbon	(–)
HFC	hydrofluorocarbon	(–)

Subscript

1	compressor suction
2’	condenser inlet
3	condenser outlet
4	evaporator inlet
5	evaporator outlet
amb	ambient
cd	condenser
CHX	cascade heat exchanger
ev	evaporator
iHX	suction line (internal) heat exchanger
int	intermediary pressure

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