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Design and Investigation of a Transcritical R744 Refrigerated Container for Military Applications

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Design and Investigation of a Transcritical R744 Refrigerated Container for Military Applications

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ABSTRACT

This paper describes the design and performance evaluation of a transcritical R744 (CO_2) multi-temperature, mobile refrigerated container system. The efficient and widespread use of R744 in both large-scale supermarket refrigeration systems and small-scale glass-door merchandisers has been shown through numerous studies and field demonstrations in recent years, indicating the suitability of this refrigerant to almost any refrigeration application. However, the extreme operating conditions of this refrigeration application intended for use in ambient temperatures up to 57°C while still maintaining a frozen temperature of -20°C make using R744 as a refrigerant a unique challenge here. The targeted use of the refrigerated container is for military applications, but a successfully developed system will show that R744 is a suitable refrigerant for a range of container applications. In order to achieve reasonable efficiency at the extreme conditions of this application, several improvements have been implemented: Improved gas cooler performance with a microchannel heat exchanger, internal heat exchange, and expansion work recovery with an ejector. The benefits of each of these improvements are discussed, and preliminary results are presented to show the realistic performance enhancement that can be achieved with each of these improvements. The results presented in this paper show that while the very high ambient temperature of this system presents a unique challenge, it also allows for very significant COP improvement using each of the above improvements methods.

1. INTRODUCTION

The use of CO_2 (R744) for refrigeration systems has been of interest for many years. Much attention has been given, particularly in recent years, to supermarket systems using transcritical CO_2 as a working fluid, with several of these systems employing parallel ejectors for expansion work recovery (Hafner *et al.*, 2014). However, the application of CO_2 ejector technology to smaller-scale systems, such as refrigerated container systems, presents unique challenges. Of particular interest in this study is the development of a refrigerated container system using CO_2 for military applications.

The use of CO_2 as a refrigerant for military applications has been of interest for several years, with one example of previous development focusing on the conversion of an R134a environmental control unit (ECU) to a CO_2 system, as described in detail by Elbel and Hrnjak (2010). The authors of this study showed that in comparison to the existing R134a ECU, the CO_2 ECU resulted in approximately 30 % higher COP and improved weight and volume compactness (per unit cooling) of approximately 60 % and 40 %, respectively; the air temperatures in this study were 52°C for the ambient air and 32°C for the indoor air. They also showed that in comparison to a commercially available R410A ECU, the CO_2 ECU system provided a 20 % improvement in COP (based on manufacturer data). The results of this study showed the very promising potential that CO_2 has to increase both efficiency and compactness of air conditioning and refrigeration systems in military applications compared to current units.

The objective of this study is to design and determine the performance of a 20 ft. ISO container with refrigeration system capable of providing cooling to both refrigerated and frozen compartments (multi-temperature container), meaning that a system with two separate evaporators would be most suitable. The refrigerated (medium-temperature, MT) compartment would occupy 3/4 of the length of container, while the frozen compartment would occupy 1/4 of the length of the container. The refrigeration unit is permanently mounted to an extension on the end of the container. Figure 1 shows an image of the existing unit with R404A and a diagram of the different compartments. The targeted design conditions and specifications of the CO_2 container system are listed in Table 1.

(a) (b)

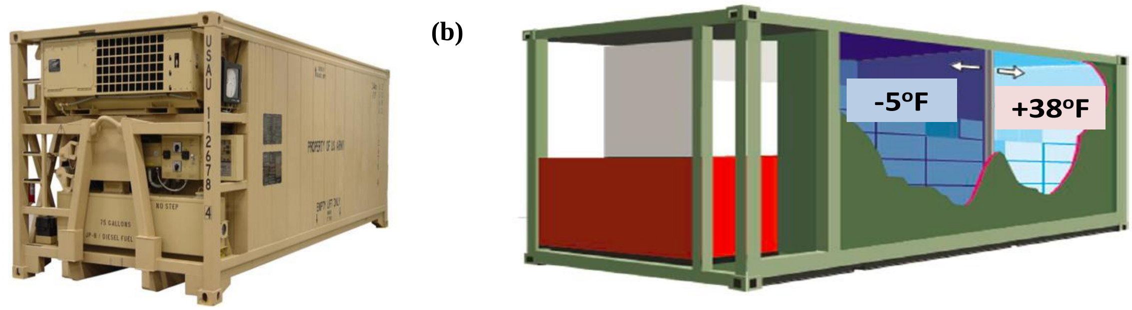

Figure 1: Refrigerated container unit used by U.S. Army: (a) image of existing unit and (b) diagram of different compartments.

Table 1: Container design conditions and specifications.

Parameter	Value	Parameter	Value
Ambient temperature	57.2°C (135°F)	COP	≥ 1.0
Refrigerated (MT) temperature	3.3°C (38°F)	Capacity	30 % reserve capacity at design condition
Frozen (LT) temperature	20.5°C (-5°F)	Refrigeration unit dimensions	1.8 m (W) x 0.6 m (D) x 0.6 m (H)
MT capacity	3.2 kW	Weight	< 540 kg (< 1200 lbs.)
LT capacity	2.5 kW		

The very high ambient temperature specified for this system (57°C) presents a significant challenge in terms of system design and performance, as transcritical CO_2 systems are known to suffer from significantly reduced capacity and efficiency at even moderately high ambient temperatures. Commercial, single-temperature, CO_2 refrigerated transport containers have recently been developed using a two-stage compressor with vapor injection at intermediate pressure for performance improvement at increased ambient temperature (Carrier, 2015). In comparison to the commercially available unit, the unit developed in this study will be multi-temperature and will be further enhanced in several ways to ensure sufficient capacity and efficiency even at very high ambient temperature. In order to achieve a COP of at least 1.0 at the extreme conditions targeted in this study, a number of improvements will need to be made to the standard transcritical CO_2 cycle. The improvements investigated in this project are the use of a microchannel gas cooler instead of a round-tube-plate-fin gas cooler, use of an internal heat exchanger (IHX), and use of an ejector for expansion work recovery. This paper presents a discussion of the challenges of using CO_2 as a refrigerant in this application and the results of a numerical study to predict the performance of a multi-temperature CO_2 container unit, identifying the improvement that is offered by using a microchannel gas cooler, adding an IHX to the cycle, and integrating an ejector into the system.

2. SYSTEM DESIGN

A schematic and a pressure-specific enthalpy diagram of the intended cycle architecture (including IHX and ejector) are shown in Figure 2. Details and potential challenges of implementing this system are discussed below.

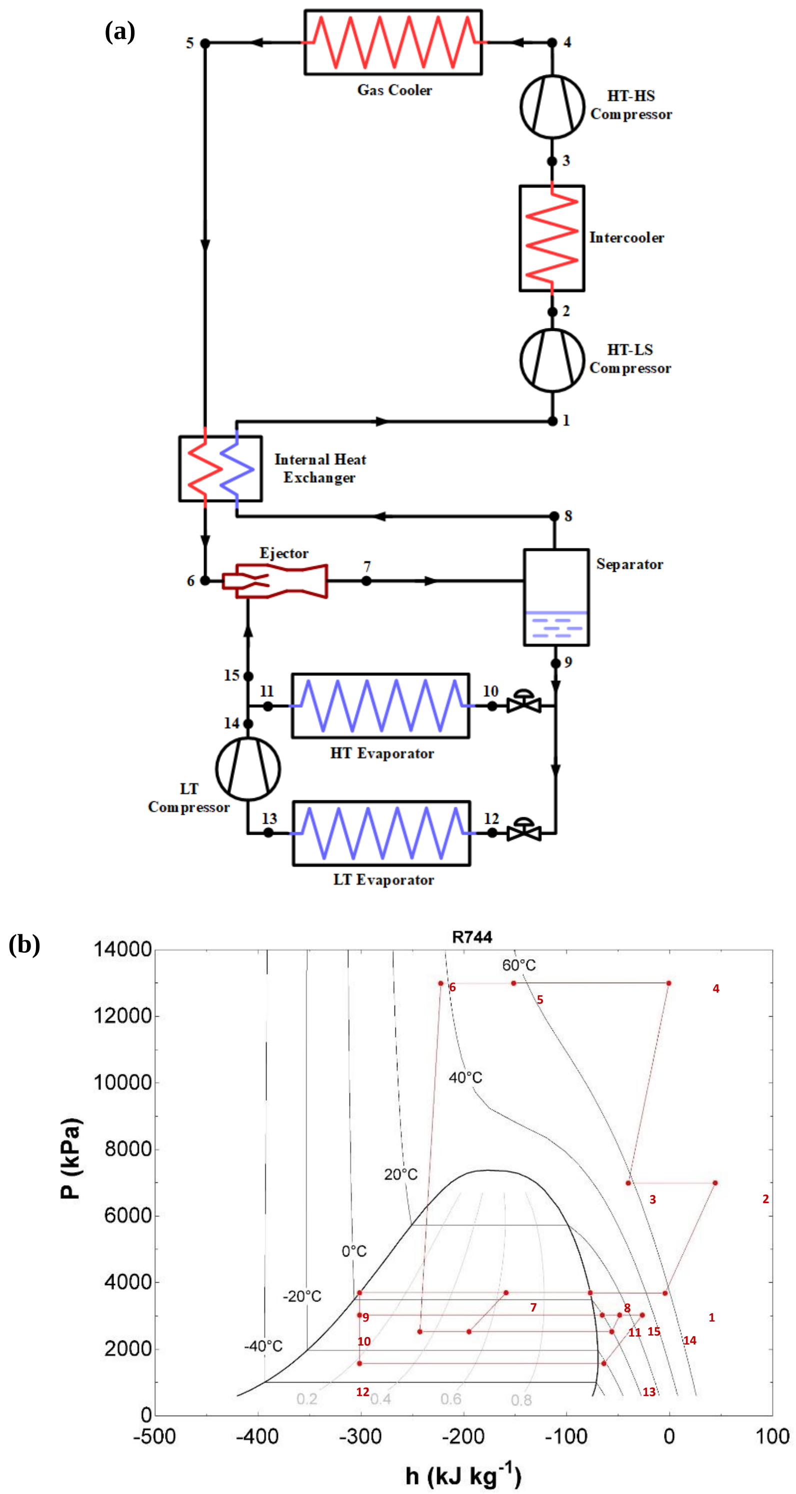

Figure 2: Multi-temperature CO_2 container (a) system schematic and (b) pressure-specific enthalpy diagram.

2.1 Compressor and Intercooler

Due to the very large pressure ratio between the LT evaporator (approximately 16 bar) and the gas cooler (130 bar maximum allowable pressure), multiple stages of compression must be employed. At a minimum a subcritical booster compressor must be used to increase pressure from the LT evaporator to the MT evaporator. Additionally, multiple stages of compression between the MT evaporator and gas cooler (with intercooling in between the stages) would further enhance performance. However, compressors capable of operating between MT evaporator pressure (approximately 30 bar) and gas cooler pressure with a single stage of compressor are commercially available. In order to achieve reasonable efficiency, this study will target the use of a two-stage compressor with an intercooler, though it remains to be seen if a suitable multi-stage compressor is commercially available that can meet the operating requirements of this application (capacity and pressure range). Given the size and weight restrictions of the refrigeration unit, it would not be practical to use two separate transcritical compressors in place of the two-stage transcritical compressor.

2.2 Heat Exchangers

Two options have been considered for the type of heat exchangers to be used: Microchannel (MC) heat exchangers and round-tube-plate-fin (RTPF) heat exchangers. Examples of these two heat exchanger types are shown in Figure 3. MC heat exchangers offer advantages over the more conventional RTPF heat exchangers because they are made of light-weight, low-cost aluminum and they offer improved air-side heat transfer performance due to greater contact between the fin and the tube. However, due to the parallel tube configuration in the inlet header, distribution of two-phase refrigerant among the parallel tubes can be an issue (not a problem for the gas cooler). Additionally, MC heat exchangers are more susceptible to fouling and can be more susceptible to frosting, both of which would be of concern for evaporators. MC heat exchangers would be an easy choice for the gas cooler, but due to the disadvantages mentioned above, round-tube heat exchangers will likely be used for the evaporators. The numerical analysis below will investigate the effect of using MC gas cooler and evaporators compared to RTPF gas cooler and evaporators, identifying the improvements in both heat exchanger and cycle performance.

(a) **(b)**

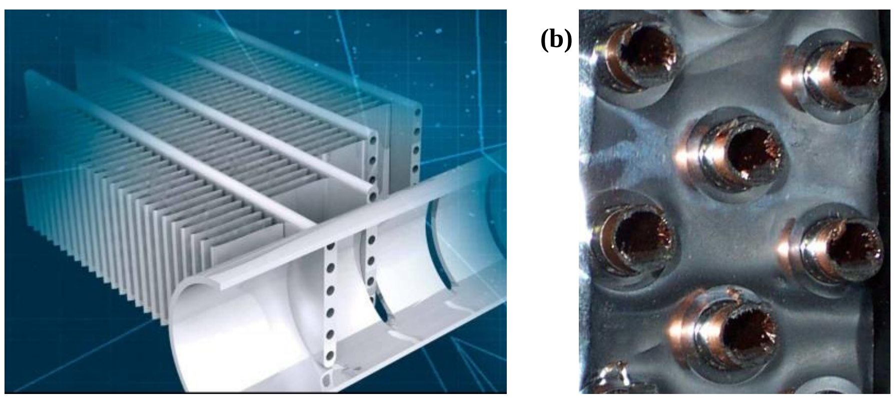

Figure 3: Examples of (a) microchannel (MC) heat exchanger and (b) round-tube-plate-fin (RTPF) heat exchanger.

2.3 Ejector

An ejector is a simple, low-cost device that uses the expansion of the high-pressure fluid from the gas cooler (or IHX) to increase the pressure of the low-pressure fluid from the evaporator; the ejector replaces high-pressure control valve in conventional, direct expansion CO_2 transcritical systems. Ejectors are most beneficial at high-ambient temperature conditions. As noted above, ejectors have been used in several recent CO_2 supermarket installations, mainly in Europe. Elbel and Lawrence (2016) provide further information on recent ejector research and its application to supermarket and other systems. When implementing ejectors in CO_2 supermarket systems, it is common to use several ejectors in parallel, each of which can be controlled on or off independently in order to efficiently control high-side pressure and flow rate for different system conditions and capacities. However, for a smaller systems such as a refrigerated container, installing a set of 4 to 6 parallel ejectors with a complex control strategy is not practical. Thus, a different type of ejector, namely an adjustable ejector in which an adjustable position needle is used to control the effective size of the ejector nozzle throat, will be used. Adjustable ejectors are known to offer lower efficiency than fixed ejectors due to additional losses associated with the needle; however, this seems to be the most reasonable method for controlling high-side pressure with an ejector for a smaller-scale refrigeration application.

2.4 Separator

An additional practical challenge of this cycle is the implementation of an efficiency liquid-vapor separator. A separator is required in this cycle at the ejector outlet to send vapor to the compressor suction and liquid to the evaporators. An efficient separator must be identified in order to prevent vapor from being sent to the evaporators and liquid from being sent to the compressor suction. At the same time, the oil in the liquid phase must be returned to the suction line. The design of efficiency, compact liquid-vapor separators and proper oil return methods for ejector refrigeration systems are topics that requires much additional research before ejectors can be applied to small-scale systems. However, for the analysis presented below, it is assumed that the separator operates with perfect separation, perfect oil return, and no pressure drop.

3. SYSTEM PERFORMANCE PREDICTION

3.1 Model description

The cycle is modeled using individual component models linked together into a cycle through the physical constraints of the system (mass and energy balances between components). The cycle shown in Figure 2 has been implemented as well as simplified cycles without an ejector and without an internal heat exchanger. The computer software Engineering Equation Solver or EES (F-Chart Software, 2016) is used to simultaneously solve the set of non-linear equations used to model the cycle. The compressors were modeled by assuming values of compression efficiency (defined as specific enthalpy difference of refrigerant assuming isentropic compression over actual specific enthalpy different of refrigerant) and mechanical efficiency (defined as power transferred to refrigerant over total input power). The fans were modeled by assuming a constant efficiency (defined as the power of an isentropic fan over actual input power). The calculated COP takes into account the power required by the fan for each heat exchanger. The ejector was modeled by using the zero-dimensional, constant-pressure mixing model of Kornhauser (1990). The ejector model assumed that mixing occurred at a pressure 1 bar lower than the suction pressure. The design conditions used of simulation are shown in Table 2. The air flow rates are relatively high but will help achieve a high COP given the extreme temperatures.

Table 2: Summary of system simulation parameters.

Parameter	Value	Parameter	Value
LT air temperature	-20.6°C	Compressor efficiencies (all)	0.70
LT air flow rate	0.43 $m^3 s^{-1}$		
MT air temperature	-3.3°C	Fan efficiency	0.30
MT air flow rate	0.43 $m^3 s^{-1}$	Ejector component efficiencies	0.75
Evaporator superheat	5 K		
Gas cooler air temperature	57.2°C		
Gas cooler air flow rate	0.98 $m^3 s^{-1}$		
Max. gas cooler pressure	130 bar		
Max. intercooler pressure	70 bar		

The heat exchangers were modeled using a finite-volume approach, in which each refrigerant tube is divided into discrete volumes or elements, and momentum and energy balances are applied to each element in order to determine the outlet of each element (and thus inlet of the next downstream element). The effectiveness-NTU method was used to determine heat transfer in each element. Refrigerant- and air-side pressure drops and heat transfer coefficients were determined based on the inlet state of each element and empirical correlations from the open literature. Dry conditions were assumed in all heat exchangers. The correlations choices are shown in Table 3.

Table 3: Summary of empirical correlations used in heat exchanger models.

Parameter	Correlation	
	MC Heat Exchangers	**RTPF Heat Exchangers**
Air-side		
Heat transfer coefficient	Park and Jacobi (2009)	Wang *et al.* (2000)
Pressure drop	Park and Jacobi (2009)	Wang *et al.* (2000)
Refrigerant-side two-phase region		
Heat transfer coefficient	Gungor and Winterton (1986)	Gungor and Winterton (1986)
Void fraction	Zivi (1964)	Zivi (1964)
Frictional pressure drop	Lee and Mudawar (2004)	Friedel (1979)
Refrigerant-side single-phase region		
Heat transfer coefficient	Gnielinski (1976)	Gnielinski (1976)
Frictional pressure drop	Churchill (1977)	Churchill (1977)

3.2 Comparison of Microchannel and Round-tube Gas Coolers

A comparison of the system performance with MC heat exchangers compared to RTPF heat exchangers of the same size is presented here. The dimensions of the heat exchangers are shown in Table 4.

Table 4: Comparison of dimensions used in MC heat exchangers and RTPF heat exchangers.

Parameter	Evaporators		Gas Cooler	
	MC	**RTPF**	**MC**	**RTPF**
Height	0.36 m	0.36 m	0.50 m	0.50 m
Width	0.61 m	0.61 m	0.81 m	0.81 m
Number of slabs/rows (parallel to air flow)	2	2	4	2
Tubes per slab	73	20	84	28
Inner diameter	-	4.8 mm	-	6.3 mm
Outer diameter	-	6.3 mm	-	4.8 mm
Ports per tube	6	-	4	-
Port hydraulic diameter	0.8 mm	-	0.9 mm	-
Tube pitch	8.2 mm	18.0 mm	9.7 mm	18.0 mm
Fin depth per slab	7.9 mm	11.0 mm	6.4 mm	16.8 mm
Fin pitch	1.4 mm	1.4 mm	1.4 mm	1.4 mm
Fin thickness	0.1 mm	0.1 mm	0.1 mm	0.1 mm

The same evaporator used for both MT and LT. All heat exchangers were configured in a cross-counterflow arrangement. Table 5 shows the total air- and refrigerant-side heat transfer areas (A_{air} and A_{ref}) of the different heat exchangers. It can be seen that the RTPF heat exchangers has about 10 – 20 % higher air-side area compared to the MC heat exchangers. Note that greater air-side area will generally translate to a heavier and more expensive heat exchanger. The RTPF heat exchangers also have about 50 – 60 % lower refrigerant-side area, which will hurt their performance compared to MC heat exchangers.

Table 5: Comparison of heat transfer areas of MC heat exchangers and RTPF heat exchangers.

	MC Heat Exchanger		**RTPF Heat Exchanger**	
	$\mathbf{A_{air}}$	$\mathbf{A_{ref}}$	$\mathbf{A_{air}}$	$\mathbf{A_{ref}}$
Evaporators	6.11 m^2	0.75 m^2	7.42 m^2	0.37 m^2
Gas Cooler	18.29 m^2	1.78 m^2	20.60 m^2	0.69 m^2

Table 6 presents a comparison of the conventional direct expansion booster cycle with intercooler (no IHX and no ejector) with all MC heat exchangers and with all RTPF heat exchangers. It can be seen that the system COP increases by a very noticeable 6.2 % when all MC heat exchangers are used instead of all RTPF heat exchangers. The performance of the heat exchangers can be measured by their overall heat transfer coefficient-area product (UA), with higher UA meaning better heat exchanger performance. It can be seen that the UA of the LT evaporator is 9.8 % higher with the MC heat exchanger; however, this yields only 0.3 bar higher evaporator pressure and contributes only 0.5 percentage points to the observed increase in COP. Furthermore, essentially no difference in UA is observed between MC and RTPF heat exchangers for the MT evaporator. This indicates that using MC heat exchangers as evaporators is not so critical for system performance. On the other hand, using an MC heat exchanger as the gas cooler yields 32.4 % higher UA and 1.0 K lower gas cooler refrigerant outlet temperature, contributing 5.6 percentage points to the observed increase in COP, indicating the importance of using a microchannel gas cooler in this application.

Table 6: Comparison of heat exchanger and conventional direct expansion booster cycle performance using MC heat exchangers and RTPF heat exchangers.

	MC Heat Exchangers	**RTPF Heat Exchangers**
$\boldsymbol{COP}$	0.567	0.534
$\boldsymbol{UA_{evap,LT}}$	0.640 kW K^{-1}	0.583 kW K^{-1}
$\boldsymbol{UA_{evap,MT}}$	0.677 kW K^{-1}	0.678 kW K^{-1}
$\boldsymbol{UA_{gc}}$	1.013 kW K^{-1}	0.765 kW K^{-1}
$\boldsymbol{P_{evap.LT}}$	15.9 bar	15.6 bar
$\boldsymbol{P_{evap.HT}}$	30.5 bar	30.5 bar
$\boldsymbol{\Delta T_{gc}}$	0.3 K	1.3 K

3.3 Improvement through Internal Heat Exchange and Expansion Work Recovery

Table 7 compares the COP, power ($\dot{W}_c$) of each compressor, heat transfer in the IHX ($\dot{Q}_{IHX}$), and power recovered by the ejector ($\dot{W}_{ejec}$) for the conventional direct expansion (DX), the cycle with the ejector in place of the high-pressure control valve (Ejec), and the addition of an IHX to each cycle. Recall that capacity in each evaporator was the same for all cycles. For reference, the 'Ejec + IHX' cycle is the cycle shown in Figure 2. It can be seen that replacing the conventional expansion valve with the ejector can yield up to 27.3 % COP improvement. This very favorable COP improvement is due to the very high ambient temperature that the system operates at. A similar COP improvement of 35.3 % can be achieved by adding an IHX to the cycle. The COP of the cycle with both IHX and ejector is the highest, offering a 68.6 % % improvement in COP compared to the DX cycle. These results demonstrate the importance of the

IHX and the ejector in achieving a sufficiently high COP, especially at the very high design ambient temperature required for this application.

Table 7: Comparison of cycle performance for cycles with and without IHX and ejector.

Cycle	COP	$\dot{W}_{c,LT}$	$\dot{W}_{c,HTLS}$	$\dot{W}_{c,HTHS}$	$\dot{Q}_{IHX}$	$\dot{W}_{ejec}$
DX	0.567	1.14 kW	4.62 kW	3.49 kW	-	-
DX + IHX	0.767	0.90 kW	3.65 kW	2.36 kW	2.05 kW	-
Ejec	0.722	0.57 kW	2.56 kW	4.51 kW	-	0.49 kW
Ejec + IHX	0.956	0.52 kW	2.73 kW	2.20 kW	3.00 kW	0.21 kW

4. CONCLUSIONS

This paper has presented an analysis of the design and performance of a multi-temperature, transcritical CO_2 refrigerated container system. The results of the numerical investigation have shown that the use of a microchannel gas cooler and the addition of an IHX and an ejector to the cycle are all important for enhancing cycle performance and achieving reasonable COP (up to 0.96 at very high ambient temperature of 57°C). It has been seen that a microchannel gas cooler improves COP by 5.6 % compared to a round-tube gas cooler. It has also been seen that an IHX can improve COP by up to 35.3 % and an ejector can improve COP by up to 27.3 %, while the combination of the two improves COP by up to 68.6 %.

In practical terms, it seems that a microchannel gas cooler would be a necessary enhancement of the system, though given the disadvantages of microchannel heat exchangers when used as evaporators and the very little cycle performance improvement they offer, it does not make sense to proceed with using microchannel evaporators. Challenges still remain in terms of achieving reasonable efficiency with an adjustable and obtaining an efficient, compact liquid-vapor separator in a small-scale refrigeration application, such as the refrigerated container in this project. Even with the addition of an IHX and an ejector, the COP is still slightly lower than the target COP of 1.0, meaning that even higher compressor, evaporator, or ejector efficiency will still be necessary in order to achieve the target COP (as the gas cooler and IHX are already achieving very high performance). It should also be noted that it is currently difficult to find multi-stage, transcritical CO_2 compressors suitable for the capacity and pressure range of this application. This would further decrease the COP compared to the target value. Nonetheless, the results show that through proper design of each component, a reasonable efficiency of this transcritical CO_2 refrigerated container unit can still be achieved even at the very high ambient condition of this project.

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