

THE PERFORMANCE OF A REVERSIBLE R744 VAPOR COMPRESSION SYSTEM WITH
AN INTERNAL HEAT EXCHANGER IN CONTINUOUS FROSTING/ DEFROSTING
CYCLES

BY

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DISSERTATION

Submitted in partial fulfillment of the requirements
for the degree of Doctor of Philosophy in Mechanical Engineering
in the Graduate College of the
University of Illinois Urbana-Champaign, 2022

Urbana, Illinois

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ABSTRACT

To study the performance of the reversible R744 heat pump (HP) system with components designed and optimized for air-conditioning (AC) mode only, a comprehensive study was conducted to examine the component- and system-level performance using a full reversible R744 heat pump system in a wide range of practical operating conditions including frosting conditions.

The performance of an integrated internal heat exchanger (IHX) and accumulator (Acc) has been studied numerically and experimentally. First, the potential benefits of introducing an IHX into reversible R744 systems are discussed and analyzed. Overall, the results of cycle analysis underestimate the efficiency improvement of introducing an IHX into an R744 vapor compression system, and the deviation between the theoretical and experimental results increases with the refrigerant temperature at the gas cooler exit. Experimental data show a more than doubled efficiency improvement due to IHX compared to the cycle analysis results when the refrigerant temperature at the gas cooler exit for both basic and IHX systems is 41.2°C. Because simplified theoretical analysis neglects the change of the compressor efficiencies and the refrigerant distribution in the evaporator. Also, a 1-D EES model of the integrated IHX/Acc was developed and it predicts the heat transfer rate and pressure drop of the low-pressure side within $\pm 10\%$, by considering the effects of oil circulating rate on heat transfer and pressure drop. To provide more detailed information on the behavior of the accumulator, a 3-D model of the accumulator was developed. The flow characteristics of the two-phase CO₂ and oil mixture in the accumulator in a typical HP operating condition were studied, considering the effects of different sizes of the oil bleed hole.

For reversible air-source AC/HP systems, one of the major concerns in a cold climate is the frosting issue, so the performance of the reversible R744 HP system in frosting conditions has

been studied, considering the environmental factors, the defrost-initiation criteria, and the orientation of the outdoor coil. The heating capacity, efficiency, frost accumulation in HP operation, as well as the defrosting capacity, efficiency, energy flow, and water drainage were studied comprehensively. The results show that with a fixed relative humidity of ambient air, the outdoor air temperature affects the capacity and efficiency of the HP system significantly, as well as the frost accumulating speed and the frosting time. When the outdoor air temperature drops from 0 to -10°C , the peak value of heating capacity reduces from 4.85 to 3.54 kW and HPF reduces from 1.61 to 1.47 in the first cycle. In the meanwhile, the frost accumulating speed decreases from 0.013 to 0.005 kg/min, and the frosting time increases from 32 to 108 minutes. However, the relative humidity of the outdoor air has a negligible impact on the capacities and efficiency of the HP system. Though it directly affects the humidity ratio and thus the frost accumulating speed and the frosting time, which increases significantly from 32 to 84 minutes when the outdoor air temperature is 0°C and the relative humidity decreases from 90% to 80%.

The heating capacity and the efficiency of the HP system drop only by less than 9% during the frosting cycles with 5 times the initial air-side pressure drop as the defrost-initiation criterion in all experimental conditions. But heating capacity and HPF drop by about 30% and 15% during the frosting cycles with 10 times the initial air-side pressure drop as the defrost-initiation criterion in all experimental conditions. Considering the operating-time-averaged performance, the 30% and 15% performance reduction can be considered mild, which indicates a longer operating time in HP mode, a lower defrosting frequency, fewer impacts on passengers' comfort, and potentially a higher periodic performance. Although the optimal defrost-initiation and termination criteria are still open to discussion and require further investigation.

In addition, the results of the defrosting energy flow study show that it is more time- and energy-efficiently to apply 10 times the initial outdoor DP_{ea} as the defrost-initiation criterion than 5 times at a cost of more temperature fluctuations of the indoor air during the defrosting cycles. However, it is possible to defrost the outdoor coil when the vehicle is parking in the parking lot or charging center and minimize the negative effects on passengers' comfort, considering the 68-minute-long operating time. Also, the vertical orientation of the outdoor coil reduces the defrosting time and energy by increasing the surface temperature of the outdoor evaporator in HP mode and improving the water drainage during the defrosting. The conclusion of this dissertation can be applied by the researchers and engineers working on designing and optimizing automotive heat pumps.

ACKNOWLEDGMENTS

I would like to express my deep gratitude to my advisor, Professor Pega Hrnjak, for his support and guidance throughout my journey of pursuing the Ph.D. degree. His passion for science and engineering inspires me to grow into a researcher and engineer. Without his help and guidance, I could not accomplish what I have done in the last five years. I would also love to thank my dissertation committee: Professor Anthony Jacobi, Professor Stefan Elbel, and Professor Tomasz Kozlowski, for their valuable input on improving my research and skills.

I am also deeply grateful for all the help and support from the Air-Conditioning and Refrigeration Center member companies at the University of Illinois at Urbana-Champaign. Their generous support makes this dissertation possible. Special thanks to Dr. Jing He from Ford (previously, now Rivian), Dr. Li, and Lisa Huang from CTS, for their generous help and suggestions.

I also like to thank my ACRC colleagues, especially Bruno, Jiu, Sugun, Hongliang, Wenzhe, and many others for their numerous help in the lab, invaluable input, and suggestions for my research. In addition, I want to thank the “Hot Pot” group members for their support and encouragement. I love the tea break, dinner, chat, and workout time we spent together! I need to thank my cat, potato, and my partners on the Focusmate website for their help during the writing process!

I would like to thank my parents, my brother, my aunt, and my dearest grandma for their unconditional love and support. Without their support and encouragement, I would not be who I am. Finally, I wish to thank my boyfriend, Dr. Yue Yu, for his help, support, encouragement, and many many jokes in all those years!

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NOMENCLATURE

Symbols:

A	Area	$[m^2]$
C	Coefficient	$[-]$
cp	Specific heat	$[kJ/(kg\cdot K)]$
d	Diameter	$[m]$
dP	Pressure drop	$[Pa \text{ or } kPa]$
E	Energy	$[kJ]$
f	Fanning friction factor	$[-]$
f_D	Darcy friction factor $=4f$	$[-]$
g	Gravitational acceleration	$[m/s^2]$
G	Mass flux	$[kg/(m^2\cdot s)]$
h	Enthalpy	$[kJ/kg]$
htc	Heat transfer coefficient	$[W/(m^2\cdot K)]$
M	Mass	$[kg]$
$\dot{m}$	Mass flow rate	$[kg/s]$
Nu	Nusselt number	$[-]$
P	pressure	$[Pa \text{ or } kPa \text{ or } MPa]$
Pr	Prandtl number	$[-]$
Q	Capacity	$[kW]$
Re	Reynolds number	$[-]$
RH	Relative humidity	$[\%]$
SH	Superheat	$[^{\circ}C]$

T	Temperature	[°C]
t	Time	[min]
th	Thickness	[mm or m]
U	Flow rate	[kg/min]
V	Velocity	[m/s]
W	Power	[kW]
x	Quality	[-]
X	Lockhart-Martinelli parameter	[-]

Abbreviations:

AC	Air-conditioning
Acc	Accumulator
AFR	Air flow rate
ASHP	Air-source heat pump
ASHARE	The American Society of Heating, Refrigerating and Air-Conditioning Engineers
CO ₂ / R744	Carbon dioxide
COP	Coefficient of performance
EEV/ EXV	Electronic expansion valve
EV	Electric vehicle
HP	Heat pump
HPF	Heating performance factor
htc	Heat transfer coefficient

ICE	Internal combustion engine
ID	Indoor
IHX	Internal heat exchanger
LOF	Liquid overfeeding
MCHX	Microchannel heat exchanger
OCR	Oil circulating rate
OD	Outdoor
PF ²	Parallel flow plate fin
PFSF	Parallel flow serpentine fin
PID	Proportional-Integral-Derivative
RTPF	Round tube plate fin
SC	subcooling
SH	superheat
TES	Thermal Energy Storage
VFD	Variable frequency drive

Subscripts:

<i>1-ph</i>	Single phase
<i>a/ air</i>	Air-side
<i>acc</i>	Acceleration
<i>Acc</i>	Accumulator
<i>basic</i>	Basic cycle/ system (without IHX)
<i>BH</i>	Bleed hole (of the J-tube inside the accumulator)

<i>c</i>	Condenser/ gas cooler
<i>cham</i>	Chamber-side
<i>coil</i>	Coil
<i>collected</i>	Collected (meltwater from the outdoor coil)
<i>cooling</i>	Cooling
<i>COP,max</i>	COP-maximizing
<i>cp/ comp</i>	Compressor
<i>defrosting</i>	Defrosting
<i>drained</i>	Drained (meltwater from the outdoor coil)
<i>dry</i>	Dry (air)
<i>dp</i>	Dew point
<i>e</i>	Evaporator
<i>exp</i>	Experimental results
<i>final</i>	Final stage
<i>frost</i>	Frost (properties)
<i>frosting</i>	Frosting
<i>fusion</i>	Fusion heat of the frost
<i>g</i>	Glycol
<i>heating</i>	Heating
<i>high</i>	High-pressure side
<i>HPF,max</i>	HPF-maximizing
<i>ini</i>	Initial stage
<i>i</i>	Inlet

<i>ID</i>	indoor
<i>IHX</i>	Internal heat exchanger
<i>J-tube</i>	J-tube inside the accumulator
<i>l</i>	Low-pressure side
<i>liq</i>	Liquid
<i>loss</i>	Loss to the ambient
<i>max</i>	Maximum
<i>min</i>	Minimum
<i>model</i>	Model results
<i>n</i>	(Air) nozzle
<i>o</i>	Outlet
<i>OD</i>	Outdoor
<i>oi</i>	Oil inlet
<i>oo</i>	Oil outlet
<i>r/ ref</i>	Refrigerant-side
<i>retained</i>	Retained (meltwater on the outdoor coil)
<i>suc</i>	Data at the suction point of the compressor
<i>water</i>	water
<i>wf</i>	Water film
<i>vap</i>	Vapor
<i>vaporized</i>	Vaporized (meltwater on the outdoor coil)
<i>x</i>	Expansion device

Greek:

α	Local void fraction	[-]
β	Orifice area ratio	[-]
ε	Relative roughness	[-]
η	effectiveness	[-]
ρ	Density	[kg/m ³]
μ	Viscosity	[Pa-s]
ω	Humidity ratio	[kg/kg _{dryair}]

CHAPTER 1: INTRODUCTION

1.1 Background and Motivation

Automobile Air-Conditioning (AC) systems have been widely applied since the 1980s. However, most of them are used for cooling only on internal combustion engine vehicles. During the past decades, Electric Vehicles (EVs) have become an alternative way to reduce air pollution and limit climate change, and more countries, states, and cities are announcing bans on internal combustion engines. Despite the rapid growth of the EV market, EVs still face the challenge of range anxiety. The situation gets worse in the cold weather since a majority of economical EVs are using low-efficiency electric heaters for providing the heat to the cabin air.

Reversing the operation of the AC system and using the Heat Pump (HP) function to warm the cabin air can improve the heating efficiency and mitigate the problem. The reversible R744 AC/ HP system has advantages like high heating efficiency, high-temperature lift, less performance drop in low-ambient-temperature conditions, and environmental protection compared to HP systems using conventional refrigerants, like R134a. However, most of the commercially available components are still designed for AC operation only. For example, the geometry of the commercially available outdoor heat exchanger is designed for the gas-cooling function only since it only works as the gas cooler and releases heat to the outdoor air in AC mode. But in the HP mode, the outdoor heat exchanger is supposed to work as an evaporator and absorb heat from the ambient air. In addition, the sizes of the accumulator, the J-tube inside it, and the oil bleed hole were designed for the operating condition of the system in AC mode. However, the performance of those components in the HP mode is not clear: How does the outdoor heat exchanger work as an evaporator, especially considering the frosting issues and water drainage? Can the existing accumulator hold the extra refrigerant when the system shifts from AC to HP mode? Is the Internal Heat Exchanger (IHX) beneficial for R744 systems in HP mode? Therefore, this thesis aims to study the performance of a reversible R744 AC/ HP system

with commercially available components in a wide range of practical operating conditions including frosting conditions.

1.2 Structure of Dissertation

Chapter 1 of this dissertation has provided the background information and the motivation of this research, as well as the structure of the dissertation. Chapter 2 reviews the relevant literature on the topics of 1) the role of IHX in R744 vapor compression systems; 2) the frosting related research, including fundamental studies, the simple geometries, the heat exchanger level, and the system level; 3) the defrosting related research, including passive and active defrosting methods, and frosting detection methods. Chapter 3 reports the performance of a reversible R744 heat pump system with and without IHX in both AC and HP modes in various dry conditions. Chapter 4 presents the numerical investigation of the benefits and behavior of the accumulator and IHX in the reversible R744 AC/ HP system in dry conditions, considering the effects of operating conditions and geometry. After obtaining the performance in dry conditions, chapter 5 shows the heating performance of the reversible R744 heat pump system in continuous frosting cycles, including the impacts of the environmental factors, the defrosting-initiation criteria, and the orientation of the outdoor heat exchanger. Chapter 6 presents the defrosting performance including water drainage of the reversible R744 heat pump system and the energy flow during continuous frosting/ defrosting cycles, including the impacts of the defrosting-initiation criteria and the orientation of the outdoor heat exchanger. Also, the distribution of the water retained on the coil is firstly reported in the open literature. In addition, the system architecture, the facility, experimental procedures, measurements, and data reduction of both dry and frosting/ defrosting experiments are described in chapter 3, 5, and 6, correspondingly. Finally, chapter 7 summarizes the key conclusions and contributions of this dissertation, as well as the recommended future work.

Appendix A presents the uncertainty analysis. Appendix B shows the local and overall images of the vertically and horizontally oriented outdoor heat exchanger during continuous frosting/defrosting cycles in various conditions.

CHAPTER 2: LITERATURE REVIEW

This chapter reviews the relevant literature on the topics of 1) the role of IHX in R744 vapor compression systems; 2) the frosting related research, including fundamental studies, the simple geometries, the heat exchanger level, and the system level; 3) the defrosting related research, including passive and active defrosting methods, and frosting detection methods. The experimental studies using R744 as refrigerant and microchannel heat exchangers are emphasized due to the similarity to our setup.

2.1 Role of IHX in R744 vapor compression systems

Internal Heat Exchangers (IHXs), also known as suction/liquid line heat exchangers, are widely applied in vapor refrigeration systems to improve system efficiency. The essence is to subcool the liquid refrigerant leaving the condenser/ gas cooler using the low-pressure low-temperature refrigerant that exits the evaporator. The benefits are increasing the specific cooling capacity of the refrigerant and preventing the liquid refrigerant from entering the compressor. However, the discharge temperature increases as the superheat at the suction point of the compressor increases, and the compressor efficiency might drop if the lubricant performance degrades due to high temperature. Overall, the potential improvement of Coefficient of Performance (COP) due to the installation of IHX depends on the tradeoff between the increase of specific cooling capacity and the increase of specific compressor work.

For different industries, engineers applied different kinds of IHX due to cost or space limitations. For domestic refrigerators and freezers, a simple expansion device- a capillary tube soldered on the external surface of the suction line, is used as a counterflow IHX. For automobile AC systems, the liquid line is commonly designed as a helix tube and immersed inside the liquid refrigerant in the accumulator for compactness. In residential and commercial systems, larger concentric-tube heat exchangers and brazed plate heat exchangers are also widely used as IHXs.

2.1.1 Theoretical studies

The typical first assessment of whether the COP increases or not is done based on the cycle analysis for the working fluid and operating conditions. The major factors are the thermodynamic and transport properties of the fluid. Several criteria of whether IHX is beneficial to the vapor compression cycles using different refrigerants have been proposed by many researchers. **Table 2.1** lists the authors, equations, and the key parameters they used.

Domanski et al. (1994) theoretically evaluated the performance effects of installing IHXs for 29 different refrigerants and four charts as examples. The results show the benefits of IHX depend on a combination of operating conditions and fluid properties with specific heat capacity being the most influential property. However, the theoretical analysis oversimplifies the cycle by using assumptions like ideal gas behaviors, isentropic compression, no-pressure-drop infinite heat exchangers, and no-pressure drop 100%-effectiveness IHX. The simulation was conducted at a reduced saturated evaporating temperature of 0.65 and a reduced saturated condensing temperature of 0.82, so the absolute temperatures in the analysis are different for all refrigerants. In addition, the authors developed the method and provided four charts of different refrigerants for evaluation, but a simplified equation or correlation is desired for general applications. Therefore, many researchers continue to explore a simple and general criterion for evaluating the potential benefits of installing IHX using only refrigerant thermodynamic properties and operating temperatures.

Apra et al. (1999) presented a simplified criterion and two application charts for R502 and R32 based on the thermodynamic analysis. The results show that IHX is only advantageous for R502. However, the authors also applied ideal gas assumptions, no-pressure-drop heat exchangers, and especially, constant isentropic efficiency for both basic and IHX cycles. In real applications, the isentropic efficiency is affected by the superheat and pressure at the suction point, so the assumptions might lead to some deviations.

Klein et al. (2000) proposed a new dimensionless group that correlates performance impacts related to IHX for eleven refrigerants considering the effects of pressure drops through the IHX. The authors conducted a simulation over a range of evaporating temperature between -40 and 10°C and condensing temperature between 10 and 60°C. The results were presented as charts (**Figure 2.1**) and correlated as a general equation to include the effects of IHX effectiveness, refrigerant properties, and the temperature lift. The paper concluded that IHXs are most useful at high temperature lifts $L = T_{cond} - T_{evap}$ and for refrigerants having a relatively small value of $D = dh_{vap}/(c_{p,l}T_c)$, and the benefits of IHX are reduced due to the pressure drop in it.

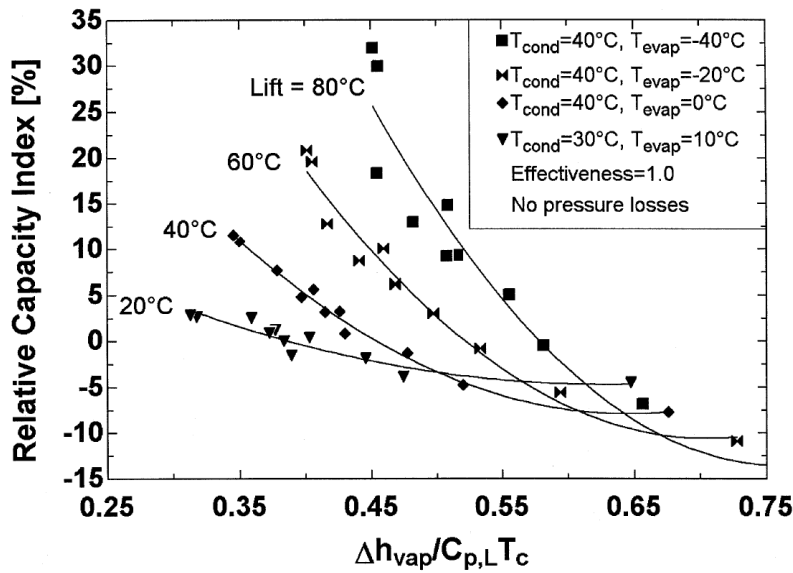


Figure 2.1: Benefits of IHX as a function of $D = dh_{vap}/(c_{p,l}T_c)$ for various operating conditions (Klein et al., 2000)

Mastrullo et al. (2007) proposed another parameter and developed a graphical method to predict the behavior of installing IHX, changing the operating conditions, refrigerants, and IHX size. In this study, 19 different refrigerants were studied with the evaporating temperature varying from -40 to 10°C, and condensing temperature changing from 25 to 50°C. The author mentioned the key property is the specific heat capacity of vapor, though latent heat, liquid specific heat, evaporating temperature, and the temperature difference between evaporating and condensing temperature also affect the potential benefit of installing IHX. They concluded that

refrigerants with a low COP and high value of $c_{p,v}$ are more positively influenced by introducing IHX. However, the authors also used assumptions including ideal gas, isentropic compression, and infinite IHX in the analysis.

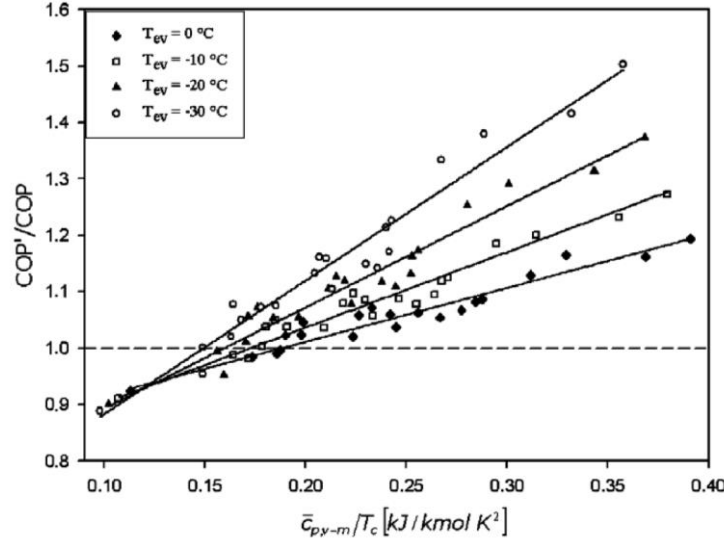


Figure 2.2: Benefits of IHX as a function of $c_{p,v-m}/T_c$ for different evaporating temperatures (Mastrullo et al., 2007)

Hermes (2013) presented a more practical thermodynamic evaluation of IHX by assuming the same cooling capacity for both basic and IHX cycles, though all previous studies assume the same evaporating pressure for both cycles. The analysis was conducted with two different evaporating temperatures ($-25/7^{\circ}\text{C}$), a fixed condensing temperature (40°C), and six different refrigerants. The author concluded that there exists an evaporating pressure that maximizes the compressor work for a fixed condensing temperature, cooling capacity, and compressor swept volume rate. However, the author used some assumptions like fixed swept volume rate and volumetric efficiency of the compressor, and constant condensing pressure in the analysis, which might not be true for some applications.

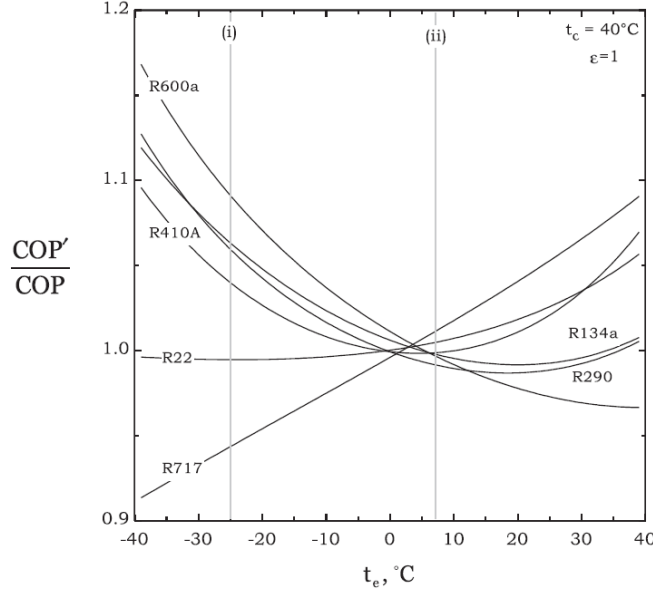


Figure 2.3: Benefits of a 100%-effective IHX for various refrigerants and evaporating temperature with constrained cooling capacity (Hermes, 2013)

Table 2.1: Theoretical criteria to evaluate the performance effect of the IHX installation

Authors	Equations	Parameters
Domanski et al. (1994)	$\frac{1}{\frac{h_{fg,e}}{c_{p,v}} - \frac{(T_c - T_e)c_{p,l}}{c_{p,v}}} > B_v$	T_c is the condensing temperature of both basic and IHX cycles. T_e is the evaporating temperature of both basic and IHX cycles. $h_{fg,e}$ is the latent heat of refrigerant at T_e. $c_{p,v}$ is the average specific heat of vapor refrigerant in the low-pressure side of IHX. $c_{p,l}$ is the average specific heat of liquid refrigerant between T_e and T_c. B_v is the average coefficient of thermal expansion in the low-pressure side of IHX.
Apra et al. (1999)	$c_p T_1 > h_1 - h_4$	c_p is the average specific heat of vapor refrigerant in the low-pressure side of IHX. T_1 is the suction temperature of the basic cycle.

Table 2.1: Theoretical criteria to evaluate the performance effect of the IHX installation (cont.)

		h_1 is the refrigerant enthalpy at the suction point of the basic cycle. h_4 is the refrigerant enthalpy at the evaporator inlet of the basic cycle.
Klein et al. (2000)	The fitted curve from simulation data: $\frac{RCI}{\varepsilon} = -3.0468 + 19.3484D - 19.091D^2 + 1.2094L + 0.02101L^2 - 5.9980DL - 0.02797DL^2 + 5.52865D^2L$	RCI is the relative capacity index: $RCI = \frac{Q_{ref,IHX} - Q_{ref,noIHX}}{Q_{ref,noIHX}} \times 100\%$ $Q_{ref,IHX}$ is the refrigeration capacity of the IHX cycle, and $Q_{ref,noIHX}$ is the refrigeration capacity of the basic cycle. ε is the effectiveness of IHX. D is a dimensionless number and equals $dh_{vap}/(c_{p,l}T_c)$, where dh_{vap} is the latent heat of refrigerant at the evaporating temperature, $c_{p,l}$ is the specific heat of saturated liquid refrigerant at the evaporating temperature, and T_c is the critical temperature of the refrigerant. L is the temperature lift and equals $T_{cond} - T_{evap}$.
Mastrullo et al. (2007)	$c_{p,v-m}/T_c > 0.2$	$c_{p,v-m}$ is the molar specific heat of refrigerant. T_c is the critical temperature of the refrigerant.
Hermes (2013)	$p_e \gg p_e^* \text{ or } p_e \ll p_e^*$	p_e is the evaporating pressure. p_e^* is the evaporating pressure which maximizes the specific compressor work for a fixed condensing pressure, yielding $\frac{c}{1+c}(k-1)\left(\frac{p_c}{p_e^*}\right)^{\frac{1}{k}} + \left(\frac{p_c}{p_e^*}\right)^{\frac{k-1}{k}} - k = 0$

In summary, several assumptions like the no-pressure drop HXs were used in all the above-referenced theoretical studies, and the general conclusion is that the refrigerant properties and operating temperatures are the dominating parameters for system performance. However, several other factors like the geometry of the Heat Exchangers (HXs), the efficiency of the compressor, and even the charge of the system also play important roles in determining the system performance. Therefore, experimental studies are required for more accurate quantitative analysis.

2.1.2 Experimental studies

A more detailed analysis considers the pressure drop in the components, the size of the heat exchangers, operating conditions (indoor/ outdoor airflow rates, temperatures, and humidity), and the compressor efficiency, which varies between different applications and different modes. For example, an indoor evaporator has one-third the size of an outdoor condenser/ gas cooler for typical residential and automobile air-conditioning systems. Also, the airflow rate through the outdoor coil is higher due to less concern for noise, and this increases the effectiveness of the outdoor coil. All the aforementioned reasons lead to the differences between the operations of vapor compression systems for different applications in AC and/ or HP modes, as well as the different roles of the IHX in the two modes.

a. Domestic refrigeration systems

On the one hand, the design of the components and the system varies for different applications due to costs and space limitations. For domestic refrigerators and freezers, the simple and low-cost expansion device- a capillary tube- is soldered on the external surface of the suction line and works as a counter-flow IHX. In the early stage, researchers focus on investigating the flow characteristics of different refrigerants to determine the length of the capillary IHX for small refrigeration systems. Staebler (1981) presented capacity balance to calculate the length of diabatic

capillary tube for R12 and R22. Bittle (1994) evaluated the capillary IHX performance with R134a and R152a experimentally, considering up to ten different design variables. The author also developed performance prediction equations for both refrigerants, which can be used as design tools for engineers. Melo et al. (2002) conducted comprehensive experiments with R600a for a range of geometries of capillary IHX and operating conditions typical of domestic refrigerators and freezers. Empirical correlations were developed to estimate the refrigerant mass flow rate through the capillary tube and the suction temperature of the compressor.

Later, different investigators studied the effects of coiling on the performance of capillary IHXs. Kuehl and Goldschmidt (1990) investigated the flow characteristics of R22 through adiabatic capillary tubes of straight and coiled arrangements for a typical domestic AC system experimentally. The data showed that the mass flow rate of the coiled capillary tube is 5% lower than it is for the straight tube. Kim et al. (2002) presented experimental results of flow characteristics of R22, R407C, and R410A in adiabatic capillary tubes with different geometric parameters and operating conditions. Dimensionless correlations were developed to predict the mass flow rate through the capillary tube within $\pm 12\%$ for test conditions. The results also showed a 5 to 16% mass flow rate reduction by introducing coiled capillary tubes. Khan et al. (2009) conducted experiments to study the flow through diabatic spiral capillary tubes, considering the geometric and arrangement of the capillary IHX. It was concluded that the refrigerant mass flow rate is also affected by the suction-line inlet superheat and heat exchange length, in addition to the capillary tube diameter, capillary tube length, coil pitch, and capillary inlet subcooling.

b. Automobile air-conditioning systems

For automobile air-conditioning systems, the compactness of components is more important than the domestic refrigeration systems due to the limited space. In this case, the Accumulator Heat Exchanger (AHX), or known as the integrated Accumulator and IHX (Acc/IHX), has been investigated by many researchers and engineers. The key benefits of AHX are flooding the evaporator, and as a consequence, increasing the use of the heat transfer area, improving the cooling capacity of the evaporator, increasing the surface temperature of the evaporator, and delaying the frosting process on the evaporator in the heat pump mode. However, introducing the integrated accumulator and IHX adds extra pressure drop and superheat at the suction point of the compressor, where the specific volume of vapor refrigerant increases and therefore the compressor work increase. Therefore, the conclusions of whether introducing the AHX benefits one specific refrigerant system, and the percentage of the improvement varies in different research papers.

Mei et al. (1994) developed an R134a Liquid Over-Feeding (LOF) mobile air conditioning system that applied an accumulator-heat exchanger. The design improved the utilization of the heat transfer area of the evaporator and reduced the compressor power per unit mass flow due to the lower compression ratio. The results indicated an increase of 20% in system cooling capacity and COP at a compressor speed of 2020 RPM. For different control variables in this study, the refrigerant charge of each system was optimized, but the opening of the metering valve was not changed or optimized. Also, the design and the geometric of the AHX used in this study were not provided. In 1997, the author proposed a liquid over-feeding refrigeration system with an integrated accumulator- expander heat exchanger in a patent (Mei and Chen, 1997). In the patent, a capillary tube with an internal diameter in the range of 0.03125 to 0.0625 inches was designed

in the form of a coil and immersed in the pool of liquid refrigerant inside the accumulator. The potential improvement of the cooling capacity was about 20 to 25% when the evaporating temperature was about 4.4 to 15.6°C.

Meyer and Wood (2001) were the first authors to describe a mathematical model of an AHX publicly. They validated the model with an R22 air-conditioning system, and the model predicts the performance of the system within -16% to 13%. The observation agrees with Mei et al. (1994) that evaporating pressure increases and compression ratio decreases after introducing AHX. As a result, the mass flow rate increases by 4%, cooling capacity increases by 6.5%, and COP increases by 7.5%. However, the refrigerant charge was fixed to be 0.83 kg for both AHX and basic systems in this study, and the dimension of the capillary tubes was not mentioned.

Kang et al. (2007) studied the effects of an AHX on the performance of an R22 refrigeration system experimentally. The data showed that the AHX system had a higher refrigerant flow rate than the basic system at a constant opening of the Electronic Expansion Valve (EXV) due to the higher subcooling. At the same EXV opening of 50%, the cooling capacity and COP of the AHX system were higher than the basic system by 7.5% and 3.2%, respectively. However, if both systems were well controlled and optimized by EXV, the cooling capacity and COP of the AHX system were only 1.7% and 1.3% higher than the basic system. In this study, the charging of both systems was not discussed and EXV opening, instead of the superheat (SH) or subcooling (SC), was used as the key control variable, which might be improved in future studies. The authors (2011) also investigated the heat transfer characteristics of AHX with different geometries using R22, R134a, and R410A under various operating conditions. The inner volume and ratio of the diameter to the height of the accumulator, and the diameter, length, and bending diameter of the

inner heat exchanger were studied. A dimensionless correlation was developed to predict the subcooling difference across the AHX. In this study, the authors focused on the heat transfer of the AHX, instead of the effects of the AHX on the refrigerant system.

c. Transcritical R744 systems

Due to environmental concerns, alternative refrigerants such as CO₂/ R744 have been widely studied before and after the turn of the century. Because of the thermodynamic and transport properties of CO₂, quite different design philosophies and control strategies should be applied for transcritical CO₂ systems compared to previously discussed conventional AC systems.

Lorentzen and Pettersen (1993) presented a transcritical CO₂ automobile AC system with an accumulator and an IHX, and the data showed the COP of the CO₂ system is slightly better than a standard R12 system. The improvement in COP by internal heat exchange is mainly due to the reduction of the throttling loss when subcooled liquid CO₂ enters the expansion device. However, this is at a cost of the increased average temperature of heat rejection and irreversibility in the gas cooler.

Boewe et al. (1999) conducted experiments with a transcritical CO₂ automobile AC system with an accumulator to study the role of IHX in air-conditioning mode. Four configurations were investigated: the basic cycle and three coaxial IHX having lengths of 1.0, 1.5, and 2.0 m. The results showed that both the capacity and COP were increased by up to 25%, and the benefits of IHX are more significant in high ambient temperature conditions. The IHX reduces the capacity- and efficiency-maximizing discharge pressure and brings them together. This helps to increase the cooling capacity when optimizing the efficiency of the AC system. The authors also developed a simulation model to predict the performance of the coaxial IHX and proposed a new design of the

microchannel-sandwich IHX, which reduces the material cost by 50% while increasing effectiveness by 10% (Boewe et al., 2001).

Kim et al. (2005) investigated the performance of a transcritical CO₂ water heater system with a coaxial IHX numerically and validated experimentally. The system model predicted the parameters within 4% of absolute average deviations. The effects of IHX length on refrigerant flow rate, compressor power, heating capacity, and efficiency were tested at different operating conditions. The results showed that the efficiency is slightly improved though the refrigerant flow rate and heating capacity are reduced as the length of IHX is increased (**Figure 2.4**). This is due to the trade-offs between the effectiveness and pressure drop in the IHX. Also, the efficiency of the system is slightly improved by introducing an IHX up to a specific discharge pressure, above which the trend is reversed.

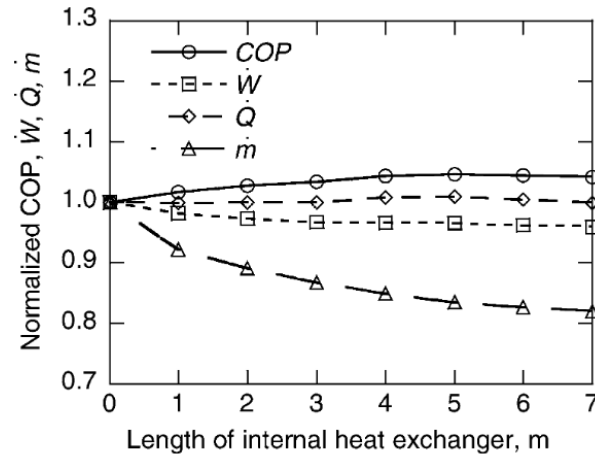


Figure 2.4: System performances to the length of the IHX (Kim et al., 2005)

Cho et al. (2007) conducted experiments to study the effects of refrigerant charge amount, compressor frequency, EXV opening, and length of the coaxial IHX on the cooling performance of a residential AC system with variable speed control. The data showed the efficiency of the basic cycle was optimized ($COP_{max} = 2.1$) at the discharge pressure of 9.2 MPa and normalized charge

of 0.282 (1.4 kg). The COP decreased as the compressor frequency increased. Also, the IHX reduced the optimal discharge pressure from 9.2 to 8.7 MPa and increased the cooling capacity and COP of the CO₂ system by 6.2 to 11.9% and 7.1 to 9.1%, respectively. Although the benefits stopped increasing when the length of IHX is above the critical value- 2.0 m. However, the refrigerant charge amount was not optimized for IHX cycles, and a fixed EXV opening instead of discharge pressure was used for control in this study. This simplified control is likely due to the application: for residential AC systems, the operating condition is relatively fixed compared to automobile AC systems.

Zhang et al. (2011) presented a thermodynamic cycle analysis and an experimental validation to study the effects of an IHX on transcritical CO₂ cycles and to find the suitable operating conditions for applying IHXs. The results showed that the COP is slightly reduced by an IHX in the subcritical mode. In the transcritical mode, a transition gas cooler exit temperature exists for a compressor discharge temperature, above which the IHX improves the cycle performance (**Figure 2.5**). However, simplified assumptions are used in the cycle analysis, like no pressure drop in the HXs, which might not be valid for different applications.

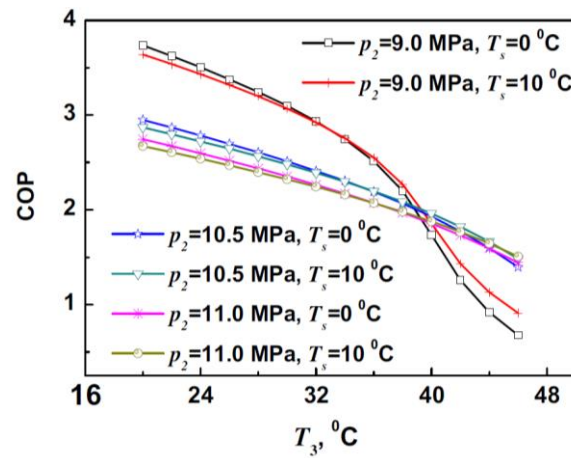


Figure 2.5: Simulation results of the COP of transcritical CO₂ cycle to the gas cooler exit temperature (T_3) at various discharge pressures (P_2) and various IHX superheat (T_s): $T_s = 0^\circ\text{C}$ represents a basic cycle, and $T_s = 10^\circ\text{C}$ represents an IHX cycle (F. Z. Zhang et al., 2011)

Cao et al. (2019) continued the study of the effects of IHX length under various temperature conditions using an air-source transcritical CO₂ water heater system. The experimental data showed that the IHX has a negligible effect on efficiency and reduces heating capacity by 2.5% when the ambient air temperature is 16°C, inlet water temperature is 17°C, and exit water temperature is 65°C. But IHX improves the efficiency by 6.4% and reduces heating capacity by 3.6% when the ambient temperature is 7 °C, inlet water temperature is 40°C, and exit water temperature is 70°C. It can be concluded that IHX benefits the system performance when the water inlet temperature is 40°C. The conclusion can be translated to and agrees with the finding of Zhang et al. (2011): IHX is beneficial when CO₂ gas cooler exit temperature is above a certain value at a fixed discharge pressure.

Although the effects of an IHX on the performance of a transcritical CO₂ refrigeration system have been studied by many researchers, the role of an IHX in a reversible CO₂ AC/ HP pump system has not been discussed in public literature. In chapter 3, the effects of an IHX in a reversible R744 AC/ HP system under various conditions are discussed. Meanwhile, the influence of the IHX on the optimal refrigerant charge amount, efficiency-maximizing control equation, and normalized heating performance are presented. The conclusion can be used to guide the design of reversible R744 air-conditioning systems for different applications.

2.2 Frosting research

As discussed in the background and motivation chapter, HP systems help to mitigate the range anxiety problem of EVs in cold weather. However, there are issues related to frosting, defrosting, and condensate removal for the outdoor evaporators of Air-Source Heat Pump (ASHP) systems. Because frost grows on the surface of the outdoor evaporators when the surface

temperature is lower than both the freezing point of water and the dew point temperature of the outdoor air during the HP operation.

2.2.1 Environmental factors

In the early stage, researchers studied the fundamental aspects of frosting- the different phases of frost growth, the properties of frost, and the effects of environmental conditions. Due to the purpose of this dissertation, it is more focused on the experimental study and the frosting in forced convection.

Hayashi et al. (1977) classified the frost formation types based on the structure of the frost in the surface temperature range of 0 to -25°C , the water concentration difference range of 0.002 to $0.010 \text{ kg/kg}_{\text{dry air}}$ and forced convection. The experiments were conducted using a flat plate and the airflow was parallel to the plate. The frost accumulation process was divided into three periods: the crystal growth period, the frost layer growth period, and the frost layer full growth periods

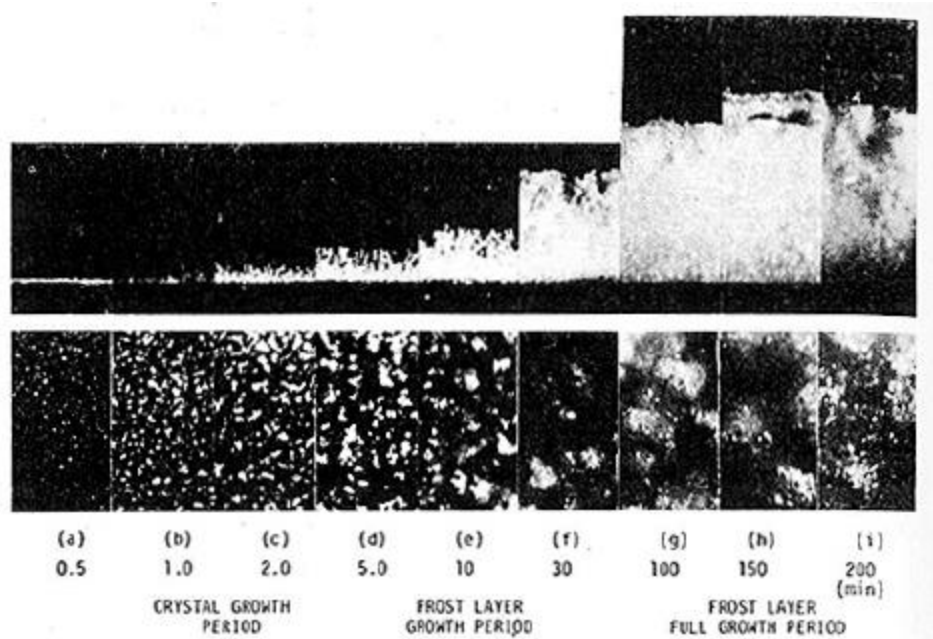


Figure 2.6: Frost formation process: $U_{\text{air}} = 4 \text{ m/s}$, $\omega_{\text{air}} = 0.0075 \text{ kg/kg}_{\text{dry air}}$, $T_s = -12.4^{\circ}\text{C}$, $\times 12$ (Hayashi et al., 1977)

(Figure 2.6). The density and effective thermal conductivity were studied and related to the frost formation types. The frost thermal conductivity was predicted using a 2D theoretical model composed of ice columns and ice-air composite parts. However, the author did not discuss the nucleation process, effects of the surface energy, and contact angle of the cold plate.

Tao et al. (1994) investigated the growth of frost on flat plates with four different materials in forced convection with Re number based on downstream location varies from 2840 to 5680. The authors also proposed two characteristic times, t_c and t_i , to show the transitions when all the condensed droplets changed from liquid to solid and when the frost layer became a uniform porous layer. But the surface roughness and the contact angle were not reported.

Na and Webb (2003) analyzed the nucleation process theoretically for the incipient frost formation and concluded that the air at the cold surface should be supersaturated for frost formation to start. Also, lower energy surfaces (high contact angle) require a higher supersaturation degree for nucleation, but surface roughness reduces the required supersaturation degree. The authors also validated their conclusions with experiments using five different kinds of surfaces.

Lee and Ro (2002) investigated the effects of environmental conditions on frost growth experimentally using vertical stainless-steel plates in the hydrodynamically and thermally developing zone. The control parameters are inlet air temperature (5 to 20°C), inlet air humidity (0.003 to 0.0055 kg/kg_{dry air}), inlet air velocity ($Re=$ 1000 to 3000), and cold plate temperature (-20 to -10°C). The data showed humidity and cold plate temperature are the dominant parameters affecting the frost growth: Higher humidity ratio and lower cold plate temperature resulted in a thicker frost layer, a greater amount of frost mass, and a less dense frost layer. But the density of the frost layer almost doubles as the air temperature increases from 5 to 20°C. The authors also

developed empirical equations to predict the mass accumulation, thickness, and density of the frost layer.

Yang and Lee (2004) conducted experiments to obtain the impact of various environmental parameters on the properties of frost formed on a cold plate. The range of the test conditions is the following: the air temperature of 5 to 15°C, air velocity of 1.0 to 2.5 m/s, absolute humidity of 0.0032 to 0.0085 kg/kg_{dry air}, and cold plate temperature of -35 to -15°C. The thickness, density, surface temperature, thermal conductivity, and average heat and mass transfer coefficient of the frost layer were correlated as functions of *Re* number, Fourier number, absolute humidity, and dimensionless temperature. However, the air inlet temperature for both studies is above 5°C and the relative humidity is lower than 80%, which might not be the most common weather in winter. Therefore, fundamental frosting studies under the conditions of an automobile or residential air-source heat pump with sub-freezing and close-to-freezing air temperatures are still limited.

2.2.2 Microscale surface features

In addition to the effects of environmental conditions, there is another mechanism affecting the frost properties and accumulation: the wettability of the substrate. Seki et al. (1985) conducted an analytical and experimental study of frost growth on a cooled plate exposed to forced convective humid air. Two different plates with contact angles to water droplets (110 and 43 degrees) were examined in the experiments. The effects of air velocity, humidity, and temperature of the air stream and the cold plate on the frost formation speed were investigated. The experimental conditions are the following: the air velocity of 1 m/s, the air inlet temperature of 10°C, the air humidity of 0.0025 to 0.0045 kg/kg_{dryair}, and the cold plate temperature of -11 to -3°C. The data showed the critical time to accumulate 0.5 mm of frost was longer on the surface with a larger contact angle, which can be converted to a slower frost growth.

Shin et al. (2003) investigated the effects of different surfaces with Dynamic Contact Angles (DCAs) of 23, 55, and 88° on frost formation. The test conditions are the following: the *Re* number of 9000, the air temperature of 12°C, the air humidity of 0.0042 kg/kg_{dryair}, and the cold plate temperature of -22°C. The results showed that the frost density and thermal conductivity increase with time. Also, the density and thermal conductivity are slightly higher for the surface with lower DCA during the first two hours, but the difference vanished afterward. The authors concluded that the effects of surface characteristics play important roles in the initial stage of frosting, but the environmental conditions dominate after two hours of frosting time. In addition, it was found that the frost layer close to the plate surface and the frost-air interface had the highest thermal conductivity, but the interior layer had a two or three times smaller thermal conductivity. This conclusion indicates that air might be trapped inside the frost layer during the fast-growing period of frost, and densification only happens on the interface.

Rahman and Jacobi (2012) investigated the water drainage after defrosting several micro-grooved brass surfaces and compared it with a bare flat baseline surface. The test conditions were controlled using a climate chamber: the air inlet temperature of -6°C, the relative humidity of 90%, and the cold plate temperature of -25°C. The micro-grooved surfaces drain up to 70% more condensate than does the baseline surface, and the sliding of the ice-water mixture along the grooves improves the defrosting efficiency. In addition, the drainage significantly increases with the width of the pillar, though it is less sensitive to the depth of the pillar. The authors also studied the effects of microgroove geometry on frost growth and properties in the early stages of a broad range of environmental conditions (Rahman and Jacobi, 2013). Micro-grooved samples with the deepest groove (122 µm) and widest pillar (187 µm) are found to have the lowest frost thickness and highest density. Then, the authors investigated the defrosting behavior of the micro-grooved

brass surfaces in continuous frosting/ defrosting cycles in a set of operating conditions (Rahman & Jacobi, 2014). The control variables include eight micro-grooved samples, a cold plate temperature of -18°C , air temperature of 20°C , relative humidity of 70%, and up to four frosting/ defrosting cycles. The microgroove texture can affect the frost properties even in the refrost cycles (**Figure 2.7**). The frost thickness and density increase in the refrost cycles and become repeatable in the 3rd cycle. However, the thickness of the frost is higher for the micro-grooved surfaces than it is for the baseline surface, and it is contrary to the density of the frost. This might be a disadvantage for applying the micro-grooved surface in the application because higher frost can block the airflow and reduce the capacity of the heat exchanger.

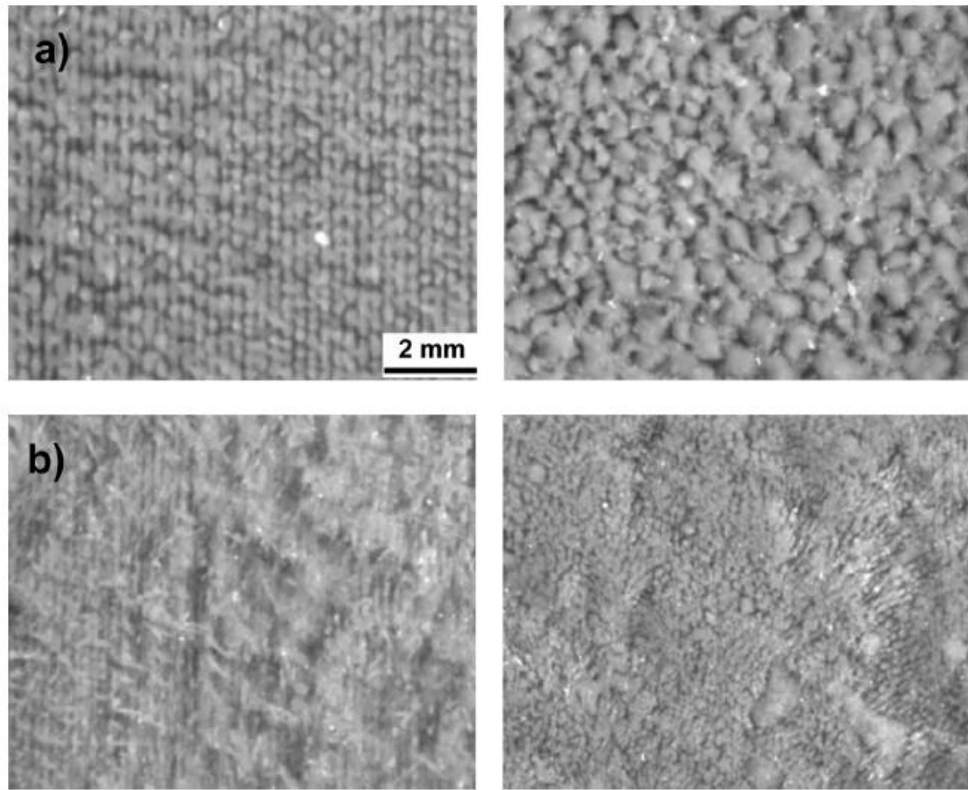


Figure 2.7: Frost structure in the a) 1st frost cycle and b) refrost cycle on micro-grooved sample (left) and bare surface (right) after 32 minute of frosting time: $T_{air} = 20^{\circ}\text{C}$, $RH_{air} = 70\%$, $T_s = -8^{\circ}\text{C}$ (Rahman & Jacobi, 2014)

Wu et al. (2017) further investigated the effects of grooved surface on the frosting and defrosting features using a flat aluminum plate, and aluminum plates with parallel and crossed grooves of 0.5 mm width (**Figure 2.8**) and 0.7 mm depth. The controlled operating conditions are the following: the cold plate temperature of -16/-10°C, the air inlet temperature of 2°C, the air humidity of 75/85%, and the air velocity of 0.92/1.16 m/s. The results showed that the parallel grooved surface drains the meltwater best and has the smallest retention ratio but the crossed grooved surface has the worst drainage performance. Also, a high retention ratio of the meltwater results in high frost thickness in the next frosting cycle. However, the author did not provide optimization of the groove geometry.

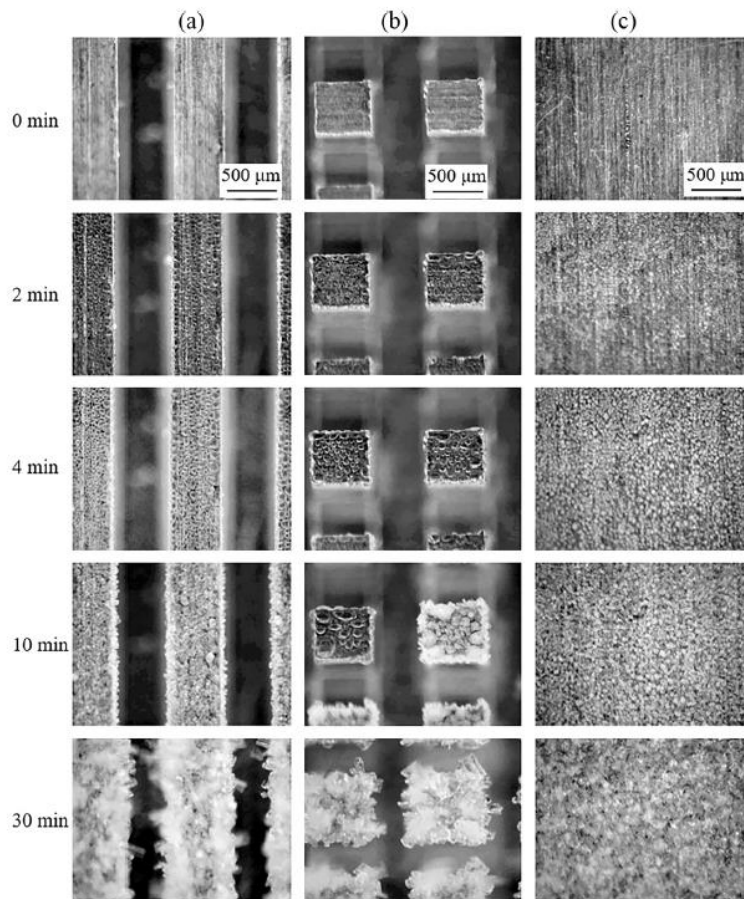


Figure 2.8: Frost accumulation on the a) surface with parallel grooves, b) surface with crossed grooves, c) flat surface: $T_{air} = 2^{\circ}\text{C}$, $RH_{air} = 85\%$, $u_{air} = 0.92 \text{ m/s}$, $T_s = -13^{\circ}\text{C}$ (Wu et al., 2017)

2.3 Defrosting research

In addition to the fundamental studies on frost properties and microscale surface features, researchers also explored different engineering methods to address the frosting issue of ASHP. The engineering methods can be divided into two major categories: passive methods and active methods. Passive methods employ different designs of the microscale surface features, the geometry of heat exchangers, and system architecture to delay the accumulation of frost, reduce the frost mass, and remove the condensate fast and efficiently. On the other hand, active defrosting methods include reverse-system, hot gas bypass, optimizing control strategies, and many other alternative methods like ultrasonic vibration and microwave, etc. Both passive and active defrosting methods are often evaluated during continuous frosting and defrosting cycles to investigate the performance for real-world applications.

2.3.1 Passive defrosting methods

a. Surface coatings on real heat exchangers/ HP systems

O'NEAL et al. (1998) conducted an experimental study to investigate the effects of different coatings on the frost/ defrost performance of a 5.3 kW heat pump. The samples are 1) a bare aluminum surface for the baseline; 2) a hydrophobic surface (coated with silicone-based polymer) with 105 to 110° contact angle for water droplets; 3) a hydrophilic surface (coated with acrylic-based polymer) with 65 to 74° contact angle for water droplets. The experimental conditions are challenging operating conditions for air-source heat pumps: air temperatures are either -1.1 or 1.7°C, and relative humidity is higher than 80%. But the velocities of indoor and outdoor airflow are not mentioned. The results showed a 38% reduction in the amount of retained water on the silicone coated heat exchanger sample compared to the uncoated heat exchanger section. Also, both types of coatings provided slower frost accumulation, faster defrosting, and

thus, higher efficiencies of the system. However, the authors mentioned that the overall impact on frost/ defrost cycle performance was smaller than 5% for both coatings.

Moallem et al. (2012) investigated the effects of hydrophilic and hydrophobic surface coatings on the frost growth and performance of five different microchannel heat exchangers during frosting conditions. The controlled experimental conditions are the following: the air inlet temperature of 1.7°C, the dew point of air of 0.6°C ($RH_{air}= 82\%$), air face velocity of 1.0 m/s, a refrigerant inlet temperature of -9.4°C, and surface temperature of -7°C at the beginning of the frosting experiments. The data showed neither hydrophobic nor hydrophilic surface coating leads to a significant extension of the frosting time (defined when air face velocity reaches 75% of the initial value). Instead, a variation of 2°C for the surface temperature due to epoxy between fins and tubes increases the frosting time by more than 80%. Although hydrophilic surface increases the cooling capacity of the heat exchanger by up to 15% at the beginning of the refrosting process.

Mahvi et al. (2021) tested the performance of aluminum microchannel heat exchangers with different surface wettability under realistic operating conditions in a full transcritical carbon dioxide heat pump. The test conditions are the following: the outdoor air temperature of 2.6/ 9.8°C, the relative humidity of 85/88%, the outdoor air face velocity of 1.5 m/s, and the initial evaporating temperature of -5°C. The superhydrophobic evaporator delay frost accumulation significantly and maintain higher heat transfer rates compared to the hydrophilic evaporator in both conditions. The delayed frost formation results in a system efficiency improvement in the first few cycles, though the benefits diminished in later cycles due to water retention and incomplete defrosting. In this paper and the paper of O'Neal et al. (1998), the authors linked the performance of the outdoor evaporator with the performance of the full heat pump system. But the air face velocity used was about 1.5 m/s, which is more applicable to stationary heat pumps than to automobile applications.

In addition, there is no active defrosting in this experiment: defrosting was simply turning off the heat pump system, which might not be practical for an automobile HP system.

b. Improved design of the outdoor coil

In addition to the surface treatment, optimizing the geometry of fins also has been widely studied in the past two decades. Xia et al. (2006) experimentally studied the air-side thermal-hydraulic performance of louvered-fin flat-tube Microchannel Heat Exchangers (MCHXs) using glycol solution as the working fluid. The experimental conditions are the following: the air temperature of -1 to -2°C, the coolant inlet temperature of -11°C, the air relative humidity of 70% to 80%, and the initial air inlet velocity of 1.0 m/s. The overall heat transfer coefficient, pressure drop, and j and f factors during one frosting cycle were compared for four MCHXs with different fin geometries. One of the key conclusions is the frosting and defrosting performance become repeatable after the third or fourth refrosting cycle.

Zhang and Hrnjak (2009) experimentally compared three types of heat exchangers: conventional Round Tube Plate Fin (RTPF), flat tube with Parallel Flow Serpentine Fin (PFSF), and a new design of Parallel Flow Plate Fin (PF²). Glycol solution was used as the working fluid and the frosting conditions were the following: the air temperature of 0°C, the coolant inlet temperature of -14.5°C, the initial air inlet velocity of 0.9, 2, 3 m/s, and relative humidity of 70 and 80%. The data showed that the RTPF heat exchanger has the longest frosting time due to the two times the large air-side surface area. The frost accumulation speed is slower for the PF² than it is for PFSF, and the orientation has negligible effects on the performance of the PF² heat exchanger. The authors also experimentally studied the effects of louver angles (15, 27, and 39 deg) and fin pitches (12, 15, and 18 fpi) on the pressure drop and overall heat transfer coefficient of serpentine fin microchannel heat exchangers in periodic frosting (Hrnjak et al., 2017). The

experimental methods are the same as in their early study and the test conditions are the following: the air temperature of 0°C, the coolant inlet temperature of -11°C, the initial air inlet velocity of 3.5 m/s, and relative humidity of 70 and 80%. The air-side pressure drop increases with louver angle for all fin pitches, but the heat transfer coefficient reaches the maximum values with 27° of louver angle for 15 and 18 fpi. Also, several (4 to 8) repetitive frosting/defrosting cycles were necessary to reach the saturation in water retention and the performance in real operation. In this case, the combination of 18 fpi and 39° had the highest overall heat transfer coefficient and the most desirable pressure drop due to the best drainage, although it was not good in dry conditions. This conclusion might indicate that the design of an outdoor heat exchanger for automobile air-conditioning systems should be optimized not only for the gas cooler in AC mode and evaporator in dry conditions but also for evaporator in frosting conditions.

Kim et al. (2014) experimentally investigated the frost growth and thermal performance degradation of microchannel heat exchangers during periodic frosting/ defrosting cycles as a function of the depth of the heat exchanger (16/ 20 mm), the fin pitch (70/ 80 FPDm), and the pitch of the microchannel tubes (7/ 8 mm). The experimental conditions are the following: the air temperature of 2°C, the air dew point of 1°C, the initial air inlet velocity of 5.0 m/s, and the coolant temperature of -8°C. Within the relatively small range of geometry, the results showed that the heat transfer rate per unit volume and the uniformity of frost growth between the front and rear sides improve as the depth of the microchannel heat exchanger reduces. Also, a larger fin pitch results in slower frosting on the front end and a slower reduction in the heat transfer rate due to the uniform frost accumulation. The increased channel pitch increases the capacity at the beginning of the frosting process, but the frost growth is less uniform, and the capacity decreases significantly later.

Park et al. (2016) experimentally investigated the thermal performance of louvered fin designs with equal and unequal louver pitch under both dry and frost conditions using the 50% glycol solution as the working fluid. The data showed that the performance difference was negligible between the two designs in the dry conditions. In the frosting conditions (the air temperature of 5.5°C , the air velocity of 0.95 m/s , the relative humidity of 72%, and fluid inlet temperature of -20°C), however, the frost accumulation at the front side was mostly delayed and the thermal performance was the highest with unequal louver pitch design in which the louver pitch decreased by 20% from the air inlet to the redirection region (**Figure 2.9**). But the effects of the fin design on the system performance are not discussed in this study.

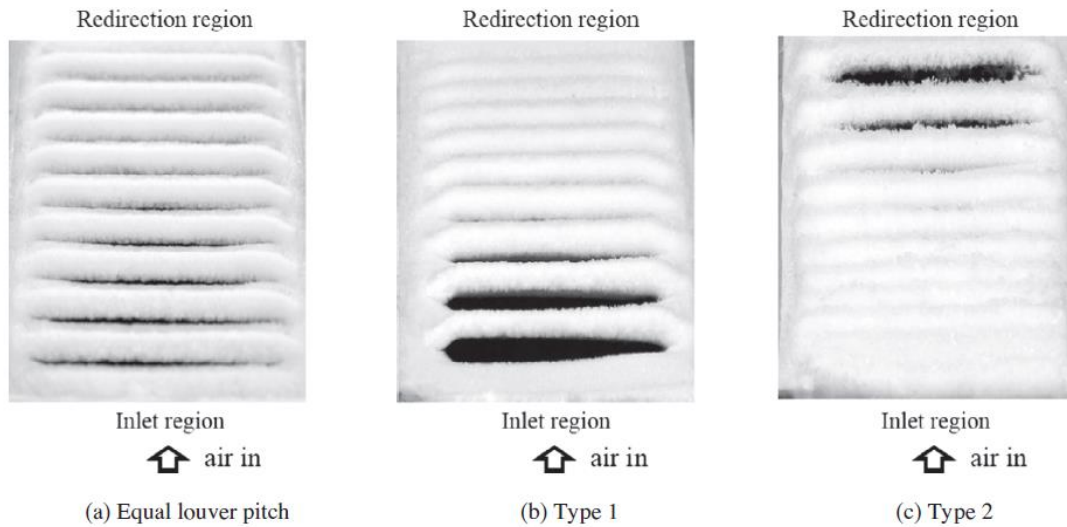


Figure 2.9: Frosting accumulation on the front side of heat exchangers with **a)** equal louver pitch, **b)** decreasing louver pitch, **c)** increasing louver pitch at 160 minutes of frosting time (Park et al., 2016)

Hong et al. (2020) proposed a novel plain louvered fin microchannel heat exchanger (MCHX) with horizontal microchannel tubes (as shown in **Figure 2.10**) to improve the frosting and defrosting performance of air-source heat pumps in electric vehicles. The test conditions are the following: the air temperature of 2°C , the relative humidity of 85%, and the air velocity of 1 to 4 m/s. The air-side areas of both the new design and the conventional one are identical. The data

showed the frosting period and peak heating capacity of the heat pump with the new fin design are 134.6% and 24.3% higher than those of the heat pump with the conventional corrugated MCHX when the initial air face velocity is 4 m/s. Also, the heat pump with a new fin design exhibited a 12.7% lower power consumption and 16.7% higher COP than the conventional one under the frosting condition. However, no direct measurement or data is showing better drainage of the new design. In addition, the initial/ frost-free surface temperature of the new design is approximately 3°C higher than it is for the conventional MCHX. Since the thermodynamic plots of the cycles

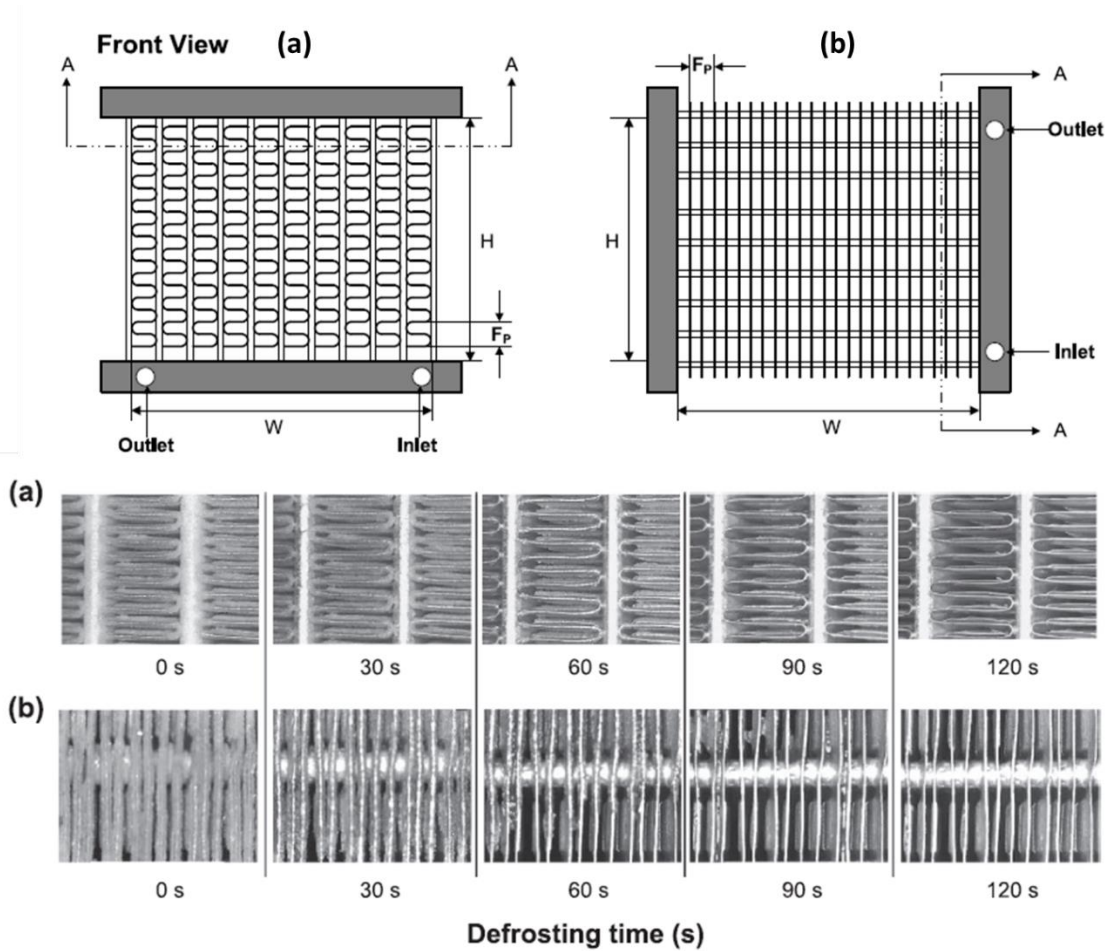


Figure 2.10: **a)** top: front view; bottom: the condensate drainage of the conventional design of microchannel heat exchanger; **b)** top: front view; bottom: the condensate drainage of the new design of plain microchannel heat exchanger: $T_{air} = 2^{\circ}\text{C}$, $RH_{air} = 85\%$, $u_{air} = 2 \text{ m/s}$ (Hong et al., 2020)

were not provided in the paper, it is difficult to evaluate if this is only resulting from the better refrigerant distribution of the new design or the refrigerant charge amounts and control of the systems.

c. Improved system architecture

Besides the optimization of the fin geometry, there are also studies improving the design of system architecture to increase the surface temperature of the outdoor evaporator and delay the frost accumulation. The significance of increasing the surface temperature of the outdoor evaporator is that an increase of 2°C for the fin temperature can increase the frosting time by more than 80% (Moallem et al., 2012). To achieve the target, several methods have been applied: accumulator heater, improving refrigerant distribution, adding an Internal Heat Exchanger (IHX), etc.

Mei et al. (2001) proposed a new design by adding a resistance heater in the accumulator of a two-ton residential heat pump to retard the frost accumulation. The operating condition includes an outdoor temperature range of 0.6°C to 5°C and 75% relative humidity. The data showed that the frequency of defrosting cycle can be reduced by a factor of 5 in the Knoxville, Tennessee, area, and the heat capacity is higher and more stable. However, the authors mentioned this method will not prevent frost accumulation on the outdoor coil when the ambient temperature is below 0°C, which is still a reasonable operating range for a heat pump. This limited the application of this method.

The surface temperature of the outdoor evaporator can also be increased by improving the refrigerant distribution inside it. Song et al. (2016) investigated the effects of Frosting Evenness Value (FEV) on the defrosting performance of an ASHP. By comparing the defrosting efficiencies (energy consumption/ energy supply for defrosting) of three circuits with FEVs of 82.6%, 90.6%,

and 96.6%, the defrosting efficiency can be improved by 6.8% when FEV increases from 82.6% to 96.6%. Thus, the defrosting performance reduces with uneven frost accumulation. Although the measurement of frost accumulation might not be very accurate since the vaporized water during the defrosting process was neglected during the calculation and FEVs were calculated using the mass of collected water on each circuit.

Cernicin et al. (2022) investigated the effects of introducing an Internal Heat Exchanger (IHX) to a transcritical CO₂ system operating in heat pump mode experimentally. The results show the efficiency of the IHX HP system is higher than the basic HP system by up to 11.6% when the heating capacities are identical for both HP systems. Also, the evaporating temperature of the IHX HP system can be higher than it is for the basic HP system by up to 2.3°C when the outdoor air temperature is 0°C. In conclusion, introducing IHX helps improve the efficiency of the CO₂ HP system and has considerable benefits for the performance in frosting conditions.

2.3.2 Active defrosting methods

Even the passive methods have the potential to delay the frost accumulation on the outdoor heat exchangers, the frost that is presented after the delay must be removed by active methods periodically for better system efficiencies. In the following section, the active defrosting methods, including reverse cycle defrosting, hot gas bypass defrosting, and alternative methods, are reviewed.

a. Reverse cycle defrosting

For reverse cycle defrosting, the indoor heat exchanger works as an evaporator and absorbs heat from indoor air/ heater, but the hot refrigerant is pumped to the outdoor heat exchanger to melt the frost. The defrost energy is supplied by the thermal energy of indoor air, the indoor heat exchanger, the indoor warm refrigerant, and the electrical power of the compressor. On the other

hand, the defrosting energy is consumed to warm up the frost to the freezing point, melt the frost, heat the retained water, vaporize a certain amount of the water, heat the outdoor heat exchanger, and heat the ambient air (heat loss). This method requires no additional devices and no extra initial cost for a reversible heat pump system. Also, reverse cycle defrosting requires the shortest defrosting time due to the sufficient energy supply. Although this method can introduce some undesired temperature fluctuations for indoor air without proper design, it is the most widely applied method.

Payne and O'Neal (1995) experimentally studied the frost accumulation and defrost cycle of a 10.6 kW air-source residential heat pump with the original scroll compressor and a reciprocating compressor. The frosting conditions were 1.7°C outdoor air temperature and 0.5°C dew point, and the defrosting was conducted by reversing the cycle. The defrost was initiated at a 20% drop in heat capacity and terminated when the temperature of the lowest liquid line of the outdoor coil reached 26.7°C. The results showed that the low-efficient reciprocating compressor produced 18% higher discharge superheat and thus defrost 5% faster than the scroll compressor with a similar capacity.

Ding et al. (2004) proposed to bypass the Thermal Expansion Valve (TXV) using a solenoid valve in an air-source heat pump water heater during the defrosting mode, to reduce the fluctuations of indoor air temperature and eliminate the risk of system shutdown due to low evaporating pressure. The authors concluded that the refrigerant rushing into the accumulator and the lag in the response of TXV are the main reasons for the unstable operation of the heat pump. Thus, the solenoid valve was used to maintain a large opening of the throttling device, facilitate the frosting process and the switchback process, and improve the cyclic efficiency.

Wenju et al. (2011) presented an experimental study on the performance of a novel Thermal Energy Storage (TES) based reverse-cycle defrosting method for air-source heat pumps. The results showed the new method reduced the defrosting time by approximately three minutes (38%) and minimized the risk of system shut down due to low suction pressure. In addition, the new method increased the mean indoor coil surface temperature by about 25°C.

Dong et al. (2012) conducted experiments to study the defrosting energy supplies and consumptions during a reverse cycle defrosting of an air-source heat pump. During the defrosting, the indoor fan was switched on to provide adequate heat. The data showed that the indoor air provides 71.8% of the total heat supplied for defrosting and 59.4% of the supplied energy was used for melting frost. Also, the maximum defrosting efficiency (energy required to melt the frost and vaporize the retained melted frost/ energy supplied to the outdoor coil) could be up to 60.1%.

Although reverse cycle defrosting has been widely applied for residential ASHP systems with tube-fin heat exchangers, the studies for ASHP systems with microchannel heat exchangers are relatively rare. Steiner and Rieberer (2013) investigated the defrosting performance of an automobile CO₂ heat pump system both experimentally and numerically. The data showed that the defrosting took less than 2 minutes with reverse cycle defrosting, and the final refrigerant inlet temperature of the outdoor coil reached 50°C. The transient simulation results indicate that there is an optimal throttle opening regarding defrosting time and efficiency for the test conditions. The authors emphasized avoiding non-uniform defrosting, where frost-free areas result in heat losses to the ambient, thus increasing the defrost time and reducing defrost efficiency.

In summary, most previous studies focus more on the air-side performance of the fin-and-tube outdoor evaporator with a residential HP system. Limited data are available for microchannel

heat exchangers in HP systems in practical operating conditions, and only a few researchers mentioned the effects of frosting on the performance of the HP system.

b. Hot gas bypass defrosting

The advantage of hot gas bypass defrosting is smaller temperature fluctuations during the defrosting process; however, the defrosting time could be longer than reverse cycle defrosting, especially when the ambient air temperature is below 0°C. Also, the defrost efficiency and cyclic efficiency for the hot gas bypass method are not necessarily higher compared to the reverse cycle defrosting method.

Cho et al. (2005) investigated and compared the performance of a showcase refrigeration system during on-off cycling and hot-gas bypass defrosting. The effects of off-period (40, 60, and 80 minutes after a normal operation of 240 minutes, as shown in **Figure 2.11**) in the on-off cycling and EXV opening (50%, 75%, and 100% of the full opening) in the hot-gas bypass defrosting on the performance of refrigeration system were analyzed. The operating conditions of the system are following: the ambient temperature of 35°C, the in-showcase temperature of 15.4°C, and the in-showcase relative humidity of 80%. The defrosting was terminated when the refrigerant temperature at the defrosting evaporator exit increased over 20°C. The data showed the hot-gas bypass method has a higher cooling capacity and smaller temperature fluctuations in frosting/defrosting conditions.

Byun et al. (2008) examined the effects of the hot gas bypass method on the frosting retarding and system performance using a 1.12 kW air-source heat pump. The results indicate that the hot gas bypass method delays the frost growth on the outdoor coil and improves the COP and heating capacity by 8.5% and 5.7% during the 210-minute operation. In addition, the optimal

bypass refrigerant flow rate was determined to be 0.2 kg/min (20% of the total refrigerant flow rate).

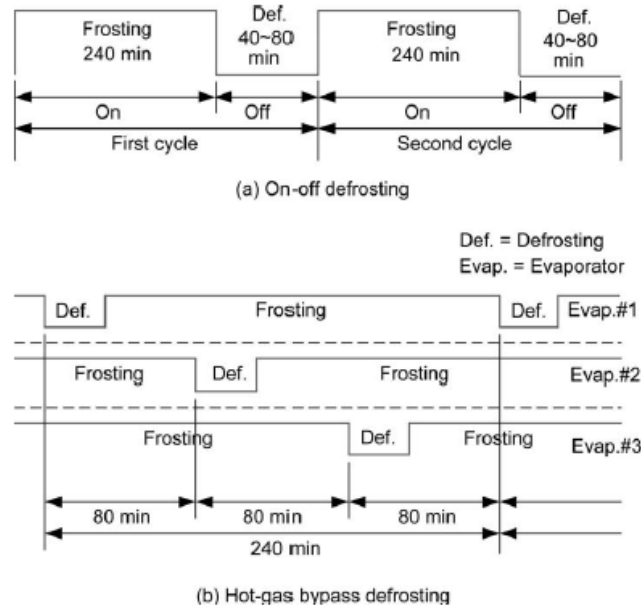


Figure 2.11: The frosting and defrosting cycle of **a)** on-off defrosting **b)** hot-gas bypass defrosting (Cho et al., 2005)

Huang et al. (2009) investigated and compared the dynamic characteristics of Reverse-Cycle Defrosting (RCD) and Hot Gas Bypass Defrosting (HGBD) using a 55-kW air source water heater heat pump. The results showed that the water temperature has a smaller fluctuation for the HGBD method. However, HGBD requires 2.89 times the defrosting time required by RCD. Because there is only one major heat source- electricity consumption of the compressor, for the HGBD method. Also, the suction and discharge superheat of HGBD is lower than those of RCD. As the liquid refrigerant rush into the accumulator, the suction superheat stays at 0°C and the discharge superheat decreases gradually. In addition, the HGBD method has a higher discharge pressure but a lower suction pressure. This might reduce the compressor efficiency and thus the cyclic efficiency of the heat pump system, which was not discussed in this paper.

Hu et al. (2014) analyzed the characteristics of hot gas bypass defrosting using an air-source CO₂ heat pump water heater under various ambient conditions. The frosting conditions are the following: the ambient air temperature of 2°C and relative humidity of 80%. The defrosting was initiated after an 81-minute of normal operation and terminated after 600 s. A rapid drop in temperature and pressure was found at the beginning of the defrosting process. The energy analysis indicates that 57.4% of the supplied energy is consumed to increase the internal energy of the gas cooler and only 35% of that is used to melt the frost. The typical defrosting efficiency in this study ranges from 30% to 40%, and it increases with increasing dry bulb temperature and decreasing relative humidity of ambient air.

Kim et al. (2015) proposed a novel defrosting method by combining the hot gas bypass defrosting with the accumulator heating, to increase the suction superheat and discharge superheat during the defrosting. In the paper, the performance of conventional reverse cycle defrosting and the proposed new method in a 16kW air-source heat pump was compared and analyzed. The frosting conditions are the following: the ambient air temperature of -8 to 2°C, the relative humidity of 84%, the indoor air temperature of 20°C, and the indoor dew point of 15°C. The data showed that the cyclic EER of the novel method is 15% higher than that of the conventional method at -5°C (considering the power input of the accumulator heater), and the defrosting time of the new method is reduced by 15%. This study shows the potential improvement of accumulator heating, but the induction heater might be replaced by an internal heat exchanger for simplification of system architecture and control.

Overall, the defrosting efficiency of the reverse cycle method is higher than that of the hot gas bypass method. Considering the simplicity of control and lower initial cost for reversible AC/HP systems, reverse cycle defrosting is recommended to be applied to electric vehicles.

c. Alternative methods

Besides the system defrosting methods, there are several alternative methods for defrosting: mechanical methods (oscillation) and chemical methods (desiccant). Wang et al. (2012) experimentally investigated the effect of frost suppression of ultrasonic vibrations on a horizontally installed hydrophilic finned-tube evaporator. An ultrasonic transducer with a resonance of 28.2 kHz and rating power of 60 W was used to excite the evaporator. The results showed the gaps between fins were nearly blocked by frost after 32 minutes of operation without ultrasonic vibration. But only 2/3 of the gaps between the fins were blocked after 92 minutes of the operation with ultrasonic vibration. Also, the basic ice layer on the surface of the fins cannot be removed by the vibration, but the frost branches on top of the ice layer can be fractured by shearing force and removed effectively by gravity.

In a following study, the authors firstly calculated the natural frequencies of frost crystals with different heights in COMSOL, then optimized the intermittent operation of the ultrasonic vibration defrosting method (Tan et al., 2014). It was found that the frost with 0.5 mm high (after growing for 4 minutes) has a natural frequency of approximately 28 kHz. The 28 kHz 60 W ultrasonic transducer was used to effectively remove the frost crystal with 0.5 mm height. Also, the optimal defrosting cycle was determined to be a 1-minute vibration after a 4-minute interval (**Figure 2.12**). However, the effects of vibration on the leakage of the system were not mentioned in the studies. In addition, the defrosting efficiency and the cyclic efficiency of the HP system were not discussed in the papers, which require further investigation.

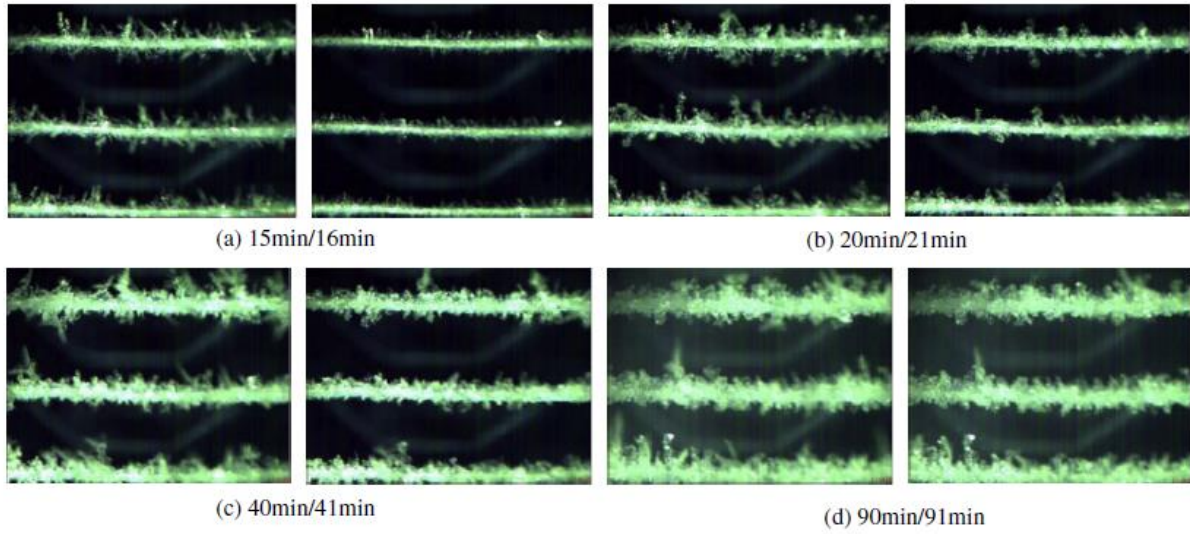


Figure 2.12: The microscopic images of frost on fins before/after ultrasonic vibration defrosting at different times (Tan et al., 2014).

To reduce the power consumption of defrosting process, some researchers used desiccants to remove water vapor from the inlet ambient air and prevent the heat pump from frosting. Wang and Liu (2005) proposed a new defrosting method using solid adsorbents to dehumidify the outdoor air before entering the evaporator. The humidity of inlet air reduces, and the air temperature rises due to the heat generated by the desiccant. The authors concluded that the air pressure loss of the adsorbent bed is negligible based on the mathematical model and the desorption can be achieved using a sunlight-powered electric heater. However, no experimental validation was provided in the paper.

Similarly, Zhang et al. (2012) further developed the idea to reduce frost formation by adding an extra desiccant-coated evaporator to the air-source heat pump water heater. During the desorption process, valve 7 is opened and valve 8 throttled. Simultaneously, air-side valve 10 is opened but 11 and 12 are closed (**Figure 2.13**). So, the regeneration air is cycled between the desiccant-coated heat exchanger 1 and the other outdoor heat exchanger to recover the desiccant. A simulation was carried out with the ambient temperature of -7 to 5.5°C and the relative humidity

of 60% to 80%. The numerical results showed the COP of the system varies from 3.3 to 3.8 and 5% to 30% higher than that of the heat pump using the hot-gas defrosting method.

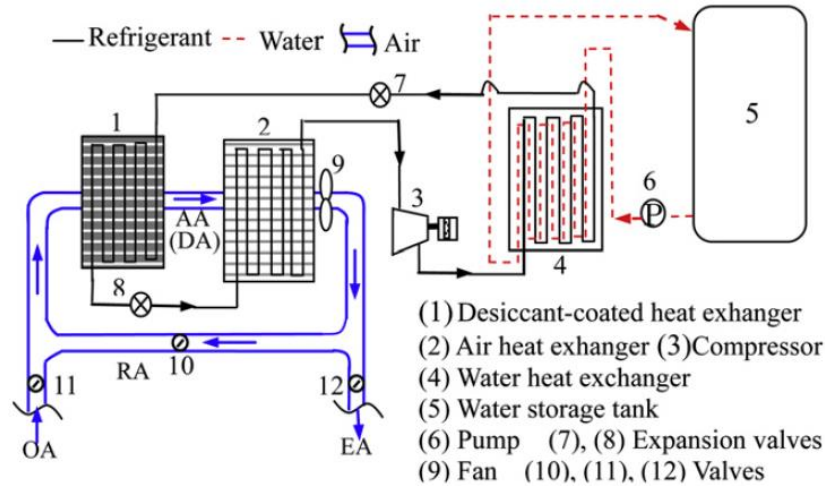


Figure 2.13: The schematic of the proposed frost-free air-source heat pump water heater (Zhang et al., 2012)

In conclusion, although the frosting and defrosting of fin-and-tube heat exchangers with residential HP systems have been widely studied, limited data are available for microchannel heat exchangers in HP systems under practical operating conditions. Especially, most previous studies focused on the air-side performance of heat exchangers only, and the effects on the HP system performance are rarely discussed. In chapters 5 and 6, the periodic performance of a reversible R744 HP system in various practical frosting conditions is discussed and analyzed. The effects of environmental conditions, the defrost-initiation criteria, and the orientation of the outdoor coil on the performance of both the outdoor coil and the HP system are investigated. The conclusion can be used to guide the design and the testing of reversible vapor compression systems for different applications.

CHAPTER 3: EXPERIMENTAL STUDY OF ROLE OF INTERNAL HEAT EXCHANGER IN AC AND HP MODES

3.1 System architecture and experimental facility

In this chapter, the role of IHX in an R744 air-source AC/ HP system was investigated experimentally. A facility with two climate chambers was calibrated and used for this purpose. To control the temperature, humidity, and airflow rate in each climate chamber, a glycol heat exchanger, a PID-controlled electric heater, a steam spray, and a wind tunnel with VFD-controlled blowers were installed inside each climate chamber. **Figure 3.1** presents the refrigerant-side sensors and the arrangement of the reversible R744 AC/ HP system inside and between the climate chambers. By controlling the ball valves 1 to 4 (black: closed; white: opened) in the system, the R744 system can be reversed from AC mode to HP mode. Using the ball valves 5 and 6, the high-pressure side of the IHX can be bypassed. The one-slab two-pass outdoor coil (gas cooler in AC mode/ evaporator in HP mode) and the Electronic Expansion Valve (EXV) for HP mode were installed inside the outdoor climate chamber. The two-slab four-pass indoor coil (gas cooler in HP mode/ evaporator in AC mode) and the EXV for AC mode were installed inside the indoor climate chamber. The mid-section circuit consists of a semi-hermetic reciprocating compressor, the integrated IHX and accumulator, and the EXV controllers. The dimensions of the key components are listed in **Table 3.1**. It has to be mentioned that a industrial compressor instead of a DC scroll compressor was used in this study due to the voltage limitation in the lab. But the other major components are designed for automobile AC applications.

To measure the steady-state performance of the reversible R744 AC/ HP system, three independent approaches were used: refrigerant-, air-, and chamber-side balances. As shown in **Figure 3.1**, pressure transducers, thermocouple probes, and a mass flow meter were installed in

the refrigerant circuit. Also, thermocouple grids, dew point sensors, air nozzles, and differential pressure transducers were installed along the wind tunnels. In addition, power transducers were used to measure the energy consumed inside the climate chamber, and the cooling capacity of the glycol heat exchangers was calculated using measured glycol mass flow rates and inlet/ exit glycol temperatures for both chambers. Considering the heat loss through the insulation walls, the cooling and heating capacities of indoor and outdoor coils were also calculated using the chamber-side energy balance. The data reduction equations of the three approaches are listed in the next section.

Table 3.1 Dimensions of key components of the R744 system

Component	Dimensions
Outdoor coil (1-slab, 2-pass: 17/ 34 tubes)	• Overall dimension (mm): 635 (H) × 467 (W) × 16 (T) • Fin height (mm): 8.0 • Fin pitch (mm): 1.3 • Fin thickness (mm): 0.1 • Louver length (mm): 7.2 • Louver pitch (mm): 0.9
Indoor coil (2-slab, 4-pass: 17 tubes in each pass)	• Overall dimension (mm): 229 (H) × 248 (W) × 35 (T) • Fin height (mm): 5.6 • Fin pitch (mm): 1.4 • Fin thickness (mm): 0.08 • Louver length (mm): 4.9 • Louver pitch (mm): 1.12
Internal heat exchanger/ Accumulator	• Overall dimension (mm): 247 (H) × 85 (D) • Counter-flow IHX for AC/ defrost mode • Parallel-flow IHX for HP mode • Nylon accumulator
Semi-hermetic reciprocating compressor	• 2-cylinder • Displacement: 2.39 m³/h (50 Hz), 2.87 m³/h (60 Hz) • 1450 rpm @ 50 Hz, 1750 rpm @ 60 Hz • 3.8 HP, 208~ 230V/ 3/ 60 Hz
Electronic expansion valve	• Actuator step range: 2500

Table 3.2 presents the details and uncertainties of the sensors used in this dissertation. All sensors were calibrated prior installed in the facility. The signals from the sensors are measured

by an HP series 75000B data logger and converted to SI-unit parameters via the VEE program. For each steady-state condition, at least 30 minutes was waited to stabilize the reading and then recorded the data for another 20 minutes with an interval of 6 seconds. **Table 3.3** lists all the experimental operating conditions in this chapter. The detailed results will be discussed in the results section.

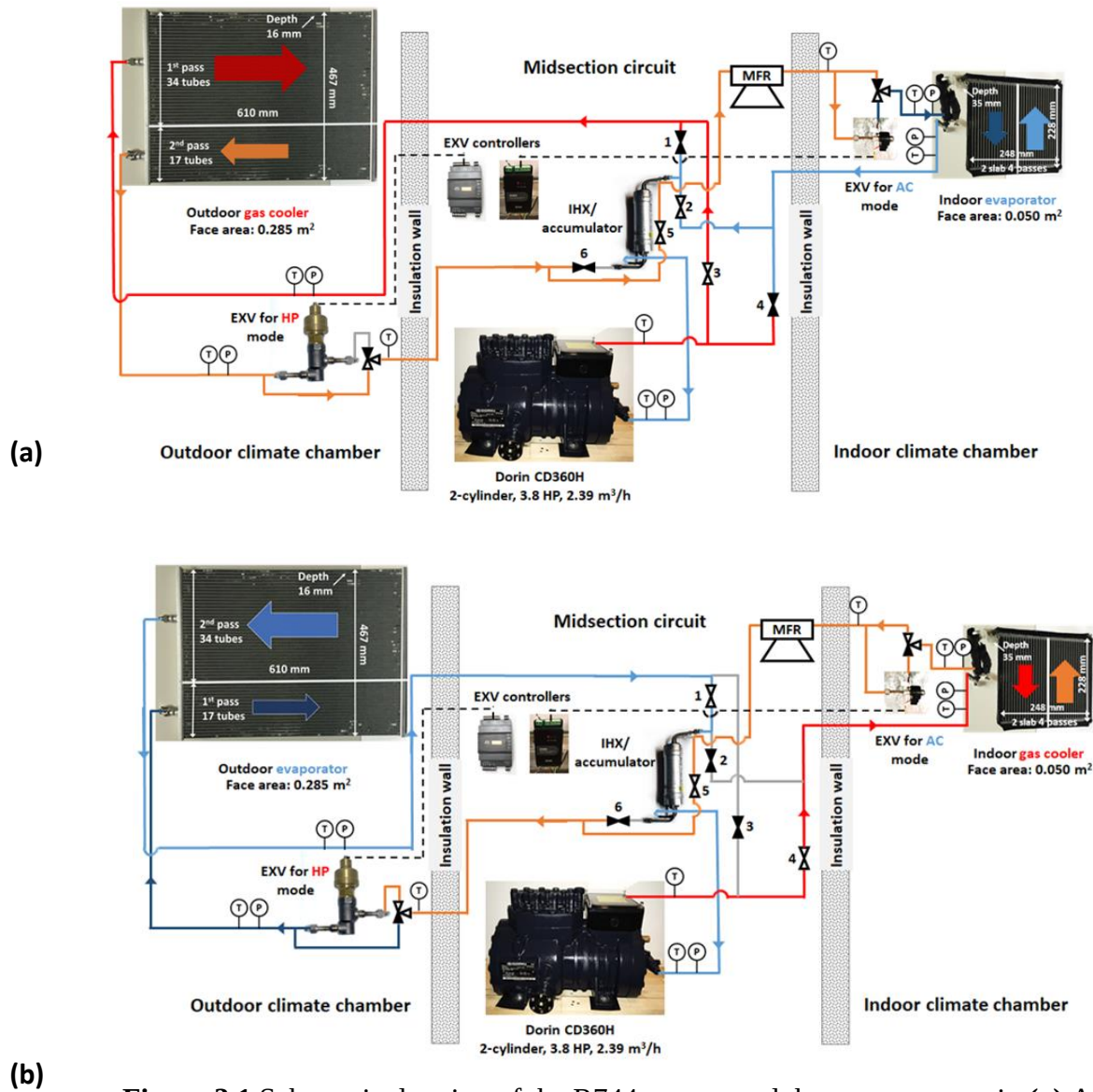


Figure 3.1 Schematic drawing of the R744 system and the measurement in (a) AC mode and (b) HP mode

Table 3.2 Uncertainties of refrigerant-, air-, and chamber-side instruments

Parameter	Sensor	Range	Uncertainty
Refrigerant/ glycol temperature [°C]	T-type thermocouple	-50 to 150	± 0.2
Refrigerant low-side pressure [MPa]	Strain gage	0 to 6.89	± 0.003
Refrigerant high-side pressure [MPa]	Strain gage	0 to 20.68	± 0.007
Refrigerant/ glycol mass flow rate [kg/h]	Coriolis-type	0 to 2180/ 0 to 6800	± 0.10% of reading
Electrical power consumption [kW]	3-phase wattmeter	0 to 6	± 0.20% of reading
Air/ wall temperature [°C]	T-type welded thermocouple	-50 to 150	± 0.2
Dew point temperature [°C]	Chilled mirror dew point sensor	-80 to 85	± 0.2
Pressure drop of air-side [Pa]	Differential pressure transducer	0 to 600	± 0.25% of full scale

Table 3.3 Operating conditions of all dry tests in AC and HP modes

(a)	The purpose of this group	Indoor			Outdoor		Compressor speed [%]
		T_{eai} [°C]	RH_{eai} [%]	V_{eai} [m/s]	T_{cai} [°C]	V_{cai} [m/s]	
Group AC-1	Optimal charge amounts	40	20	2.5	40	3.0	70
Group AC-2	Control equations	25 to 40			25 to 40		60 to 75
Group AC-3	Steady-state scenario	25			25 to 40		Various values to match $T_{eao} = 5^{\circ}\text{C}$
Group AC-4	Pull-down scenario	25 to 40			40		Various values to match $T_{eao} = T_{eai} - 5^{\circ}\text{C}$

(b)	The purpose of this group	Outdoor			Indoor		Compressor speed [%]
		T_{eai} [°C]	RH_{eai} [%]	V_{eai} [m/s]	T_{cai} [°C]	m_{cai} [kg/min]	
Group HP-1	Optimal charge amounts	0	40	3.0	10	6.0	80

Table 3.3 Operating conditions of all dry tests in AC and HP modes (cont.)

Group HP-2	Control equations	-5 to 10			20		50 to 75
Group HP-3	Steady-state scenario	-10 to 10			20		Various values to match $T_{cao} = 50^{\circ}\text{C}$
Group HP-4	Warm-up scenario	5			5 to 20		Various values to match $T_{cao} = T_{cai} + 30^{\circ}\text{C}$

3.2 Measurement and data reduction

a. In AC mode

The cooling/ heating capacities of indoor/ outdoor coil in the R744 AC system with and without IHX were calculated using refrigerant-, air-, and chamber-side balances. The thermodynamic properties were calculated using commercial software- Refprop 9.0 (Lemmon. et al., 2010). For the indoor evaporator, the three balances are shown below:

$$Q_{cooling,ref} = (1 - OCR)\dot{m}_r(h_{ero} - h_{eri}) + OCR \cdot \dot{m}_r(h_{eoo} - h_{eoi}) \quad \text{Eqn (1)}$$

$$Q_{cooling,air} = \dot{m}_{eai,dry}(h_{eai} - h_{eao}) \quad \text{Eqn (2)}$$

$$Q_{cooling,cham} = \sum P_{id,elec} - Q_{id,loss} - \dot{m}_{id,g}(T_{id,go} - T_{id,gi}) \quad \text{Eqn (3)}$$

where, OCR (Oil Circulation Rate) was measured by the sampler installed across the mass flow meter on the liquid line. In this project, the OCR is measured to be 2% and the effects on system capacity are negligible. $\dot{m}_r$ was measured by the mass flow meter, and it is actually the mass flow rate of the oil and CO₂ mixture. Refrigerant enthalpy h_{eri} equals the refrigerant enthalpy at EXV inlet h_{xri} , assuming isenthalpic expansion. Refrigerant enthalpy h_{ero} , oil enthalpies h_{eoi} and h_{eoo} were determined from the pressures and temperatures measured at the inlet (T_{eri} and P_{eri}) and outlet (T_{ero} and P_{ero}) of the indoor coil. The refrigerant-side calculation for the evaporator is only valid

when there is at least 3°C of superheat at the evaporator exit. Otherwise, the cooling capacity and related refrigerant properties are calculated using the other two approaches.

The indoor air mass flow rate $\dot{m}_{eai,dry}$ was calculated using measured air temperature (T_{ean}), dew point (T_{dpen}), and pressure drop (DP_{en}) at the indoor nozzles. The enthalpies of air h_{eai} and h_{eao} were determined using air temperatures at the indoor coil inlet T_{eai} and exit T_{eao} . In the calculation, dry air properties were used because dew point measurements showed there was no water condensation across the indoor coil.

The electrical power consumption $P_{id,elec}$ of each device inside the indoor chamber was measured using power transducers. The heat loss from the indoor climate chamber to the ambient $Q_{id,loss}$ was calculated using the measured UA value of the climate chamber. The mass flow rate of glycol flowing into the indoor chamber $\dot{m}_{id,g}$ and the inlet and exit glycol temperatures ($T_{id,gi}$ and $T_{id,go}$) were used to calculate the cooling capacity of the indoor glycol heat exchanger.

Similarly, three approaches were used to calculate the heating capacity of the outdoor gas cooler:

$$Q_{heating,ref} = (1 - OCR)\dot{m}_r(h_{cri} - h_{cro}) + OCR \cdot \dot{m}_r(h_{coi} - h_{coo}) \quad \text{Eqn (4)}$$

$$Q_{heating,air} = \dot{m}_{cai,dry}(h_{cao} - h_{cai}) \quad \text{Eqn (5)}$$

$$Q_{heating,cham} = \dot{m}_{od,g}(T_{od,go} - T_{od,gi}) + Q_{od,loss} - \sum P_{od,elec} \quad \text{Eqn (6)}$$

Also, for the IHX cycle, the capacity of the IHX was calculated using the high-pressure refrigerant-side, since it is in the supercritical or subcooled zone:

$$Q_{IHX,ref} = (1-OCR)\dot{m}_r(h_{cro} - h_{xri}) + OCR \cdot \dot{m}_r(h_{coo} - h_{xoi}) \quad \text{Eqn (7)}$$

Where, refrigerant enthalpy h_{xri} and oil enthalpy h_{xoi} were determined from the pressure and temperature measured at the inlet of the expansion valve used in AC mode. The low-pressure side capacity of IHX equals the high-pressure side capacity, assuming no heat loss to the ambient.

The Coefficient of Performance (COP) of the R744 air-conditioning system with and without IHX was calculated as follows:

$$COP = \frac{Q_{cooling,ave}}{P_{comp}} \quad \text{Eqn (8)}$$

Where, $Q_{cooling,ave}$ is the average of ref-, air-, and chamber-side cooling capacity of indoor evaporator. P_{comp} is the measured electrical power consumption of the VFD-controlled compressor.

The uncertainty analysis was conducted using a MATLAB program integrated with the data reduction program, following the method of Moffat (1988). According to the method, the uncertainty in a calculated value R, which is a function of N independent measurements ($X_1, X_2, X_3, \dots, X_N$), can be calculated as **Eqn (9)**:

$$\delta R = \left(\sum_{i=1}^N \left(\frac{\partial R}{\partial X_i} \delta X_i \right)^2 \right)^{\frac{1}{2}} \quad \text{Eqn (9)}$$

Where uncertainty of each measurement is determined using **Eqn (10)**:

$$\delta X_i = \left(\delta X_{i,A}^2 + \delta X_{i,B}^2 \right)^{\frac{1}{2}} \quad \text{Eqn (10)}$$

Where, $\delta X_{i,A}$ is the type-A uncertainty of measurement X_i , which was calculated as the standard deviation of each group of data recorded every 20 minutes. The type-B uncertainty $\delta X_{i,B}$ was either provided by the manufacturer or by the calibration: for example, it is $\pm 0.2^\circ\text{C}$ for the refrigerant temperatures after the calibration of the type-T thermocouples with RTD sensors (PT100).

For the indoor cooling capacity, the uncertainties were 2.2%, 6.2%, and 0.5% for ref-, air-, and chamber-side approaches. Overall, the most reliable approach is the chamber-side due to the simplicity of energy conservation, and the airside is the least accurate one because of the high volumetric flow rate of indoor air.

b. In HP mode

Similarly, the heating capacity of the indoor gas cooler was calculated following:

$$Q_{\text{heating,ref}} = (1 - OCR)\dot{m}_r(h_{\text{cri}} - h_{\text{cro}}) + OCR \cdot \dot{m}_r(h_{\text{coi}} - h_{\text{coo}}) \quad \text{Eqn (11)}$$

$$Q_{\text{heating,air}} = \dot{m}_{\text{cai,dry}}(h_{\text{cao}} - h_{\text{cai}}) \quad \text{Eqn (12)}$$

$$Q_{\text{heating,cham}} = \dot{m}_{\text{id,g}}(T_{\text{id,go}} - T_{\text{id,gi}}) - (\sum P_{\text{id,elec}} + Q_{\text{id,loss}}) \quad \text{Eqn (13)}$$

where, refrigerant enthalpy h_{cri} , oil enthalpies h_{coi} and h_{coo} were determined from the pressures and temperatures measured at the inlet (T_{cri} and P_{cri}) and outlet (T_{cro} and P_{cro}) of the indoor coil. For the transcritical cycle, refrigerant enthalpy h_{cro} was determined from T_{cro} and P_{cro} . The indoor air mass flow rate $\dot{m}_{\text{cai,dry}}$ was calculated using measured air temperature (T_{can}), dew point (T_{dpcn}), and pressure drop (DP_{cn}) at the indoor nozzles. The enthalpies of air h_{cai} and h_{cao} were determined using air temperatures at the indoor coil inlet T_{cai} and exit T_{cao} . In the calculation, dry air properties were used because it is an air-heating process, and no condensation happens. The electrical power consumption $P_{\text{id,elec}}$ of each device inside the indoor chamber was measured using power

transducers. The heat loss through the insulation wall $Q_{id,loss}$ was calculated using the measured UA value of the climate chamber. The mass flow rate of glycol flowing into the indoor chamber $m_{id,g}$ and the inlet and exit glycol temperatures ($T_{id,gi}$ and $T_{id,go}$) were used to calculate the cooling capacity of the glycol heat exchanger.

Also, the outdoor cooling capacity was calculated as below:

$$Q_{cooling,ref} = (1 - OCR)\dot{m}_r(h_{ero} - h_{eri}) + OCR \cdot \dot{m}_r(h_{eoo} - h_{eoi}) \quad \text{Eqn (14)}$$

$$Q_{cooling,air} = \dot{m}_{eai,dry}(h_{eai} - h_{eao}) \quad \text{Eqn (15)}$$

$$Q_{cooling,cham} = \sum P_{od,elec} + Q_{od,loss} - \dot{m}_{od,g}(T_{od,go} - T_{od,gi}) \quad \text{Eqn (16)}$$

In the IHX cycle, the capacity of IHX was calculated using the high-pressure side of the refrigerant-side, since it is in the supercritical or subcooled zone:

$$Q_{IHX,ref} = (1 - OCR)\dot{m}_r(h_{cro} - h_{xri}) + OCR \cdot \dot{m}_r(h_{coo} - h_{xoi}) \quad \text{Eqn (17)}$$

where, refrigerant enthalpy h_{xri} and oil enthalpy h_{xoi} were determined from the pressure and temperature measured at the inlet of the expansion valve used in HP mode. The low-pressure side cooling capacity of the IHX equals the high-pressure heating capacity, assuming no heat loss to the ambient.

The Heating Performance Factor (HPF) of the R744 heat pump system with and without IHX was calculated as:

$$HPF = \frac{Q_{heating,ave}}{P_{comp}} \quad \text{Eqn (18)}$$

where, $Q_{heating,ave}$ is the average of ref-, air-, and chamber-side heating capacity of indoor gas cooler. P_{comp} is the measured electrical power consumption of the VFD-controlled compressor.

Similarly, the uncertainty analysis was conducted using **Eqn (9)** and **Eqn (10)**. For the indoor heating capacity and the HPF of the system, the average uncertainties that include ref-, air-, and chamber-side approaches were 6.9% and 8.1%, respectively.

3.3 Charging experiments and high-side pressure controls in AC/ HP modes

In this section, the role of IHX in a reversible R744 heat pump system operating in both AC and HP mode was studied experimentally. To compare the performance of the system with and without the IHX, two groups of experiments were conducted to determine the optimal charge amounts and the efficiency-maximizing control equations for both basic and IHX systems. Then, the performance of both well-tuned systems was compared and analyzed under the different steady-state conditions in two scenarios.

3.3.1 AC mode with/ without IHX

a. Charging experiments: Group AC-1

The effects of the refrigerant charge amount on the performance of the R744 AC system with and without IHX are presented in the section. **Figure 3.2(a)** and **(b)** show the cooling capacity, compressor power consumption, COP, and refrigerant quality at the evaporator exit for both the basic system and IHX system as the charge amount increases. **Figure 3.2 (c)** and **(d)** present the refrigerant mass flow rate, pressure, and temperature of the refrigerant at different locations of each system during the charging experiments.

The charging experiments were conducted following the procedure in the SAE standard (SAE J2765, 2008). The ambient air temperature T_{cai} and in-cabin air temperature T_{eai} were 40°C.

The in-cabin humidity was controlled to eliminate condensation happening on the in-cabin evaporator. The outdoor air velocity was 3.0 m/s, and the in-cabin air flow rate was 2.5 m/s in the charging experiments. The compressor speed was fixed at 70% of the full speed (1450 rpm). During the charging procedure, CO₂ was added to the system in front of the Acc/IHX in increments of 50g, then stabilized for at least 30 minutes before recording data for another 20 minutes. For each charge amount, at least three different high-side pressures were tested to find the COP-maximizing pressure, which is presented in **Figure 3.2(c) and (d)** as P_{cro} .

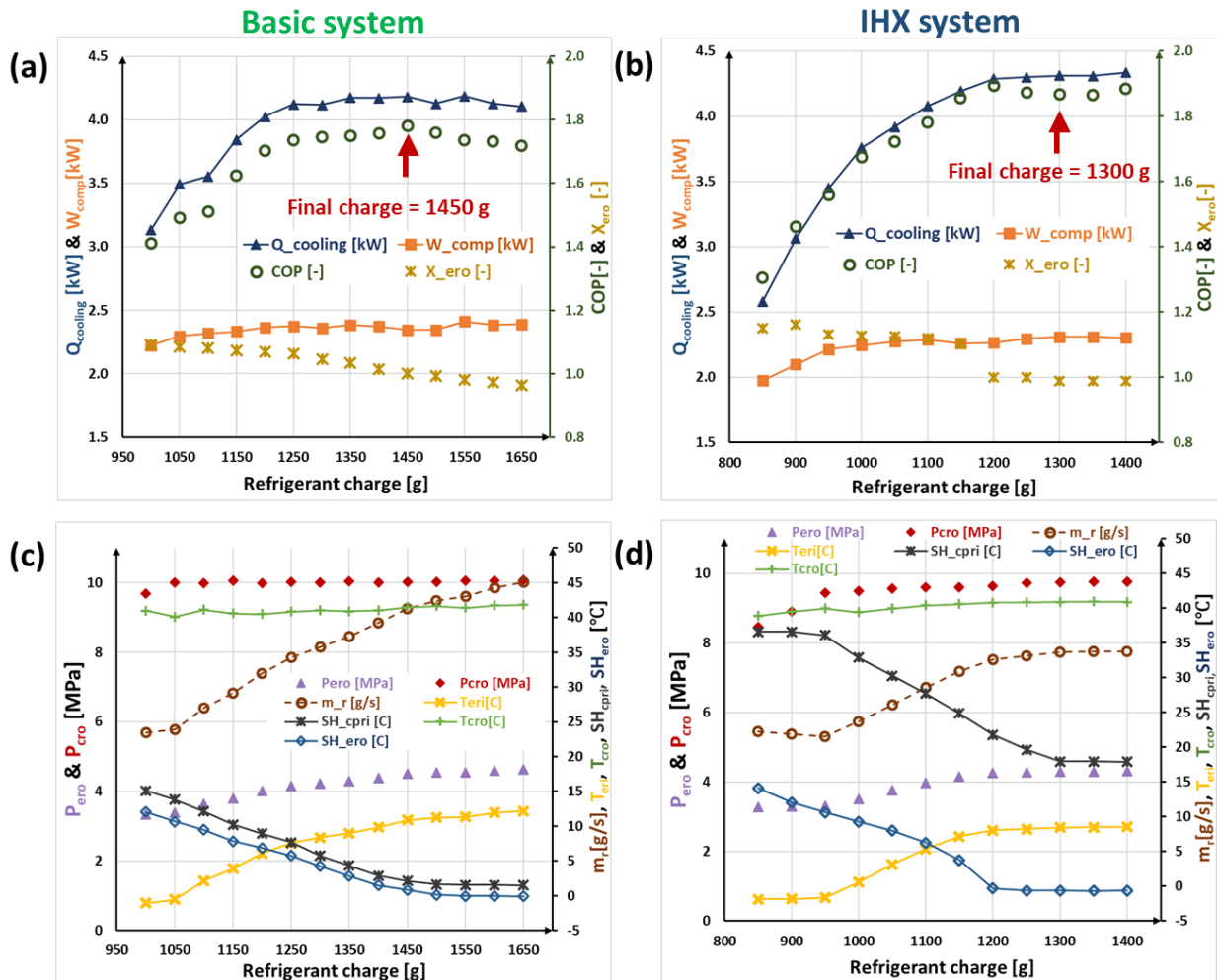


Figure 3.2: Results of charging experiments (Group AC-1) for the basic and IHX AC systems

For both basic and IHX systems, the cooling capacity $Q_{cooling}$ and COP increase as the refrigerant charge amount increases and reach a plateau as the charge amount increases to 1350 g for the basic system and 1200 g for the IHX system, as shown in **Figure 3.2(a) and (b)**. The compressor power consumption W_{comp} has a similar trend but reaches the plateau earlier than cooling capacity and COP. In other words, W_{comp} is sensitive to charge amount when the system is extremely undercharged, but it becomes stable when the refrigerant charge amount is close to the proper charge amount. Also, the evaporator exit quality X_{ero} is decreasing below 1.0 as the charge amount increases to 1500 g for the basic system and 1300 g for the IHX system. The qualities indicate the evaporator is flooded and the heat transfer area is fully used. Also, it shows two-phase refrigerant enters the accumulator, and the system is well charged. Finally, the refrigerant charge amount was optimized to be 1450 g for the basic system to avoid too much liquid refrigerant entering the reciprocating compressor. In the meanwhile, it is chosen to be 1300 g for the IHX system, because liquid refrigerant is stored in the accumulator and the performance becomes stable starting at this point. Comparing the performance of both systems with the final charge, introducing IHX improves the $Q_{cooling}$ and COP by 3.2% and 4.9% at the test condition. This also reduces the charge amount by 150 g, which is 10.3% of the total charge amount of 1450g.

For both systems, the gas cooler exit temperatures T_{cro} are stabilizing at about 41°C when the systems are well-charged; however, the COP-maximizing pressure P_{cro} is 10.1 MPa for the basic system and 9.7 MPa for the IHX system in the test condition. This agrees with previous studies that the COP-maximizing pressure will be reduced by introducing IHX into a transcritical CO₂ system (D. Boewe et al., 1999; Cho et al., 2007). The major difference between the two systems during the charging procedure is: the refrigerant mass flow rate m_r increases and reaches a plateau of 33.8 g/s for the IHX system, but it continues increasing as refrigerant is added to the

basic system. This is because vapor CO₂ with fixed superheat SH_{cpr} leaves the low-pressure side of IHX and enters the compressor for the IHX system during the charging procedure, as shown in **Figure 3.2(d)**. But two-phase refrigerant with low quality ($x_{cpr} = x_{ero}$ for the basic system) enters the compressor for the basic system during the charging procedure (**Figure 3.2(a)**), which might cause a failure of the reciprocating compressor. In this case, the volumetric efficiency of the compressor is significantly increased for the basic system, and the refrigerant mass flow rate is proportional to the charge amount, as shown in **Figure 3.2(c)**.

b. Control equations: Group AC-2

To achieve the optimal system performance in a wide range of operating temperatures, control equations, instead of a fixed EXV opening, were developed for both basic and IHX systems to maximize efficiency. The high-side pressure control theory was discussed in detail and a graphical method was developed using ideal cycle analysis in the paper of Inokuty (1928). Due to the trade-off between the specific cooling capacity and specific compressor work, there exists one COP-maximizing high-side pressure for a fixed refrigerant gas cooler exit temperature. In practice, the equation is developed using experimental data due to various performances of different gas coolers and compressors, like the approaching temperature and isentropic efficiency, etc.

Figure 3.3(a) and **(b)** show the COP curves as functions of the refrigerant pressure at the gas cooler exit P_{cro} for the basic and the IHX system. The control variables including the operating temperatures, airflow rates, charge amounts, and compressor speeds are listed in the charts. The maximum COP and related T_{cro} are highlighted in red and then used to develop the control equations for both systems, as presented in **Figure 3.3(c)**. The results show that both basic and IHX systems reach the maximum efficiency operating in subcritical modes with 1.61 and 1.01°C

of subcooling at the condenser exit when the ambient temperature T_{cai} is 25°C. For higher ambient-temperature conditions, the maximum COPs are obtained in transcritical modes for both systems. Especially, the simple linear control equations developed work well for both subcritical and

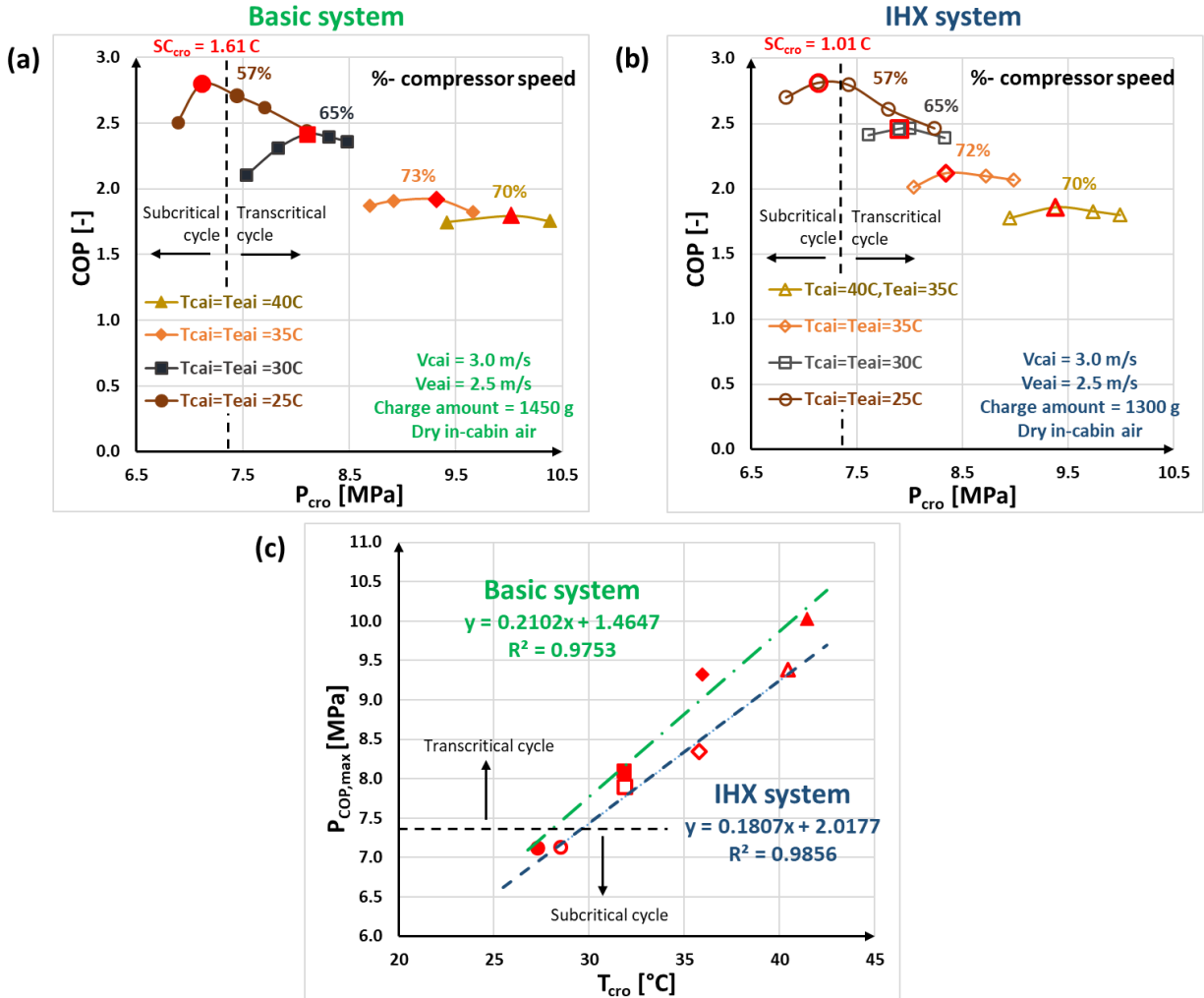


Figure 3.3: The experimental performance data and high-side pressure control equations for the basic and IHX AC systems (Group AC-2)

transcritical modes. Also, **Figure 3.3(c)** shows that introducing IHX reduces $P_{COP,max}$ by 0.24 to 0.67 MPa, and this reduction is more significant when the ambient temperature is increased. This indicates an easier control and improved safety result from introducing an IHX in the R744 AC system.

3.3.2 HP mode with/ without IHX

a. Charging experiments: Group HP-1

The effects of the refrigerant charge amount on the performance of the R744 HP system with and without IHX are presented in the section. **Figure 3.4(a)** and **(b)** show the heating capacity, compressor power consumption, and HPF for the basic system and IHX system as the charge amount increases. **Figure 3.4(c)** and **(d)** present the refrigerant mass flow rate, pressure, and temperature of the refrigerant at different locations of each system during the charging experiments.

The temperatures for ambient and in-cabin air were kept the same ($T_{eai} = 0^{\circ}\text{C}$, $T_{cai} = 10^{\circ}\text{C}$). The humidity in the outdoor chamber was controlled to prevent frosting on the evaporator surface. The outdoor air velocity was 3.0 m/s, and the in-cabin air flow rate was 6.0 kg/min in the charging experiments. The compressor speed was fixed at 80% of the full speed (1450 rpm). During the charging procedure, CO_2 was added to the system in front of the Acc/IHX in increments of 50g and then stabilized for at least 30 minutes before recording data for another 20 minutes. For each charge amount, at least three different high-side pressures were tested to find the COP-maximizing pressure, which is presented in **Figure 3.4(c)** and **(d)** as P_{cro} .

For the basic system, the heating capacity $Q_{heating}$ and compressor power consumption W_{comp} increase as the refrigerant charge amount increases and reach a maximum of 1000 g CO_2 . After that, both parameters start decreasing, but heating capacity is more sensitive to charge amount when the system is overcharged. However, due to similar trends in both parameters, HPF has a small plateau from 850 to 1000 g, as shown in **Figure 3.4(a)**. The system with IHX is less sensitive to the charge amount. In the other words, the heating capacity of the IHX system is more stable than it is for the basic system even when the IHX system is undercharged. The plateau for

heating capacity, compressor power consumption, and HPF is from 850 to 1050 g, as shown in **Figure 3.4(b)**. Now, the system is well charged, and a two-phase refrigerant enters the accumulator. With the last point, the accumulator is flooded, and superheat decreases.

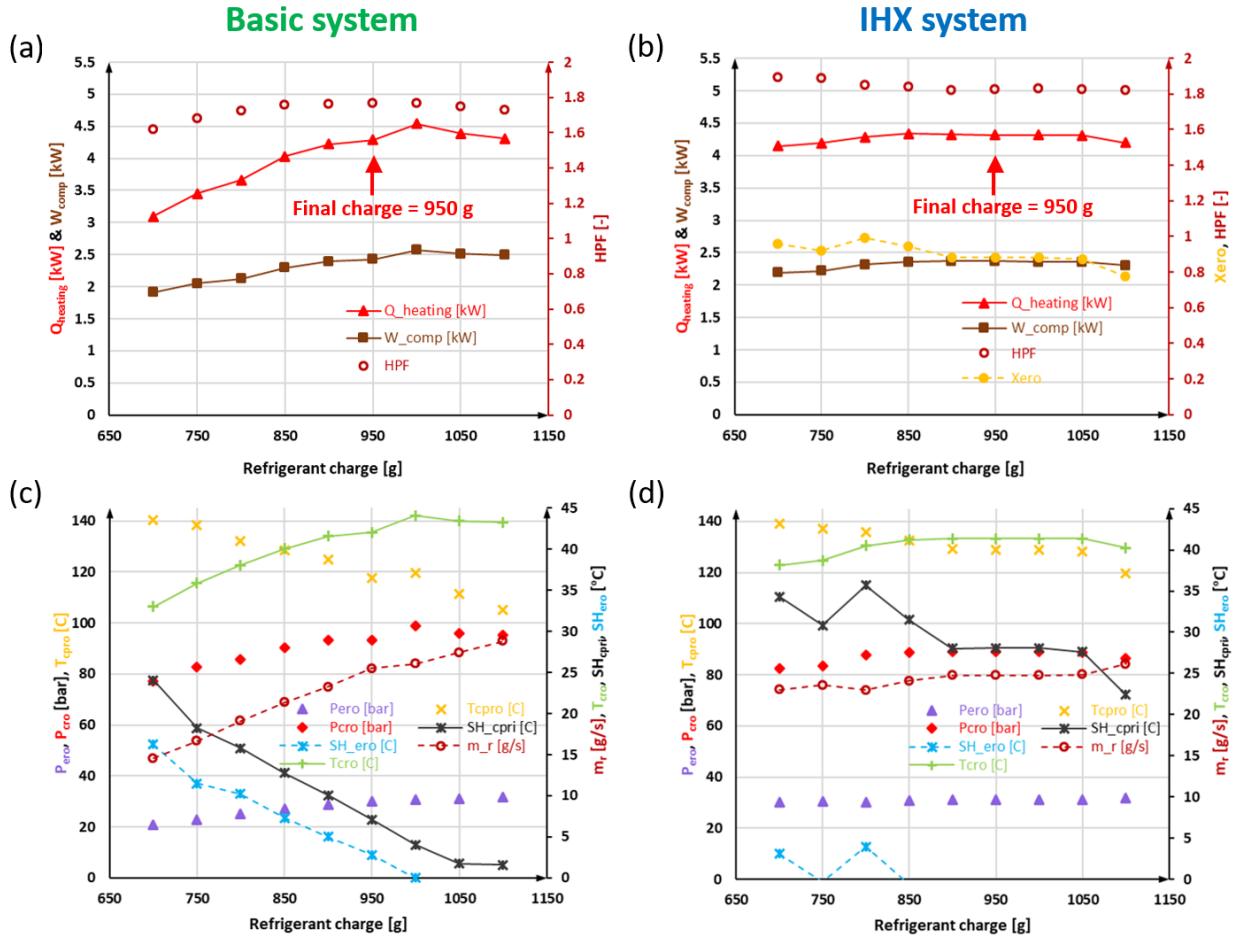


Figure 3.4: Results of charging experiments (Group HP-1) for the basic and IHX HP systems

At the beginning of the charging experiments for both systems, the gas cooler exit temperature T_{cro} is low, and that is beneficial for heating efficiency. For an undercharged system, mass flow rate m_r and high-side pressure P_{cro} are lower, and the limitations in compressor work (high T_{cpro}) are easier to achieve. With adding more charge amount, those parameters start to increase, T_{cpro} decreases, and m_r has more effect on the heating capacity and HPF than T_{cro} . The sensitivity of those parameters is the main difference in the system with and without IHX. For a

basic system, all the parameters are changing very fast with every charge amount, but for an IHX system it is easier to keep a constant value. For example, the mass flow rate for the whole amount range increases two times for the basic system, but only increases 12% for the IHX system, as shown in **Figure 3.4 (c) and (d)**. This is likely due to the more stable evaporating pressure P_{ero} and the suction superheat SH_{cpr} of the IHX system compared to the basic system (**Figure 3.4(c) and (d)**), so that the refrigerant density at the suction point of the compressor is more stable for the IHX system. To avoid sending too much liquid refrigerant into the compressor, another limitation (except $T_{cpr} < 140^{\circ}\text{C}$) is the superheat at the compressor suction. For that reason, the basic system is limited to further adding refrigerant after the 1100g charge due to the constant decrease of the superheat at the evaporator outlet SH_{ero} and the compressor inlet SH_{cpr} .

Finally, the refrigerant charge amount was optimized to be 950 g for the basic system to avoid too much liquid refrigerant entering the reciprocating compressor. In the meanwhile, 950 g is also chosen for the IHX system, because liquid refrigerant is stored in the accumulator and the performance becomes stable starting at this point. Comparing the performance of both systems with the final charge, introducing IHX keeps the $Q_{heating}$ almost the same and improves HPF by 2.8% at the test condition. From this point of view, adding IHX provides no significant benefits to heating capacity and efficiency. However, it provides a much more stable operation of the system.

b. Control equations: Group HP-2

To achieve the optimal HP system performance in a wide range of operating temperatures, control equations, instead of a fixed EXV opening, were developed for both basic and IHX systems. Due to the trade-off between the specific cooling capacity and specific compressor work, there exists one COP-maximizing high-side pressure for a fixed refrigerant gas cooler exit temperature.

A similar rule should exist for HPF-maximizing high-side pressure in the HP mode. For that reason, the same logic and procedure were applied in HP mode, too.

Figure 3.5(a) and **(b)** show the HPF curves as functions of the refrigerant pressure at the gas cooler exit P_{cro} for the basic and IHX systems. The control variables including the operating temperatures, airflow rates, charge amounts, and compressor speeds are listed in the charts. The maximum HPF and related T_{cro} are highlighted in red and then used to develop the control equations for both systems, as presented in **Figure 3.5(c)**.

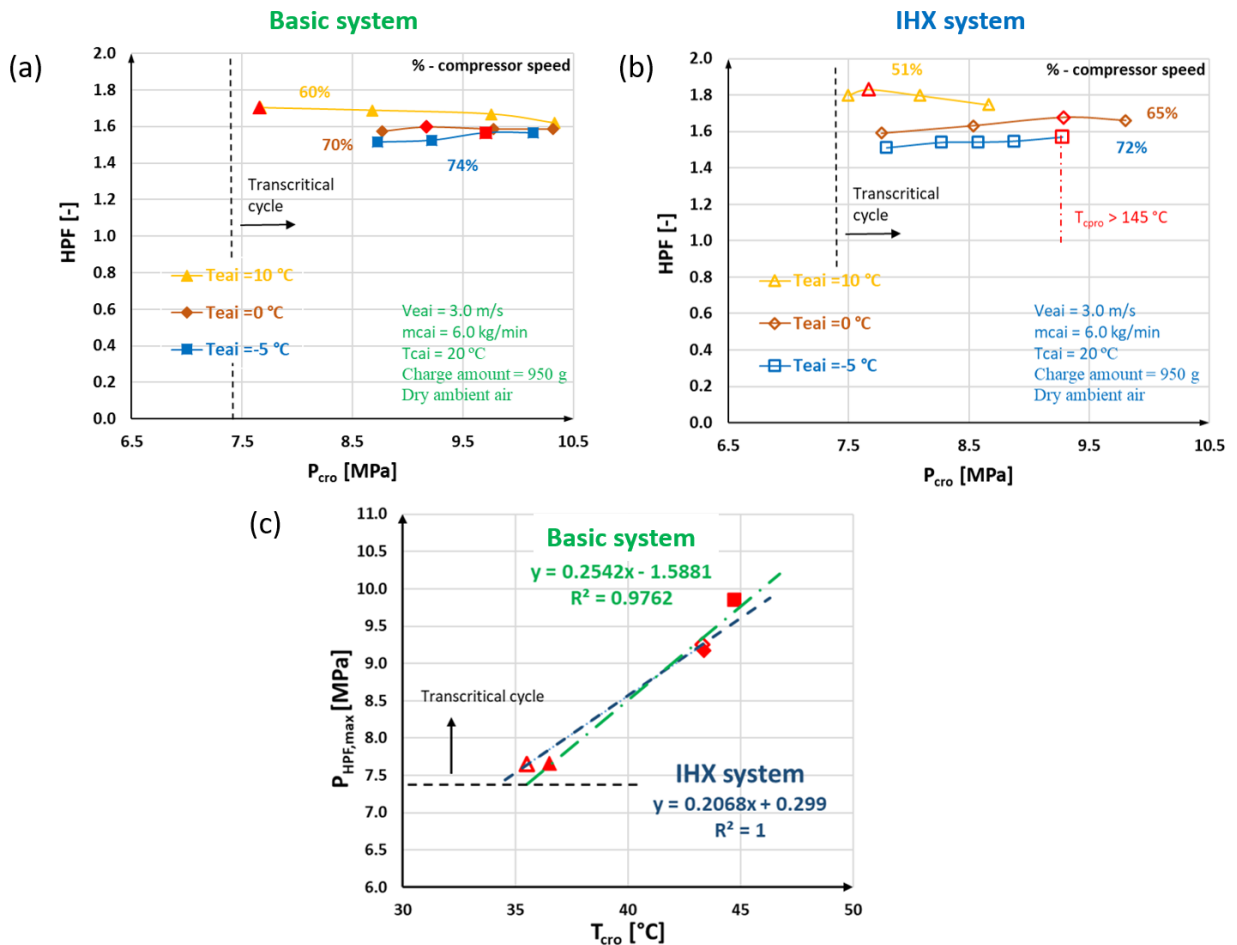


Figure 3.5: The experimental performance data and high-side pressure control equations for the basic and IHX HP systems

The results show that HPF-maximizing control equations could be limited by practical limitations like the discharge temperature of the compressor, especially in high lift conditions (when $T_{eai} = -5^{\circ}\text{C}$). Overall, the deviation between the control equations is smaller in HP than it is in AC mode. Because the airflow rate is relatively lower, and the air inlet temperature is relatively fixed for the indoor gas cooler in steady-state scenarios. The IHX changes the slope of the $P_{HPF,max} - T_{cro}$ line, and after some point, the basic system demands higher HPF-maximizing pressure for higher gas cooler exit temperatures.

3.4 The benefits of IHX in AC/ HP modes under various conditions

3.4.1 AC mode with/ without IHX: Group AC-3 and AC-4

After fine-tuning the basic and IHX systems for AC application, experiments were conducted in the steady-state and pull-down scenarios to investigate the role of IHX in an R744 AC system. In the steady-state scenario, the indoor air inlet (T_{eai}) and exit (T_{eao}) temperatures are fixed to be 25 and 5°C (in other words, the cooling capacity is fixed), but the ambient air temperature T_{cai} varies from 25 to 40°C . In the pull-down scenario, T_{cai} is 40°C , T_{eai} changes from 40 to 25°C , and the indoor air exit temperature T_{eao} is always 20°C lower than T_{eai} (in other words, the cooling capacity is fixed). The compressor speed is adjusted to match the fixed cooling capacity, and all other control parameters are listed in the charts in **Figure 3.6**.

Figure 3.6(a) and **(b)** show the experimental results of the COP of both basic and IHX systems and the COP improvement by introducing IHX. For low ambient temperature conditions ($T_{cai} = T_{eai} = 25^{\circ}\text{C}$), both systems operate in subcritical mode and IHX has a negligible effect (only 1.2% improvement) on the system efficiency. However, IHX improves the system efficiency from

5.9% to 13.8% when both systems are operating in transcritical mode and at higher ambient temperatures.

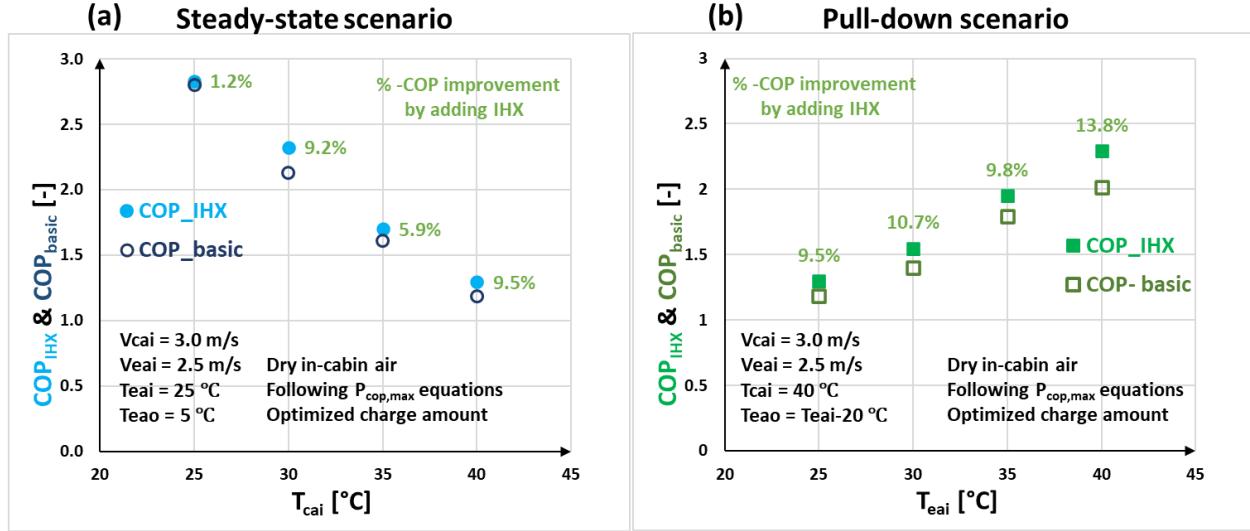


Figure 3.6: The performance data and benefits of IHX in steady-state and pull-down scenarios

Figure 3.7(a) and **(b)** show the experimental results of the P-h plots of both basic and IHX systems in typical subcritical and transcritical operating conditions. For the subcritical condition ($T_{cai} = T_{eai} = 25$ °C), introducing IHX reduces refrigerant quality at evaporator inlet x_{ero} and thus improves the refrigerant distribution in the evaporator. However, IHX has insignificant effects on both evaporating and condensing pressure. As a result, it has limited impacts on reducing the compression ratio and improving compressor efficiencies. In addition, IHX reduces throttling loss negligibly by reducing the temperature at the EXV inlet from 27.82 to 19.68 °C, because the inlet of the EXV is in the subcooled liquid zone for both systems. On the other hand, IHX reduces the refrigerant density at the compressor suction point from 100.3 to 78.79 kg/m³ and thus the refrigerant flow rate from 26.72 to 23.24 g/s. Overall, IHX has limited benefits for R744 AC systems operating in subcritical mode. On the contrary, for the transcritical condition ($T_{cai} = T_{eai} = 40$ °C), introducing IHX significantly lower the high-side pressure and raises the evaporating

pressure by 8% and 7%, respectively. As a result, the compression ratio reduces from 2.21 to 1.89, and the volumetric efficiency and mass flow rate increase, but the compressor power consumption reduces by 15%. Also, the throttling loss is reduced as the compression ratio decreases. In conclusion, IHX improves the performance of the R744 AC system significantly in transcritical mode.

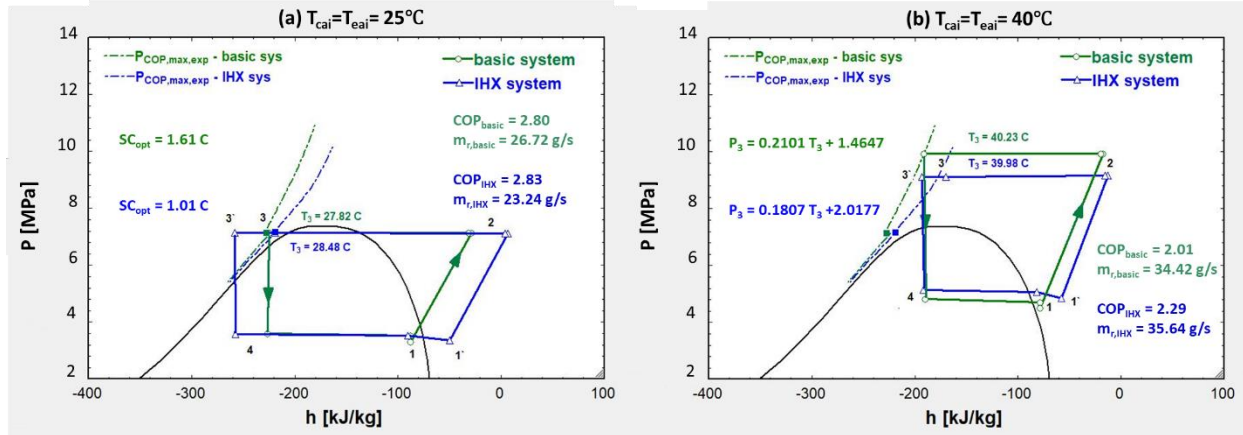


Figure 3.7: The experimental P-h plots of both basic and IHX systems in (a) subcritical and (b) transcritical modes

3.4.2 HP mode with/ without IHX: Group HP-3 and HP-4

Similar to AC mode, experiments were conducted in the steady-state and warm-up scenarios to investigate the role of IHX in an R744 HP system after the fine-tuning. In the steady-state scenario, the indoor air inlet (T_{cai}) and exit (T_{cao}) temperatures are fixed to be 20 and 50°C (fixing the heating capacity), and the ambient air temperature T_{eai} varies from -10 to 10°C, with increments of 5°C. In the warm-up scenario, T_{eai} is 5°C, T_{cai} changes from 5 to 20°C, and the indoor air exit temperature T_{cao} is always 30°C higher than T_{cai} (fixing the heating capacity). The compressor speed is adjusted to match the heating capacity, and all other control parameters are listed in the charts in **Figure 3.8**.

Figure 3.8(a) and **(b)** show the experimental results of the HPF of both basic and IHX systems and the HPF improvement in percentage by introducing IHX. For low ambient temperature conditions ($T_{eai} = -10^\circ\text{C}$), a system with IHX has lower efficiency due to compressor limitations of high discharge temperature. In other words, the high-pressure side control cannot be reached due to the limitation of the discharge temperature. However, in a relatively mild ambient temperature condition, like when $T_{eai} = -5^\circ\text{C}$ and above, IHX improves the system efficiency by 0.7% to 9.8%, and this improvement increases as T_{eai} increases (**Figure 3.8(a)**). Also, during a typical warm-up process with the ambient air $T_{eai} = 5^\circ\text{C}$, IHX improves the system efficiency by up to 10.5%, as shown in **Figure 3.8(b)**. The improvement also increases with the indoor air temperature T_{cai} .

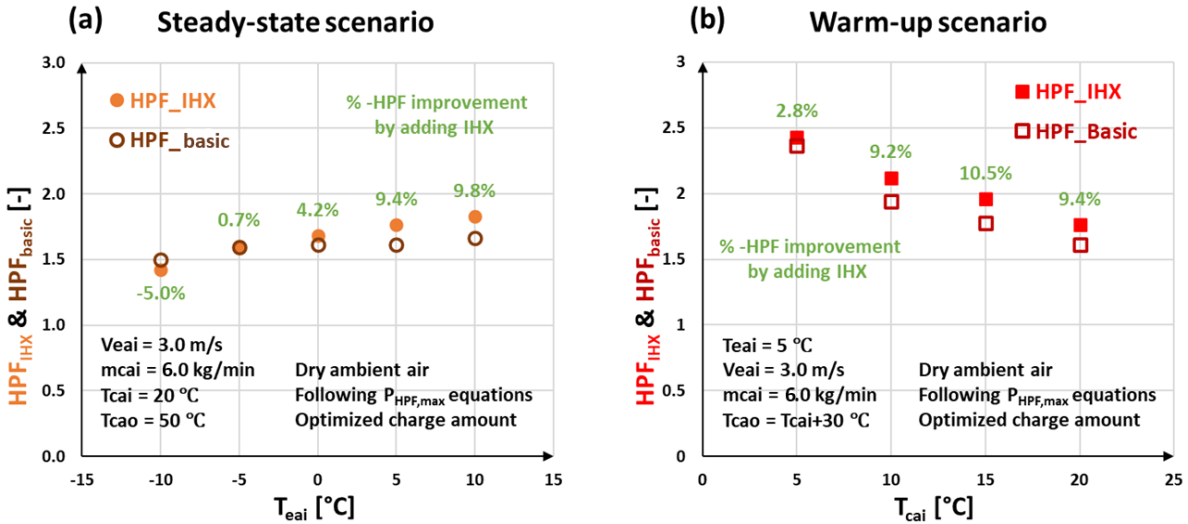


Figure 3.8: The performance data and benefits of IHX in steady-state and warm-up scenarios

Introducing the IHX to the transcritical R744 HP system not only yields a higher efficiency but also contributes to a safer operation of the system. It is known that the low side of the IHX prevents the liquid refrigerant from entering the compressor by heat exchanging with the warm refrigerant exiting the gas cooler. In addition, it has been experimentally confirmed in this dissertation that an R744 HP system with an IHX operates with higher evaporating temperatures

than a basic R744 HP system. As a result, adding an IHX can bring potential benefits like delaying frost accumulating on the surface of the outdoor evaporator. **Figure 3.9(a)** and **(b)** show the evaporating temperatures for both basic and IHX systems in both steady-state and warm-up scenarios. Evaporating temperatures are compared to the dew point temperature T_{dpei} , which is assumed to be the dew point with 90% Relative Humidity (RH) in various outdoor air temperature conditions. The larger difference between the evaporating temperature and T_{dpei} would lead to a faster frosting on the outdoor coil surface. IHX could not fully prevent frosting, especially at the lower ambient temperatures, but it can delay frost growing in the most demanding conditions when T_{eai} is between 0 to 10°C. Also, the evaporating temperature increases faster with T_{eai} and T_{cai} for the IHX system than it does for the basic system. The difference between the evaporating temperatures of the two HP systems could be up to 6.1°C. Though when the ambient temperature T_{eai} equals 0°C, the evaporating temperature for the IHX system is higher than it is for the basic HP system by 2.3°C.

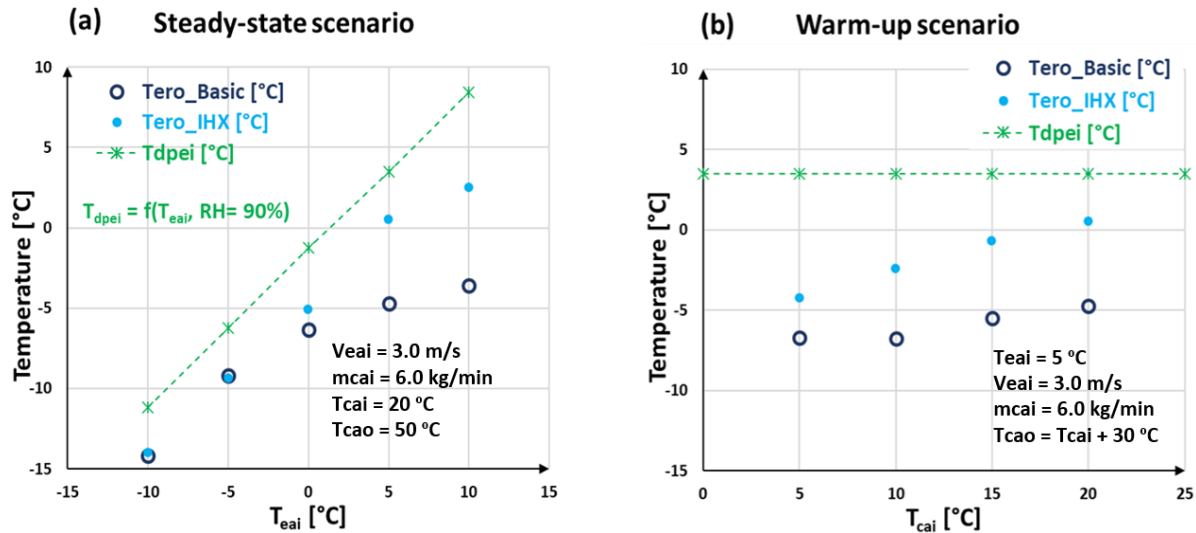


Figure 3.9: The performance data and potential benefits of delaying frosting by introducing an IHX in frosting conditions

In conclusion, a reversible R744 AC/ HP system was built and calibrated to study the role of IHX in both AC and HP operations. The results show:

a. In AC mode

- The IHX AC system requires less refrigerant charge amount than the basic AC system in the most demanding condition- when the outdoor air temperature T_{cai} is 40°C.
- Introducing an IHX reduces $P_{COP,max}$ for the R744 AC system by 0.24 to 0.67 MPa, and this reduction is more significant when the ambient temperature T_{cai} is increased.
- IHX has a negligible effect on system efficiency (COP) when the R744 AC system is operating in subcritical mode, but it improves system efficiency by up to 13.8% when the AC system is operating in transcritical mode and with higher ambient temperatures.

b. In HP mode

- The IHX HP system requires the same refrigerant charge amount as the basic HP system in the defined condition for the charging experiment: outdoor temperature $T_{eai} = 0^{\circ}\text{C}$, indoor temperature $T_{cai} = 10^{\circ}\text{C}$.
- Introducing an IHX does not change $P_{HPF,max}$ much, and this is likely due to the relatively fixed inlet air temperature and the lower airflow rate for the gas cooler in HP mode than they are in the AC mode.
- IHX has a negligible effect on system efficiency (HPF) when the HP system is operating with lower outdoor air temperature T_{eai} , due to the limitation of the compressor discharge temperature and the high-side pressure cannot reach the optimal value. However, IHX can improve the system efficiency (HPF) by up to 10.5% when the HP system is operating in transcritical mode with higher ambient temperatures.

- Introducing an IHX potentially delays frost accumulating on the outdoor evaporator due to an increased evaporating temperature, which can be higher than it is for the basic HP system by 2.3°C when the outdoor air temperature T_{eai} is 0°C.

CHAPTER 4: NUMERICAL INVESTIGATION OF BENEFITS AND BEHAVIOR OF THE INTEGRATED INTERNAL HEAT EXCHANGER AND ACCUMULATOR

In this chapter, the numerical investigation of the benefits and behavior of the accumulator and IHX in the reversible R744 AC/ HP system is presented. The numerical study includes 1) a simple cycle analysis of the role of IHX in different applications; 2) the 1-D modeling of the integrated IHX/Acc using EES (Klein, 2021); 3) the CFD modeling of the accumulator using ANSYS Fluent, considering the effects of the diameter of the oil bleed hole.

4.1 Cycle analysis of the role of IHX

Figure 4.1 shows the temperature-enthalpy (T-h) plots of both sub- and trans-critical CO₂ cycles with and without IHX. The major processes for both cycles are the following:

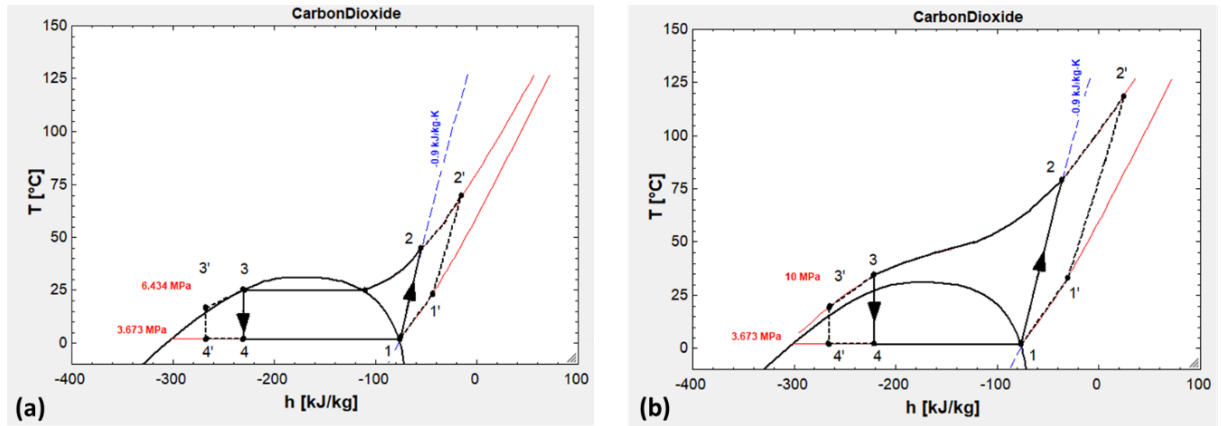


Figure 4.1: T-h plots for CO₂ systems with and without IHX in **(a)** subcritical mode **(b)** transcritical mode

1-2 (1'-2'): Isentropic compression in the compressor

2-3 (2'-3): Heat exchanging with ambient air in the condenser/gas cooler

3-4 (3'-4'): Isenthalpic expansion in the expansion device

4-1 (4'-1): Heat exchanging with indoor air in the evaporator

1-1' and 3-3': Heat exchanging in the low- and high-pressure side of IHX

4.1.1 The assumptions

The role of IHX in an R744 vapor compression cycle was studied theoretically for both sub- and transcritical modes using EES programs (Klein, 2021). The following assumptions were used for subcritical cycles (**Figure 4.1(a)**):

- a. The vapor compression cycles are in steady states.
- b. Saturated vapor refrigerant leaves the evaporator ($x_1=1$).
- c. The compression process is isentropic ($s_1=s_2$ and $s_1'=s_2'$).
- d. Saturated liquid refrigerant leaves the condenser ($x_3=0$).
- e. The expansion process is isenthalpic ($h_3=h_4$ and $h_3'=h_4'$).
- f. The pressure drop is negligible for all heat exchangers.
- g. The heat loss to the ambient is negligible for all components.
- h. The changes in kinetic and potential energy are negligible.
- i. The effectiveness of IHX η_{IHX} is fixed to be 0.25/0.50/1.00.

The only different assumptions used for transcritical cycles (**Figure 4.1(b)**) are: Instead of assumption d (fixed condensing temperature T_3 below the critical temperature of CO₂ and $x_3=0$), the refrigerant temperature at the gas cooler exit T_3 is fixed to be higher than the critical

temperature of CO₂, and the high-side pressure P_3 is optimized to be the COP-maximizing high side pressure for both basic and IHX cycles in transcritical modes.

4.1.2 The equations

The equations used in the cycle analysis are the following:

- 1) The specific cooling capacity of refrigerant for basic cycle

$$q = h_1 - h_4 \quad \text{Eqn (19)}$$

- 2) The specific compression work for the basic cycle

$$w = h_2 - h_1 \quad \text{Eqn (20)}$$

- 3) The COP for basic cycle

$$COP_{basic} = \frac{q}{w} \quad \text{Eqn (21)}$$

- 4) The improvement of specific cooling capacity due to IHX

$$dq = (h_1 - h_{4'}) - (h_1 - h_4) = h_4 - h_{4'} \quad \text{Eqn (22)}$$

- 5) The increased part of specific compression work due to IHX

$$dw = (h_{2'} - h_{1'}) - (h_2 - h_1) \quad \text{Eqn (23)}$$

- 6) The COP for IHX cycle

$$COP_{IHX} = \frac{q + dq}{w + dw} \quad \text{Eqn (24)}$$

7) The effectiveness of the IHX

$$\eta_{IHX} = \frac{(T_1' - T_1)}{(T_3 - T_1)} \quad \text{Eqn (25)}$$

The effectiveness of the IHX is calculated using the temperature change of the low-pressure side because the specific heat of saturated vapor at the exit of the evaporator is smaller than it is for either saturated liquid CO₂ at the exit of the condenser or supercritical CO₂ at the exit of the gas cooler.

4.1.3 The variables in the cycle analysis

To investigate the effects of the operating conditions and the effectiveness of IHX on the potential benefits of IHX, the following parameters were studied in the cycle analysis:

1. The heat source is assumed to be an air-conditioner, refrigerator, and freezer. Therefore, the evaporating temperature ranges from -40 to 12°C. Although it might not be efficient and practical to use a single-stage cycle for the freezer, it is still listed in the results for comparison.
2. The heat sink for all cycles is ambient air at various temperatures. Therefore, the condensing temperature or gas cooler exit temperature T_3 is assumed to be either 21/25/29°C for subcritical mode or 35/40/45°C for transcritical mode.

3. Also, for transcritical CO₂ cycles, the high-side pressure is optimized based on maximizing the COP for each combination of T_1 , T_3 , and η_{IHX} . Only the COP-maximizing P_3 is presented in the results section.

Table 4.1 summarizes the simulation conditions for the cycle analysis.

Table 4.1: Operating conditions of the cycle analysis

	Application	T_1 [°C]	T_3 [°C]		η_{IHX} [-]	P_3 [MPa]	
			Sub-	Trans-		Sub-	Trans-
Group 1	Air-conditioner	2 to 12	21/25/29	35/40/45	0.25/0.50/1.00	$P_{sat}(T_3)$	$P_{COP,max}$
Group 2	Refrigerator	-15 to 2					
Group 3	Freezer	-40 to -15					

4.1.4 The results of the cycle analysis

Figure 4.2 shows the ratio between the COP of the IHX cycle and the COP of the basic cycle in both sub- and transcritical modes with various operating temperatures and the effectiveness of IHX. The ratio COP_{IHX}/COP_{basic} represents the potential improvement of system efficiency: if the ratio is larger than 1, it is beneficial to install IHX for this condition and vice versa. In the one group of nine charts in **Figure 4.2**, the evaporating temperature T_1 increases from -40 to 12°C from left to right for the three columns of charts. The different colors of the legends represent different condensing or gas cooler exit temperatures T_3 : it is 21, 25, and 29°C for subcritical mode, and it is 35, 40, and 45°C for transcritical mode. The effectiveness of the IHX η_{IHX} is 0.25, 0.50, and 1.00 for the three rows of charts, respectively.

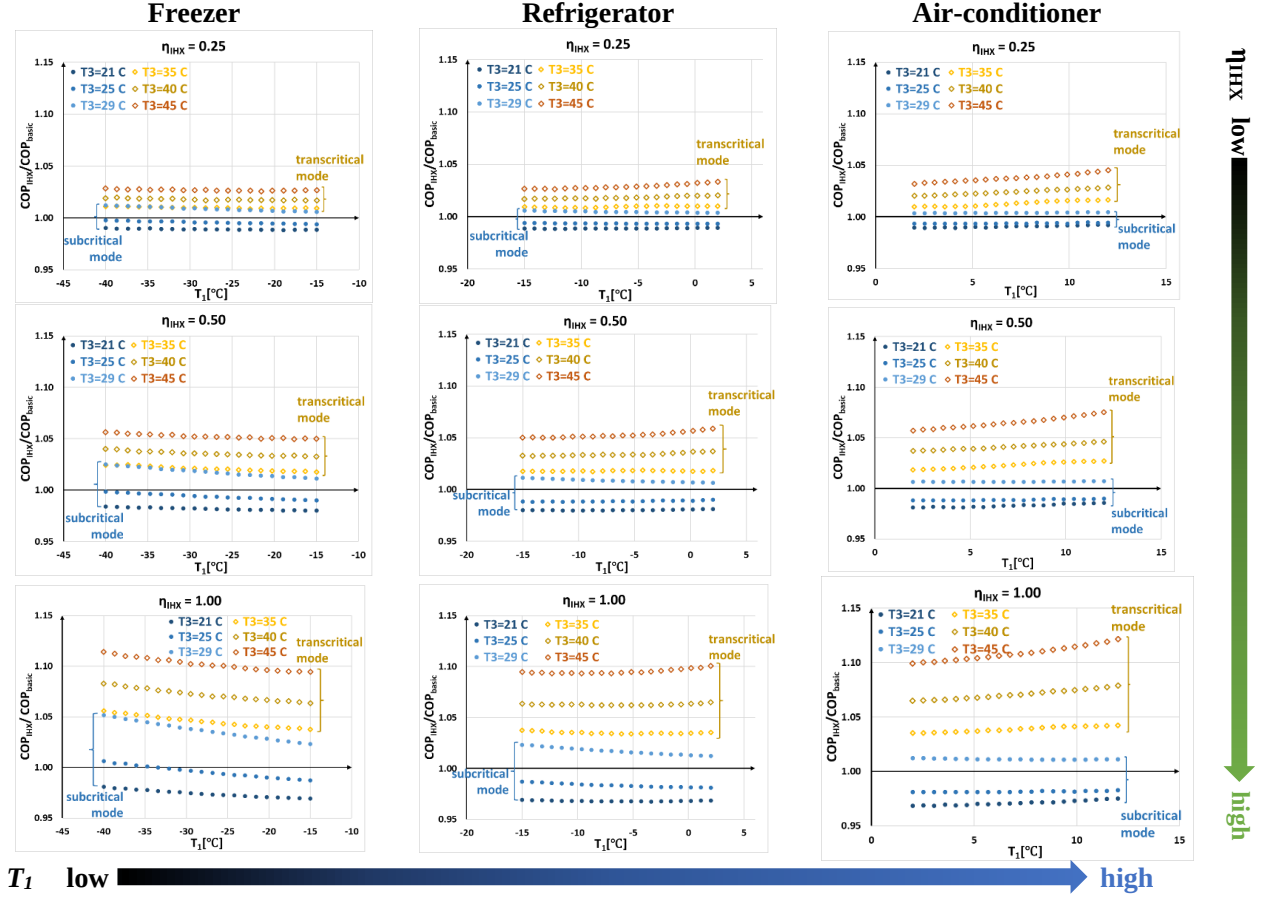


Figure 4.2: The ratio of COPs of IHX and basic CO₂ cycles in sub- and transcritical modes with various operating temperatures and η_{IHx}

a. Effects of the operating temperatures T_3 and T_1

The effects of the condensing/gas cooler exit temperature T_3 on the ratio COP_{IHx}/COP_{basic} is dominating compared to the other two variables: T_1 and η_{IHx} . As T_3 increases from 21 to 45°C, the ratio COP_{IHx}/COP_{basic} changes from 0.97 to 1.12 when T_1 is 12°C and η_{IHx} is 1.00 (the right bottom chart in **Figure 4.2**). Also, the ratio is higher than 1 when T_3 is higher than or equal to 29°C as shown in all nine charts in **Figure 4.2**. This indicates that IHX is beneficial to the system efficiency when T_3 is approaching or higher than the critical point of CO₂.

The effects of T_3 can be explained by the proof as follows:

- i. The ratio COP_{IHX}/COP_{basic} should be higher than 1 so that IHX improves the efficiency of the CO_2 cycle

$$\frac{COP_{IHX}}{COP_{basic}} > 1 \quad \text{Eqn (26)}$$

- ii. By combining **Eqn (21)** and **Eqn (24)**

$$\frac{COP_{IHX}}{COP_{basic}} = \frac{\frac{q+dq}{w+dw}}{\frac{q}{w}} = \frac{\frac{q+dq}{w+dw}}{\frac{q}{w}} = \frac{1+\frac{dq}{q}}{1+\frac{dw}{w}} > 1 \quad \text{Eqn (27)}$$

- iii. **Eqn (27)** can be converted to

$$1 + \frac{dq}{q} > 1 + \frac{dw}{w} \quad \text{Eqn (28)}$$

$$\frac{dq}{q} > \frac{dw}{w} \quad \text{Eqn (29)}$$

$$\frac{dq}{dw} > \frac{q}{w} = COP_{basic} \quad \text{Eqn (30)}$$

Therefore, when the efficiency of the basic cycle COP_{basic} is low, there are more potential benefits to introduce an IHX into the CO_2 cycle. Also, the theoretical efficiency of the Carnot refrigeration cycle is

$$COP_{R,carnot} = \frac{T_1}{T_3 - T_1} \quad \text{Eqn (31)}$$

As a result, the COP_{basic} decreases, and the COP improvement due to introducing IHX increases when T_3 increases. In addition, the effects of T_3 are related to the high throttling loss of CO_2 when the refrigerant at the expansion device inlet for the basic cycle is near or higher than the critical point. Introducing an IHX can decrease the temperature of the refrigerant at the expansion device inlet and reduce the throttling loss significantly. Also, the high T_3 can be translated into a few scenarios for different applications: 1) the inlet air temperature of the outdoor condenser/gas cooler is or higher than 29°C; 2) the size of the outdoor condenser/gas cooler is limited by the onboard space, and therefore, the approaching temperature is high.

On the other hand, the impacts of evaporating temperature T_1 (no matter positive or negative effects) on the ratio of COPs is negligible in subcritical mode and small in transcritical mode, especially when the effectiveness of IHX is relatively low: The ratios are changing by less than 0.02 when T_1 decreases from 12 to -40°C and η_{IHX} equals 0.25, as shown by the first row of charts in **Figure 4.2**. In addition, the trends of COP_{IHX}/COP_{basic} curves are different for sub- and transcritical modes. In subcritical mode, the ratio COP_{IHX}/COP_{basic} monotonically increases from 1.01 to 1.05 when T_1 decreases from 12 to -40°C, η_{IHX} equals 1.00 and T_3 equals 29°C, as shown by the light blue lines in the last row of charts in **Figure 4.2**. In transcritical mode, however, the ratio COP_{IHX}/COP_{basic} firstly decreases from 1.12 to 1.09 and then increases back to 1.11, when T_1 decreases from 12 to -40°C, η_{IHX} equals 1.00 and T_3 equals 45°C, as shown by the orange lines in the last row of charts in **Figure 4.2**.

b. Effects of the effectiveness of the IHX η_{IHX}

The effects of the effectiveness of IHX η_{IHX} on the ratio COP_{IHX}/COP_{basic} is the second significant among the studied parameters, following the impacts of T_3 . As shown from the first to

the last row in **Figure 4.2**, both the positive and negative effects on the ratio $\text{COP}_{\text{IHX}}/\text{COP}_{\text{basic}}$ by introducing IHX are more significant when η_{IHX} increases from 0.25 to 1.00: the maximum benefit increases from 4.5% to 12.2%, and the maximum decrease of COP reduces from -1.1% to -3.2%.

In conclusion, the impacts of IHX on COP are always positive or negligible in a transcritical cycle based on the simple cycle analysis, and these benefits increase with T_3 and η_{IHX} . However, only a high-effectiveness IHX can improve the COP of the CO_2 subcritical cycle by up to 5% when the difference between the evaporating temperature T_1 and condensing temperature T_3 is large (the left bottom chart in **Figure 4.2**). Because the $\text{COP}_{\text{basic}}$ decreases as T_3 increases and the throttling loss is increasing with the pressure and temperature differences between low- and high-pressure sides of the CO_2 cycle. Overall, IHX benefits the COP most when $\text{COP}_{\text{basic}}$ is low and T_3 is high due to the environment temperature or space limitation based on the cycle analysis.

4.1.5 Comparison of theoretical and experimental results

In this section, the experimental results in chapter 3 are compared with the cycle analysis results. **Figure 4.3** presents the comparison of the theoretical and experimental results of the ratio $\text{COP}_{\text{IHX}}/\text{COP}_{\text{basic}}$ and the T-h plots. Due to the experimental limitation, the comparison includes three typical operating conditions with the same evaporating temperature ($T_1 = 0^\circ\text{C}$), same IHX effectiveness ($\eta_{\text{IHX}} = 0.50$), but different condensing/gas cooler exit temperatures (T_3 varies from 29 to 41°C). Because the two-phase refrigerant exits the evaporator, T_{ero} in the experiments is considered as T_1 in the cycle analysis, and the deviation is less than 0.2°C for all data in **Figure 4.3(a)**. T_{cro} in the experiments is considered as T_3 in the cycle analysis and used as the x-axis in **Figure 4.3(a)**. Although the effectiveness of the IHX η_{IHX} is fixed to be 0.50 in the cycle analysis,

η_{IHX} varies from 0.49 to 0.82 for different conditions in the experiments, because it depends on the mass flow rate and the refrigerant quality/superheat at the evaporator exit.

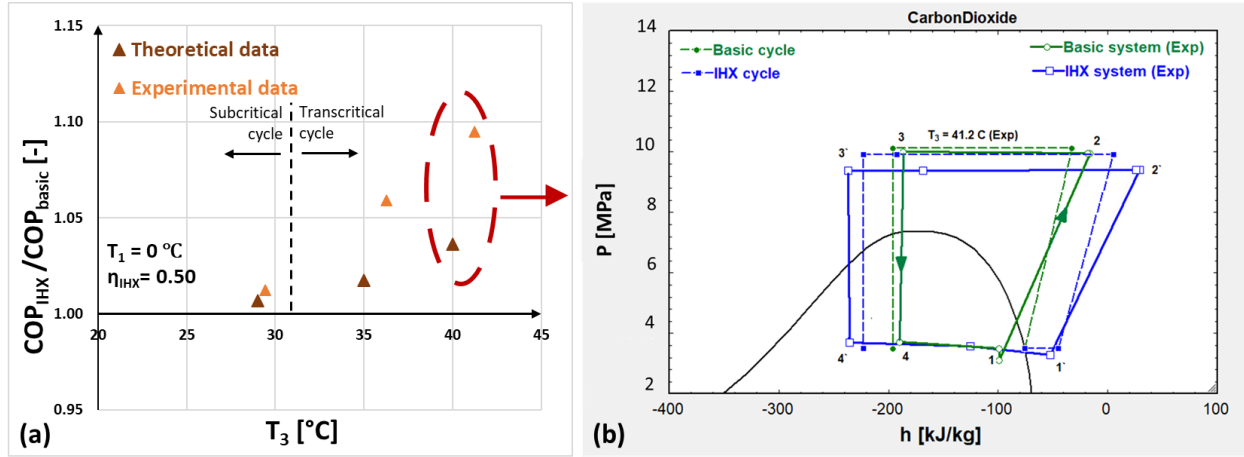


Figure 4.3: Comparison of results of experiments and cycle analysis of (a) COP_{IHX}/COP_{basic} (b) T-h plots

Overall, the results of cycle analysis underestimate the COP improvement of introducing an IHX into an R744 vapor compression system, and the deviation between the theoretical and experimental results increases with T_3 (**Figure 4.3(a)**). Experimental data show a more than doubled COP improvement due to IHX when T_3 for both basic and IHX systems are 41.2°C. Because simplified theoretical analysis neglects the following factors: First, the optimal high-side pressure P_3 drops more by introducing IHX in experimental results. As shown in **Figure 4.3(b)**, P_3 drops by 0.63 and 0.23 MPa for experimental and theoretical results when adding IHX. Second, the compressor isentropic efficiency increases from 0.47 to 0.59 as the high-side pressure and compression ratio decreases in experiments, though it is isentropic compression in the cycle analysis. Third, the refrigerant distribution is improved, and the pressure drop across the evaporator dP_{er} decreases from 0.21 to 0.12 MPa by adding IHX, since both the inlet and exit quality of the evaporator decrease by 0.21 and 0.10. However, the pressure drop of refrigerant is considered negligible in the cycle analysis. In general, it is difficult to accurately predict the

aforementioned factors for a wide range of operating conditions, and sometimes those factors are neglected in numerical studies.

4.2 1-D model of the integrated IHX and accumulator

4.2.1 Apparatus and Test Conditions

Figure 4.4 shows the section view of the integrated IHX/Acc used in this dissertation. The internal plastic cylinder serves as the accumulator, and a helical tube is arranged in the gap between the accumulator and the stainless-steel housing shell and forms a force-fit contact. The two-phase refrigerant from the evaporator outlet flows through the inlet tube and enters the accumulator and then it is separated into liquid and vapor. In the normal operating conditions, only vapor flows through the gas filter on the top-end of the J-tube, but the mixture of liquid refrigerant and oil flow through the bleed hole on the bend of the J-tube. Then, the two-phase refrigerant exits the accumulator and flows radially on the top cover of the accumulator, entering the spiral flow channel between the accumulator, housing shell, and the tube coil. After being heated by the high-pressure side refrigerant inside the helical tube, the low-pressure side vapor flows towards the bottom of the IHX and is drawn off through the low-pressure outlet tube. On the other hand, the high-pressure side refrigerant from the gas cooler enters via the inlet tube at the bottom and flows upwards inside the spiral tube, forming a counter-flow internal heat exchanger. The key parameters of this IHX/ Acc are shown in **Table 4.2**.

Experiments were conducted over a wide range of indoor and outdoor temperatures, airflow rates, and different compressor speeds, as listed in **Table B.1** and **Table B.2** in Appendix B). For each AC or HP operating condition, the EXV opening was controlled to

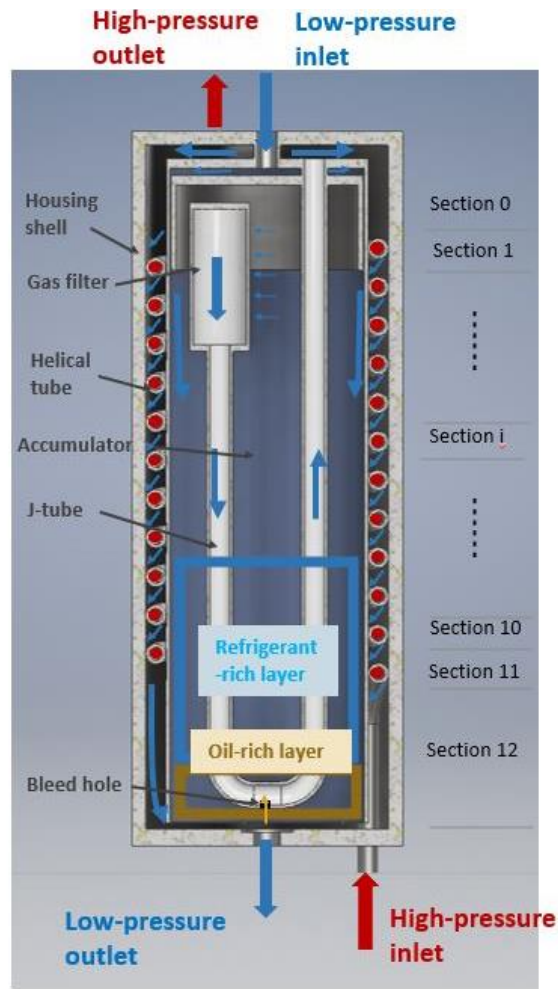


Figure 4.4: Section view of the integrated IHX/Acc used in this dissertation (blue and red arrows represent the flow direction of low- and high-pressure side refrigerant)

Table 4.2: Key dimensions of the integrated IHX and accumulator

Parameter	Dimension
Overall height (mm)	247
Overall diameter (mm)	80
IHX high-side tube length (mm)	2800
IHX high-side tube spiral numbers× pitch (mm)	11×11.4
IHX high-side tube OD× thickness (mm)	6×1
Accumulator effective volume (mL)	404

modify the high-side pressures, which bracketed the maximum efficiency and capacity in most of the conditions. However, in some cases, the max values could not be achieved due to the limitation of discharge temperature (140°C) and pressure (12 MPa) of the compressors. In addition, the refrigerant mass flow rate, oil circulating rate (OCR), and inlet and outlet refrigerant states of the integrated IHX/Acc in AC and HP modes are listed in **Table B.3** and **Table B.4** (Appendix B). Especially, the OCR of each condition was measured using an oil sampler during the experiments. Also, the uncertainty of the measurement instruments is listed in **Table 3.2** and the relative uncertainty of the heat transfer rate of the IHX Q_{IHX} is 3.3%. In the model, the effect of oil concentration on the heat transfer and pressure drop is taken into consideration, because 4% of oil concentration can decrease the heat transfer coefficient and increase the pressure drop significantly, as proposed by researchers (Gao and Honda, 2006; Kim et al., 2010; Li et al., 2014). The qualities at low-pressure side inlet $x_{IHX,l,i}$ are calculated using measured refrigerant states at the suction of the compressor and the IHX capacity Q_{IHX} calculated based on the high-pressure side states.

4.2.2 1-D model of the integrated IHX and accumulator

a. Internal heat exchanger model

The internal heat exchanger is simulated using the finite volume method, similar to the models in the literature (J. Li and Hrnjak, 2017; W. Li and Hrnjak, 2018). As shown in **Figure 4.4**, the IHX is divided into 13 coils, and heat transfer occurs in section 1 to 11. In the model, the state of refrigerant is checked before each calculation: for higher accuracy, each coil will be divided into five elements if the low-pressure side refrigerant is two-phase.

There are a few assumptions in the modeling: First, the equations neglect heat conduction between the high-pressure side and liquid refrigerant inside the accumulator. The contact thermal

resistance between the helical tube and accumulator plus the conduction thermal resistance of the plastic (Nylon) accumulator wall is about 11 times higher than the thermal resistance between the low- and high-pressure side refrigerant in the IHX. Secondly, the heat conduction through the housing shell is negligible.

The low-pressure side refrigerant is most likely two-phase fluid exiting the accumulator. Based on the calculation, the low-pressure side inlet quality $x_{IHX,l,i}$ varies from 0.85 to 0.95 in normal operating conditions. However, in some cases, like start-up conditions in heat pump mode, the accumulator might be flooded and $x_{IHX,l,i}$ can be as low as 0.60. But the cold stream is warmed up by the hot stream and the superheat at the compressor suction point is always higher than 3 °C. For the high-pressure side in transcritical cycles, the refrigerant is either supercritical or subcooled liquid. Therefore, for all aforementioned single-phase regions, like the supercritical or subcooled CO₂ in the high-side tube, or the subcooled liquid or superheated vapor CO₂ in the low-side flow channel, the heat transfer coefficients of pure refrigerant are calculated using the Gnielinski correlation (Gnielinski, 1976):

$$Nu_{1-ph} = \frac{(f_D/8)(Re_{1-ph}-1000)Pr_{1-ph}}{1+12.7(f_D/8)^{0.5}(Pr_{1-ph}^{2/3}-1)} \quad \text{Eqn (32)}$$

$$0.5 < Pr_{1-ph} < 2000, 2300 < Re_{1-ph} < 500000$$

Then, the correction of oil effects on the heat transfer coefficient (htc) should be considered. For example, the decreasing ratio of heat transfer coefficient of the mixture against the pure CO₂ is applied in the high-pressure side gas cooling htc calculation (Dang et al., 2007). (Table 2, ID= 4 mm). On the other hand, for the two-phase evaporating region, the convective heat transfer coefficient of pure refrigerant is firstly calculated using the Wattelet-Chato correlation (Wattelet,

1994), and then corrected with the decreasing ratio from the paper of Katsuta et. al (Katsuta et al., 2008):

$$h = F_{cv} h_{cv} \quad \text{Eqn (33)}$$

$$h_{cv} = 0.023 \frac{k_{liq}}{D} \text{Re}_{liq}^{0.8} \text{Pr}_{liq}^{0.4} \quad \text{Eqn (34)}$$

$$F_{cv} = 1 + 1.925 X_{tt}^{-0.83} \quad \text{Eqn (35)}$$

where, X_{tt} is the Lockhart-Martinelli parameter.

$$X_{tt} = \left(\frac{1-x}{x} \right)^{0.9} \left(\frac{\rho_{vap}}{\rho_{liq}} \right)^{0.5} \left(\frac{\mu_{liq}}{\mu_{vap}} \right)^{0.1} \quad \text{Eqn (36)}$$

In this study, the nucleate boiling heat transfer is negligible in the IHX, because $x_{IHX,l,i}$ is higher than the onset of dry-out quality. Especially, due to the oil concentration (2-4%) in the R744 system, the dry-out point moves to the low-quality region. According to Katsuta et.al (Katsuta et al., 2008), when the evaporating temperature is 0 or 10 °C, mass flux $G = 400 \text{ kg}/(\text{m}^2\text{-s})$ and heat flux $q = 10 \text{ kW}/\text{m}^2$, the onset of dry-out is 0.5 and 0.8 with the oil concentration of 4% and 2%, respectively. Therefore, for all the operating conditions, forced convection is the dominating heat transfer mechanism in the low-pressure side of the IHX. Also, both Katsuta et al. (Katsuta et al., 2008) and Li's (Li et al., 2014) data confirmed that the post-dryout heat transfer is not affected by the oil addition. Therefore, in the post-dryout two-phase zone, the effect of OCR is neglected in the model.

The total pressure drop is composed of two parts: the frictional part and the acceleration part. For the first part, the Fanning friction factors of pure refrigerant for the single-phase region are calculated using the Churchill correlation (Churchill, 1977):

$$f = 8 \left[\left(\frac{8}{\text{Re}_{1-ph}} \right)^{12} + \left[\left(2.457 \ln \left(\frac{1}{\left(\frac{7}{\text{Re}_{1-ph}} \right)^{0.9} + 0.27 \varepsilon} \right) \right)^{16} + \left(\frac{37530}{\text{Re}_{1-ph}} \right)^{16} \right]^{\frac{3}{2}} \right]^{\frac{1}{12}} \quad \text{Eqn (37)}$$

where, ε is the relative roughness.

The friction factors of pure refrigerant in the two-phase zone are calculated using the Souza correlation (Souza, 1995):

$$f = f_{lo} F \quad \text{Eqn (38)}$$

where, f_{lo} is the friction factor assuming only liquid flowing through using Churchill correlation.

Factor F is calculated using the following equations:

$$\gamma = \left(\frac{v_{vap}}{v_{liq}} \right)^{0.5} \left(\frac{\mu_{vap}}{\mu_{liq}} \right)^{0.125} \quad \text{Eqn (39)}$$

$$F = 1 + (\gamma^2 - 1) X^{1.75} (1 + 0.952 \gamma X_{tt}^{0.4126}) \quad \text{Eqn (40)}$$

The acceleration pressure drop is calculated using the homogeneous model, and the void fraction is determined by the Zivi correlation (Zivi, 1964):

$$\alpha = \frac{1}{1 + \left(\frac{1-x}{x} \right) \left(\frac{\rho_{vap}}{\rho_{liq}} \right)^{0.67}} \quad \text{Eqn (41)}$$

$$\Delta P_{acc} = G^2 \left\{ \left[\frac{x_o^2}{\rho_{vap} \alpha_o} + \frac{(1-x_o)^2}{\rho_{liq} (1-\alpha_o)} \right] - \left[\frac{x_i^2}{\rho_{vap} \alpha_i} + \frac{(1-x_i)^2}{\rho_{liq} (1-\alpha_i)} \right] \right\} \quad \text{Eqn (42)}$$

Then, the ratios between the pressure drop of the CO₂-oil mixture and the pressure drop of pure CO₂ are applied in the model to compensate for the effects of oil droplets entrained in the gas phase as quality increases, based on the data from Katsuta et al. (2008).

b. Accumulator model

The accumulator model aims to simulate the pressure loss inside the accumulator, mainly the pressure drop of the two refrigerant streams: one is the vapor (in some cases two-phase) refrigerant flows through the top end of the J-tube, and the other is the mixture of lubricant and liquid CO₂ flows through the oil bleed hole, as the following equation:

$$P_{Acc} - dP_{Jtube,contra} - dP_{Jtube,fri} = P_{Acc} + dP_{stat,liq,Acc} - dP_{BH,mix} \quad \text{Eqn (43)}$$

where, P_{Acc} is the vapor pressure inside the accumulator; $dP_{Jtube,contra}$ is the contraction pressure loss calculated based on the equations in Chapter 2 of Chen's book (Chen and Tseng, 2012); $dP_{Jtube,fri}$ is the frictional pressure drop inside the J-tube between the top inlet and the bleed hole, which is calculated in the same way as in the IHX; $dP_{stat,liq,Acc}$ is the hydrostatic pressure difference calculated with this equation:

$$dP_{stat,liq,Acc} = \rho_{liq} g (h_{liq} - h_{BH}) \quad \text{Eqn (44)}$$

where, ρ_{liq} is the saturated liquid CO₂ density with the vapor pressure of P_{Acc} ; g is the gravitational acceleration; h_{liq} is the liquid level inside the accumulator to be solved, and h_{BH} is the height of the bleed hole from the bottom of the accumulator. Here, it is assumed the addition of oil has a marginal impact on the density of the liquid inside the accumulator, and a detailed discussion is in the results section.

$dP_{BH,mix}$ is the pressure drop of the liquid mixture across the bleed hole, which is calculated by the following equations proposed by Miller (1978):

$$dP_{BH,mix} = 0.5K\rho_{mix}U_{mix}^2 \quad \text{Eqn (45)}$$

where, K is the dimensionless pressure drop, calculated as a function of orifice area ratio $\beta = (d_{BH}/D_{Acc})^2$ and thickness ratio t_{tube}/d_{BH} by following equations (d_{BH} is the orifice/bleed hole diameter; D_{Acc} is the accumulator internal diameter; t_{tube} is the thickness of J-tube wall):

$$K = \frac{C_0(1-C_c\beta^2)^2}{C_c^2}; \beta = \left(\frac{d_{BH}}{D_{Acc}}\right)^2 \quad \text{Eqn (46)}$$

$$C_0 = 0.5 + \frac{0.178}{4\left(\frac{t_{tube}}{d_{BH}}\right)^2 + 0.355}; 0.1 \leq \frac{t_{tube}}{d_{BH}} \leq 3 \quad \text{Eqn (47)}$$

Where, C_0 is a correction factor depending on the orifice thickness ratio t_{tube}/d_{BH} and C_c is the contraction coefficient from Kays and London's book (Kays and London, 1984).

Additionally, the density of mixture ρ_{mix} is affected by the local pressure at the bottom of the accumulator, and the local concentration of oil, consequently. The detail will be discussed later in the discussion chapter. U_{mix} is the average velocity of the mixture through the bleed hole.

4.2.3 Validation of the model

Model results are compared with a group of experimental data of the integrated IHX/Acc tested with R744 AC/HP systems (with different compressors, but other components are the same), in which the IHX capacity varies from 0.4 to 3.0 kW. In most cases, the model predicts the heat transfer rate, IHX effectiveness, and DP within $\pm 10\%$, as shown in **Figure 4.5**.

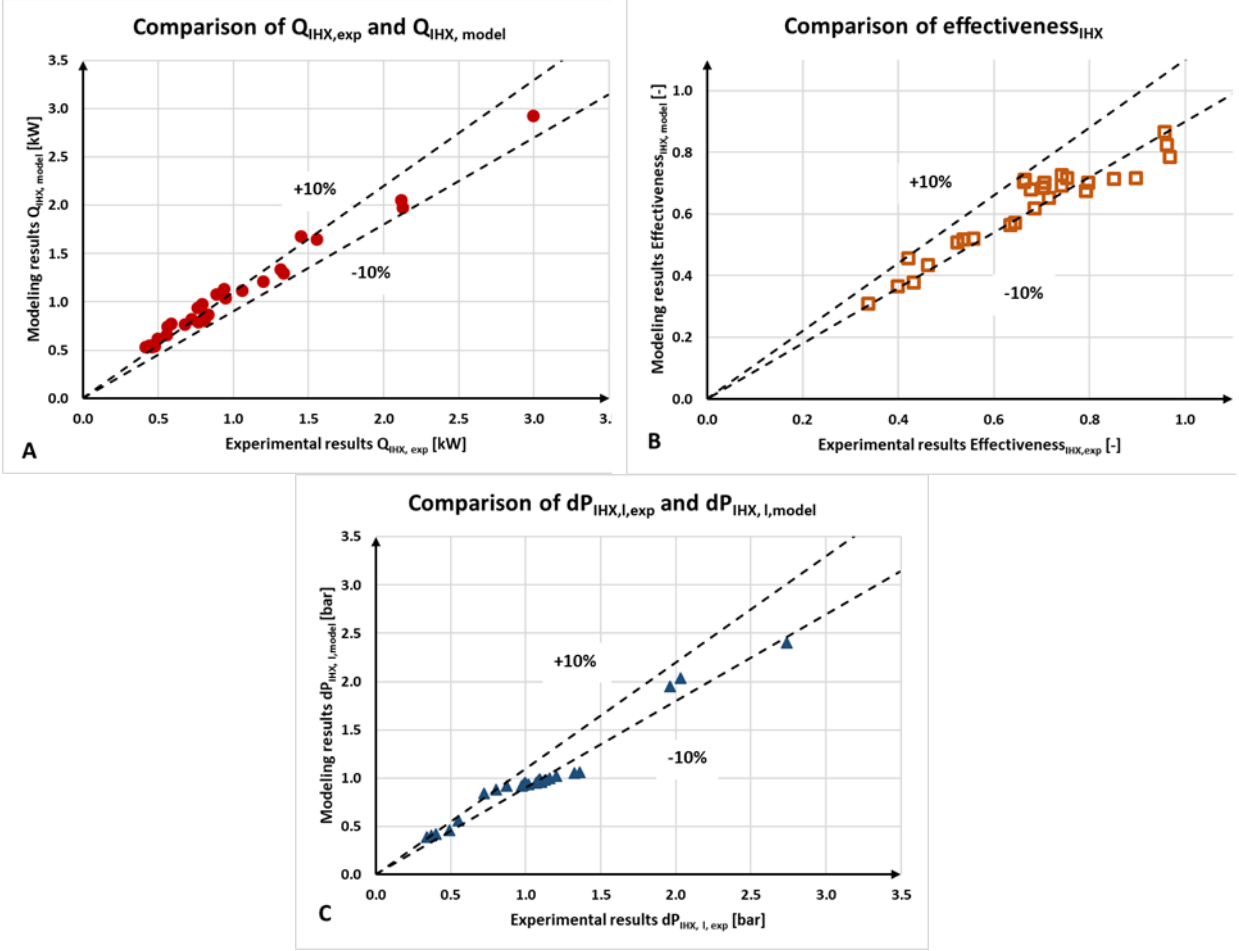


Figure 4.5: The comparison of modeling and experimental results: **A.** IHX heat transfer rate; **B.** IHX effectiveness; **C.** IHX and Acc low-side pressure drop.

The effectiveness of the IHX is defined as below:

$$effectiveness_{IHX} = \frac{Q_{IHX}}{Q_{IHX,max}} = \frac{h_{h,i} - h_{h,o}}{h_{h,i} - h_{h,o,min}} \quad \text{Eqn (48)}$$

Where, $h_{high,o,min}$ is determined by the high-side outlet pressure and low-side inlet temperature. The refrigerant enthalpy, instead of temperature, is used to calculate the effectiveness. Because the specific heat of the hot stream varies significantly in transcritical cycles, especially near the critical point. Therefore, it is not suitable to calculate the effectiveness as a function of temperatures.

4.2.4 Results and discussions

In this section, the performance of this integrated IHX/Acc is evaluated in two aspects: the first is the improvement of the IHX installation on the refrigeration capacity with different mass flow rates, evaporator outlet qualities, and operating temperatures; the second is the accumulator behavior in both AC and HP modes.

a. The performance of the Internal heat exchanger

The heat transfer rates and sensible parts of that are presented in **Figure 4.6** for different relative mass flow rates (in other words, different scales of the system) and operating temperatures in AC/ HP modes. The data labels next to the sensible heat transfer rate are the percentages of that out of the total heat transfer rates. The value of sensible heat transfer is considered as the increase of refrigeration capacity. Both the value and the portion of the sensible part are indicating the quantitative improvement by installing the IHX in real applications. The dot in the middle of each line represents the experimental result with a unity mass flow rate, and other dots are numerical results to show the performance of the IHX in a smaller or larger system. In the system with a different scale, the model assumes the same gas cooler and evaporator exit states for each operating condition.

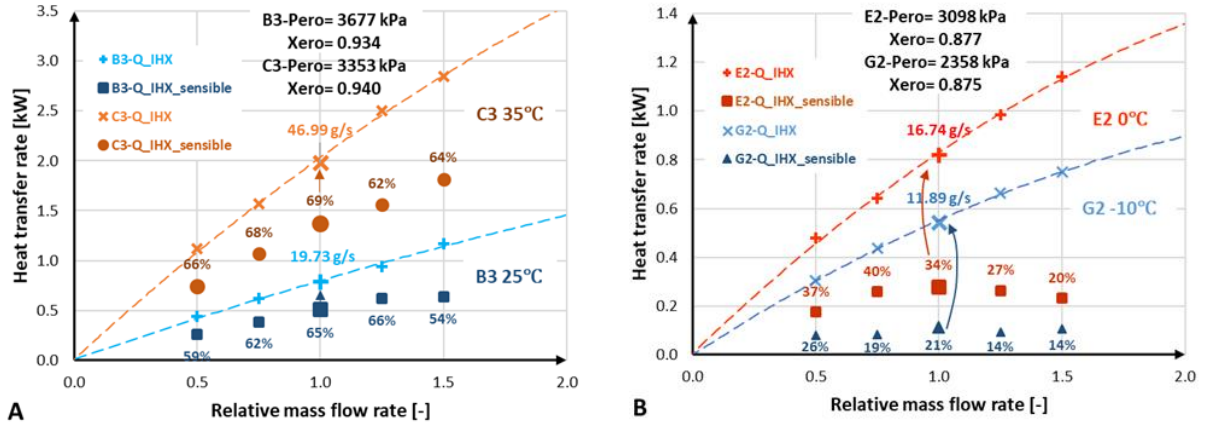


Figure 4.6: The heat transfer rate of IHX in **A:** AC and **B:** HP mode under various operating conditions (detailed operating conditions are listed in Appendix B)

1) Effects of operating conditions

The results indicate the total heat transfer rate of the IHX is approximately proportional to the relative mass flow rate in all four cases, but the operating temperature has a more significant impact on the sensible part. The benefits of IHX are significant in AC mode, especially in high-temperature operating conditions. As shown in **Figure 4.6**, the average percentages of sensible heat transfer are 66% and 61% in C3 (ambient temperature is 35°C) and B2 (25°C) conditions, but 32% and 19% in E2 (0°C) and G2 (-10°C) conditions. The main reasons are the different inlet states of the cold and hot streams in the IHX. **Figure B.1** in Appendix B shows the local temperatures and heat transfer rates of IHX in both AC/ HP modes. On the one hand, the inlet of the cold stream has a lower quality in HP mode (0.871) than in AC mode (0.940). Because the accumulator exit quality x_1 (or evaporator exit quality) is controlled by the bleed-hole diameter and is affected by the liquid level and mixture properties in the accumulator. The liquid level represents the extra refrigerant stored inside the accumulator, which increases when the operating mode switches from AC to HP. Feng and Hrnjak (2016) studied the charge optimization for an R134a automobile AC/ HP system and found that the AC mode required 300 g more charge than the HP mode. The main reason is the difference in internal volume between the outdoor condenser

(AC) and the indoor condenser (HP), although the liquid line's internal volume contributed to half of the imbalance. The optimal charge also varies with operating temperatures within each mode.

Because of the charge imbalance between the AC and HP modes, the accumulator in a reversible mobile air-conditioning system is required to contain the extra refrigerant in HP mode. In this study, the charge determination experiment of HP mode was conducted in the E2 condition, when both the ambient $T_{OD,a,i}$ and indoor air inlet temperature $T_{ID,a,i}$ was $-5\text{ }^{\circ}\text{C}$. The capacity- and efficiency- maximizing charge was determined to be 0.95 kg. However, a charge of 1.10 kg was determined to match the requirement of AC mode. Therefore, the liquid level in the accumulator is higher in HP mode than it is in AC mode, and the accumulator exit quality is consequently lower in HP mode.

On the other hand, the inlet state of the hot stream varies when the operating temperature changes. For example, in hot summer (C3) the refrigerant exit outdoor gas cooler is in a high-temperature and high-pressure supercritical state and has a higher specific heat. Oppositely, the refrigerant exit indoor gas cooler is likely to be sub-cooled liquid in cold winter (G2), especially during the warm-up process. Therefore, the hot stream has less potential to warm the cold stream to superheated vapor and brings benefits for the specific cooling capacity in HP mode.

2) Effects of relative mass flow rates

In this section, the relative mass flow rate is assumed to be proportional to the scale of the AC/ HP system, except for the integrated IHX/Acc, the size of which is fixed in the modeling. As shown in **Figure 4.6-A**, when the relative mass flow rate increases from 0.5 to 1.5, the relative benefit of IHX (in other words, the percentage of the sensible heat) varies slightly (about 8%) in AC mode and drops by 12% in G2 (HP mode). Also, the percentage peaks with the unity mass

flow rate in the B3 and C3 conditions in HP mode, but it peaks with a lower relative mass flow rate in the E2 and G2 conditions in AC mode. This indicates the IHX is better sized for the AC application than it is for the HP application.

In addition, the sensible heat transfer capacity increases from 0.74 to 1.82 kW in the C3 condition but varies from 0.08 to 0.11 kW in the G2 condition with the same change of relative mass flow rate. The much smaller capacity in the HP mode is due to the lower temperature of the hot stream as discussed earlier. **Figure B.1** (Appendix) shows the local temperatures of IHX in C3 and G2 conditions with different relative mass flow rates. As the relative mass flow rate increases from 0.5 to 1.5, the low-pressure side exit superheat drops from 24 to 14°C in the C3 condition and drops from 9 to 2 °C in the G2 condition. In the meanwhile, the effectiveness of IHX decreases from 0.43 to 0.35 in the C3 condition and from 0.81 to 0.64 in the G2 condition. Overall, the sensible heat transfer capacity and the IHX performance are less sensitive to mass flow rate compared to that of operating temperatures or inlet qualities.

3) Correction of the IHX benefits considering low-side pressure drop

Similarly, the low-side refrigerant pressure drop, and the related density drop of the refrigerant at the suction point of the compressor $F_{\rho,suc}$ are presented in **Figure 4.7** for different relative mass flow rates and operating conditions in AC/ HP modes. The factor of the relative density reduction $F_{\rho,suc}$ is defined as follows:

$$F_{\rho,suc} = \frac{\rho_{suc}(P_{ero} - dP_{IHX,l}, SH_{cpri}) - \rho_{suc}(P_{ero}, SH_{cpri})}{\rho_{suc}(P_{ero}, SH_{cpri})} \times 100\% \quad \text{Eqn (49)}$$

where, ρ_{suc} is determined by the inputs in the parenthesis. The factor $F_{\rho,suc}$ could be used to correct the previous evaluation of the benefits of IHX installation in **Figure 4.6**. Because the refrigerant

mass flow rate in a real system is reduced after introducing an IHX into the system since the refrigerant density at the suction point of the compressor and the refrigerant mass flow rate decrease. Assuming the effect of low-leg pressure drop on the compression ratio is negligible, and the volumetric efficiency is constant before and after installing the IHX, consequently. Then, the improvement of the refrigeration effect presented in **Figure 4.6** could be corrected with the factor $F_{\rho,suc}$ shown in **Figure 4.7**. Besides, the refrigeration capacity drop in the evaporator due to the reduction of mass flow rate should also be calculated for the overall evaluation. The quantitative evaluation of the overall improvement of the IHX after the correction can be expressed as a sum of two terms:

$$Q_{benefit} = Q_{ref,IHX} (1 + F_{\rho,suc}) - \left(\frac{Q_{ref,evap}}{1 + F_{\rho,suc}} - Q_{ref,evap} \right) \quad \text{Eqn (50)}$$

where, the $Q_{ref,IHX}$ is the increase of the refrigeration effect due to the installation of IHX listed in **Figure 4.6**. $Q_{ref,evap}$ is the experimental evaporator capacity of the system with the IHX/Acc.

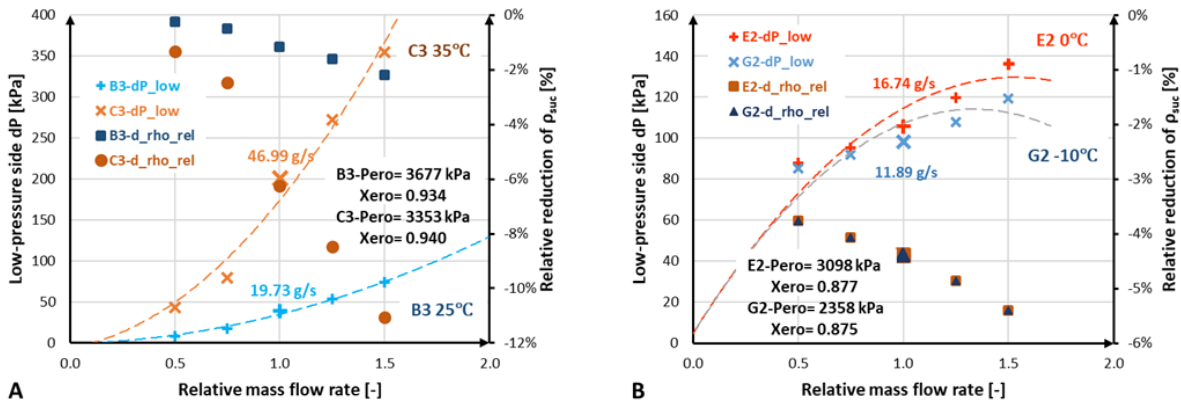


Figure 4.7: The low-side pressure drop in IHX/ Acc and the relative refrigerant density reduction $F_{\rho,suc}$ at the suction point of the compressor in **A:** AC mode and **B:** HP mode under various operating conditions (detailed operating conditions are listed in Appendix B)

Figure 4.8 shows the overall effect of the IHX/Acc on the refrigeration capacity after the correction in different conditions with various relative mass flow rates. The results illustrate that the overall benefits of IHX decrease when the ambient temperature decreases. In the C3 condition,

the highest possible improvement is 0.92 kW, but it is about zero in the G2 condition. Moreover, the improvement has a peak value as the relative mass flow rate changes, and this can be calculated using the IHX/Acc model. Therefore, for further optimization of a reversible R744 AC system, the model can be applied to maximize the benefits of IHX in certain operating conditions.

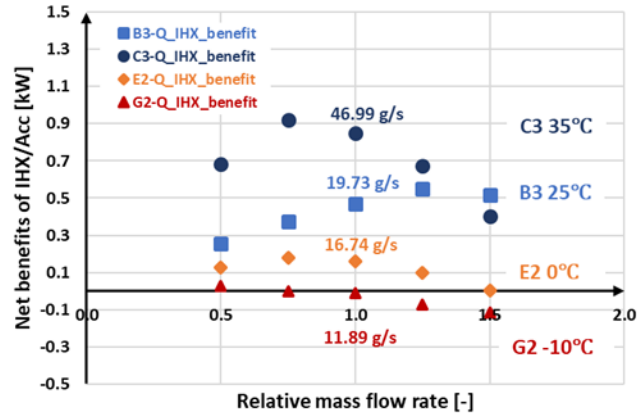


Figure 4.8: The net effects of the IHX/Acc on the refrigeration capacity in AC and HP modes under various operating conditions (detailed operating conditions are listed in Appendix B)

b. The behavior of the accumulator

With a fixed design of the accumulator, the evaporator exit quality is controlled by the bleed-hole diameter and affected by liquid level (refrigerant charge amount) and liquid properties in the accumulator. The liquid level difference between HP and AC is determined by the internal volume of indoor and outdoor coil volumes and the liquid line volume, which is not stressed here. The liquid mixture properties, especially the density and viscosity, change with the pressure inside the accumulator, as shown in **Figure 4.9**.

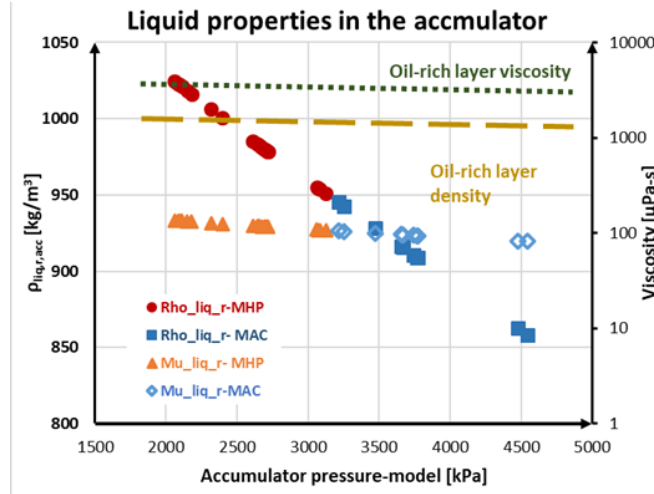


Figure 4.9: The properties of liquid refrigerant and oil mixture in the accumulator (C. J. Seeton & Hrnjak, 2006)

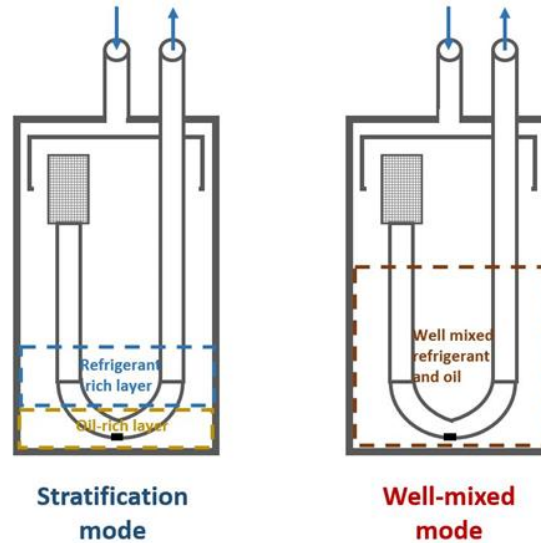


Figure 4.10: The liquid mixture inside the accumulator in stratification and well-mixed modes

In AC mode, the evaporating temperature varies from 0 to 10 °C, and the saturated pressure inside the accumulator is between 32 to 45 bar. In contrast, the evaporating temperature of CO₂ in HP mode can be as low as -20 °C, and the saturated pressure in the accumulator can be as low as 20 bar. **Figure 4.9** shows the liquid mixture properties inside the accumulator in both AC and HP modes with different saturated pressures. The distribution of oil over the height of the accumulator can be different, as shown in **Figure 4.10**. In the stratification mode (mostly in AC mode), the oil accumulates at the bottom of the cylinder, and the J-tube functions well. However, in low-

temperature conditions of MHP mode, the refrigerant can be well mixed with the oil, and the J-tube has difficulties in returning oil to the compressor.

As described previously, the effect of oil distribution was considered: first, the local liquid density and viscosity were obtained based on the data in Seeton and Hrnjak's report (C. J. Seeton & Hrnjak, 2006) as a function of the saturated pressure near the bleed hole. The pressure drop coefficient of the bleed hole is a function of the Re number and the local liquid properties, consequently. Additionally, the effect of oil addition on average liquid density inside the accumulator is marginal, so the hydrostatic pressure is calculated using the saturated liquid refrigerant density.

Figure 4.11 presents the results of the dimensionless liquid level estimated using the accumulator model. The black dash line is the trend line representing the behavior of the accumulator. The liquid level increases as the ambient temperature decreases, and when the high-side pressure decreases. Further work, like visualization of the accumulator, is required to validate the modeling results.

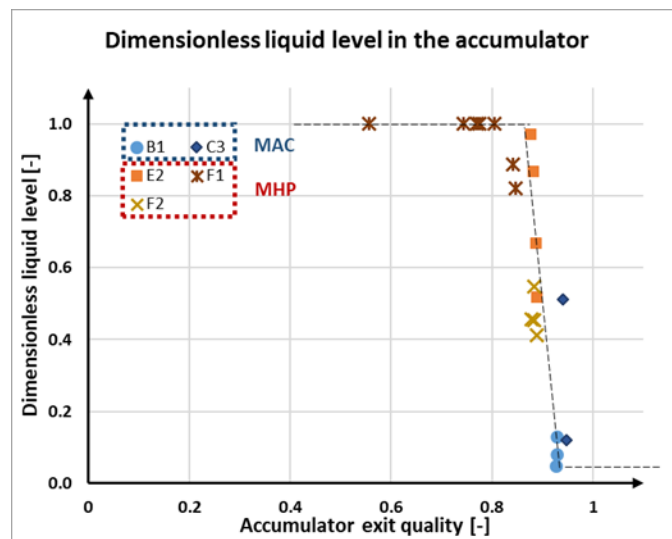


Figure 4.11: Modeling results of dimensionless liquid level in the accumulator

In conclusion, the 1-D model of the integrated CO₂ IHX/Acc predicts the heat transfer rate and pressure drop of the low-pressure side within $\pm 10\%$, by considering the effects of oil circulating rate on heat transfer and pressure drop. Also, the benefits and negative effects of introducing an IHX are analyzed in typical AC/ HP operating conditions. Results show that the integrated IHX/Acc provides higher benefits in AC mode, especially in high ambient temperature conditions. Finally, the modeling results of the accumulator behavior are presented. Further works are required to validate the model and optimize the design of the integrated IHX/ Acc for better overall performance in reversible R744 systems.

4.3 CFD modeling of the accumulator

In this section, the flow characteristics of the two-phase refrigerant and oil mixture are presented, aiming to provide more local data in addition to the 1-D modeling results of the accumulator in the last section. Volume of Fluid (VOF) model in commercial software ANSYS Fluent is used to track the liquid-vapor interface.

4.3.1 The 3-D model

The geometry of the IHX/Acc was presented in **Figure 4.4** and **Table 4.2**. An updated section view of the 3-D model of the integrated IHX and accumulator is shown in **Figure 4.12**. The internal nylon cylinder serves as the accumulator (orange part in **Figure 4.12**). A helical tube is arranged in the gap between the accumulator and the stainless-steel housing shell and forms a force-fit contact. The red and blue arrows represent the high- and low-pressure streams and form a counter-flow heat exchanger in air-conditioning mode.

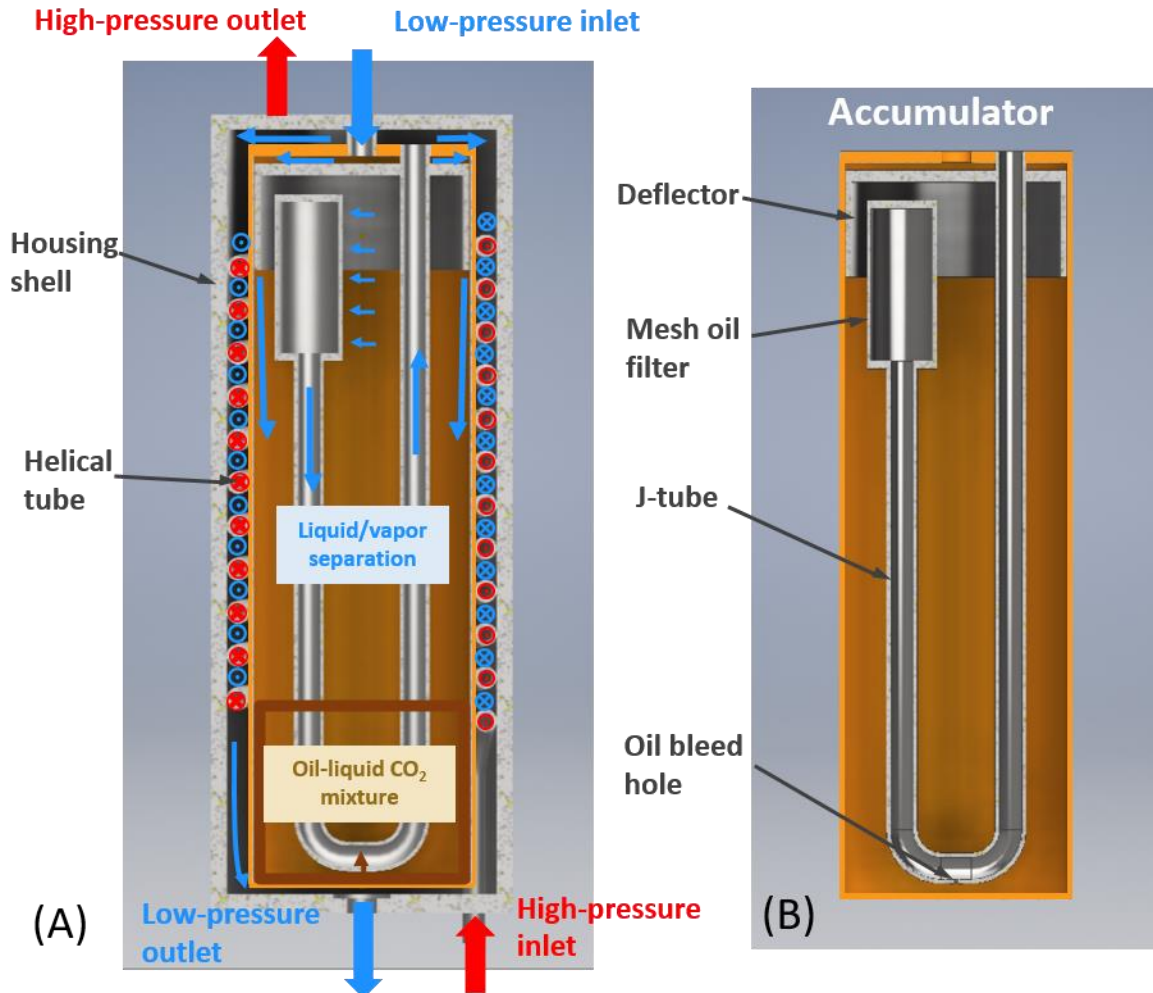


Figure 4.12: Section view of the 3-D model of (A) the integrated IHX and accumulator (B) accumulator only

For the accumulator, the two-phase refrigerant from the evaporator outlet flows through the inlet tube at the top center and enters the accumulator. Then, the refrigerant flows radially on the top of the deflector, then downwards along the small gap between the deflector and the accumulator wall. Afterward, the refrigerant separates into vapor and liquid in the space above the liquid level. In the normal operating conditions, vapor refrigerant and a small number of liquid droplets flow through the mesh oil filter on the top-end of the j-tube, but most of the liquid mixture flows through the oil bleed hole at the bottom-bend of the j-tube. Finally, the mixture of vapor and liquid flows upwards along the j-tube, exits the accumulator, and enters the IHX.

4.3.2 The quality of the mesh

The simulation object in this section is limited to the accumulator, and **Table 4.3** shows the key dimensions of the 3-D accumulator model. Due to the complexity of the flow domain, tetrahedron mesh was generated and three layers of prism were added along the wall of the j-tube to capture the interface between the liquid and vapor phases and the flow behavior in the boundary layer.

Table 4.3: Key dimensions of the accumulator model

Internal diameter [mm]	Internal height [mm]	Accumulator internal volume [mL]	J-tube internal diameter D_j [mm]	Oil bleed hole diameter D_{BH} [mm]
51.7	194	407	6	1.0/ 1.2/ 1.4/ 1.6/ 1.8

For the mesh independence, three sets of mesh files with 1.85, 4.30, and 9.99 million cells were tested, and the 4.30 million cell mesh (0.78 million nodes) was chosen due to the balance between accuracy and calculation cost. **Figure 4.13** shows the cross-section view of the mesh with 4.30 million cells and the details of the inlet of the j-tube and near the oil bleed hole. For this set of mesh file, the minimum and maximum cell volumes are $3.45\text{e-}15$ and $6.59\text{e-}9 \text{ m}^3$. Also, the minimum, maximum, and mean quality numbers of the mesh are 0.3081, 0.9996, and 0.7305 in the meshing software ANSYS ICEM. Also, the minimum skew value equals 0.3105. Generally, the quality of the mesh is good for the simulation.

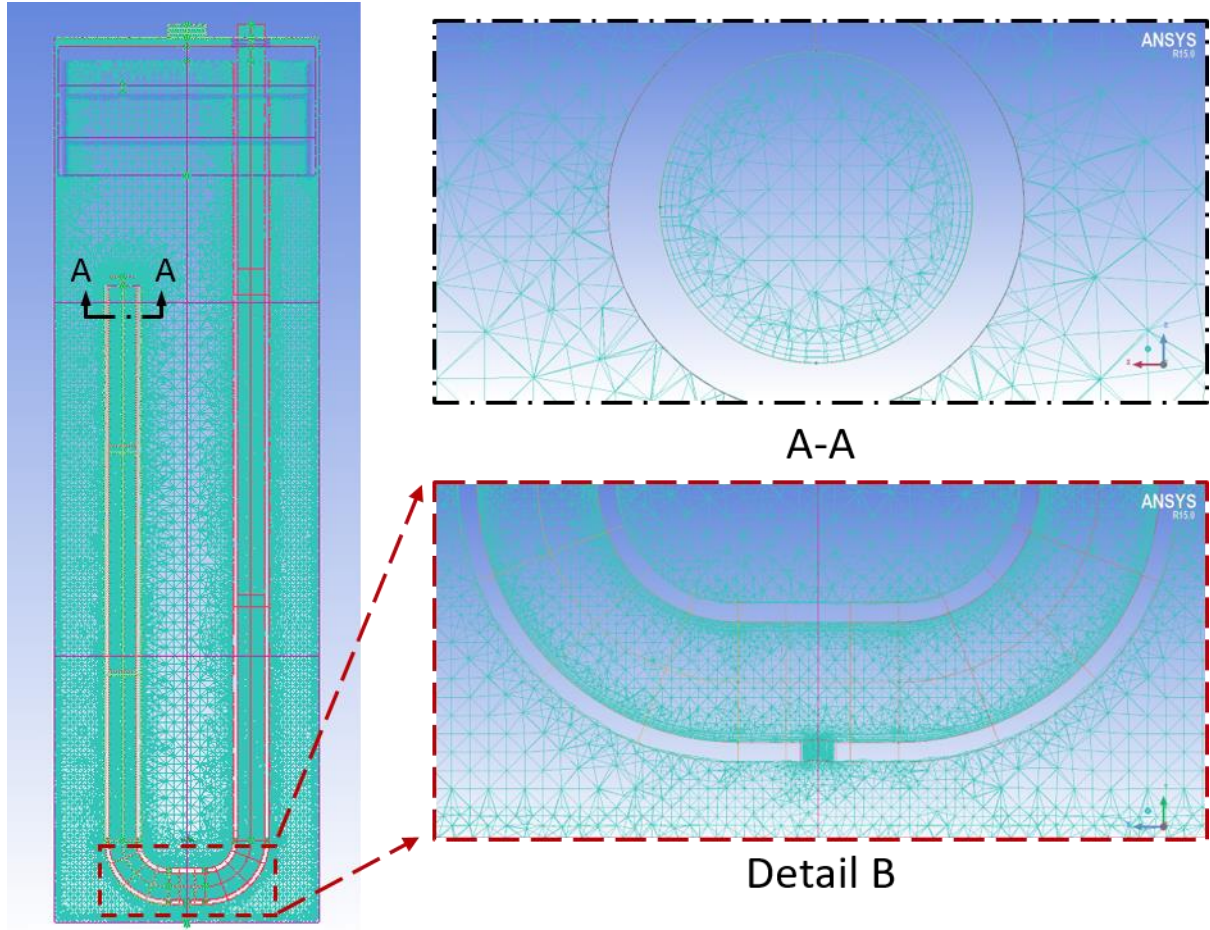


Figure 4.13: Section view of the volume mesh of the accumulator model and details of the prism on the wall of the j-tube

4.3.3 VOF model and other settings

VOF (Volume of Fluid) model is a surface tracking method designed for multiple immiscible fluids where the interface is of interest. Compared with the Eulerian model, it solves one single set of momentum equations and reduces the calculation cost. Also, VOF model is more numerical-stable than the Eulerian model. Therefore, VOF model is used in this dissertation to simulate the flow characteristics inside the accumulator.

For the turbulence model, realizable k - ϵ model and standard wall function are applied. The wall y -plus value ranges from 120 to 700 and agrees well with the requirement of the wall function.

The materials include vapor CO₂ and the liquid mixture of POE oil and CO₂. The density, viscosity, and surface tension of the materials are from Seeton's dissertation (2009). The vapor phase is considered the primary phase, and the liquid mixture is the second phase.

Besides, Pseudo Transient method is used for steady-state cases for better numerical stability. The convergence criterion of the simulation requires the residuals for all the variables to be lower than 1e-2, the net mass flow rate of inlet and outlet to be lower than 0.2 g/s, and the integration of the mass imbalance over the whole domain to be lower than 1e-15 kg.

4.3.4 Boundary and initial conditions

The mass flow rate boundary condition is adopted for the vapor and liquid inlets and the pressure boundary conditions are chosen for the outlet. For the initial condition, different liquid levels were iterated to converge faster for each geometry.

4.3.5 Simulation results and discussion

a. Effects of the diameter of the oil bleed hole on the liquid level

In this section, the effects of the diameter of the oil bleed hole on the liquid level and local pressure drop in steady-state are analyzed. **Figure 4.14** shows the simulation results for the liquid level and the streamlines inside the accumulator with different diameters of the oil bleed hole D_{BH} : 1.0, 1.2, 1.4, 1.6, and 1.8 mm. The red represents the liquid mixture of POE oil and liquid CO₂, and the blue represents the vapor CO₂. The boundary conditions of all the cases are the same: the inlet mass flow rates of vapor and liquid mixture are 26.80 and 3.43 g/s, and the oil circulating rate (OCR) is 2.0%. The static pressure at the exit is 3016 kPa. Gravity is in the negative y-direction. Those data are from the experimental data of a CO₂ HP system with the integrated accumulator and IHX in the 0°C ambient condition.

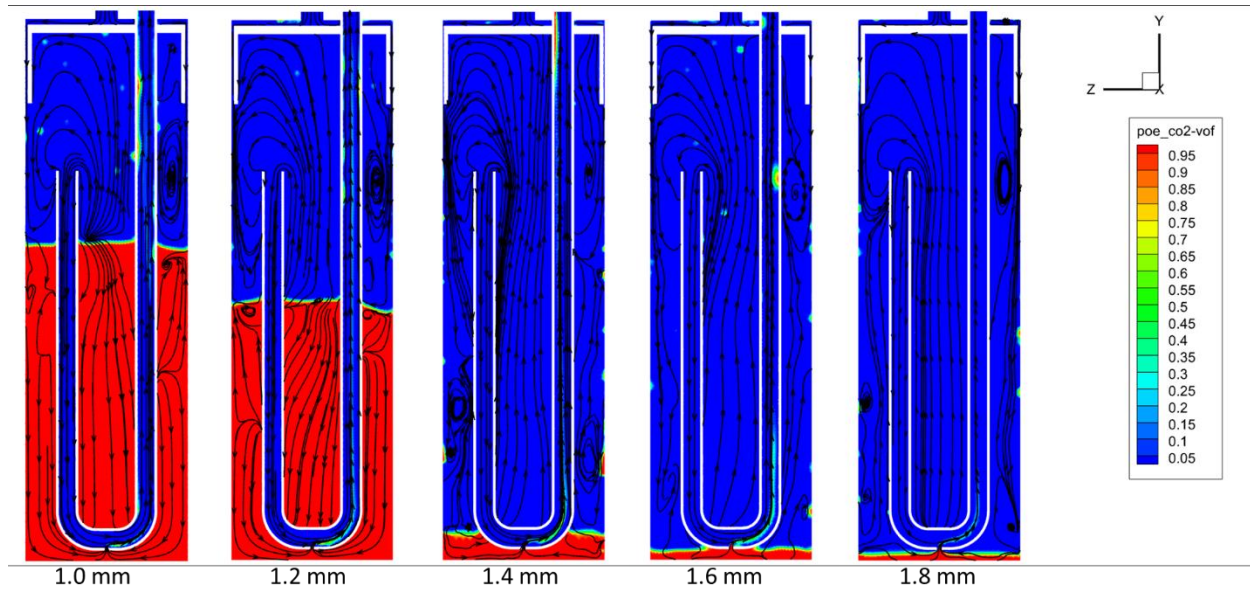


Figure 4.14: The volume fraction and streamlines inside accumulator with different sizes of oil bleed hole in y-z plane (parallel to the j-tube axis plane)

The simulation results show that the liquid level in the accumulator decreases as the diameter of the oil bleed hole D_{BH} increases (**Figure 4.14**). The masses of the liquid mixture retained in the accumulator are 287.3, 236.9, 26.9, 10.7 and 8.4 g for D_{BH} are 1.0, 1.2, 1.4, 1.6, and 1.8 mm, respectively. Also, the mass-weighted average volume fraction was performed over the inlet surfaces of the j-tube at the top and the oil bleed hole at the bottom, as shown in **Table 4.4**. The volume fractions of the liquid mixture on the inlet surface of the j-tube at the top are lower than $1e-03$ for almost all geometries, except for the 1.0 mm case. It can be concluded that for most of the cases only vapor enters the top inlet of the j-tube in steady-state in the current HP operating condition. However, 1.6 mm might be too large for the diameter of the oil bleed hole, because only a small amount of liquid mixture was retained in the accumulator, and even some vapor flows through the bleed hole. As seen in **Table 4.4**, the volume fractions of the liquid mixture on the inlet of the oil bleed hole are $8.64e-01$ and $7.15e-01$ for the 1.6- and 1.8-mm cases, but they are $1.00e+00$, $9.79e-01$, and $9.99e-01$ for the 1.0, 1.2, and 1.4 mm cases, respectively.

Table 4.4: Mass-weighted average volume fraction on the inlet surfaces of j-tube and oil bleed hole (0 to 1: vapor to liquid)

Cases	liquid VOF on j-tube inlet	Liquid VOF on oil bleed hole inlet
1.0 mm	1.608e-03	1.000e+00
1.2 mm	2.360e-05	9.790e-01
1.4 mm	1.045e-05	9.989e-01
1.6 mm	5.641e-04	8.647e-01
1.8 mm	1.290e-04	7.147e-01

Figure 4.14 and **Figure 4.15** also illustrate the streamlines of the two-phase flow in y-z and x-y planes. As seen in **Figure 4.15**, the two-phase mixture falls along the wall and gradually separates, and only vapor flows upwards in the middle part of the accumulator. Due to the block of the right part of the j-tube, there are two local vortexes between it and the wall (**Figure 4.15**). This might reduce the separation efficiency when the liquid level increases. Thus, the geometry of the j-tube can be modified to smooth the flow and reduce the vortex. On the other hand, the liquid mixture flows downwards to the oil bleed hole after the separation. When the liquid part flows through the oil bleed hole, it forms a jet flow and is then bent by the perpendicular vapor flow. Then, as the two-phase mixture flows upwards along the j-tube, it first forms an annular flow and breaks up into large drops due to the surface tension, as shown in **Figure 4.14**. Similar trends can be observed in **Figure 4.15**: vortexes exist in the bulk vapor under the deflector due to the two inversely vertical flows: upwards in the middle and downwards near the wall. In the 1.0- and 1.2-mm cases, vortexes exist in the bulk liquid, too.

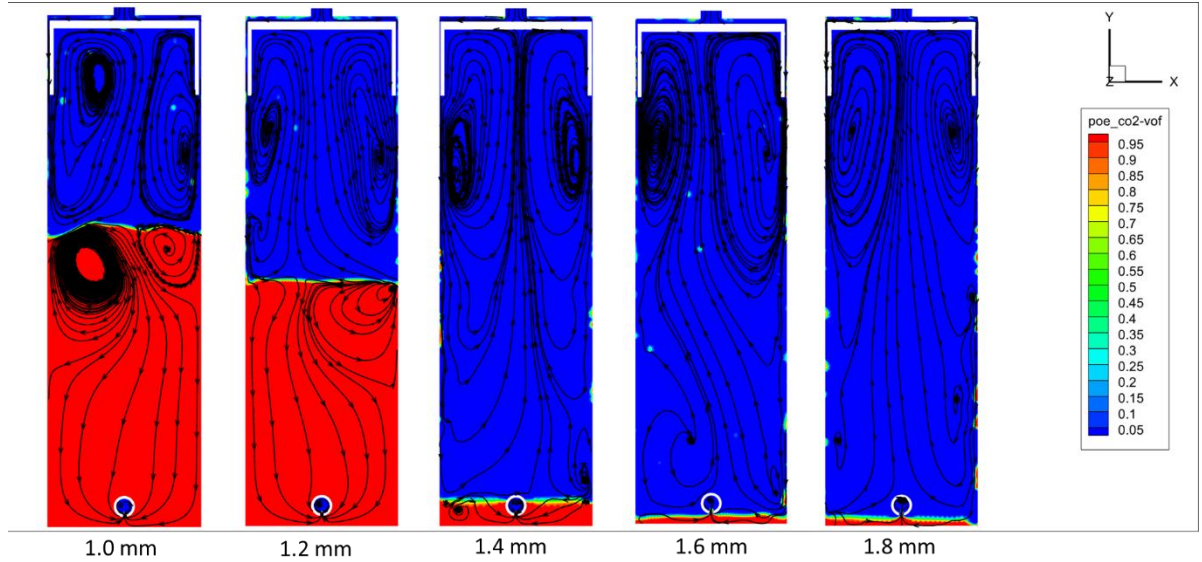


Figure 4.15: The volume fraction and streamlines inside accumulator with different sizes of oil bleed hole in the x-y plane (perpendicular to the j-tube axis plane)

Figure 4.16 presents the flow pattern of the liquid mixture on the internal wall of the accumulator with different diameters of the oil bleed hole. The liquid flows through the thin gap between the deflector and the wall of the accumulator in a fingering shape. Later, the liquid breaks up into droplets due to the surface tension and the disturbance of the vapor flow.

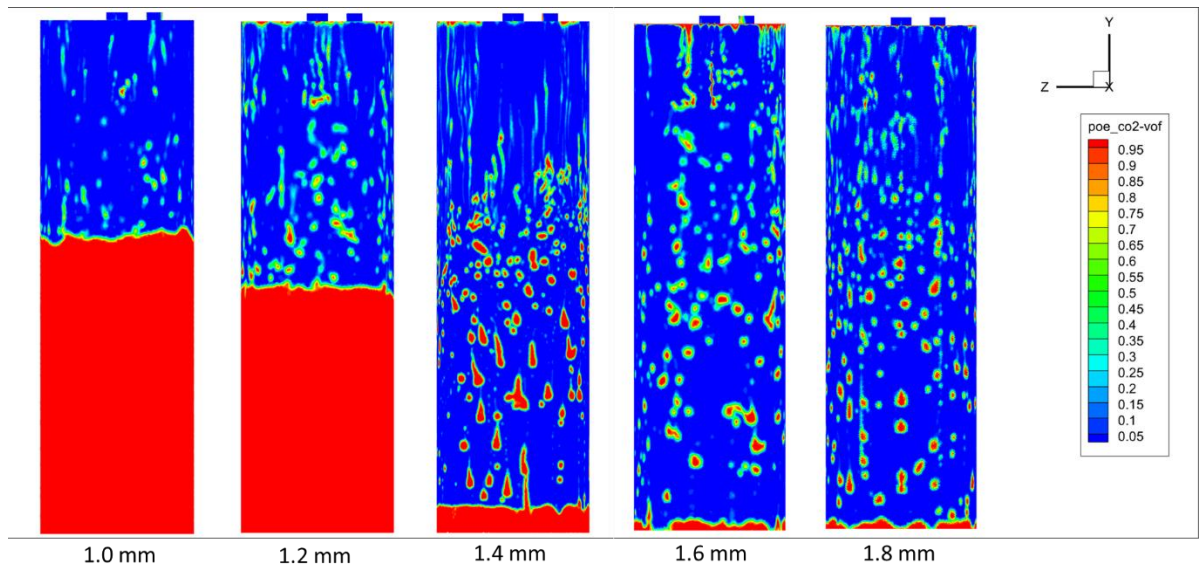


Figure 4.16: The volume fraction on the internal wall of the accumulator with different sizes of oil bleed hole

b. Effects of the diameter of the oil bleed hole on the local pressure drop

Figure 4.17 presents the static pressure in the accumulator with different sizes of the oil bleed hole in the y-z plane. The static pressure contours show that the major pressure drop occurs along the j-tube due to the high velocity of the vapor and related friction. Also, there are other kinds of pressure drops due to the bend, sudden contraction, and expansion. For example, when

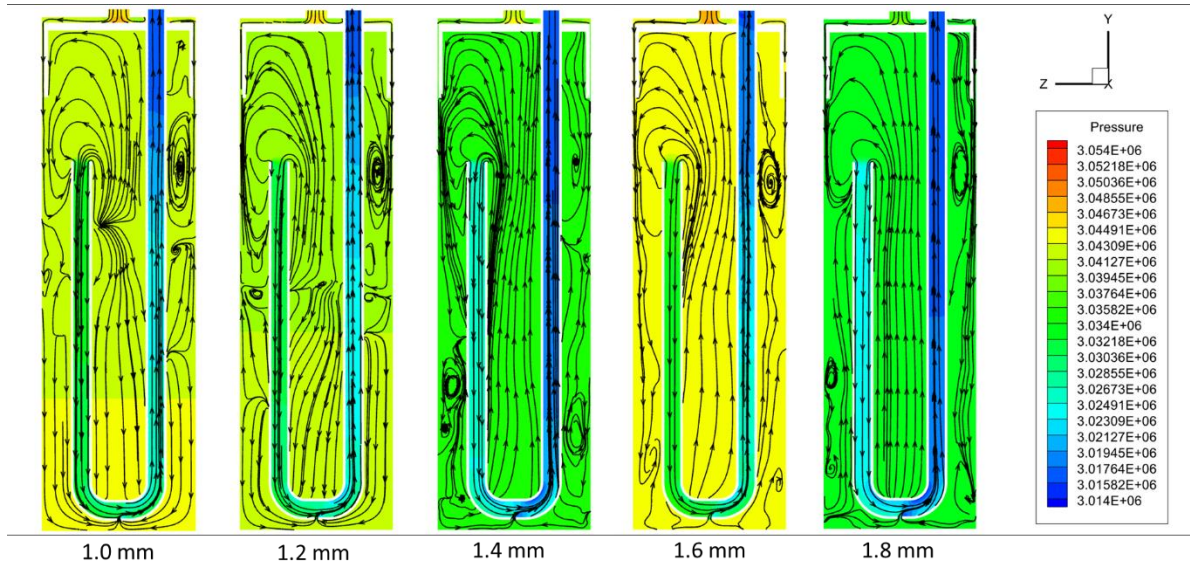


Figure 4.17: The static pressure (in Pa) inside the accumulator with different sizes of the oil bleed hole in the y-z plane

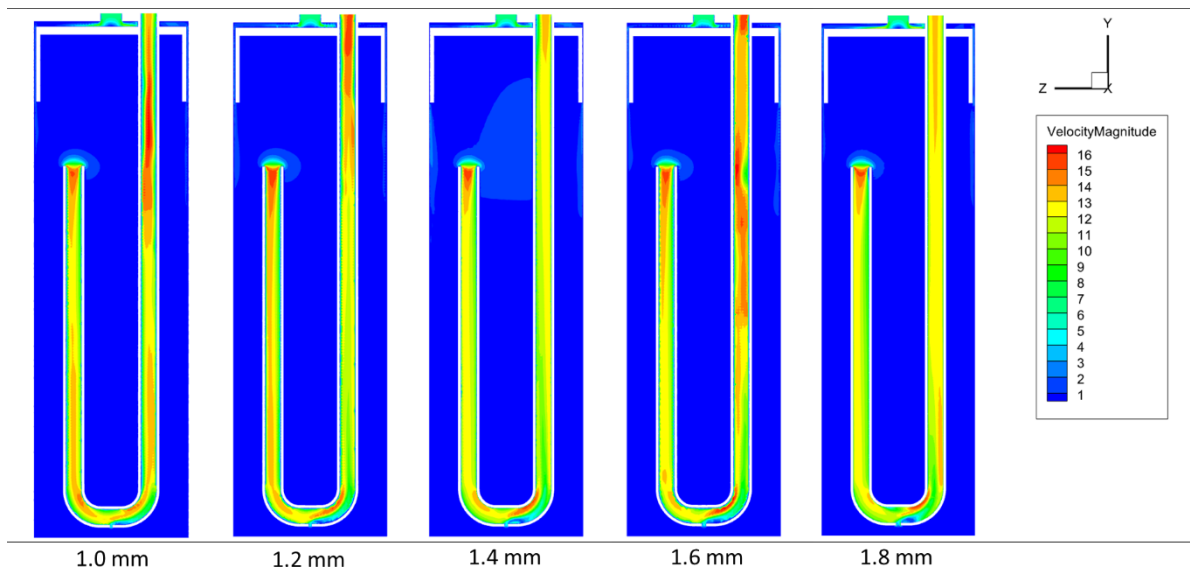


Figure 4.18: The velocity (in m/s) inside the accumulator with different sizes of the oil bleed hole in the y-z plane

the two-phase impinging flow hits the top surface of the deflector, the velocity is forced to change the direction and creates a stagnation pressure drop of about 4 kPa. Another example is the pressure drop at the top inlet of the j-tube due to sudden contraction. As shown in **Figure 4.18**, the velocity of the bulk vapor in the accumulator is about 2 m/s, but it increases to 15 m/s as vapor enters the j-tube. The static pressure drops by 10 to 13 kPa to compensate for this increase in dynamic pressure.

Figure 4.19 presents the local pressure drop along a streamline across the oil bleed hole with different sizes. Also, the mass flow rate across the oil bleed hole of all the cases can be calculated. Besides, the pressure drops across the oil bleed hole are listed in **Table 4.5**, so that the pressure drop coefficient K can be calculated using the following equation:

$$\Delta P = K \frac{1}{2} \rho V^2 \quad \text{Eqn (51)}$$

where, ΔP is the static pressure drop in Pa; ρ is the mass-weighted average density in kg/m³, and V is the mass-weighted average velocity in the cross-section in m/s. It can be concluded that for a pure liquid mixture, the pressure drop coefficient varies a little from 2.57 to 2.69, but it is 1.51 for the two-phase flow with 1.8 mm D_{BH} . A sensitivity study can be conducted to summarize more general empirical values or equations for the local pressure drop coefficient of the oil bleed hole.

Table 4.5: Pressure drop across the oil bleed hole

Cases	Static pressure outside the BH [Pa]	Static pressure inside the BH [Pa]	Pressure drop [Pa]	Pressure drop coefficient
1.0 mm	3043383.89	3029639.74	13744.15	2.69
1.2 mm	3041927.02	3027545.48	14381.54	2.66
1.4 mm	3035605.26	3022319.11	13286.15	2.57
1.6 mm	3043574.53	3029294.71	14279.82	2.43
1.8 mm	3033093.93	3020568.81	12525.12	1.51

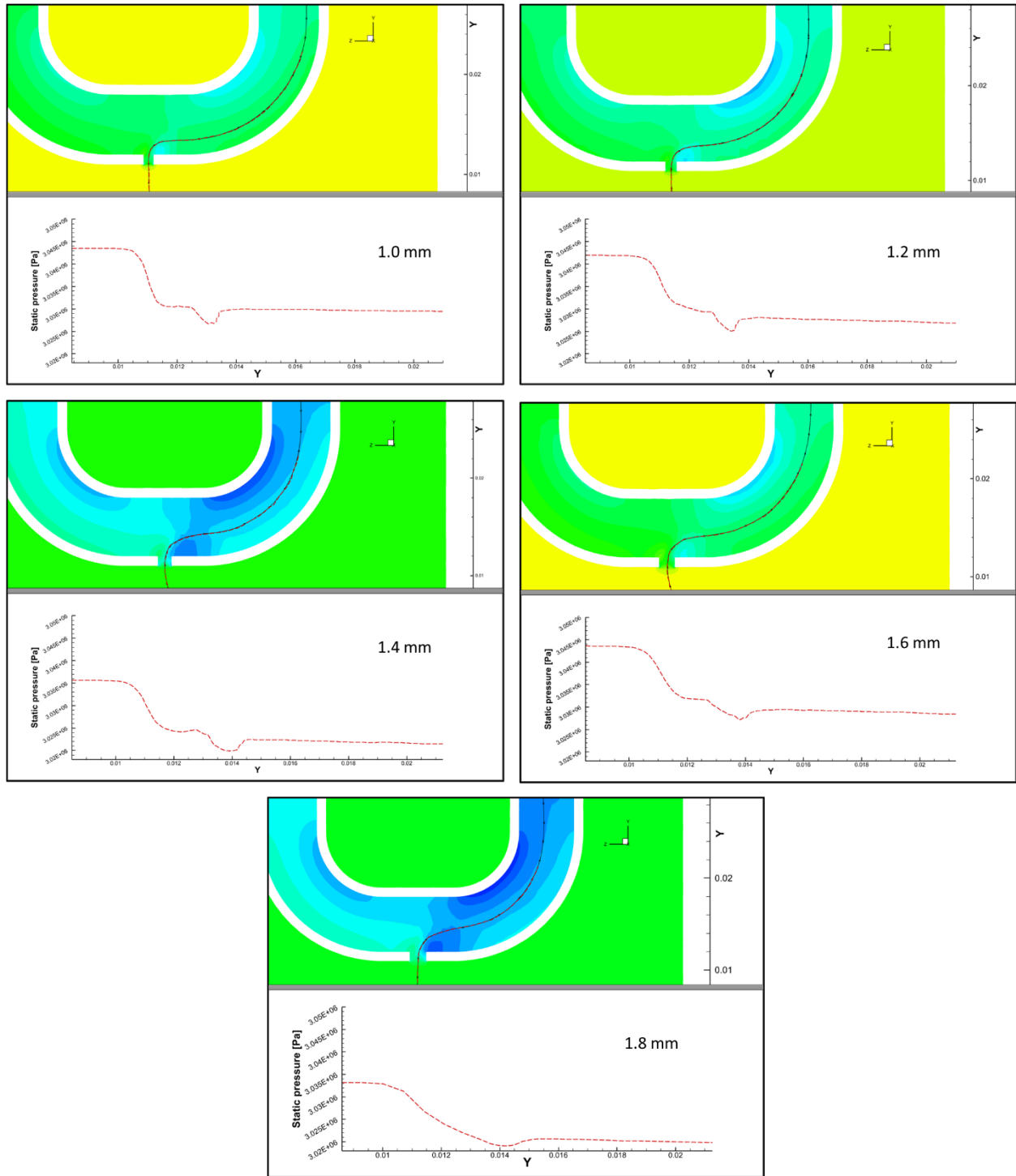


Figure 4.19: Pressure drop across the oil bleed hole with different sizes

In conclusion, a CFD simulation is conducted to investigate the flow characteristics of the two-phase CO_2 and POE mixture in the accumulator in a typical HP operating condition, considering the effects of different sizes of the oil bleed hole. The results show:

- The liquid level in the accumulator decreases as the diameter of the oil bleed hole D_{BH} increases. The masses of the liquid mixture retained in the accumulator are 287.3, 236.9, 26.9, 10.7 and 8.4 g for D_{BH} are 1.0, 1.2, 1.4, 1.6, and 1.8 mm, respectively.
- Only vapor enters the top inlet of the j-tube for almost all geometries in the current operating condition, except for the 1.0 mm case. Also, a 1.6 mm oil bleed hole is too large because only a small amount of liquid mixture was retained in the accumulator and even vapor CO_2 flows through the bleed hole.
- The flow pattern inside the accumulator is analyzed, which includes the vortexes of the vapor CO_2 and bulk liquid when the liquid level is high, the break-up of the jet flow of liquid mixture after the oil bleed hole, and the instability of the fingering flow along the wall.
- The static pressure contours show that the major pressure drop occurs along the j-tube due to the high velocity of the vapor and related friction.
- Also, the pressure drop across the oil bleed hole and the pressure drop coefficients K are calculated for all cases: K varies little from 2.57 to 2.69 for pure liquid mixture, but it is 1.51 for two-phase flow with 1.8 mm D_{BH} .

CHAPTER 5: PERFORMANCE OF THE R744 HEAT PUMP IN CONTINUOUS FROSTING CYCLES

To investigate the performance of the reversible R744 heat pump system in a wide range of operating conditions, including practical frosting conditions, the reversible HP system was tested in continuous frosting/defrosting cycles. The impacts of the environmental factors, the defrosting-initiation criteria, and the orientation of the outdoor heat exchanger on the system heating performance are discussed and analyzed in this chapter.

5.1 System architecture and experimental facility

In chapter 3, **Figure 3.1** shows the schematic of the reversible R744 heat pump system in both AC (defrost) and HP modes with the images of the components, and **Table 3.1** reports the dimensions of the key components of the HP system. All the components and the measurements are the same for the experiments in chapter 3, 5, and 6 of this dissertation. The main differences are: 1) the outdoor coil is installed with horizontally oriented microchannel tubes in chapter 3 for dry tests, but it is installed both horizontally and vertically in chapter 5 and 6 for the frosting/defrosting experiments, as shown in **Figure 5.1**; 2) the performance of the HP system was calculated using three approaches in the dry tests, but only air- and refrigerant-approaches are used for frosting and defrosting experiments because they are transient processes. The chamber-side calculation is inaccurate due to the large thermal mass of the chamber and equipment.

Figure 5.2 presents the air- and refrigerant-side measurements of the reversible R744 heat pump system in both HP and defrost modes. The performances of the reversible CO₂ system during continuous frosting/defrosting cycles were experimentally studied using two climate chambers: outdoor and indoor climate chambers. Each climate chamber is composed of a wind tunnel with

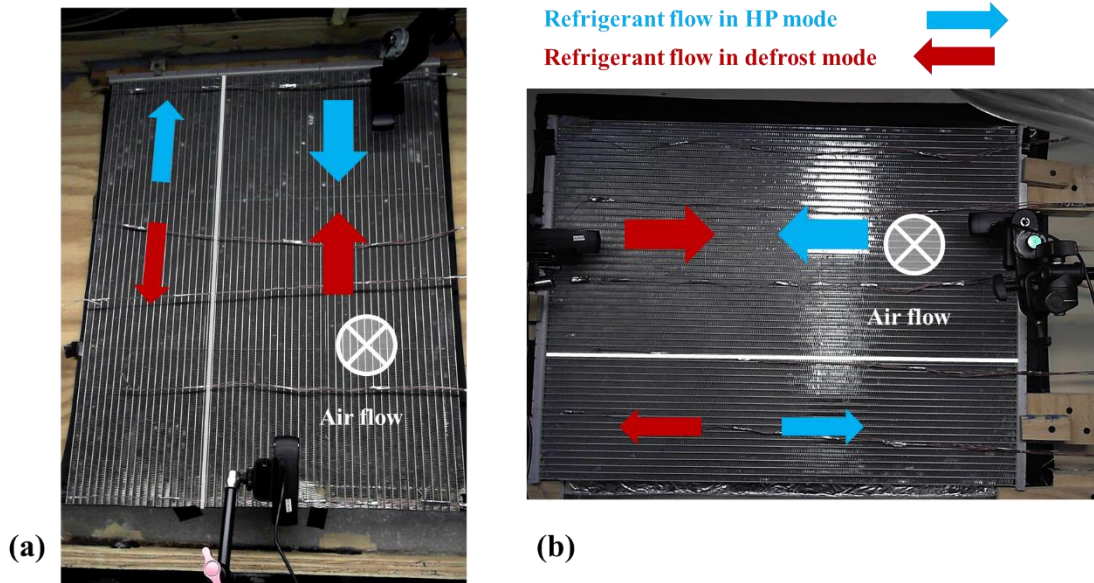


Figure 5.1: The same outdoor microchannel heat exchanger installed (a) vertically and (b) horizontally: 17 tubes in 1st pass and 34 tubes in 2nd pass in HP mode

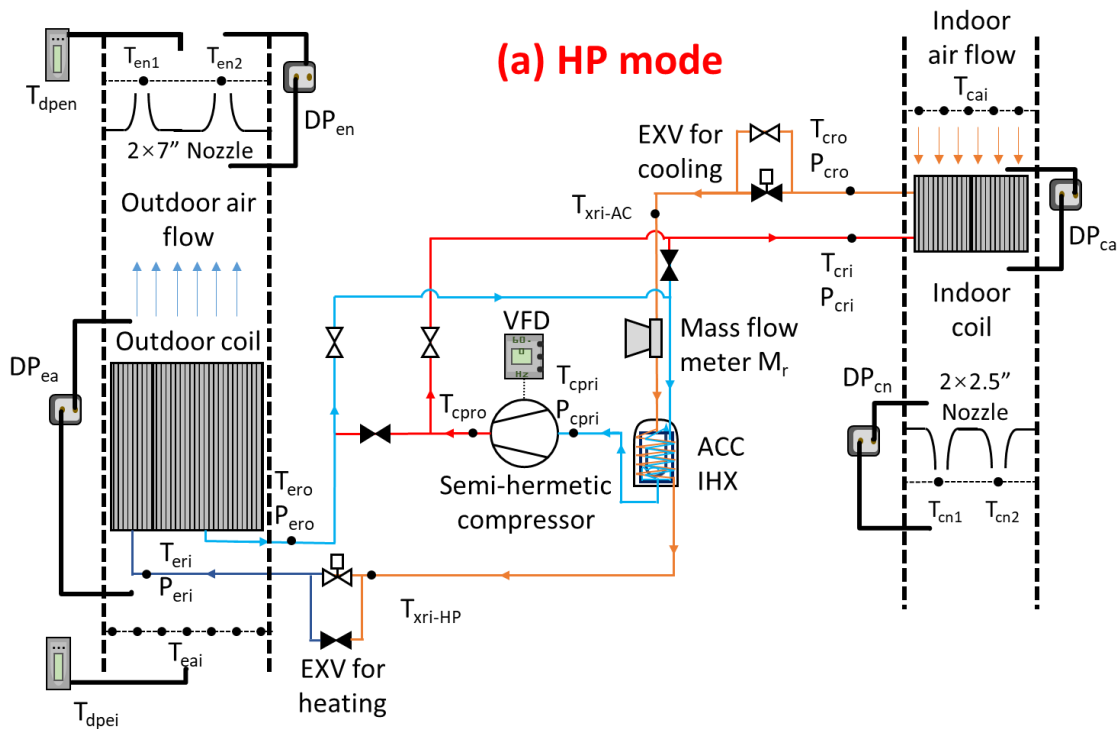


Figure 5.2: The instrumentation for the air- and refrigerant-side of the reversible CO₂ system in (a) HP mode and (b) defrost mode

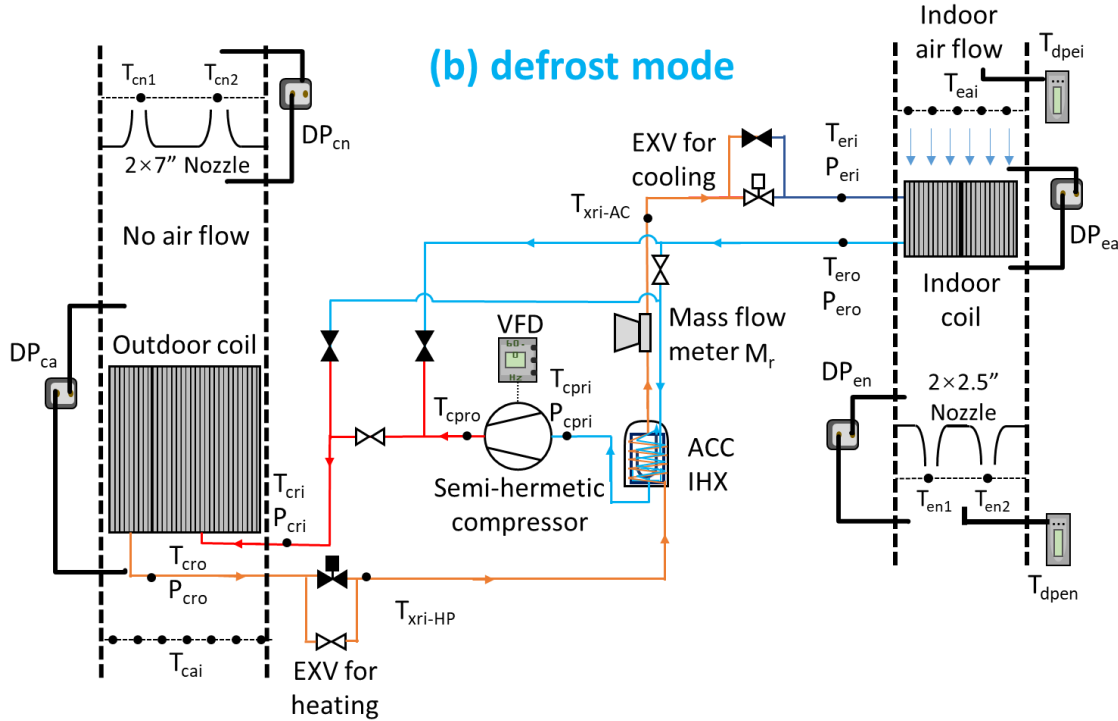


Figure 5.2: The instrumentation for the air- and refrigerant-side of the reversible CO₂ system in **(a)** HP mode and **(b)** defrost mode (cont.)

ASHARE standard air nozzles, Variable-Frequency Drive (VFD) controlled blowers, a Proportional-Integral-Derivative (PID) controlled electric heater, a glycol chiller integrated with a fan, and a steam spray with a needle valve. In addition, sensors for temperature, pressure, and humidity measurements were installed correspondingly in the CO₂ vapor compression system and both climate chambers. The air inside each climate chamber is conditioned by the electric heater and the glycol heat exchanger for temperature and by the steam spray for humidity. The states of inlet air for each heat exchanger are measured using thermocouple grids and a chilled mirror dew point sensor, and the states of exit air are measured by a thermocouple and a dew point sensor installed at the exit of the air nozzle. The conditioned air is pulled through the wind tunnel by the VFD-controlled blowers, and the pressure drops across air nozzles are monitored during the periodic cycles to calculate the airflow rates. The refrigerant states at the inlet and exit of each component are determined by the temperature and pressure measurements when the local

refrigerant is in the single-phase zone. The detailed calculation is presented in the data reduction section. In addition, the compressor power was directly measured using a three-phase watt transducer. Also, the ranges and uncertainties of the instrumentation are listed in **Table 3.2**.

5.2 Experimental procedure and experimental conditions

The performances of the reversible CO₂ system in periodic frosting/defrosting cycles are studied using the following steps: 1) preconditioning, 2) frosting, 3) defrosting by reversing the cycle, 4) water collection, 5) refrosting for periodic performance. The operating conditions of the experiments are listed in **Table 5.1**.

Table 5.1: Operating conditions of frosting/ defrosting experiments

Microchannel tube orientation	T_{eai} [°C]	RH_{ei} [%]	$V_{eai,ini}$ [m/s]	T_{cai} [°C]	U_{cai} [kg/min]	# of frosting cycle	Defrost initiation criterion	Defrost termination criterion
Vertical	0/-5/-10	80/90	3.0	20	7.0	1/ 2/ 3	$5/10 \times DP_{ea,ini}$	Outdoor $T_{cro} = 45^{\circ}\text{C}$
Horizontal	0	90				1/ 2	$10 \times DP_{ea,ini}$	

1) Preconditioning:

First, the CO₂ system, the glycol loop, and the electric heaters are used to precondition the chambers. To keep the outdoor coil free from frost during the preconditioning, the glycol loop was first switched on and the dew point in the outdoor climate chamber was monitored until it reached -5°C. Then, the CO₂ system was switched on and the compressor speed and EXV opening were controlled to make sure the evaporating temperature of the CO₂ system was higher than the dew point of the air in the outdoor chamber- to keep the surface of the evaporator frost-free. After outdoor and indoor air inlet temperatures T_{eai} and T_{cai} reached the target values, the EXV was controlled to provide the efficiency-maximizing high-side pressure for the transcritical CO₂ cycle.

This is to simulate the preconditioning of the vehicle when it is parked in the relatively dry garage before driving.

2) frosting

After all measurements reached steady states and a 20-minute operation, steam was added into the outdoor chamber until it stabilized at the desired relative humidity RH_{ei} (this usually lasts a maximum of two minutes). Then, experimental data and frost distribution images of the outdoor evaporator were recorded at an interval of 6 seconds until the air-side pressure drop across the outdoor evaporator DP_{ea} reached the criterion of initiating defrosting. During the frosting procedure, the blower speeds were fixed at constant values to provide the air velocity $V_{eai,ini}$, and U_{cai} . As frost accumulated on the outdoor coil, the outdoor air velocity V_{eai} could drop to approximately 1 m/s when the final DP_{ea} reached 10 times the initial DP_{ea} . Also, the compressor speed (1450 rpm) was controlled by VFD at a constant frequency of 50 Hz, and the high-side pressure P_{cro} was controlled by EXV to a value as close to the initial efficiency-maximizing pressure as possible with the limitation that the discharge temperature T_{cpro} is lower than 140°C. The EXV opening was fixed during the frosting process.

3) Defrosting

Once the air-side pressure drop across the outdoor evaporator DP_{ea} reached the defrost-initiation criterion, the HP system was switched off and the pressure inside the system equalized. Then, the six ball valves in the system were adjusted and the cycle was reversed to defrost mode. During the defrosting cycle, the outdoor blowers were off to reduce the heat loss to the ambient air, and the indoor blowers were still on and kept at the same speed in frosting mode. The benefit is a higher heat supply from the indoor air and reduced defrosting time. The disadvantage is a

lower indoor air temperature during the defrosting and negative impacts on the comfort level. However, if the defrosting is conducted while the EV is parking in the parking lot or charging center, the effects on passengers' comfort can be minimized. Compressor speed was reduced to 1160 rpm and the opening of the second EXV (for defrost mode) was fixed at 50% through the whole defrosting. The transient data were recorded at an interval of 6 seconds and the defrosting images were recorded at an interval of 2 seconds because the frost melt completely within 10 to 40 seconds in the conditions studied. Finally, the defrosting was terminated once the refrigerant exit temperature at the outdoor gas cooler T_{cro} reached 45°C.

4) Water collection

The water collection was conducted once the final defrosting was finished for each day. For example, three continuous frosting/defrosting cycles were completed at 0°C and 90% RH with 5 times the initial DP_{ea} as the defrost-initiation criterion. On the first day, only one frosting/defrosting cycle was finished, and then the retained and drained water after one cycle were collected and recorded. On the second day, two continuous frosting/defrosting cycles were completed, and then the retained and drained water after the second cycle were collected. On the third day, the frosting/defrosting cycles were repeated for three times and then water were collected. So, all data for this specific condition were collected and the performance is repeatable.

To collect the meltwater, the compressor, blowers, heaters, and steam sprays were switched off after defrosting. The water retained on the outdoor evaporator was collected using compressed air and paper towels: one side window of the outdoor wind tunnel was opened and the air spray was used to blow the retained water out. In the meanwhile, 6 to 10 layers of paper towels were gently pushed on the other side of the outdoor coil to absorb the water. The mass of the paper

towels before and after collecting the retained water was measured, and the difference was the water collected. However, based on the mass and energy imbalance during the frosting and defrosting process, it is found that there should be a thin water film retained on the outdoor coil after this collecting, and the mass of this thin water film and the total mass of retained water $M_{retained}$ can be estimated. Also, the drained water $M_{drained}$ was collected using a slightly inclined pan installed under the outdoor coil, a hose, and a 500 mL beaker.

Then, the difference between the total frost accumulation M_{frost} and the sum of $M_{retained}$ and $M_{drained}$ was considered as the vaporized water $M_{vaporized}$ during the defrosting process.

5) refrosting

If it is not the final defrosting cycle of one day, the frosting/defrosting procedures 2) and 3) were repeated without collecting the retained water in procedure 4) to evaluate the periodic performance of the HP system.

5.3 Data reduction and uncertainty analysis

For the data reduction during frosting cycles, all the experimental data used to determine thermodynamic properties are the averaged values of data recorded in two minutes. Because the frosting is a slow process and can be considered a quasi-steady state.

During the frosting cycle, the cooling capacity of the outdoor evaporator was calculated only based on the air-side approach using **Eqn (52)**, because there was always a two-phase refrigerant at the exit of the evaporator due to the existence of the accumulator, as shown in **Figure 5.3**.

$$Q_{cooling} = \dot{m}_{eai,dry}(h_{eai} - h_{eao}) \quad \text{Eqn (52)}$$

where, the mass flow rate of the dry air $\dot{m}_{eai,dry}$ was determined from measured air dry-bulb temperature (T_{en} = average of T_{en1} and T_{en2}), dew point T_{dpen} , and pressure drop across the outdoor nozzles DP_{en} . The enthalpies of air h_{eai} and h_{eao} were determined using air dry-bulb temperature and dew point at the inlet of the outdoor evaporator (T_{eai} and T_{dpei}) and at the exit of outdoor nozzles (T_{en} and T_{dpen}). Assuming an isenthalpic expansion process, the refrigerant state at the inlet of the outdoor evaporator can be determined using the measured local refrigerant temperature and pressure. Also, the refrigerant state at the exit of the outdoor evaporator can be calculated using the mass flow rate of refrigerant $\dot{m}_r$, the air-side cooling capacity $Q_{cooling}$, and measured local refrigerant pressure and temperature.

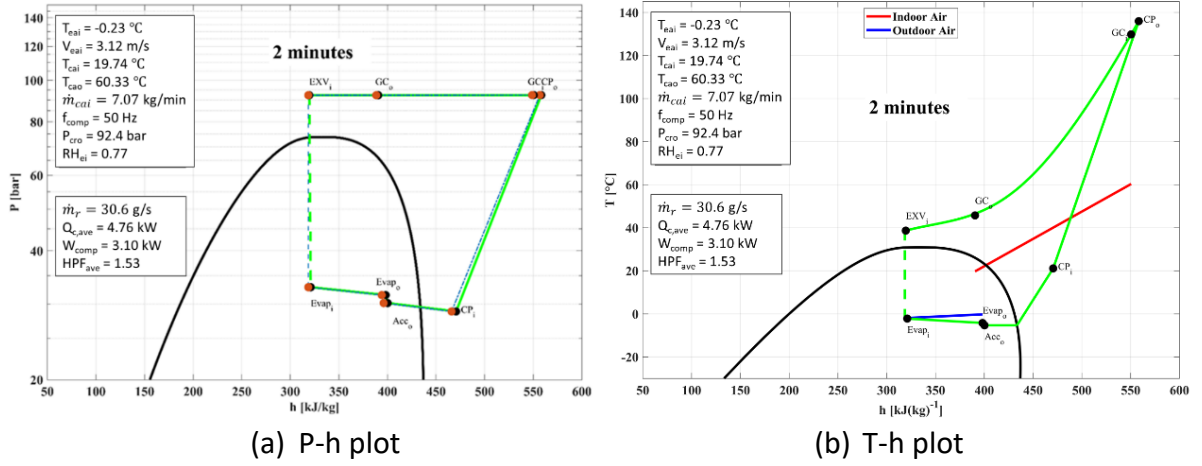


Figure 5.3: The P-h and T-h plots of the first 2-minute of the first frosting cycle with vertically oriented outdoor coil in the 0°C and 80% RH outdoor condition

In addition, the mass of the frost accumulated on the outdoor evaporator during one frosting cycle was determined by integrating the product of humidity ratio difference at the inlet and exit of the outdoor coil and the dry air mass flow rate, as **Eqn (53)**:

$$Q_{cooling} = \dot{m}_{eai,dry}(h_{eai} - h_{eao}) \quad \text{Eqn (53)}$$

where, $t_{frosting}$ is the time of a frosting cycle in second. The humidity ratios ω_{eai} and ω_{eao} were determined using air dry-bulb temperatures and dew points measured at the outdoor evaporator inlet and the nozzles' exits, respectively.

The heating capacity of the indoor gas cooler was calculated based on both air- and refrigerant-side approaches, following **Eqn (54)** and **Eqn (55)**:

$$Q_{heating,air} = \dot{m}_{cai,dry}(h_{cao} - h_{cai}) \quad \text{Eqn (54)}$$

$$Q_{heating,ref} = (1-OCR)\dot{m}_r(h_{cri} - h_{cro}) + OCR \cdot \dot{m}_r(h_{coi} - h_{coo}) \quad \text{Eqn (55)}$$

where, the indoor air mass flow rate $\dot{m}_{cai,dry}$ was calculated using measured air dry-bulb temperature (T_{cn} = average of T_{cn1} and T_{cn2}), and pressure drop (DP_{cn}) across the indoor nozzles. The enthalpies of air h_{cai} and h_{cao} were determined using air dry-bulb temperatures at the indoor gas cooler inlet T_{cai} and the nozzles' exits T_{cn} , assuming dry air because there is no water condensation at the indoor gas cooler. For the refrigerant side, OCR (Oil Circulation Rate) is measured by a sampler installed across the mass flow meter on the liquid line. $\dot{m}_r$ was directly measured by the mass flow meter, and it is actually the mass flow rate of the oil and CO₂ mixture. Refrigerant enthalpies h_{cri} and h_{cro} , oil enthalpies h_{coi} and h_{coo} were determined from the pressures and temperatures measured at the inlet (T_{cri} and P_{cri}) and outlet (T_{cro} and P_{cro}) of the indoor gas cooler. In this project, the OCR is measured to be 2% and the effects on system capacity are negligible: the black dots and thick green line in **Figure 5.3(a)** represent the pure refrigerant properties, and the orange dots and thin blue line represent the oil and CO₂ mixture properties. The two sets of P-h plots almost overlap with each other, and the most different part is at the compressor suction point. In the results section, only the average value of the heating capacity is presented as $Q_{heating}$:

$$Q_{heating} = 0.5 \times (Q_{heating,air} + Q_{heating,ref}) \quad \text{Eqn (56)}$$

Similar to the gas cooler, the heating capacity of the IHX on the high-pressure side was calculated based on the refrigerant-side approach (**Eqn (57)**), since both heat exchangers are in the supercritical zone:

$$Q_{IHX,ref} = (1 - OCR) \dot{m}_r (h_{cro} - h_{xri}) + OCR \cdot \dot{m}_r (h_{coo} - h_{xoi}) \quad \text{Eqn (57)}$$

where, the refrigerant and oil enthalpies (h_{xri} and h_{xoi}) were determined based on the pressure and temperature measured at the inlet of the expansion valve used in HP mode. Assuming the heat loss through the insulation to the ambient is negligible, the cooling capacity of IHX in the low-pressure equals $Q_{IHX,ref}$. Then, the refrigerant state at the accumulator exit can be calculated by combining the state of superheated vapor at the compressor suction point and the energy balance of IHX. As shown in **Figure 5.3**, the calculated enthalpies at evaporator exit h_{ero} and the accumulator exit $h_{acc,ro}$ agree well during the first two minutes of the frosting cycle. This indicates that the liquid level in the accumulator is stable at the beginning of the frosting cycle. Although the liquid level inside the accumulator increases as h_{ero} drops with $Q_{cooling}$ during the later stage of the frosting process. Because of the design of the J-tube (Zhang and Hrnjak, 2020), the refrigerant quality at the accumulator exit $x_{acc,ro}$ and $h_{acc,ro}$ stays almost constant before the accumulator gets flooded. A clear deviation between h_{ero} and $h_{acc,ro}$ can be observed after a significant frost accumulation during one frosting process, as shown by the T-h diagram in the results section.

The Heating Performance Factor (HPF) represents the efficiency of the HP system and is calculated following:

$$HPF = \frac{Q_{heating}}{W_{comp}} \quad \text{Eqn (58)}$$

where, W_{comp} is the directly measured power consumption of the compressor.

The uncertainty analysis was conducted using a MATLAB program integrated with the data reduction program, following the method of Moffat (1988). According to the method, the uncertainty in a calculated value R , which is a function of N independent measurements ($X_1, X_2, X_3, \dots, X_N$), can be calculated as following:

$$\delta R = \left(\sum_{i=1}^N \left(\frac{\partial R}{\partial X_i} \delta X_i \right)^2 \right)^{\frac{1}{2}} \quad \text{Eqn (59)}$$

where, the uncertainty of each measurement is determined using **Eqn (60)**:

$$\delta X_i = \left(\delta X_{i,A}^2 + \delta X_{i,B}^2 \right)^{\frac{1}{2}} \quad \text{Eqn (60)}$$

where, $\delta X_{i,A}$ is the type-A uncertainty of measurement X_i , which was calculated as the standard deviation of each group of data recorded every two minutes. The type-B uncertainty $\delta X_{i,B}$ was either provided by the manufacturer or by the calibration: for example, it is $\pm 0.2^\circ\text{C}$ for the refrigerant temperatures after the calibration of the type-T thermocouple probes with RTD sensors (PT100).

For the outdoor cooling capacity based on the air-side, the relative uncertainty was 5.1%. The relative uncertainties of heating capacity are 1.3% and 4.0% for refrigerant- and air-side approaches, respectively. Overall, the refrigerant side is more accurate in our experiments, because the airflow rates are high and the air-side temperature differences (for example, $T_{eai} - T_{eao} < 3^\circ\text{C}$) are not significantly larger than the uncertainty of the thermocouples.

5.4 Effects of environmental conditions: outdoor temperature and humidity

In this dissertation, 0°C and 90% RH is the most demanding condition considering the operating time of the heat pump system. Due to the high face velocity of the outdoor airflow ($V_{eai} > 3.0 \text{ m/s}$), the large air-side heat transfer area of the outdoor coil, and the two-phase refrigerant at

the exit of the outdoor evaporator, the temperature difference between air inlet temperature T_{eai} and the evaporating temperature T_{ero} is smaller than 3°C. As a result, the surface temperature of the outdoor evaporator is higher than 0°C when the air inlet temperature T_{eai} is 2°C. In other words, there is no frosting issue when T_{eai} is 2°C for our HP system, which is unique compared to many other frosting research on stationary systems. On the other hand, because the humidity ratio drops by three times as the ambient air temperature T_{eai} reduces from 0 to -10°C, frost accumulates much slower in colder conditions. In summary, 0°C and 90% RH is the most important experimental condition for the HP system in this dissertation.

5.4.1 Effects of ambient air temperature T_{eai}

Figure 5.4 shows the effects of ambient air temperature T_{eai} varying from -10 to 0°C on the performance of the CO₂ HP system and the thermal-hydraulic performance of the vertically oriented outdoor microchannel heat exchanger during continuous frosting and defrosting cycles. The relative humidity was fixed at 90% and the defrosting was initiated when the outdoor air-side pressure drop DP_{ea} reached 5 times the initial value $DP_{ea,ini}$. In addition, **Figure 5.5** shows the images of the vertically oriented outdoor evaporator during three continuous frosting cycles with 5 times initial DP_{ea} as defrost-initiation criterion in the 0°C and 90% RH condition.

a. The general trend of one frosting cycle

For a typical frosting cycle, water condenses on the coldest spots of the outdoor coil surface: starting from the second header (due to the low air-side heat transfer coefficient outside

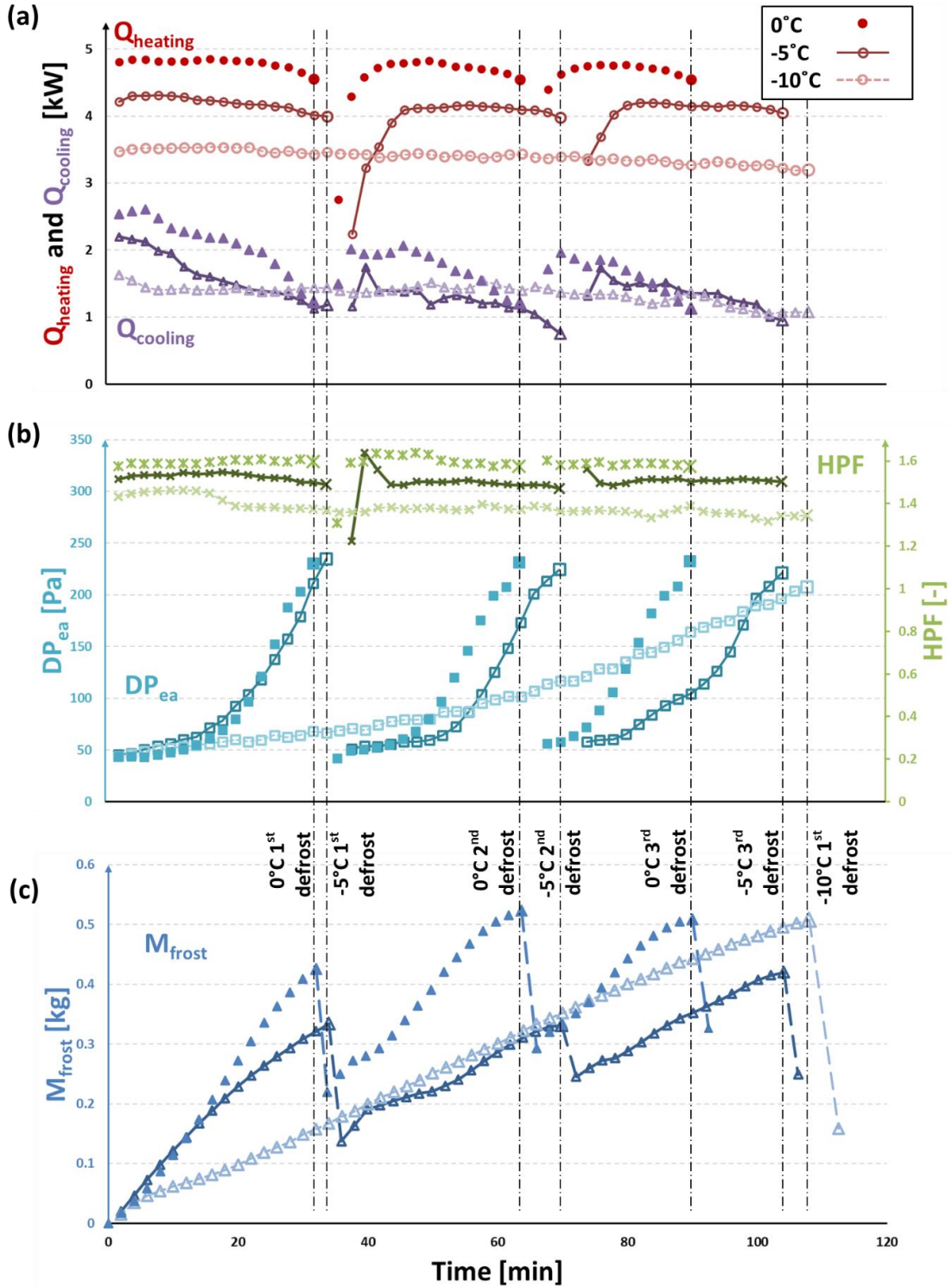


Figure 5.4: The effects of outdoor air temperature on (a) the heating and cooling capacity of the HP system; (b) the air-side pressure drop of the outdoor evaporator and the efficiency of the HP system; (c) the mass of frost accumulated on the vertically oriented outdoor evaporator, during continuous frosting cycles with 5 times initial DP_{ea} as defrost-initiation criterion in 90% RH conditions. Only the defrost initiations are labeled with vertical dash-dot lines.

Table 5.2: Mass of frost and meltwater for continuous frosting/defrosting cycles
(vertical outdoor coil, $5 \times DP_{ea,ini}$ and 90% RH outdoor condition)

Parameter	0°C			-5°C			-10°C
	1 st cycle	2 nd cycle	3 rd cycle	1 st cycle	2 nd cycle	3 rd cycle	1 st cycle
M_{frost} [kg]	0.426	0.523	0.509	0.334	0.331	0.420	0.507
$M_{drained}$ [kg]	0.043	0.111	0.155	0.008	0.016	0.108	0.113
$M_{retained}$ [kg]	0.219	0.293	0.327	0.138	0.245	0.249	0.159
$M_{vaporized}$ [kg]	0.164	0.119	0.028	0.189	0.069	0.062	0.236

the header and the low local refrigerant quality in the header), then the microchannel tube, and finally the louvered fins due to the fin efficiency. Also, the frost distributes on the outdoor evaporator unevenly by the end of a frosting cycle, as shown in the top-right image in **Figure 5.5**: first, there are two frost-free triangles at the left (4 tubes wide) and right bottom (9 tubes wide) of the second pass of the outdoor evaporator due to the uneven distribution of the two-phase refrigerant in the second pass. More liquid enters the middle tubes of the second pass. Secondly, the thickness of the frost is higher at the top than it is at the bottom of the second pass since refrigerant evaporates along the microchannel tubes. This is shown by the local frost images at the end of one frosting cycle in **Figure 5.5**.

Figure 5.6 presents the T-h diagram of the CO₂ HP system at the beginning and the end of the first frosting cycle in different ambient temperature conditions. As frost accumulates on the outdoor evaporator, it blocks the airflow area and increases the thermal resistance between air and refrigerant. This leads to a decreased $Q_{cooling}$ and x_{ero} , as well as a significant refrigerant migration into the accumulator. At the beginning of frosting cycles, x_{ero} is most likely higher than $x_{acc,ro}$ because steam is added into the outdoor chamber, and the capacity of the outdoor evaporator increases, as shown in **Figure 5.6**. During the frosting cycles, x_{ero} drops more than 0.16 for all three conditions, but the accumulator was able to maintain the quality $x_{acc,ro}$ in the conditions of

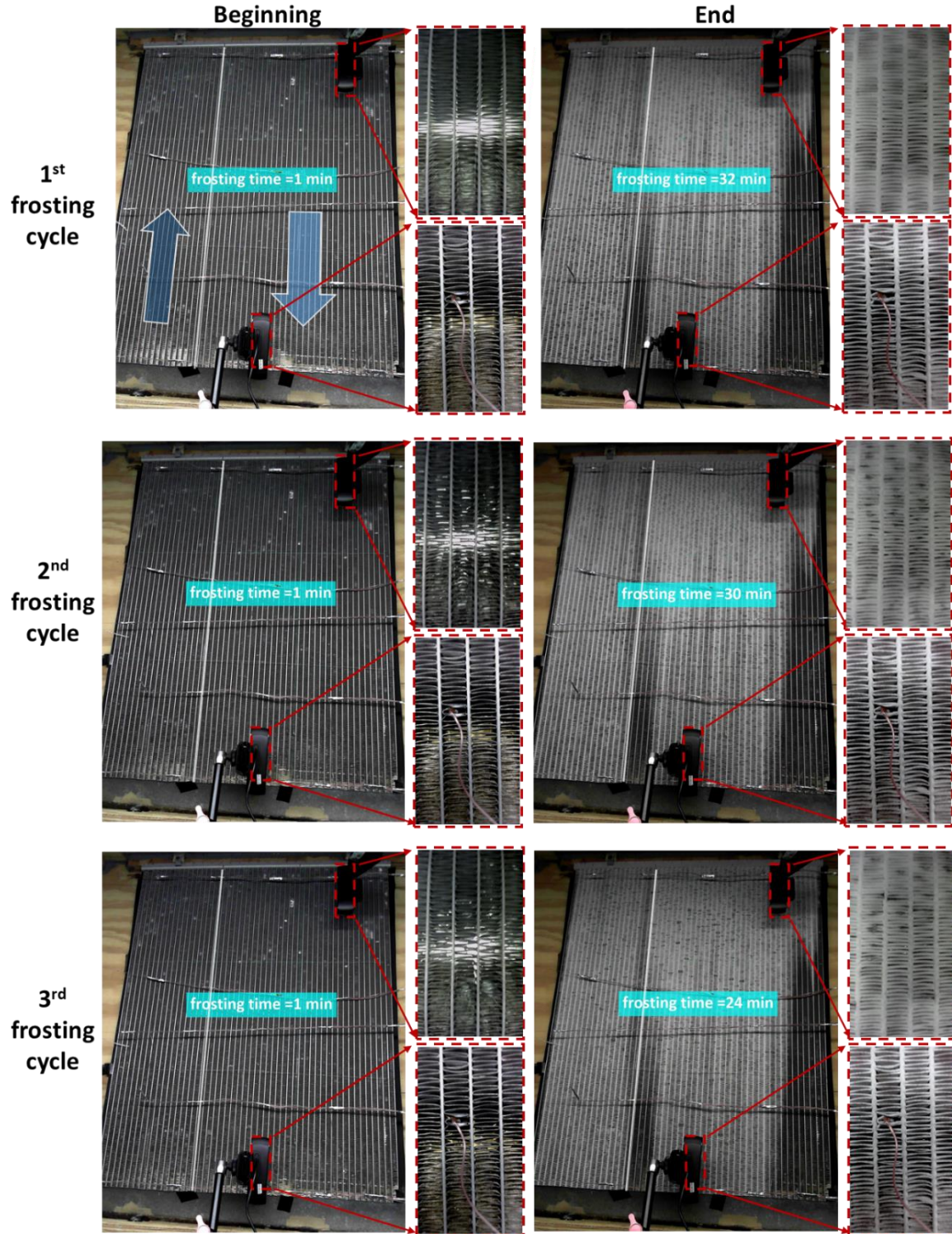


Figure 5.5: The images of the vertically oriented outdoor evaporator at the beginning and the end of three continuous frosting cycles with 5 times initial DP_{ea} as defrost-initiation criterion in the 0°C and 90% RH outdoor condition. The refrigerant flow direction is shown using blue arrows in the left top image, and the local frost images of the right-top and left-bottom corner of the second pass are presented in red rectangles.

0 and -5°C as liquid refrigerant migrates into the accumulator. Although too much liquid refrigerant leaves the accumulator in the -10°C frosting condition and it requires an accumulator with a larger effective volume.

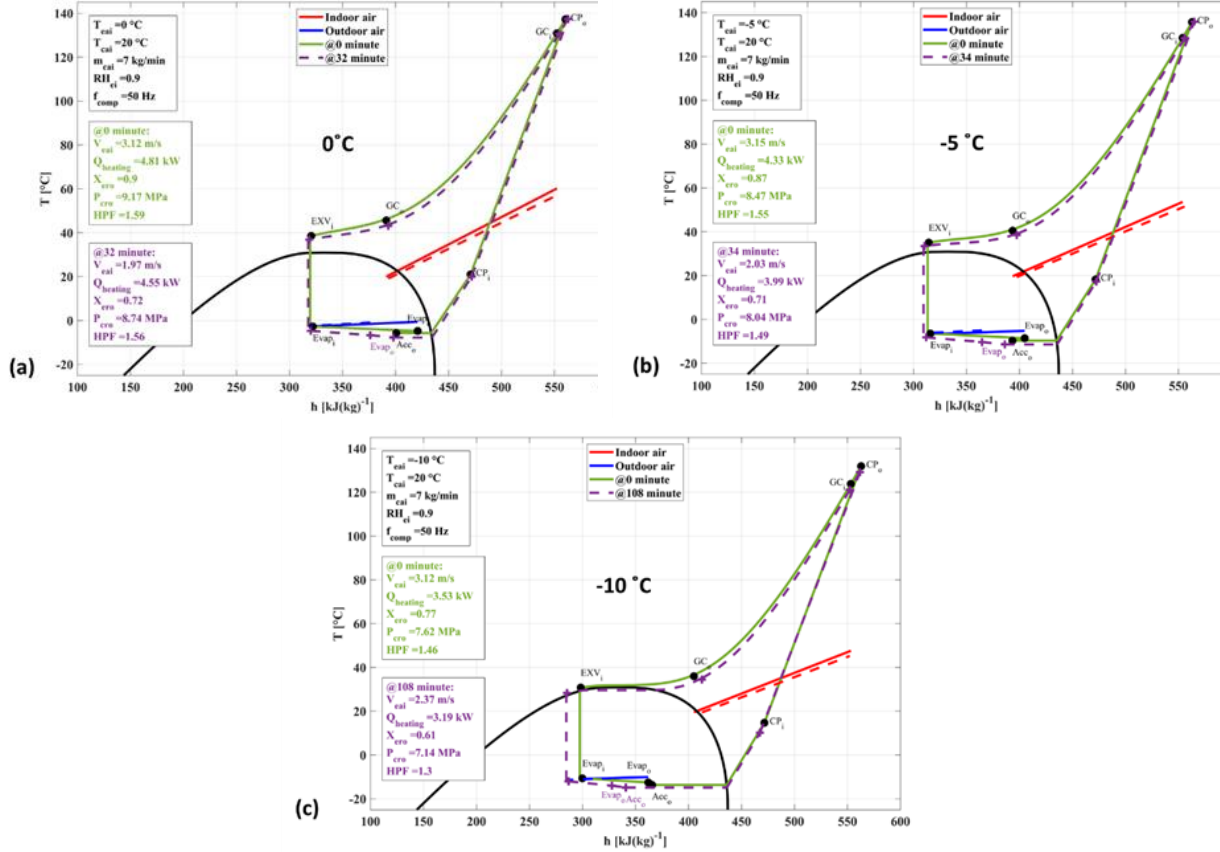


Figure 5.6: The temperature-enthalpy diagram of the CO₂ heat pump system with the vertically oriented outdoor coil at the beginning and the end of the first frosting cycle with 5 times initial DP_{ea} as defrost-initiation criterion in 0/-5/-10°C and 90% RH conditions.

In our experimental conditions, the heating capacity Q_{heating} and the efficiency (Heating Performance Factor, $\text{HPF} = Q_{\text{heating}}/W_{\text{comp}}$) drop less than 10% during one frosting cycle with 5 times $DP_{ea,ini}$ as the defrost-initiation criterion (**Figure 5.4 (a) and (b)**). This indicates the potential of a longer operating time of the HP system if a larger performance reduction can be tolerated. The benefit of a longer operating time is that the defrosting can be conducted when the vehicle is

parking in the parking lot or charging center, and the impacts of reverse-cycle defrosting on passengers' comfort can be minimized.

b. The periodic performance

The periodic performance of the HP system can be affected by both the restart-up of the system and the retained meltwater on the outdoor evaporator after each reverse-cycle defrosting. As shown in **Figure 5.4(a)**, it took 2 and 8 minutes, respectively, for the cooling and heating capacity ($Q_{cooling}$ and $Q_{heating}$) of the HP system to reach the peak value in the re-frosting cycles because of the thermal mass of the heat exchangers and tubing.

The long blue dash lines in **Figure 5.4 (c)** show the water removal during each defrosting process: the beginning point is the total mass of frost accumulation by the end of each frosting cycle M_{frost} , and the end is the total mass of water retention on the outdoor coil after each defrosting cycle $M_{retained}$. Based on the images of the microchannel tubes and fins at the beginning of re-frosting cycles (**Figure 5.5**), it is clear that the defrosting with the current defrost-termination criterion cannot completely remove all meltwater by drainage and vaporization. More than 0.130 kg of meltwater stays on the outdoor coil after the first defrosting. Also, the mass of retained water $M_{retained}$ increases up to 0.327 kg with the cycle number in our experimental conditions. As a result, the air-side pressure drop DP_{ea} at the beginning of each frosting cycle increases from 43.6 to 56.2 Pa in the 0°C condition from the first to the third cycle (**Figure 5.4(b)**). Moreover, **Figure 5.4(a)** shows that the peak value of the cooling capacity $Q_{cooling}$ decreases by 25% in the 0°C condition from the first to the third cycle due to the extra thermal resistance of the water film on the outdoor coil.

In addition, the frost accumulation on the coil by the end of each frosting cycle M_{frost} increases with the cycle number. **Table 5.2** presents the mass of the frost accumulation M_{frost} and the distribution of $M_{retained}$, $M_{drained}$, and $M_{vaporized}$ for various ambient air temperature conditions. As the number of cycles increases from one to three, M_{frost} increases from 0.426 to 0.509 kg in the 0°C condition (**Table 5.2**). Because retained water droplets and film freeze and high-density ice, instead of low-density frost, grows on the surface of the outdoor evaporator in the re-frosting cycles. As shown in **Figure 5.5**, there are frozen droplets on the middle part of the louvered fins at the right-top corner of the outdoor evaporator by the end of re-frosting cycles in the local frost images. Following the increased M_{frost} , the retained and drained water ($M_{retained}$ and $M_{drained}$) increased from 0.219 to 0.327 kg and from 0.043 to 0.155 kg in the 0°C condition. On the other hand, the mass of vaporized water $M_{vaporized}$ decreases from 0.164 to 0.028 kg with the cycle number in 0°C condition, as shown in **Table 5.2**. This is due to the increased mass, thickness, and thermal resistance of the water film during the defrosting process. The surface temperature of the water film decreases significantly and so does the kinetic energy of the water molecules on the air-water interface. Similar trends can be observed in the -5°C outdoor condition.

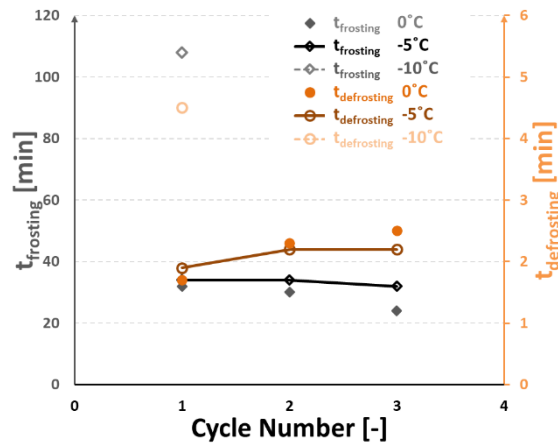


Figure 5.7: The effects of outdoor air temperature on the frosting and defrosting time of the vertically oriented outdoor coil during continuous cycles with 5 times initial DP_{ea} as defrost-initiation criterion in the 90% RH condition.

Due to the increased water retention, the time of the frosting cycle $t_{frosting}$ reduces for the refrosting cycles. As shown in **Figure 5.7**, $t_{frosting}$ decreases from 32 to 24 minutes from the first to the third frosting cycle at the 0°C condition. In the meanwhile, since M_{frost} increases with the cycle number, the energy required to defrost and the defrosting time $t_{defrosting}$ also increase with the cycle number. The trends can be observed in 0 and -5°C conditions, as shown in **Figure 5.7**. As the total operating time increases, heating capacity, efficiency, and $t_{frosting}$ reduce but $t_{defrosting}$ increases. Thus, the periodic performance of the heat pump system drops with the total operating time.

c. The effects of ambient air temperature T_{eai}

The ambient air temperature T_{eai} affects the capacity, efficiency, frost accumulating speed, and thus the operating time of the HP system. As shown in **Figure 5.4(a)**, the peak value of $Q_{heating}$ reduces from 4.85 to 3.54 kW when the ambient air temperature drops from 0 to -10°C during the first frosting cycle. Similarly, the peak value of $Q_{cooling}$ decreases from 2.61 to 1.67 kW and HPF reduces from 1.61 to 1.47 (**Figure 5.4(a) and (b)**). As T_{eai} drops, the evaporating temperature T_{ero} and pressure P_{ero} decrease. So, the vapor density at the suction point of the compressor and the mass flow rate $\dot{m}_r$ decrease. Thus, the heating and cooling capacity, as well as the efficiency of the HP system reduce with T_{eai} . **Figure 5.6** presents more details of the system operation in different T_{eai} conditions.

The ambient air temperature also affects the water content significantly in our test conditions. As the ambient air temperature T_{eai} was 0, -5, and -10°C and the relative humidity RH_{ei} was fixed to be 90% in the experimental conditions, the humidity ratio was 3.4, 2.2, and 1.4 g water per kg dry air, respectively. Following the decrease of the humidity ratio, a significant reduction in frost accumulating speed was recorded, as shown in **Figure 5.4(c)**. In the first 32

minutes, the frost accumulating speed was 0.013, 0.010, and 0.005 kg/min, which was almost proportional to the humidity ratio, when T_{eai} was 0, -5, and -10°C. The frost images in the left column in **Figure 5.8** also present a considerably slower frost accumulation at the 20th minute in the -10°C and 90% RH condition, but the frost growth is similar at the same moment in 0°C and -5°C conditions.

However, the operating time $t_{frosting}$ during the first frosting cycle was 32, 34, and 108 minutes (**Figure 5.7**). On the one hand, it has the shortest operating time for our HP system in the outdoor condition of 0°C and 90% RH, so it is the most demanding condition for defrosting. On the other hand, the operating time $t_{frosting}$ was not completely inversely proportional to the frost accumulating speed since the final mass M_{frost} was not identical for different T_{eai} conditions. This is likely due to the different average density of the frost in different T_{eai} conditions: Assuming the volume of the frost on the outdoor coil were the same by the end of the first frosting cycles (since the DP_{ea} are approximately the same), the density of the frost in different T_{eai} conditions can be compared directly using the M_{frost} at the end of first frosting cycles. It is obvious that the frost density is lowest when T_{eai} was -5°C, then follows the frost density when T_{eai} was 0°C, and it is highest for T_{eai} was -10°C. This conclusion is also supported by the frost images: As shown by the right column in **Figure 5.8**, both the surface area covered by white frost and the thickness of the frost are smallest at -10°C at the end of the first frosting cycle, and it is similar for the 0°C and -5°C conditions. Combined with the M_{frost} at the end of the first frosting cycles in **Table 5.2**, the frost density trend can be validated again.

In the literature, the density of the frost is well studied in the operating range of the freezer, refrigerator, and stationary heat pump system. But more research should be conducted in the future

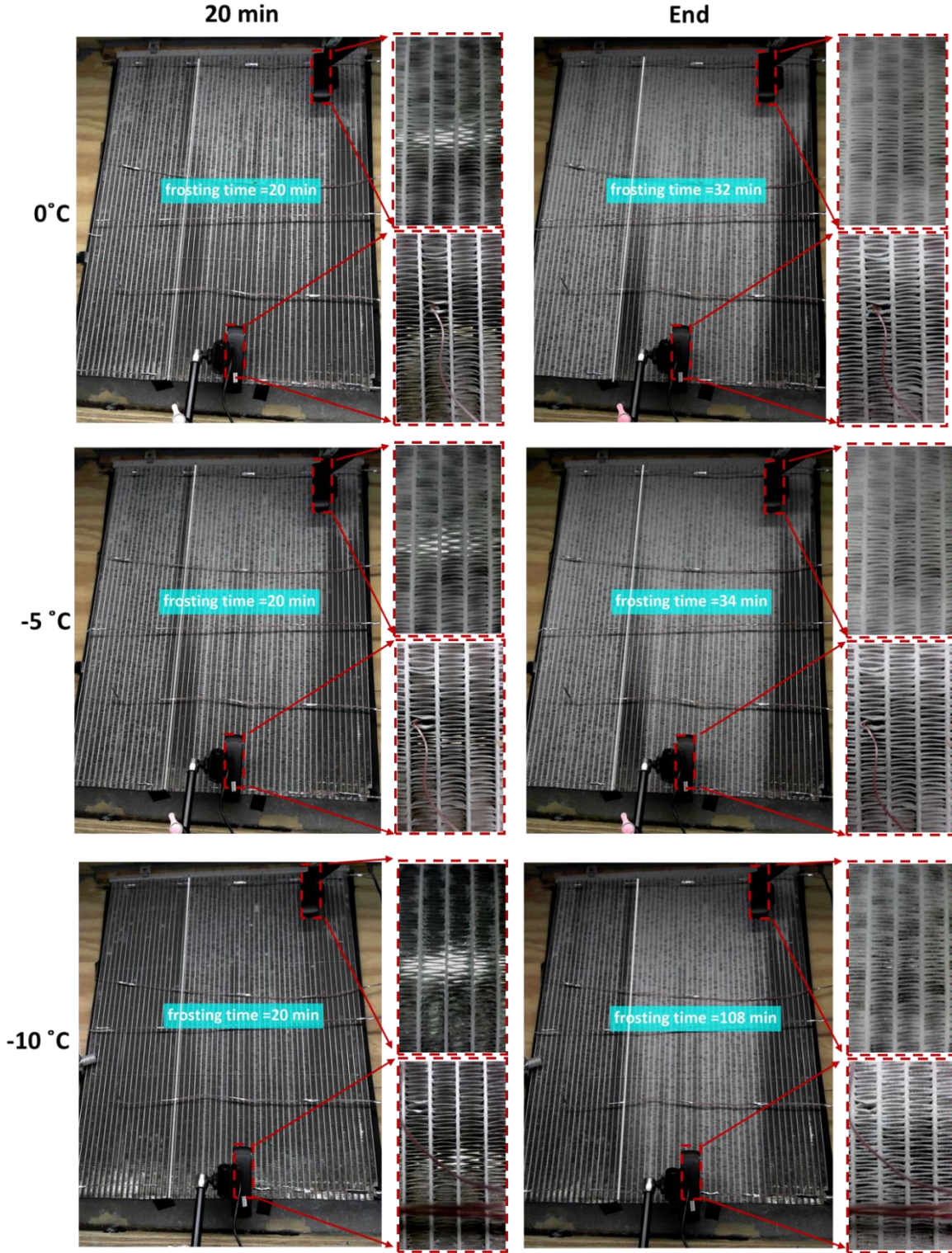


Figure 5.8: The images of the vertically oriented outdoor evaporator at the 20th minute and the end of the first frosting cycle with 5 times initial DP_{ea} as defrost-initiation criterion in 0/-5/-10°C and 90% RH conditions. The local frost images of the right-top and left-bottom corner of the second pass are presented in red rectangles.

to explore the properties of frost in the operating conditions of heat pumps. In other words, the conditions that the outdoor air face velocity is as high as the driving speed of a car, both ambient dry-bulb temperature and dew point are below but close to the freezing point of water, and the cold surface temperature is only 2 to 4 °C lower than the dry-bulb temperature of ambient air.

5.4.2 Effects of relative humidity of ambient air RH_{ei}

Figure 5.9 shows the effects of the relative humidity of outdoor air RH_{ei} (80% and 90%) on the performance of the CO₂ HP system and the thermal-hydraulic performance of the outdoor microchannel heat exchanger during continuous frosting and defrosting cycles in 0°C dry-bulb ambient conditions.

The relative humidity of ambient air RH_{ei} has a negligible impact on the capacities and efficiency at the beginning and end of one frosting cycle: they are identical at the beginning and drop the same amount during one frosting cycle for both 80% and 90% RH conditions (**Figure 5.9(a) and (b)**). However, RH_{ei} has a significant impact on the frost accumulating speed, the frost density, the final frost accumulation m_{frost} , and the time $t_{frosting}$ and $t_{defrosting}$. As shown in **Figure 5.9(c)**, the frost accumulating speed was 0.010 kg/min in the first 32 minutes in the 80% RH condition, and this speed dropped further to 0.005 kg/min later. But it was 0.013 kg/min in the first cycle in the 90% RH condition. This is because the humidity ratio decreases from 3.4 to 3.0 g of water per kg dry air when RH_{ei} decreases from 90% to 80% at 0°C. **Figure 5.10** presents the frost accumulation on the outdoor evaporator at the 20th minute and the end of the first frosting cycle for both 80% and 90% RH conditions when T_{eai} is 0°C. At the 20th minute, the frost grows significantly slower in 80% RH condition than it does in 90% RH condition. The frost mass

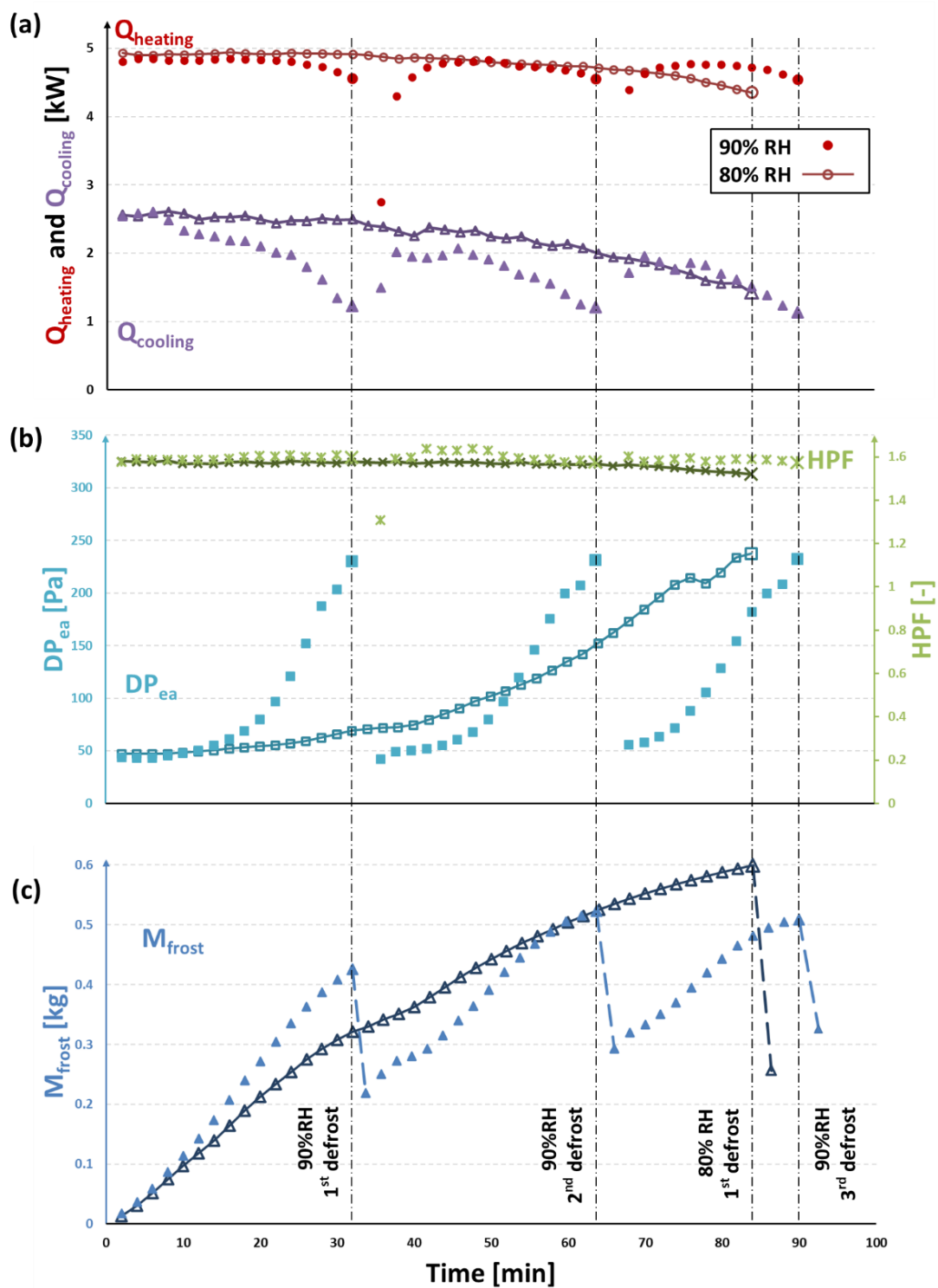


Figure 5.9: The effects of outdoor air relative humidity on (a) the heating and cooling capacity of the HP system; (b) the air-side pressure drop of the outdoor evaporator and the efficiency of the HP system; (c) the mass of frost accumulated on the vertically oriented outdoor evaporator, during continuous frosting cycles with 5 times initial DP_{ea} as defrost-initiation criterion in 0°C ambient conditions

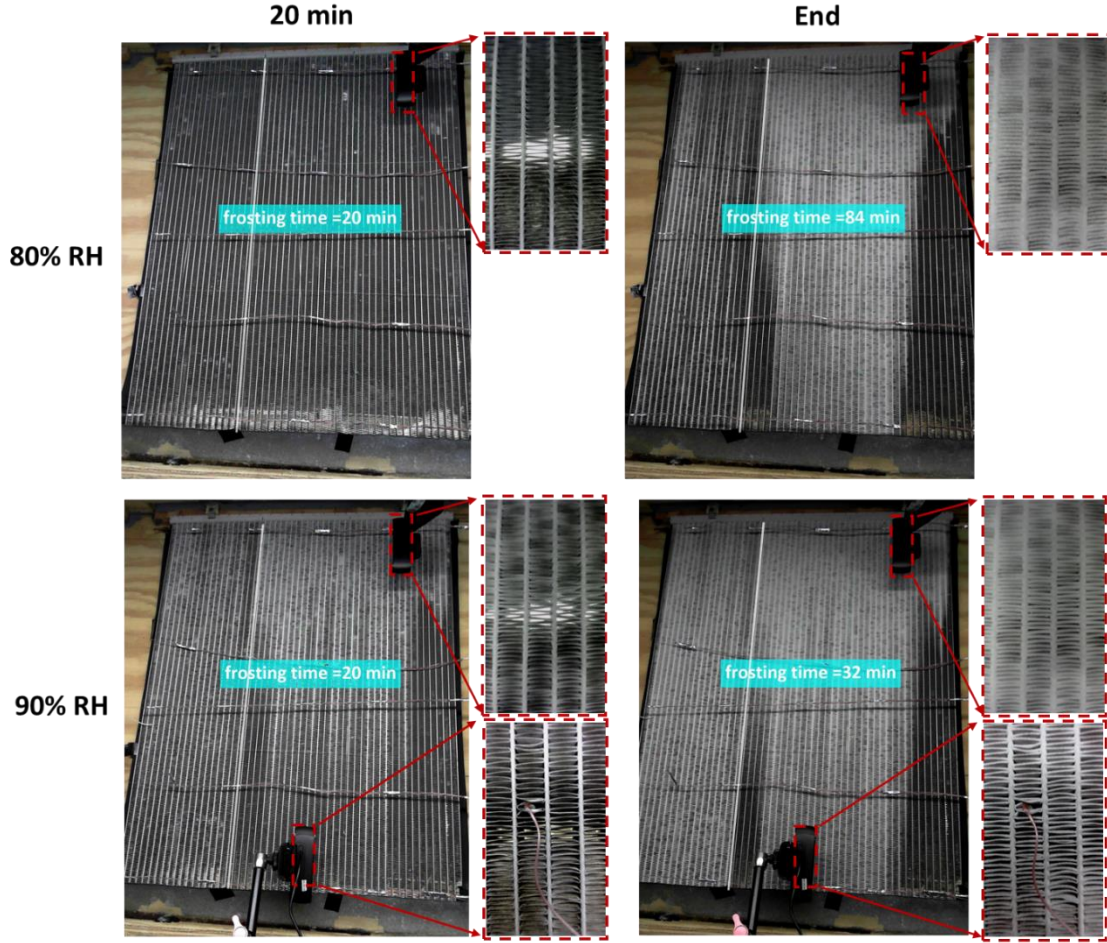


Figure 5.10: The images of the vertically oriented outdoor evaporator at the 20th minute and the end of the first frosting cycle with 5 times initial DP_{ea} as defrost-initiation criterion in 0°C and 80%/90% RH conditions. The local frost images of the second pass are presented in red rectangles

accumulation m_{frost} at the end of the first frosting cycle increases from 0.426 to 0.599 kg when RH_{ei} decreases from 90% to 80% at 0°C (**Figure 5.9(c)**). This indicates a higher frost density with lower RH_{ei} , which is also supported by the frost images at the end of the first frosting cycle in the right column of **Figure 5.10**. Also, the frosting time $t_{frosting}$ increases significantly from 32 to 84 minutes, and the defrosting time $t_{defrosting}$ increases slightly from 1.7 to 2.4 minutes when RH_{ei} decreases from 90% to 80% at 0°C. In conclusion, the frosting time $t_{frosting}$ can be affected by several parameters like the ambient air temperature, humidity ratio, and the density of the frost,

but the defrosting time $t_{defrosting}$ is mostly impacted by the mass of frost at the end of one frosting cycle m_{frost} .

5.5 Effects of defrost-initiation criteria: 5 vs. 10 times initial DP_{ea}

Figure 5.11 shows the effects of different defrost-initiation criteria (5 or 10 times the initial DP_{ea}) on the performance of the transcritical R744 HP system and the thermal-hydraulic performance of the outdoor microchannel evaporator in the 0°C and 90% RH outdoor condition during continuous frosting cycles. Similarly, **Figure 5.12** and **Figure 5.13** present the corresponding data in the -5°C and -10°C outdoor conditions. In addition, **Table 5.3** reports the mass of the frost accumulation, the mass of the water drainage, the mass of the water retention, and the mass of the water vaporization in various air temperatures and 90% RH outdoor conditions during continuous frosting cycles.

Table 5.3: Mass of frost and meltwater for continuous frosting/defrosting cycles (vertical outdoor coil and 90% RH outdoor condition)

Parameter	0°C and 90% RH						-5°C and 90% RH						-10°C and 90% RH	
	5× $DP_{ea,ini}$			10× $DP_{ea,ini}$			5× $DP_{ea,ini}$			10× $DP_{ea,ini}$			5× $DP_{ea,ini}$	10× $DP_{ea,ini}$
	1 st cycle	2 nd cycle	3 rd cycle	1 st cycle	2 nd cycle		1 st cycle	2 nd cycle	3 rd cycle	1 st cycle	2 nd cycle		1 st cycle	1 st cycle
M_{frost} [kg]	0.426	0.523	0.509	0.520	0.591		0.334	0.331	0.420	0.500	0.556		0.507	0.688
$M_{drained}$ [kg]	0.043	0.111	0.155	0.084	0.278		0.008	0.016	0.108	0.164	0.296		0.113	0.333
$M_{retained}$ [kg]	0.219	0.293	0.327	0.305	0.298		0.138	0.245	0.249	0.245	0.239		0.159	0.212
$M_{vaporized}$ [kg]	0.164	0.119	0.028	0.131	0.016		0.189	0.069	0.062	0.092	0.022		0.236	0.142

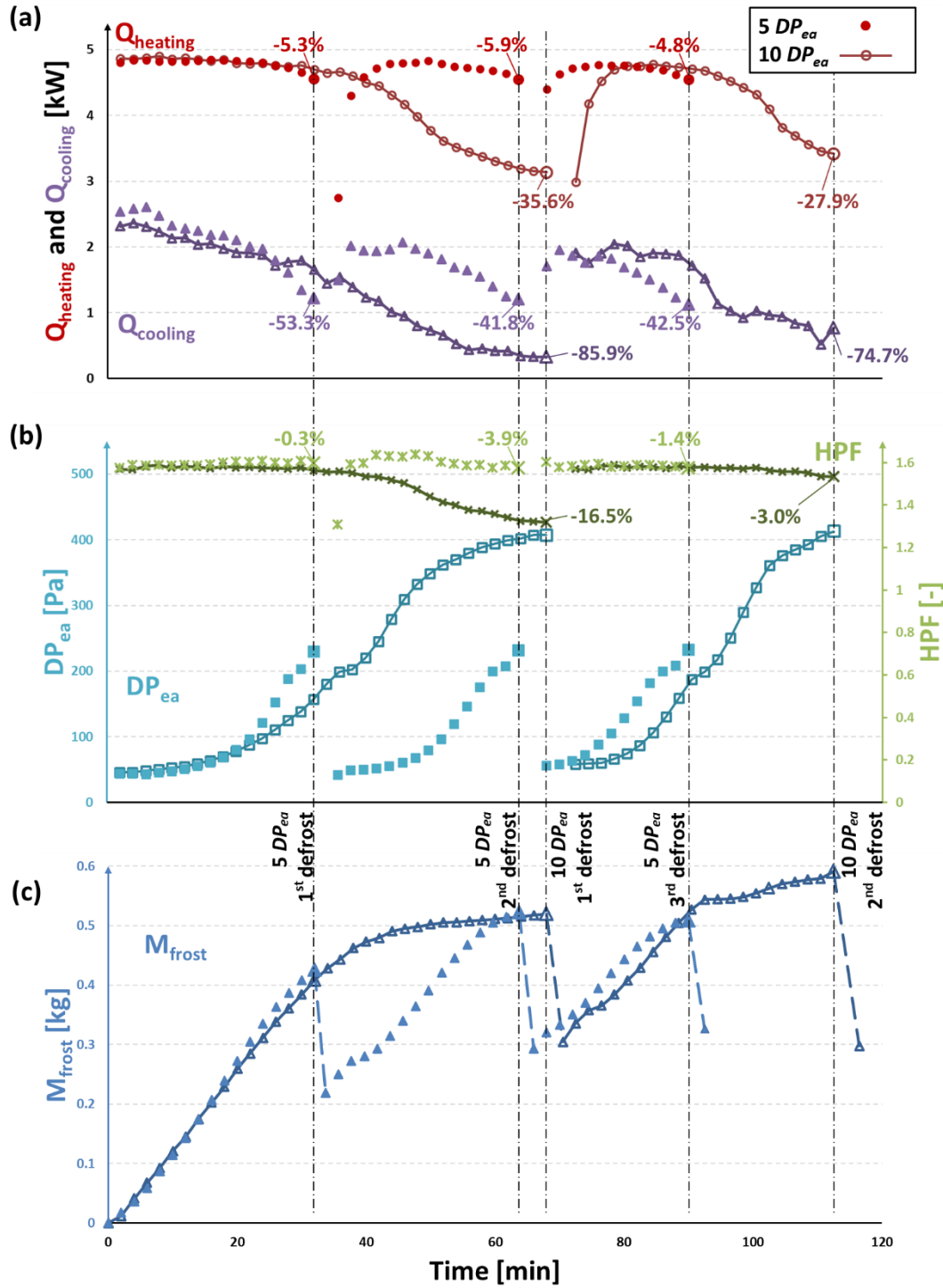


Figure 5.11: The effects of different defrost-initiation criteria on **(a)** the heating and cooling capacity of the HP system; **(b)** the air-side pressure drop of the vertically oriented outdoor evaporator and the efficiency of the HP system; **(c)** the mass of frost accumulated on the outdoor evaporator, during continuous frosting cycles in the 0°C and 90% RH condition. The defrost initiations are labeled with vertical dash-dot lines and the percentages of the performance reductions by the end of each frosting cycle are labeled.

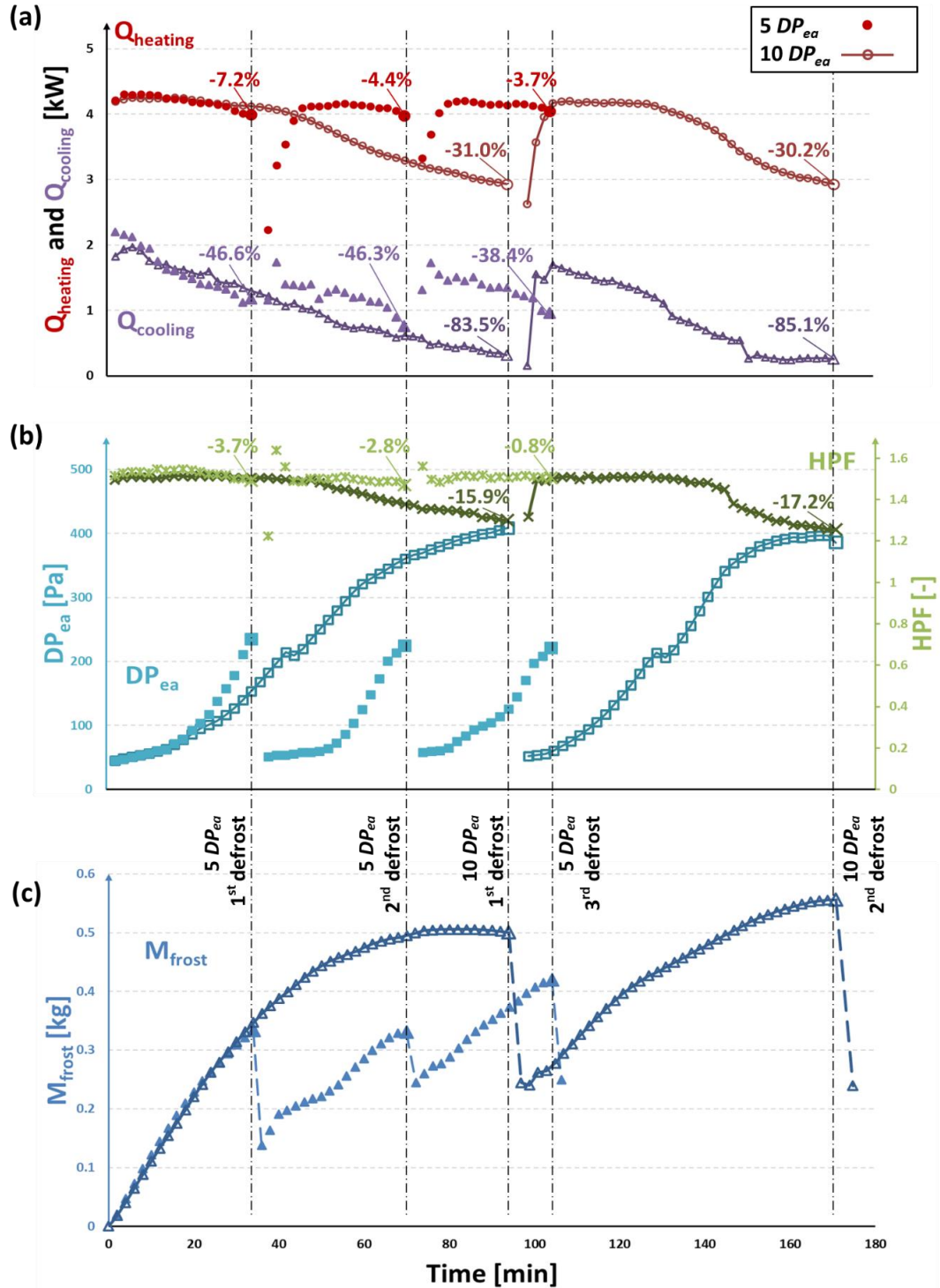


Figure 5.12: The effects of different defrost-initiation criteria on **(a)** the heating and cooling capacity of the HP system; **(b)** the air-side pressure drop of the vertically oriented outdoor evaporator and the efficiency of the HP system; **(c)** the mass of frost accumulated on the outdoor evaporator, during continuous frosting cycles in the -5°C and 90% RH condition. The defrost initiations are labeled with vertical dash-dot lines and the percentages of the performance reductions by the end of each frosting cycle are labeled.

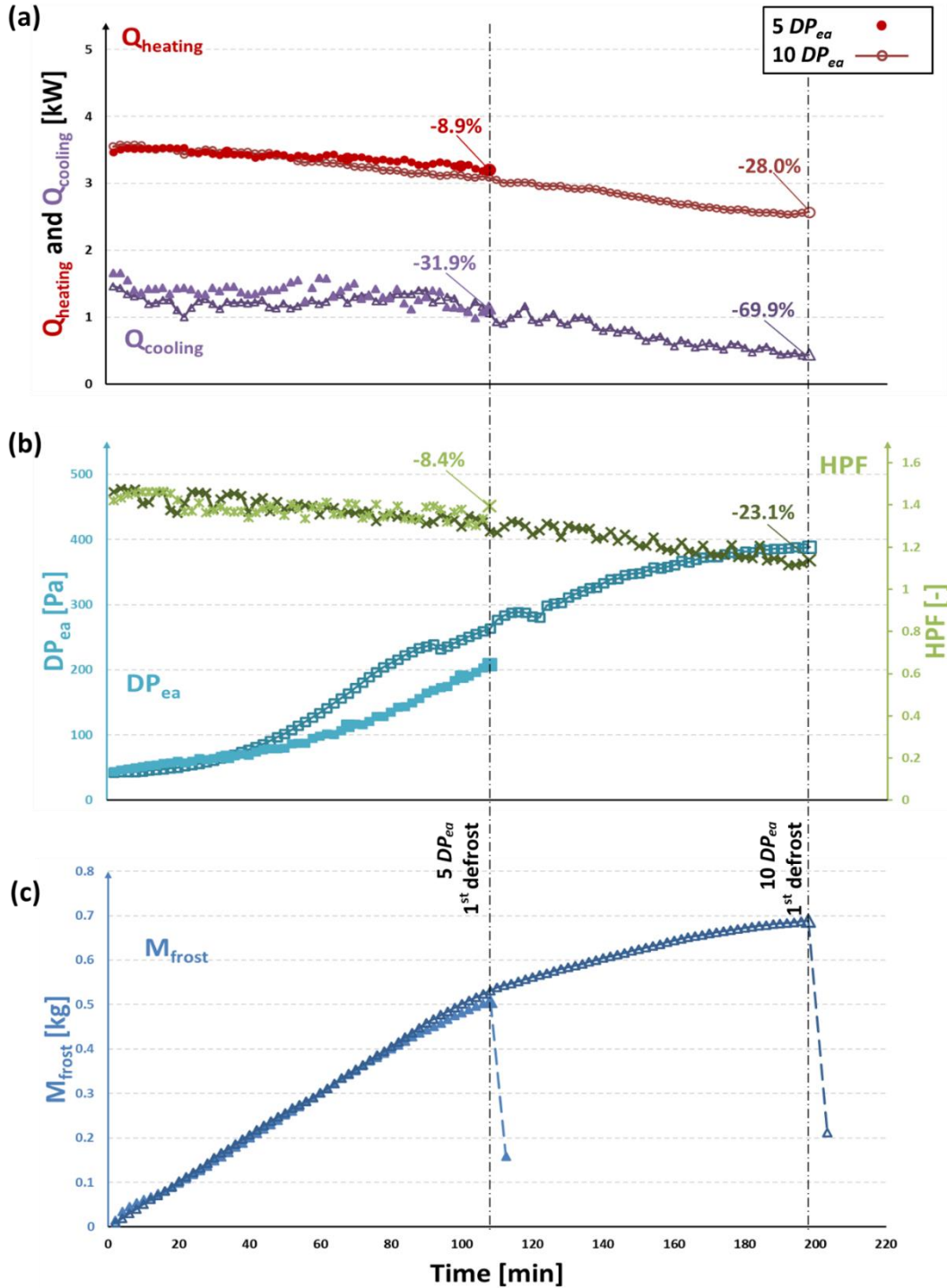


Figure 5.13: The effects of different defrost-initiation criteria on (a) the heating and cooling capacity of the HP system; (b) the air-side pressure drop of the vertically oriented outdoor evaporator and the efficiency of the HP system; (c) the mass of frost accumulated on the outdoor evaporator, during continuous frosting cycles in the -10°C and 90% RH condition. The defrost initiations are labeled with vertical dash-dot lines and the percentages of the performance reductions by the end of each frosting cycle are labeled.

5.5.1 Effects on the HP system performance

a. In the first frosting cycle

At the end of the first frosting cycle in 0°C outdoor condition, the heating capacity $Q_{heating}$ reduces from the initial value of 4.808 to 4.552 kW (only 5.3%) using 5 times the initial DP_{ea} as defrost-initiation criterion, but it drops from to 3.135 kW (35.6%) using 10 times the initial DP_{ea} as defrost-initiation criterion (**Figure 5.11(a)**). Similarly, the cooling capacity of the outdoor evaporator $Q_{cooling}$ decreases by 53.3% and 85.9% using 5 and 10 times the initial DP_{ea} as defrost-initiation criterion in the 0°C outdoor condition. In the meanwhile, the efficiency HPF stays almost constant for 5 times the initial DP_{ea} , and it drops from 1.58 to 1.32 (16.5%) for 10 times the initial DP_{ea} . However, the operating time $t_{frosting}$ increases significantly from 32 to 68 minutes, which is slightly more than double when the defrost-initiation criterion changes from 5 to 10 times the initial DP_{ea} (as shown in **Figure 5.15(a)**).

The data during the continuous frosting/defrosting cycles in the -5 and -10°C outdoor conditions show similar trends compared to the data in the 0°C outdoor condition. In the first frosting cycle of -5 °C and 90% RH outdoor condition (**Figure 5.12(a)**), $Q_{heating}$ and HPF reduce by 7.2% and 3.7% for 5 times the initial DP_{ea} , and they decrease by 31.0% and 15.9% for 10 times of the initial DP_{ea} as defrost-initiation criterion. $Q_{cooling}$ decreases by 46.6% and 83.5% using 5 and 10 times the initial DP_{ea} as defrost-initiation criterion, respectively. Also, the operating time $t_{frosting}$ almost increases by three times from 34 to 94 minutes, as shown in **Figure 5.15(b)**. In the only frosting cycle of -10 °C and 90% RH outdoor condition (**Figure 5.13(a)**), $Q_{heating}$ and HPF reduce by 8.9% and 8.4% for 5 times the initial DP_{ea} , and they decrease by 28.0% and 23.1% for 10 times of the initial DP_{ea} as defrost-initiation criterion. $Q_{cooling}$ decreases by 31.9% and 69.9% using 5 and

10 times the initial DP_{ea} as defrost-initiation criterion, respectively. Also, the operating time $t_{frosting}$ almost doubles from 108 to 198 minutes, as shown in **Figure 5.15(c)**.

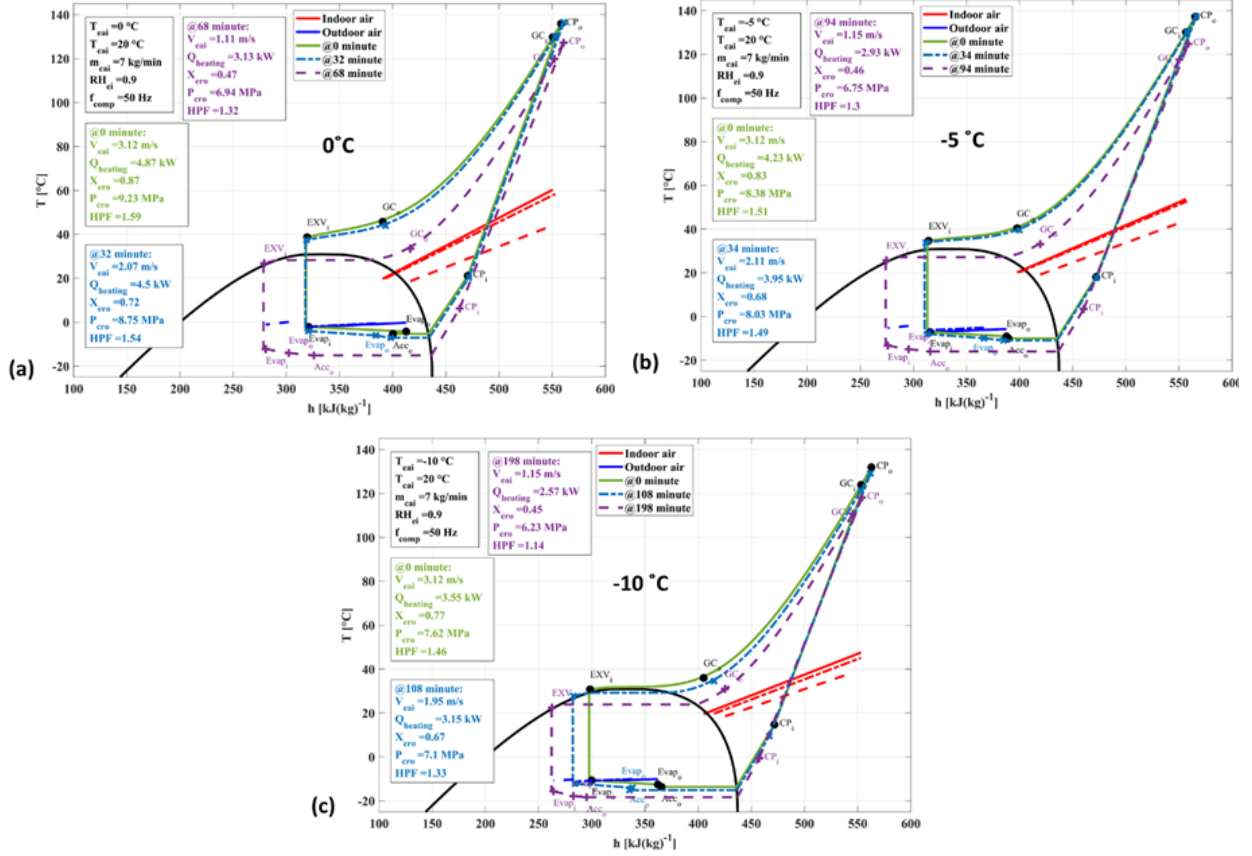


Figure 5.14: The temperature-enthalpy diagram of the CO₂ heat pump system at the beginning and the end of the first frosting cycle with 5 and 10 times initial DP_{ea} as defrost-initiation criterion in 0/-5/-10 °C and 90% RH outdoor conditions.

In all conditions, the reduction of the efficiency HPF is smaller than it is for the capacity $Q_{heating}$. Because the compressor power W_{comp} reduces with the mass flow rate $\dot{m}_r$ and the refrigerant density ρ_{cpr} at the suction point of the compressor, as the frost accumulates on the outdoor evaporator and the evaporating temperature and pressure drops. Also, the cooling capacity $Q_{cooling}$ drops almost with constant speed during each frosting cycle, but the heating capacity $Q_{heating}$ stays relatively stable in the early stage of the frosting cycles and drops faster later. This is

most likely because the accumulator holds extra refrigerant during the early stages of the frosting cycle, but it gets flooded later and cannot maintain the suction superheat.

Figure 5.14 illustrates the shifts of the T-h diagram of the CO₂ HP cycle during the first frosting cycle in various temperatures and 90% RH outdoor conditions. The refrigerant states at the beginning and the end of the frosting cycles using 5 and 10 times the initial DP_{ea} as defrost-initiation criterion are presented using different colors- green (black dots and labels), blue, and purple lines. Some key parameters like the outdoor air velocity V_{ei} and the evaporator exit quality x_{ero} at different moments are also listed on the plots.

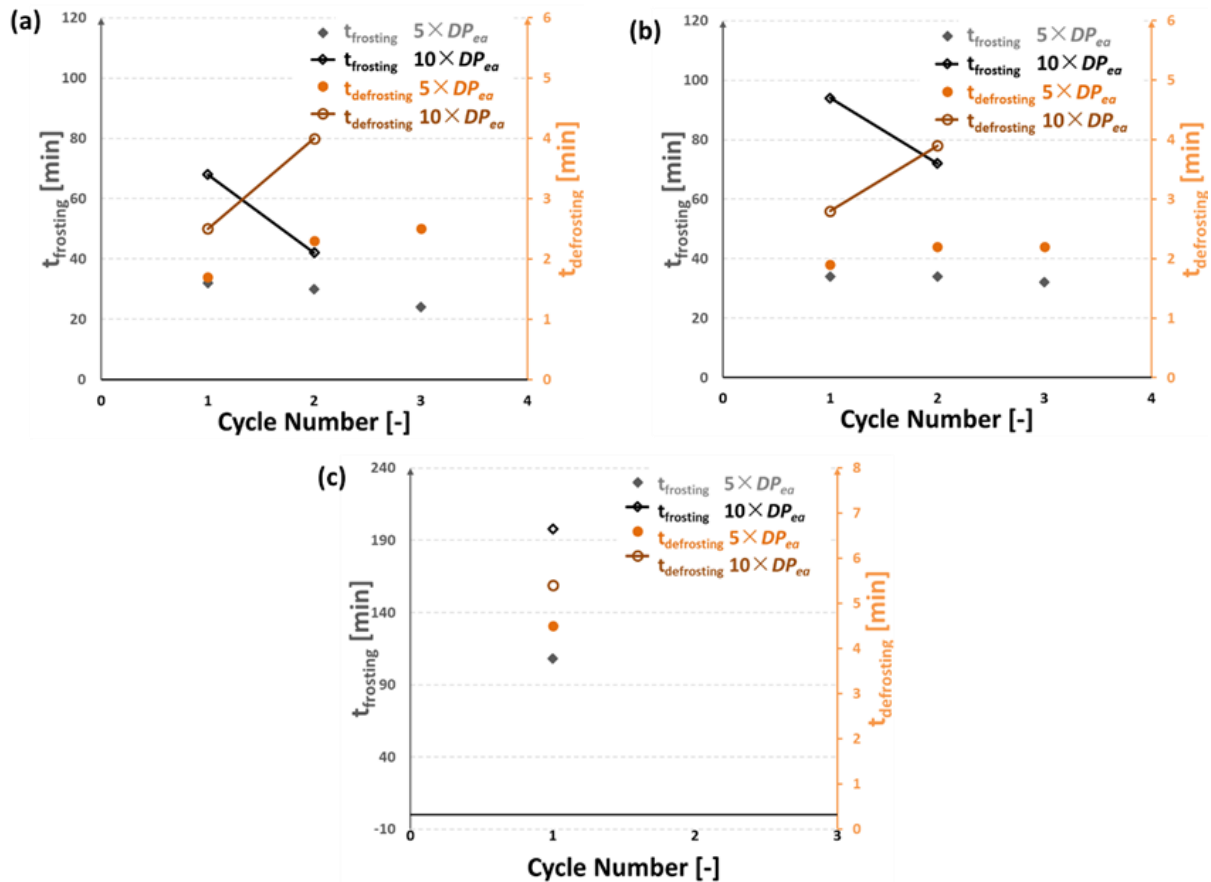


Figure 5.15: The effects of defrost-initiation criteria on the frosting and defrosting time during continuous frosting/defrosting cycles in various temperatures and 90% RH outdoor conditions: (a) 0°C; (b) -5°C; (c) -10°C.

As mentioned earlier, the performance reduction is negligible when the air-side pressure drop DP_{ea} reaches 5 times the initial value. This is also presented by the almost overlapped blue and green T-h diagrams in all three conditions, as shown in **Figure 5.14**. The major difference between the blue and green T-h diagrams is the refrigerant quality at the evaporator exit x_{ero} drops by 0.15, 0.15, and 0.10 in the 0, -5, and -10°C and 90% RH outdoor conditions, as the frost grows on the outdoor evaporator and the cooling capacity drops. Also, this indicates liquid refrigerant migrates from the high-pressure side into the accumulator. Because x_{ero} drops but the quality at the accumulator exit $x_{acc,ro}$ stays the same from the beginning until DP_{ea} reaches 5 times the initial value, and the high-side pressure drops slightly at the same time. Therefore, the accumulator holds the extra refrigerant due to frosting and the high-effectiveness IHX keeps the suction superheat, and thus the discharge temperature T_{cpro} is almost constant when DP_{ea} reaches 5 times the initial value in all three conditions. However, as the frost continues growing and densifying on the outdoor coil, the evaporating pressure and temperature, as well as the cooling capacity further decrease. The accumulator is completely flooded when DP_{ea} reaches 10 times the initial value, as shown by the purple T-h diagrams in **Figure 5.14**. Both x_{ero} and $x_{acc,ro}$ drop by about 0.4 and 0.3 in the three conditions from the beginning until DP_{ea} reaches 10 times the initial value. At the same time, the pressure and temperature at the compressor suction point decrease so the mass flow rate $\dot{m}_r$ drops with the reduced refrigerant density ρ_{cpri} at the suction point of the compressor. The compressor discharge pressure p_{cpro} also drops below the critical pressure of CO₂ and heating capacity drops significantly.

b. In the refrosting cycles

Similar to the performance reduction in the first frosting cycle, $Q_{heating}$ and HPF drop less than 6% and much more significantly with 5 and 10 times the initial DP_{ea} as the defrost-initiation criteria, respectively, by the end of the re-frosting cycles in 0°C outdoor condition (**Figure 5.11(a)**). In the -5°C outdoor condition, the performance reduction along the re-frosting cycles is smaller than 5% and higher than 15% with 5 and 10 times the initial DP_{ea} as the defrost-initiation criteria, as shown in **Figure 5.12(a)**.

On the other hand, the peak values of the heating capacity $Q_{heating}$ reduces slightly by 1.7% and 2.5% from the first frosting cycle to the last re-frosting cycles with 5 and 10 times the initial DP_{ea} as the defrost-initiation criteria in 0°C outdoor condition, respectively (**Figure 5.11(a)**). Similarly, the peak value of $Q_{heating}$ reduces by 2.5 % and 0.7% from the first to the last frosting cycles with 5 and 10 times the initial DP_{ea} as the defrost-initiation criteria in the -5°C outdoor condition (**Figure 5.12(a)**). This shows that the current defrost-termination criterion (terminating defrosting when refrigerant exit temperature from the outdoor coil reaches 45°C) is effective to recover the HP system performance for both defrost-initiation criteria.

However, the peak value of $Q_{cooling}$ drops by 24.9% and 13.1% from the first frosting cycle to the last re-frosting cycles with 5 and 10 times the initial DP_{ea} as the defrost-initiation criteria in the 0°C outdoor condition (**Figure 5.11(a)**). Also, the operating time $t_{frosting}$ reduces by 8 and 26 minutes from the first to the last frosting cycle with 5 and 10 times the initial DP_{ea} as the defrost-initiation criterion, respectively (**Figure 5.15**). It shows a similar trend for the -5 °C outdoor condition (**Figure 5.12(a)**). This indicates that the imperfectness of water removal leads to the reduced operating time $t_{frosting}$ in the refrosting cycles in all conditions.

For the defrosting time in the 0°C outdoor condition, it takes 1.7, 2.3, and 2.5 minutes to defrost the outdoor coil for the three continuous cycles using 5 times initial DP_{ea} as the defrost-initiation criterion. It takes 2.5 and 4.0 minutes to defrost for the two continuous cycles using 10 times initial DP_{ea} as the defrost-initiation criterion. Percentage-wise, the defrosting time is 7.0% and 5.7% for using 5 and 10 times the initial DP_{ea} as the defrost-initiation criterion in the experiments. Similar trends can be observed in -5 and -10°C outdoor conditions.

Overall, the capacity and efficiency of the CO₂ HP system drop less than 9% during the frosting cycles with 5 times the initial DP_{ea} as the defrost-initiation criterion in all our experimental conditions and all the continuous cycles. Also, the capacity and efficiency drop by about 30% and 15% during the frosting cycles with 10 times the initial DP_{ea} as the defrost-initiation criterion. The operating-time-averaged performance reduction is even lower than the aforementioned values. This indicates the HP system has the potential to operate for a longer period if this mildly reduced capacity can be tolerated, although the optimal defrost-initiation criterion is still open to discussion. The advantage is that the driver will be able to defrost the outdoor coil using indoor thermal energy when the passengers are not in the vehicle. For example, after a 68-minute commute, which is beyond the average commuting time for Americans (50 minutes), defrosting is conducted when the electric vehicle is in the parking lot or charging center. In this scenario, the effects of defrosting on passenger comfort can be minimized, and the heat in the cabin and even the electricity from the grid can be utilized for defrosting. Also, prolonging the operating time can significantly reduce the percentage of defrosting time over the operating time, and potentially improve the overall system efficiency.

5.5.2 Effects on the frost accumulation and water retention

a. In the first frosting cycle

As shown in the left-top image in **Figure 5.16**, in the first 20 minutes of the first frosting cycle in the 0°C outdoor condition, dense frost grows on the cold surfaces like the microchannel tubes and the base of the louvered fins. The air-side pressure drop DP_{ea} increases slowly (1.8 Pa/min) in this early stage (as shown in **Figure 5.11(b)**) because the frost height is relatively low. However, from the 20th to the 50th minute, DP_{ea} increases at a rate of 8.9 Pa/min due to the fast increase of frost height and the blockage of airflow. After the 50th minute, the rate of DP_{ea} change reduces again to 3.3 Pa/min, because most of the front surface of the outdoor coil is covered by the frost and frost densifies instead of growing on the outdoor coil. The mass of frost accumulation M_{frost} , on the other hand, increases almost linearly with a rate of 11.83 g/min in the first 40 minutes and then increases by 1.68 g/min from the 40th to the 68th minute, as shown in **Figure 5.11(c)**. This also indicates the non-uniformity of the frost density in the height direction and the transient process. Similar trends can be observed in the first frosting cycles in both -5 and -10°C outdoor conditions, as shown in **Figure 5.12** and **Figure 5.13**.

By the end of the first frosting cycle, it accumulates 0.426 and 0.520 kg of frost in the 0°C outdoor condition, with 5 and 10 times the initial DP_{ea} as the defrost-initiation criteria, respectively (**Table 5.3**). As a result, the time required for defrosting $t_{defrosting}$, the drained water and retained water are both higher (2.5 vs 1.7 minutes, 0.084 vs 0.043 kg, and 0.305 vs 0.219 kg) for the longer operation- 10 times the initial DP_{ea} . It vaporizes more water in the first defrosting cycle for the shorter operation- 5 times the initial DP_{ea} . In author's previous paper, the relation between more

water vaporization and thinner water film on the outdoor coil during defrosting was discussed (Zhang and Hrnjak, 2022).

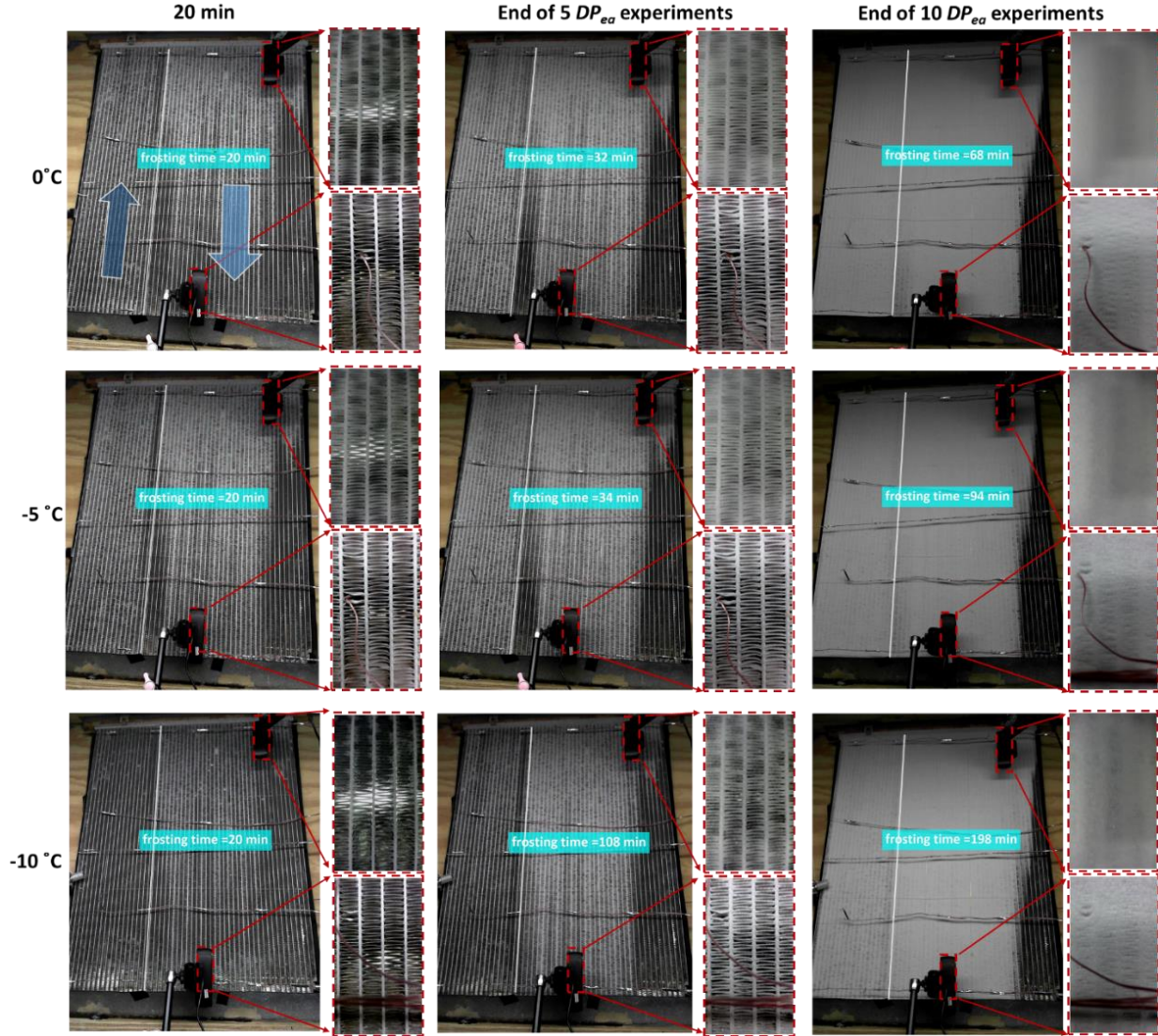


Figure 5.16: The images of the outdoor evaporator at the 20th minute, at the end of the **first** frosting cycle with 5 and 10× $DP_{ea,ini}$ as defrost-initiation criterion in various temperatures and 90% RH outdoor conditions. The refrigerant flow direction in HP mode is shown using blue arrows in the left top image, and the local frost images of the right-top and left-bottom corners of the second pass are presented in red rectangles.

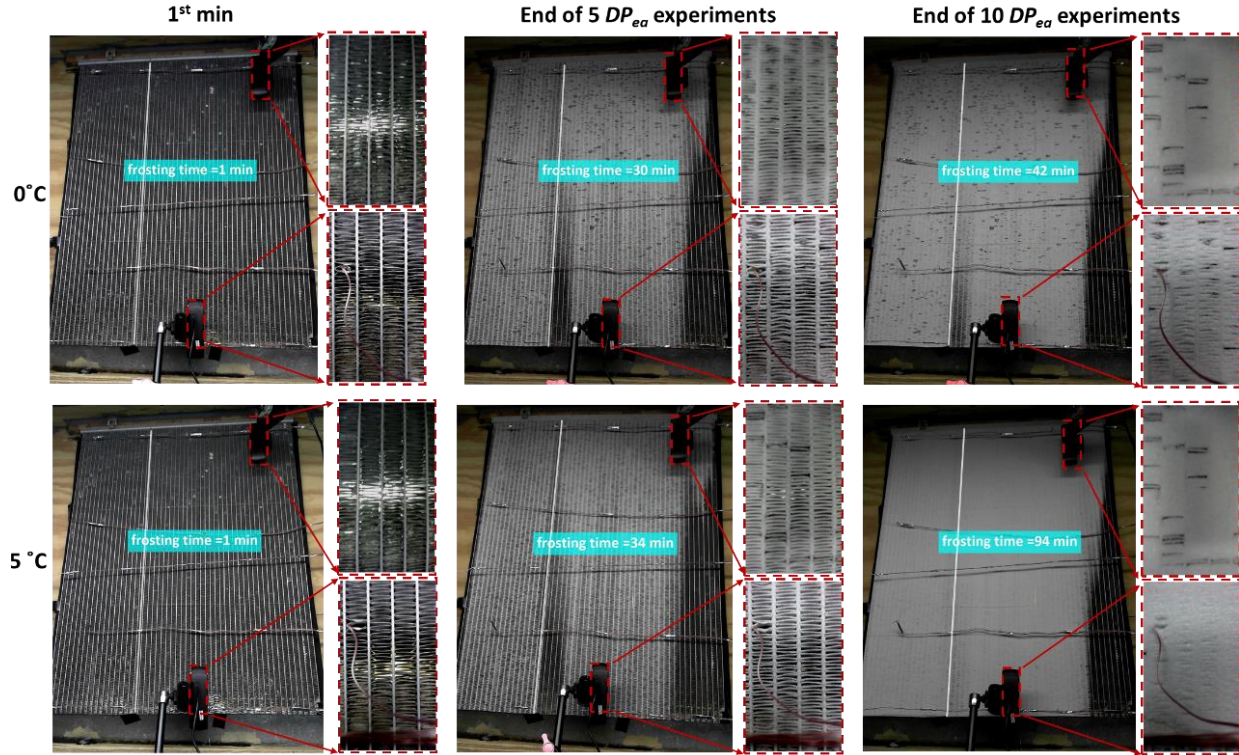


Figure 5.17: The images of the outdoor evaporator at the first minute, at the end of the **second** frosting cycle with 5 and $10 \times DP_{ea,ini}$ as defrost-initiation criterion in 0 and -5°C and 90% RH outdoor conditions. The local frost images of the right-top and left-bottom corners of the second pass are presented in red rectangles.

In summary, when DP_{ea} reaches 5 times the initial value, the frost is still growing in the height direction, the frost accumulates with the maximum speed, and more water vaporizes during the defrosting cycle afterward. When DP_{ea} reaches 10 times the initial value, the frost is barely growing, and more water remains on the outdoor coil after the defrosting because of the longer operating time and more frost accumulation in HP mode.

b. In the refrosting cycles

The air-side pressure drop across the outdoor evaporator DP_{ea} at the beginning of each cycle increases from 43.7 to 56.2 Pa and 58.0 Pa, respectively, with 5 and 10 times the initial DP_{ea} as the defrost-initiation criterion in the 0°C outdoor condition (**Figure 5.11(b)**). Similar trends can

be observed in the -5°C outdoor condition (**Figure 5.12(b)**): DP_{ea} at the beginning of each frosting cycle increases from 45.6 to 58.2 Pa and 52.3 Pa, respectively, with 5 and 10 times the initial DP_{ea} as the defrost-initiation criterion. This shows that a certain amount of water remains on the outdoor evaporator after the defrosting, and the water cannot be removed by the outdoor airflow in HP mode.

Figure 5.17 presents the images of the outdoor coil at the beginning of and at the end of the second frosting cycle with different defrost-initiation criteria in both 0°C and -5°C outdoor conditions. The images clearly show the retained water between the microchannel tubes and fins. Also, those retained water droplets turn into transparent ice instead of white frost in the second frosting cycle, as shown in **Figure 5.17**. In addition, the images show that the amount of ice/retained water is larger in the 0°C than it is in the -5°C outdoor condition, which supports our measurement in **Table 5.3**.

The long blue dash lines in **Figure 5.11(c)**, **Figure 5.12(c)**, and **Figure 5.13(c)** represent the water removal during defrosting cycles, which is the combination of water drainage and vaporization. The beginning point is the total mass of frost M_{frost} by the end of a frosting cycle, and the end is the total mass of water retention on the outdoor coil after a defrosting cycle $M_{retained}$. It is clear that $M_{retained}$ increases with M_{frost} in the refrosting cycles and stabilizes at about 0.3 kg in the 0°C outdoor condition (**Figure 5.11(c)**). Though $M_{retained}$ stabilizes after two frosting cycles with 5 times the initial DP_{ea} as defrost-initiation criteria and only one frosting cycle with 10 times the initial DP_{ea} . A similar trend can be seen in -5°C outdoor condition (**Figure 5.12(c)**), though $M_{retained}$ stabilizes at 0.25 kg in this condition.

5.6 Effects of the orientation of the outdoor coil: horizontal vs. vertical

Figure 5.18 shows the effects of the orientation of the outdoor microchannel heat exchanger (with vertical or horizontal microchannel tubes) on the performance of the CO₂ HP system and the thermal-hydraulic performance of the outdoor microchannel heat exchanger during two continuous frosting and defrosting cycles in 0°C and 90% RH outdoor conditions. For both experiments, the defrosting was initiated when DP_{ea} reaches 10 times the initial value.

5.6.1 Effects on the HP system performance

By the beginning and the end of the first frosting cycle, the heating capacity $Q_{heating}$ and the efficiency HPF are identical for the HP system with both horizontally and vertically oriented outdoor coil (**Figure 5.18**): $Q_{heating}$ drops from 4.89 to 3.12 (35.6%) kW, and HPF decreases from 1.58 to 1.32 (16.5%) for both orientations, even the operating time $t_{frosting}$ are different. The cooling capacity $Q_{cooling}$ of the horizontally installed outdoor evaporator is 0.56 kW (24.9%) higher than that of the vertically installed coil by the beginning of the first frosting cycle. This is likely resulting from the worse refrigerant distribution of the horizontally installed outdoor evaporator. As presented in **Figure 5.19(a)**, the refrigerant-side pressure drop across the outdoor evaporator DP_{er} is approximately 0.16 bar (11.9%) higher in the early stage of the first frosting cycle for the horizontally installed coil than it is for the vertically installed coil. As a result of the higher pressure drop, the evaporating temperature T_{ero} is about 0.4°C lower for the horizontally installed coil than it is for the vertically oriented evaporator (**Figure 5.19(b)**). The evaporator exit refrigerant temperature T_{ero} is used as the evaporating temperature because two-phase refrigerant exits the evaporator and enters the integrated IHX/Acc during the frosting cycles in all test conditions. On the one hand, the IHX can evaporate extra liquid (quality ranging from 0.85 to 0.95) leaving the

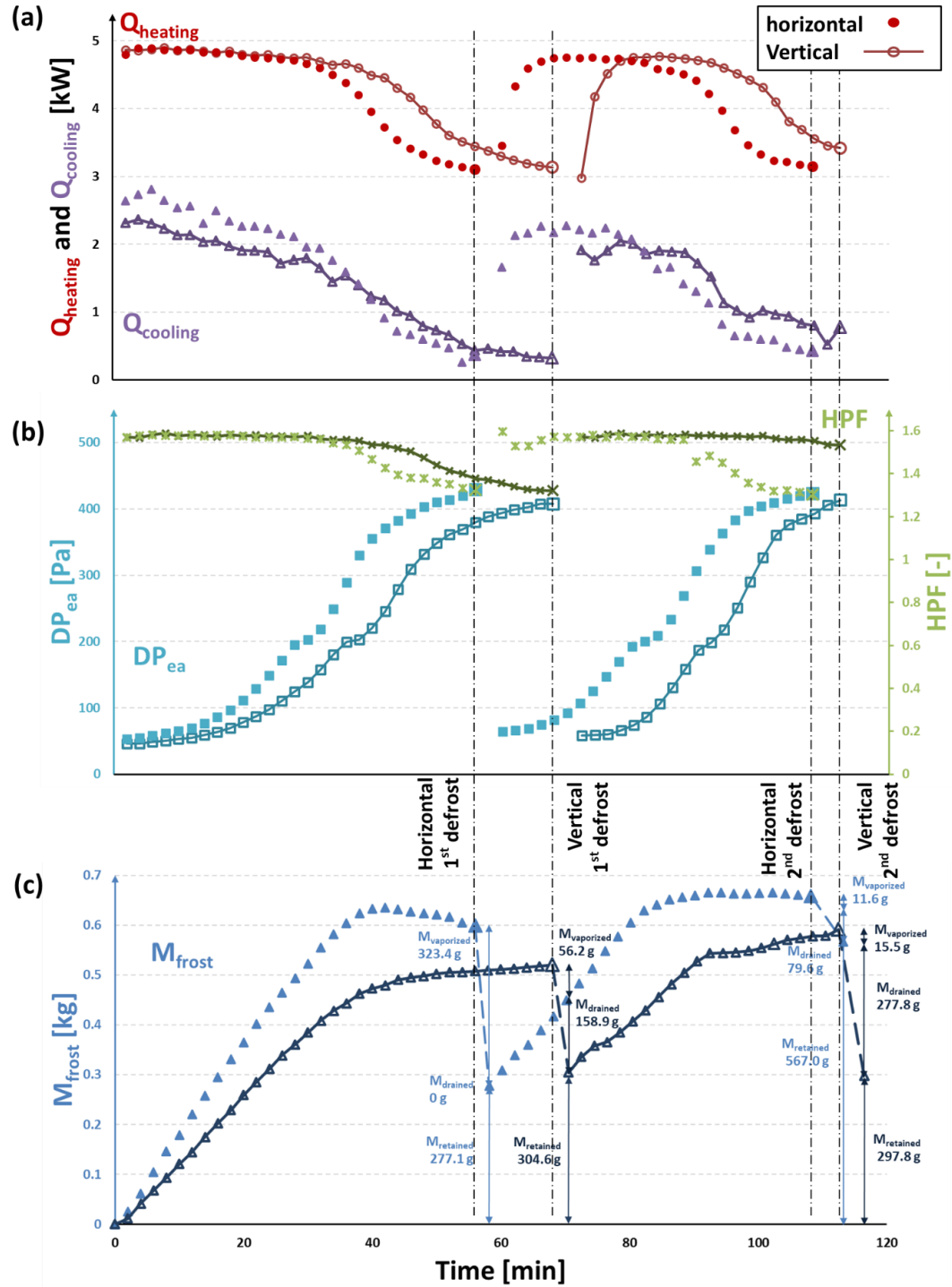


Figure 5.18: The effects of the outdoor coil orientation on (a) the heating and cooling capacity of the HP system; (b) the air-side pressure drop of the outdoor evaporator and the efficiency of the HP system; (c) the mass of frost accumulated on the outdoor evaporator and the meltwater after defrosting, during continuous cycles with 10 times initial DP_{ea} as defrost-initiation criterion in 0°C ambient conditions. Only the defrost initiations are labeled with vertical dash-dot lines.

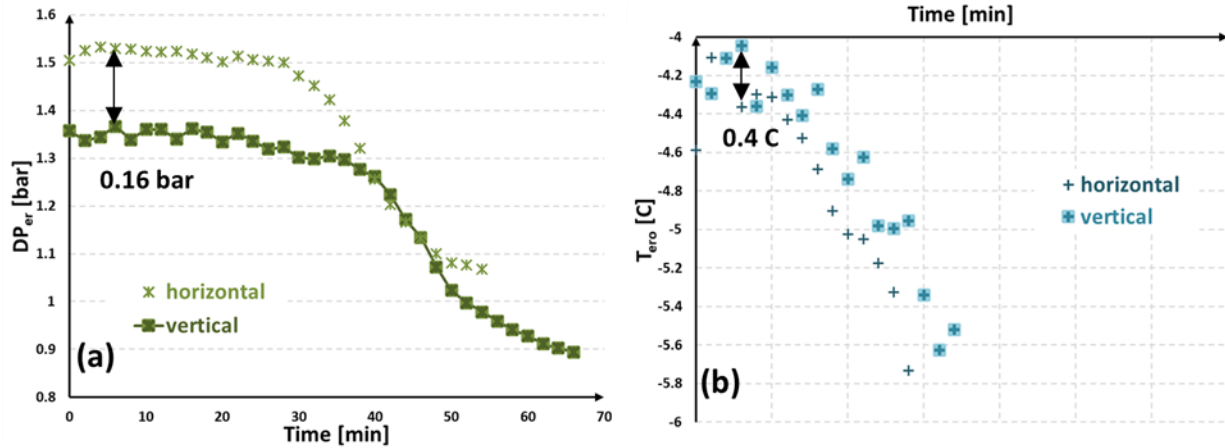


Figure 5.19: The effects of the outdoor coil orientation on (a) the refrigerant pressure drop across the outdoor evaporator DP_{er} ; (b) the evaporator exit temperature T_{ero} . Two-phase refrigerant exits the evaporator in both experiments.

evaporator. On the other hand, the refrigerant charge amount of the HP system was determined to be 950 g in the condition of the outdoor air temperature T_{eai} of 0°C. In this case, the outdoor evaporator is flooded, and the refrigerant-side heat transfer area is well-utilized in HP mode. With a lower evaporating temperature T_{ero} , the horizontally oriented outdoor evaporator has a 24.9% higher $Q_{cooling}$ during the early stage of the first frosting cycle. However, the percentage of latent load is higher, frost accumulates faster, and DP_{ea} reaches 10 times the initial value by the 54th minute for the horizontally installed coil, which is 12 minutes earlier than the vertically installed coil.

In conclusion, the vertical orientation of the evaporator improves the refrigerant distribution, reduces the refrigerant-side pressure drop, and increases the evaporating temperature. Therefore, it prolongs the operating time of the HP system $t_{frosting}$ by 12 minutes (18.2%) in the 0°C and 90% RH outdoor condition for the first frosting cycle with 10 times of the initial DP_{ea} as the defrost-initiation criterion.

5.6.2 Effects on the frost accumulation and water retention

As shown in **Figure 5.18(c)**, the mass of the frost M_{frost} accumulates faster in the first 40 minutes than it does in the later stage of the first frosting cycle for both orientations: the densification of frost happens later with a slower rate compared to the growth in the early stage. Also, the frost grows faster on the surface of the horizontally installed outdoor evaporator due to the lower surface temperature: it accumulates at a rate of 16.7 g/s for the horizontal outdoor evaporator though it accumulates at a rate of 12.0 g/s for the vertical outdoor evaporator. **Figure 5.20** also shows the more severe and thicker frost on the horizontal outdoor coil at the 40th minute in the first frosting cycle. This also agrees with our analysis that the evaporating temperature is lower for the horizontal outdoor coil. The peak value of M_{frost} is 0.64 kg at the 40th minute for the horizontally oriented evaporator in the first frosting cycle, though it is 0.52 kg at the 68th minute for the vertically oriented evaporator. The higher mass of frost for the horizontally installed evaporator might be due to the lower surface temperature and higher frost density.

In addition, the retained water $M_{retained}$ is comparable for both orientations after the first defrosting cycle, as shown in **Figure 5.18(c)**. **Figure 5.21** also shows the retained water on the outdoor coil at the 1st minute, the frost at the 30th minute, and the end of the second frosting cycles in the 0°C and 90% RH outdoor condition for both orientations. The main difference is the retained water stays on the middle of the louvered fins for the vertically oriented outdoor coil, but it flows downwards and stays by the root of the fins for the horizontally oriented outdoor coil. It is likely that the retained water can be spread along the tube depth direction and become less obvious on the front surface in the latter case. However, $M_{retained}$ is 0.567 kg for the horizontally oriented outdoor coil after the second defrosting cycle, and it is almost twice the amount of $M_{retained}$ for the

vertically oriented outdoor coil (0.298 kg). Also, the water drainage is obviously better for the vertically oriented outdoor coil: it drains 0.159 and 0.278 kg after the two continuous defrosting cycles for the vertical outdoor coil, though it drains 0 and 0.080 kg for the horizontal outdoor coil.

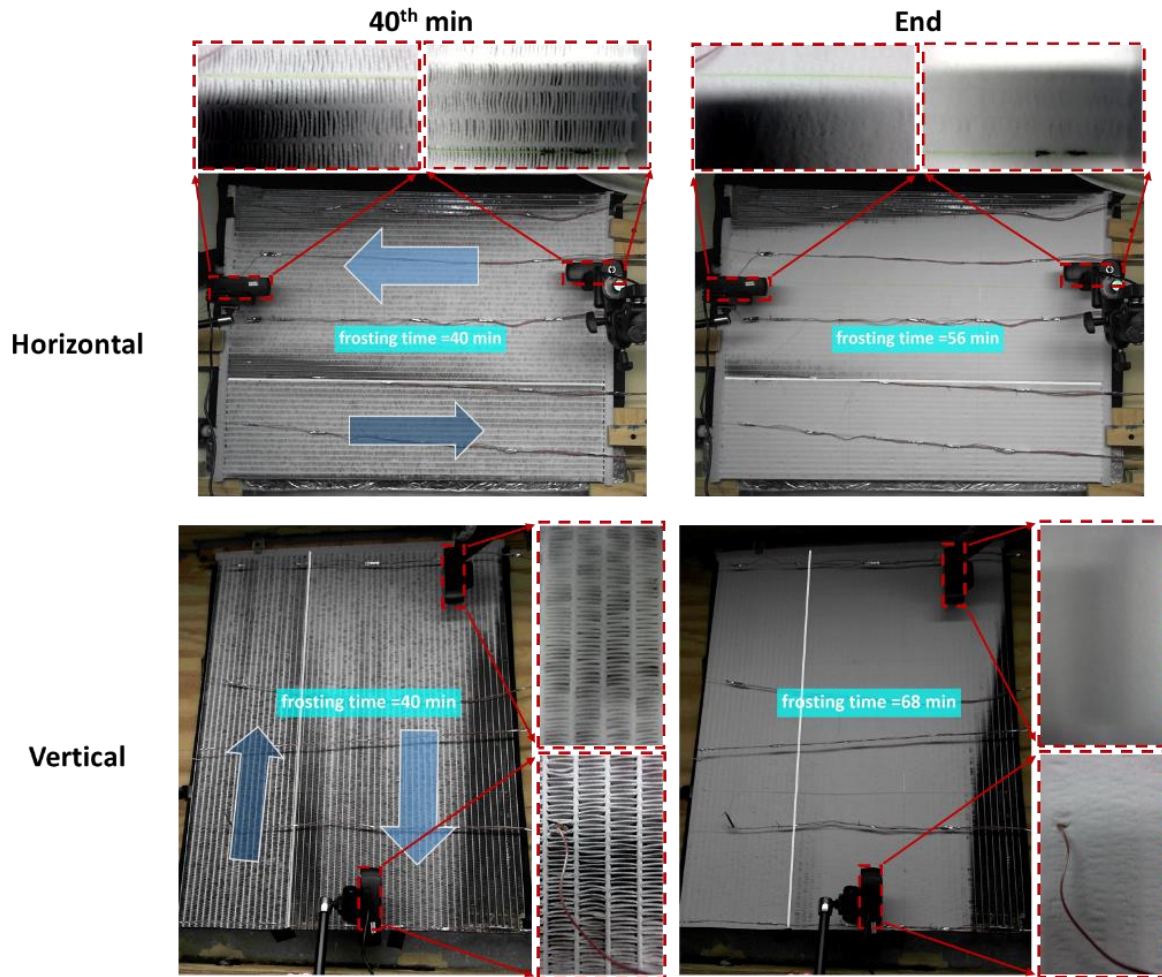


Figure 5.20: The images of the horizontally and vertically installed outdoor evaporator at the 40th minute and the end of the first frosting cycle with 10 times initial DP_{ea} as defrost-initiation criterion in the 0°C and 90% RH outdoor condition. The local frost images of the second pass are presented in red rectangles.

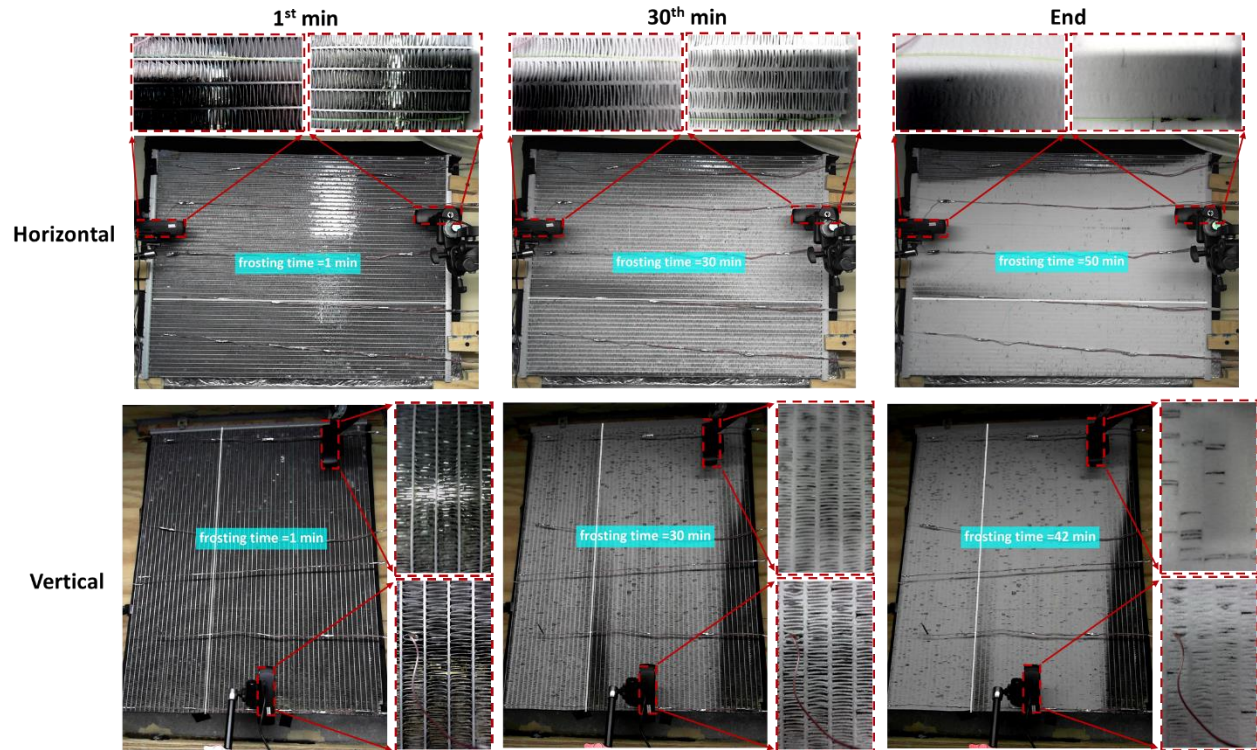


Figure 5.21: The images of the horizontally and vertically installed outdoor evaporator at the 1st minute, the 30th minute, and the end of the second frosting cycle with 10 times initial DP_{ea} as defrost-initiation criterion in the 0°C and 90% RH outdoor condition. The local frost images of the second pass are presented in red rectangles.

For the defrosting time, it takes 2.5 and 4.0 minutes to defrost the vertically oriented outdoor coil for the two continuous cycles. However, it takes 2.6 and 5.2 minutes to defrost the horizontally oriented outdoor coil for the two continuous cycles. Especially, it takes up to 5.2 minutes to melt the frost and warm up a large amount of the retained water on the horizontal outdoor coil during the second defrosting cycle, so that the refrigerant exit temperature of the outdoor gas cooler reaches the pre-selected value of 45°C. Percentagewise, the defrosting time is 7.0% and 5.6% for the horizontally and vertically oriented outdoor coil in the two continuous frosting/defrosting cycles or approximately two hours of operation.

In this chapter, the effects of environmental factors, different defrost-initiation criteria, and outdoor coil orientations on the performance of an CO₂ HP system and the thermal-hydraulic

performance of the outdoor evaporator are experimentally investigated in various practical operating conditions and several continuous frosting/ defrosting cycles. The results show:

- Although the evaporator is flooded during the HP operation, frost grows unevenly on the outdoor evaporator due to the maldistribution of two-phase refrigerant and the evaporation of liquid refrigerant along the microchannel tubes.
- As frost accumulates on the outdoor evaporator, the cooling capacity $Q_{cooling}$ and the quality of refrigerant at the evaporator exit x_{ero} drop and liquid refrigerant migrates into the accumulator. Therefore, the effective volume of the accumulator should be optimized to hold extra liquid refrigerant in the wide operating range of a reversible HP system.
- With fixed relative humidity of ambient air (RH_{ei}), the ambient air temperature T_{eai} affects the capacity and efficiency of the HP system significantly, as well as the frost accumulating speed and the frosting time $t_{frosting}$. When T_{eai} drops from 0 to -10°C, the peak value of $Q_{heating}$ reduces from 4.85 to 3.54 kW and HPF reduces from 1.61 to 1.47 in the first cycle. In the meanwhile, the frost accumulating speed decreases from 0.013 to 0.005 kg/min, and $t_{frosting}$ increases from 32 to 108 minutes.
- With fixed T_{eai} , however, RH_{ei} has a negligible impact on the capacities and efficiency of the HP system. Though it directly affects the humidity ratio and thus the frost accumulating speed and $t_{frosting}$, which increases significantly from 32 to 84 minutes when T_{eai} is 0°C and RH_{ei} decreases from 90% to 80%.
- The heating capacity $Q_{heating}$ and the efficiency HPF of the HP system drop only by less than 9% during the frosting cycles with 5 times the initial DP_{ea} as the defrost-

initiation criterion in all experimental conditions. But $Q_{heating}$ and HPF drop by about 30% and 15% during the frosting cycles with 10 times the initial DP_{ea} as the defrost-initiation criterion in all experimental conditions. Considering the operating-time-averaged performance, the 30% and 15% performance reduction can be considered mild, which indicates a longer operating time in HP mode, a lower defrosting frequency, fewer impacts on passengers' comfort, and potentially a higher periodic performance. Although the optimal defrost-initiation and termination criteria are still open to discussion and require further investigation.

- The retained water $M_{retained}$ gets saturated at 0.3 kg with a vertically oriented outdoor coil after two/one frosting cycles with 5/10 times the initial DP_{ea} as the defrost-initiation criterion in the 0°C and 90% RH outdoor condition. A similar trend can be seen in -5°C outdoor condition, though $M_{retained}$ stabilizes at 0.25 kg in this condition.
- The vertical orientation of the outdoor coil prolongs the operating time of the HP system $t_{frosting}$ from 56 to 68 minutes (+18.2%) in the 0°C and 90%RH outdoor condition for the first frosting cycle using 10 times the initial DP_{ea} as the defrost-initiation criterion. Because the vertical orientation improves the refrigerant distribution, reduces the refrigerant-side pressure drop, and increases the evaporating as well as the surface temperature of the outdoor coil.
- Also, the outdoor coil in the vertical orientation retains only 0.298 kg of water after the second defrosting, which is 58.7% less than it does in the horizontal orientation. As a result, the defrosting time for the vertical outdoor coil reduces by 1.3 minutes.

Considering the total two-hour operation, the vertical orientation reduces the defrosting time from 7.0% to 5.6%.

In conclusion, the result of this section shows that the performance of the CO₂ HP system drops mildly using 10 times the initial DP_{ea} as the defrost-initiation criterion in all experimental conditions, and the percentage of the defrosting time $t_{defrosting}$ drops compared to it when using 5 times the initial DP_{ea} as the defrost-initiation criterion. Also, the HP system with a vertically installed outdoor coil has a better performance in both frosting and defrosting cycles because of the better distribution of two-phase refrigerant and better water drainage. The conclusion of this work can be applied by the researchers and engineers working on designing and optimizing automotive heat pumps.

CHAPTER 6: DEFROSTING PERFORMANCE OF THE R744 HEAT PUMP AND THE ENERGY FLOW IN CONTINUOUS FROSTING/ DEFROSTING CYCLES

In this chapter, the effects of different defrost-initiation criteria and orientations of the outdoor coil on the defrosting energy flow during several continuous defrosting cycles are presented with a reversible R744 HP system. The frost accumulation, meltwater retention and distribution, meltwater drainage, and vaporization are discussed. Also, the meltwater distribution on the outdoor coil is the first time reported in the open literature to the authors' knowledge. Also, the energy supply and consumption of continuous defrosting processes under practical operating conditions are calculated and analyzed.

6.1 System architecture and experimental facility

The reversible CO₂ ASHP system used for continuous frosting/defrosting experiments is well described in chapter 5, as well as the facility.

6.2 Experimental procedure and experimental conditions

The performances of the reversible CO₂ system in periodic frosting/defrosting cycles are studied using the following steps: 1) preconditioning, 2) frosting, 3) defrosting by reversing the cycle, 4) water collection, 5) refrosting for periodic performance. The detailed descriptions of procedures are presented in chapter 5. **Table 6.1** shows the operating conditions of the frosting/defrosting experiments in this section.

6.3 Measurement and data reduction

The air- and refrigerant-side measurements of the CO₂ HP system and the details of the sensors are well described in chapter 5. The data reduction of the defrosting performance and the energy flow will be presented in the following section. Starting this chapter, the subscription ID

and OD are used to represent the indoor and outdoor components to separate the calculation in defrosting mode from the HP mode.

Table 6.1: Operating conditions of frosting/defrosting experiments in this section

Microchannel tube orientation	T_{eai} [°C]	RH_{ei} [%]	$V_{eai,ini}$ [m/s]	T_{cai} [°C]	U_{cai} [kg/min]	# of frosting cycles	Defrost initiation criteria	Defrost termination criterion
Vertical	0	90	3.0	20	7.0	1/2/3	$5/10 \times DP_{ea,ini}$	Outdoor $T_{cro} = 45 \text{ °C}$
Horizontal						1/2	$10 \times DP_{ea,ini}$	

6.3.1 Data reduction of the defrosting performance of the HP system

For the defrosting process, the cooling capacity of the indoor coil was calculated using only air-side measurements, following **Eqn (61)**. Because of the two-phase refrigerant at the indoor coil exit and the fast change of the capacity, neither refrigerant- nor chamber-side measurements can be used for calculation.

$$Q_{ID,e,air} = \dot{m}_{ID,dryair} (h_{ID,eai} - h_{ID,eao}) \quad \text{Eqn (61)}$$

where, the airflow rate $\dot{m}_{ID,dryair}$ was calculated using the measured air temperature $T_{ID,ean}$ and the dew point $T_{ID,dpen}$ behind the indoor air nozzles, and the air-side pressure drop $DP_{ID,en}$ across the indoor nozzles. The enthalpies of air in front of and leaves the indoor evaporator $h_{ID,eai}$ and $h_{ID,eao}$ were determined using air temperatures and dew points at the indoor coil inlet ($T_{ID,eai}$ and $T_{ID,dpei}$), and the indoor nozzles' exits ($T_{ID,ean}$ and $T_{ID,dpen}$). In addition, the expansion process is considered as an isenthalpic expansion process in all conditions.

As shown in **Figure 6.1**, the quality of refrigerant at the evaporator exit $x_{ID,ero}$ is lower than that at the accumulator exit $x_{acc,ro}$ (which is calculated based on the energy balance of the low- and high-pressure sides of the IHX) at the 60th second of the first defrosting cycle. This indicates a significant amount of the liquid refrigerant migrates from the cold outdoor coil and liquid line into the accumulator in a short period. Also, this explains why the refrigerant- and chamber-side measurements cannot be used to calculate the cooling capacity: the refrigerant migration is too fast to be considered a quasi-steady-state during the defrosting process.

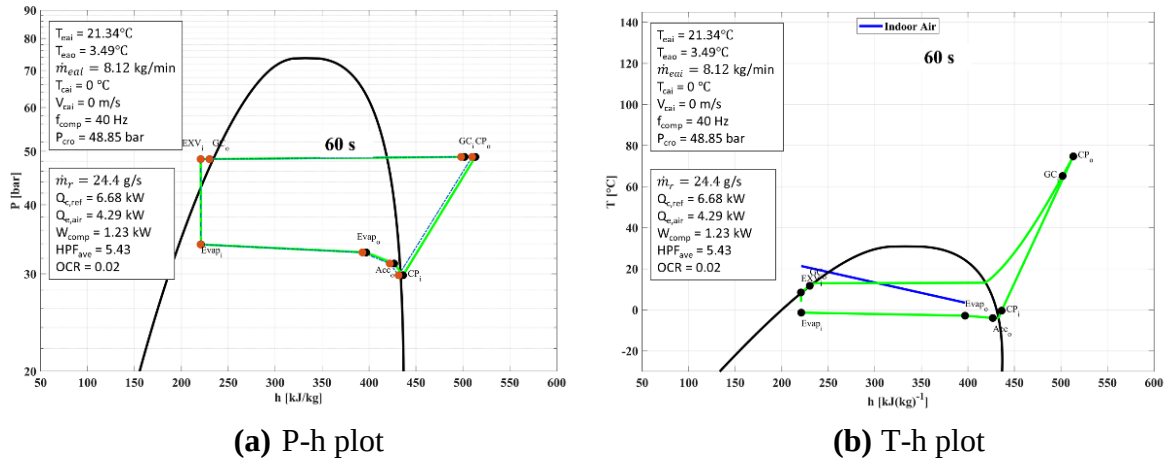


Figure 6.1: The P-h and T-h plots of the 60th second of the first defrosting cycle in the 0°C and 90% RH outdoor condition with vertically oriented outdoor coil

The heating capacity of the outdoor gas cooler was calculated only using the refrigerant-side:

$$Q_{OD,c,ref} = (1 - OCR) \dot{m}_r (h_{OD,cri} - h_{OD,cro}) + OCR \cdot \dot{m}_r (h_{OD,coi} - h_{OD,coo}) \quad \text{Eqn (62)}$$

where, OCR is measured by the oil sampler installed across the mass flow meter on the liquid line. OCR is measured to be 2%. The black dots and thick green line in **Figure 6.1(a)** represent the pure refrigerant properties, and the orange dots and thin blue dash-dot line represent the oil and CO_2 mixture properties. The two sets of P-h plots almost overlap with each other, and the most different

part is at the compressor suction point. $\dot{m}_r$ was measured by the mass flow meter, and it is actually the mass flow rate of POE oil and CO₂ mixture. The refrigerant enthalpies $h_{OD,cri}$ and $h_{OD,cro}$, oil enthalpies $h_{OD,coi}$ and $h_{OD,coo}$ were determined from the pressures and temperatures measured at the outdoor condenser/gas cooler inlet ($T_{OD,cri}$ and $P_{OD,cri}$) and outlet ($T_{OD,cro}$ and $P_{OD,cro}$).

Like the heating capacity of the outdoor condenser/gas cooler, the heating capacity of the IHX was calculated using the refrigerant properties on the high-pressure side, following **Eqn (63)**:

$$Q_{IHX,high} = (1 - OCR)\dot{m}_r(h_{OD,cro} - h_{ID,xri}) + OCR \cdot \dot{m}_r(h_{OD,coo} - h_{ID,xoi}) \quad \text{Eqn (63)}$$

where, the refrigerant and oil enthalpy ($h_{ID,xri}$ and $h_{ID,xoi}$) were determined using the pressure and temperature measured at the inlet of the expansion valve used in defrosting/AC mode. Also, the cooling capacity of the low-pressure side of the IHX equals the heating capacity of the high-pressure side of the IHX, assuming there is no heat loss to the ambient through the insulation. Thus, the state of the refrigerant at the accumulator exit can be determined based on the cooling capacity of the IHX low-pressure side and the measured superheated refrigerant state at the compressor suction point.

In chapter 5, **Figure 5.1** shows the flow directions of refrigerant and air for the vertically and horizontally orientated outdoor coil in both HP and defrost modes. The blue/red arrows represent the refrigerant flow in HP/defrosting mode. The white arrows represent the airflow direction. The green dots are the local surface temperature measuring points using T-type thermocouples. The outdoor coil was evenly divided into 9 blocks using green fishing lines, and the meltwater was collected, measured, and recorded separately for each block to show the distribution of water retention after defrosting. The total mass of all collected water from the 9 blocks was named $M_{collected}$. However, since there exists a thin water film on the outdoor coil even

after the water is collected with compressed air, the actual water retention $M_{retained}$ is higher than the total mass collected from the coil $M_{collected}$. Assuming a 0.015 mm water film evenly distributed on the whole air-side surface of the outdoor coil after collecting, the mass of this water film on the outdoor coil $M_{OD,wf,coil}$ can be estimated by using the following:

$$M_{OD,wf,coil} = A_{OD,a,coil} th_{OD,wf,coil} \rho_{water} = 0.1074kg \quad \text{Eqn (64)}$$

where, $A_{OD,a,coil}$ is the air-side surface area of the outdoor coil- 7.18 m²; $th_{OD,wf,coil}$ is the thickness of the water film that stays on the outdoor coil after the water collection with compressed air and paper towel, which is 0.015 mm. ρ_{water} is the density of water. Therefore, the actual water retention $M_{retained}$ is calculated using **Eqn (65)**:

$$M_{retained} = M_{collected} + M_{OD,wf,coil} \quad \text{Eqn (65)}$$

Similarly, the mass of the water film on the internal surface of the pan and hose used to collect the drained water can be estimated. In the following sections, the corrected values of $M_{retained}$ and $M_{drained}$ will be presented and analyzed.

Then, the difference between the total amount of accumulated frost M_{frost} and the sum of retained water and drained water ($M_{retained} + M_{drained}$) was considered as the mass of water vaporized $M_{vaporized}$ during the defrosting process, following **Eqn (66)**:

$$M_{vaporized} = M_{frost} - M_{retained} - M_{drained} \quad \text{Eqn (66)}$$

In the calculation, $M_{retained}$ and $M_{drained}$ can be underestimated in the first one or two defrosting cycles, and $M_{vaporized}$ can be overestimated, consequently, when comparing the mass and energy balances. But the accuracy of $M_{retained}$ and $M_{drained}$ is improved with the increased total

frosting/defrosting time because the coil gets saturated with thin water films after collecting the retained water with compressed air and paper towels in the following defrosting cycles.

6.3.2 Data reduction of the energy supply during defrosting

There were three major energy supplies during the defrosting: the heat from indoor warm coil metal, the heat from indoor 20°C air, and energy from the compressor power, as shown in **Figure 6.2** on the left. The heat supplied by the indoor air was calculated by integrating $Q_{ID,e,air}$ over the defrosting time $t_{defrosting}$, as shown in **Eqn (67)**:

$$E_{ID,e,air} = \int_0^{t_{defrosting}} Q_{ID,e,air} dt \quad \text{Eqn (67)}$$

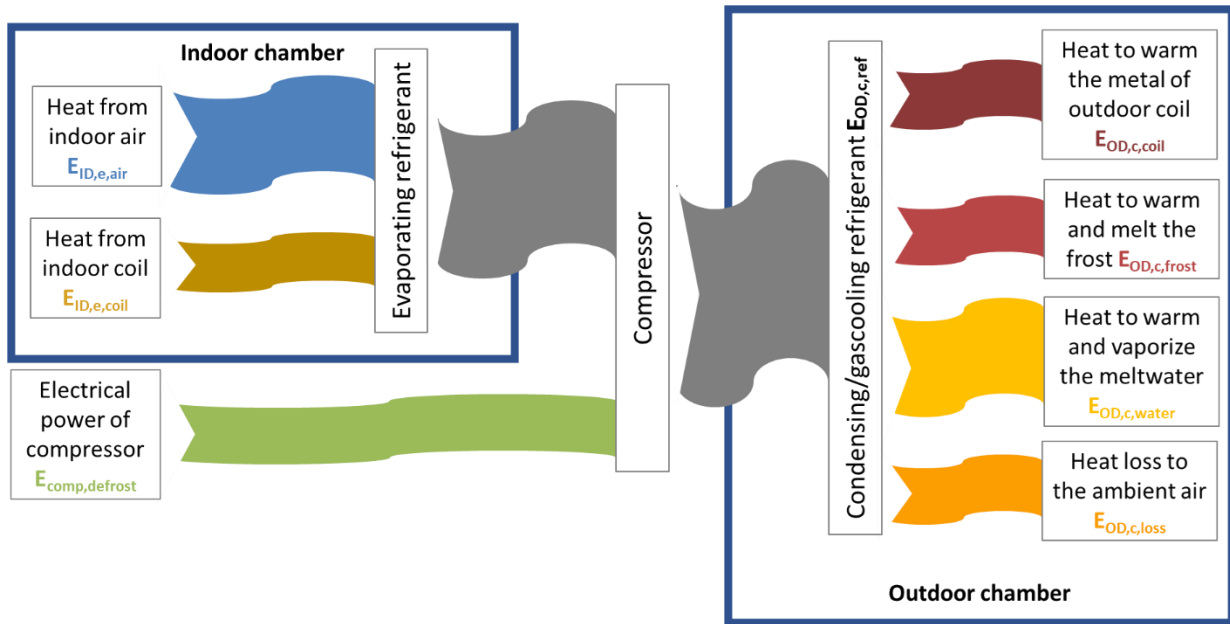


Figure 6.2: The energy flow during the defrosting process

The heat supplied by the indoor coil metal can be calculated using the initial and final average temperatures of the indoor evaporator during the defrosting, as shown in **Eqn (68)**:

$$E_{ID,e,coil} = M_{ID,coil} cp_{al} \left(ave(T_{ID,cri}, T_{ID,cro})_{frost, final} - 0.5 \times (T_{ID,eri} + T_{ID,ero})_{defrost, final} \right) \quad \text{Eqn (68)}$$

where, $M_{ID, coil}$ is the mass of the indoor aluminum coil. cp_{al} is the specific heat of aluminum. $T_{ID,cri}$ and $T_{ID,cro}$ are the refrigerant inlet and exit temperatures of the indoor gas cooler during the last frosting cycle. Also, the $ave(T_{ID,cri}, T_{ID,cro})$ represents the calculation of the average coil temperature following a logarithmic temperature profile along the flow direction using $T_{ID, cri}$ and $T_{ID,cro}$ as the starting and ending points. The subscripts *frost,final* represents the final states of the previous frosting cycle, and *defrost,final* represents the final states of the current defrosting cycle. Also, the transient capacity of the heat supply by the indoor warm coil $Q_{ID,e,coil}$ can be calculated as the derivative of $E_{ID,e,coil}$ with respect to time.

The compressor power W_{comp} during defrosting was directly measured by a watt transducer, and the energy supplied by the compressor was calculated following:

$$E_{comp,defrost} = \int_0^{t_{defrosting}} W_{comp} dt \quad \text{Eqn (69)}$$

The total energy supplied for defrosting $E_{total,supply}$ was calculated by adding the three terms together:

$$E_{total,sup ply} = E_{ID,e,air} + E_{ID,e,coil} + E_{comp,defrost} \quad \text{Eqn (70)}$$

6.3.3 Data reduction of the energy consumption during defrosting

As shown in **Figure 6.2** on the right, the energy consumption is divided into four parts: the heat used to warm up the cold outdoor coil metal, the heat used to preheat the frost to 0°C and to melt the frost, the heat used to warm and vaporize the meltwater during the defrosting and water

drainage, and the heat loss to the cold ambient air. **Figure 6.2** presents the overall energy flow diagram for the defrosting progress.

First, the total amount of all four parts of energy consumption was calculated by integrating $Q_{OD,c,ref}$ over the defrosting time $t_{defrosting}$ using **Eqn (71)**:

$$E_{OD,c,ref} = \int_0^{t_{defrosting}} Q_{OD,c,ref} dt \quad \text{Eqn (71)}$$

The energy used to warm the cold metal of the outdoor coil was calculated using **Eqn (72)**:

$$E_{OD,c,coil} = M_{OD,coil} cp_{al} \left(0.5 \times (T_{OD,cri} + T_{OD,cro})_{defrost, final} - 0.5 \times (T_{OD,eri} + T_{OD,ero})_{frost, final} \right) \quad \text{Eqn (72)}$$

where, $M_{OD,coil}$ is the mass of the outdoor coil. $T_{OD,cri}$ and $T_{OD,cro}$ are the refrigerant inlet and exit temperatures of the outdoor coil/condenser at the *final* stage of the current *defrosting* cycle. $T_{OD,eri}$ and $T_{OD,ero}$ are the refrigerant inlet and exit temperatures of the outdoor coil/evaporator at the *final* stage of the previous *frosting* cycle. Similarly, the transient heating load of heating the metal of outdoor coil $Q_{OD,c,coil}$ can be calculated as the derivative of $E_{OD,c,coil}$ with respect to time.

The energy used to preheat and melt the frost $E_{OD,c,frost}$ can be calculated using **Eqn (73)**:

$$E_{OD,c,frost} = M_{frost} cp_{frost} \left(0 - \left(\frac{T_{OD,eri} + T_{OD,ero}}{2} \right)_{frost, final} \right) + M_{frost} h_{frost, fusion} \quad \text{Eqn (73)}$$

where, cp_{frost} is the specific heat of frost- 2.108 kJ/(kg-K). $T_{OD,eri}$ and $T_{OD,ero}$ are the refrigerant inlet and exit temperatures of the outdoor coil/evaporator at the *final* stage of the previous *frosting* cycle. $h_{frost, fusion}$ is the latent heat to melt the frost- 333.5 kJ/kg. The transient heating load of

preheating and melting the frost $Q_{OD,c,coil}$ can be calculated as the derivative of $E_{OD,c,coil}$ with respect to time.

The energy used to heat and vaporize the water $E_{OD,c,water}$ was estimated using **Eqn (74)**:

$$\begin{aligned} E_{OD,c,water} = & M_{drained} cp_{water} (10 - 0) \\ & + M_{retained} cp_{water} \left(0.5 \times (T_{OD,cri} + T_{OD,cro})_{defrost, final} - 0 \right) \\ & + M_{vaporized} \left[cp_{water} (50 - 0) + h_{water, vaporization} \right] \end{aligned} \quad \text{Eqn (74)}$$

where, cp_{water} is the specific heat of water, and $h_{water, vaporization}$ is the latent heat to vaporize water at 50 °C. In the calculation, the drained water was assumed to leave the outdoor coil at 10°C. The average of the *final* temperature of $T_{OD,cri}$ and $T_{OD,cro}$ is used to represent the final temperature of retained water. The vaporized water was assumed to be heated up to 50 °C and then vaporized. It is clear that the accuracy of $E_{OD,c,frost}$ and $E_{OD,c,water}$ in **Eqn (73)** and **Eqn (74)** considerably rely on the accuracy of the mass measurements. Unfortunately, the mass of meltwater can be inaccurate during the first one to two frosting/defrosting cycles due to the imperfectness of the water collection methods.

Based on energy conservation, the heat loss to the outdoor ambient air was calculated using **Eqn (75)**:

$$E_{OD,c,loss} = E_{OD,c,ref} - E_{OD,c,frost} - E_{OD,c,coil} - E_{OD,c,water} \quad \text{Eqn (75)}$$

However, it should be noted that the accuracy of $E_{OD,c,loss}$ in **Eqn (75)** depends on the accuracy of $E_{OD,c,water}$, and thus depends considerably on the accuracy of $M_{vaporized}$ due to the large latent heat of water vaporization. As mentioned previously, $M_{vaporized}$ can be overestimated in the first one or two defrosting cycles. As a result, $E_{OD,c,water}$ in **Eqn (74)** might be significantly

overestimated and $E_{OD,c,loss}$ in **Eqn (75)** can be underestimated in the first one or two defrosting cycles. But the accuracy is improved in the following frosting/ defrosting cycles, as the surface of the outdoor coil gets saturated with the water film.

6.4 Effects of defrost-initiation criteria: 5 vs. 10 times initial DP_{ea}

In this section, the effects of two different defrost-initiation criteria on the defrosting capacity, efficiency, and meltwater distribution of the reversible R744 HP in continuous defrosting cycles under the outdoor condition of 0°C and 90% RH are discussed and analyzed.

6.4.1 The overall performance of continuous defrosting cycles

A series of continuous frosting/defrosting experiments were conducted following previously described experimental procedures in chapter 5, and the experimental data of defrosting processes were processed following **Eqn (61)** to **Eqn (75)**. **Figure 6.3** presents the accumulated mass of frost and meltwater on the vertically oriented outdoor coil during continuous frosting/defrosting cycles with the vertically oriented outdoor coil using different defrost-initiation criteria in the outdoor condition of 0°C and 90% RH.

As shown in **Figure 6.3**, the defrosting time $t_{defrosting}$ is 1.7, 2.3, and 2.5 minutes for the continuous three defrosting cycles with 5 times the initial outdoor DP_{ea} as the defrost-initiation criterion. But it is 2.5 and 4.0 minutes for the continuous two defrosting cycles with 10 times the initial outdoor DP_{ea} as the defrost-initiation criterion. Also, the water retention $M_{retained}$ is 219.3 g, 292.8 g, and 326.6 g after continuous three defrosting cycles using 5 times the initial outdoor DP_{ea} as the defrost-initiation criterion, and it is 304.6 g and 297.8 g after defrosting using 10 times the initial outdoor DP_{ea} as the defrost-initiation criterion. Overall, the defrosting time and water retention on the outdoor coil increase gradually with the number of the frosting/defrosting cycles.

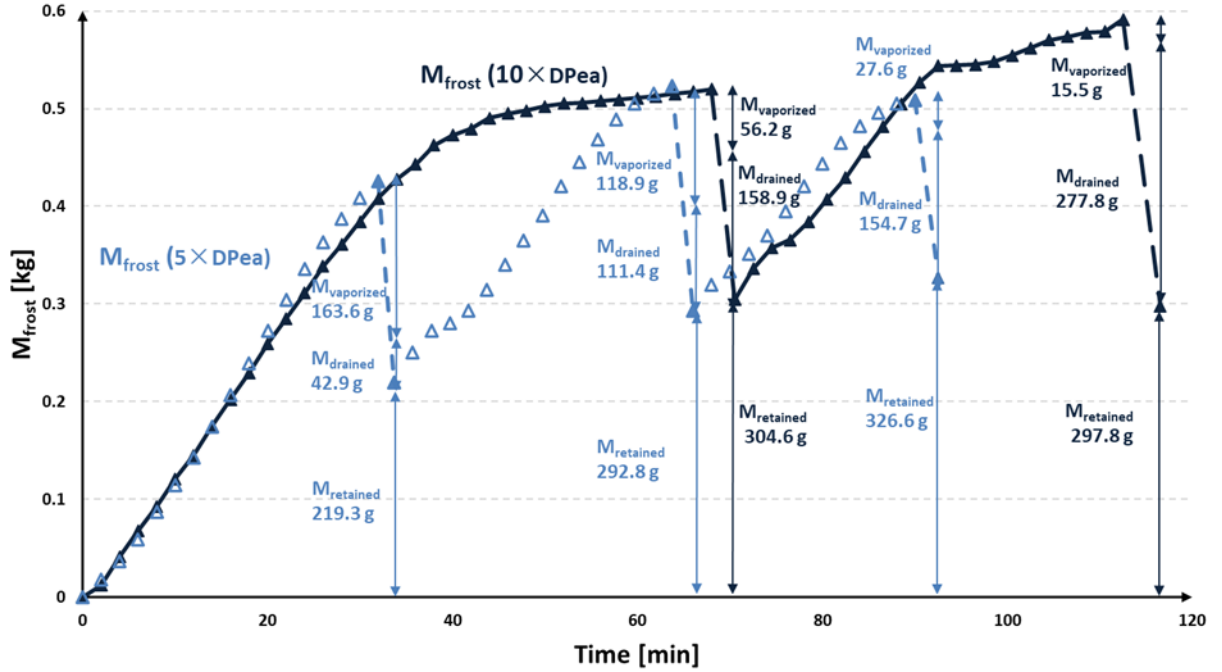


Figure 6.3: The effects of different defrost-initiation criteria on the mass of frost accumulated and the meltwater on the vertically oriented outdoor evaporator during continuous frosting/defrosting cycles in the 0°C and 90% RH outdoor condition. The long blue dash lines represent the water removal, and the masses of the vaporized, drained, and retained water are labeled by the end of each defrosting cycle.

This is due to the retained water turning into ice instead of high-porosity frost at the beginning of the next frosting cycle, so the density of the re-accumulated frost increases near the coil surface. This increases the mass of the frost accumulated, the time and energy required to defrost, and the water retention in the next re-frosting cycles. In addition, the final water retention is 326.6 g for the third defrosting cycle using 5 times the initial outdoor DP_{ea} as the defrost-initiation criterion and 297.8 g for the second defrosting cycle using 10 times the initial outdoor DP_{ea} as the defrost-initiation criterion, respectively. This indicates the retained water gets saturated with approximately 300 g for the vertically installed outdoor coil. Further refrosting cycles are likely to show repeatable performance.

Similar to the water retention $M_{retained}$, the drained water $M_{drained}$ increases gradually with the number of the frosting/defrosting cycles (**Figure 6.3**): it drains 42.9 g, 111.4 g, and 154.7 g for the continuous three defrosting cycles using 5 times the initial outdoor DP_{ea} as the defrost-initiation criterion, and it drains 158.9 g and 277.8 g after two continuous defrosting cycles using 10 times the initial outdoor DP_{ea} as the defrost-initiation criterion. On the contrary, the mass of the vaporized water $M_{vaporized}$ reduces from hundreds of grams to tens of grams with the increased number of the frosting/defrosting cycles (**Figure 6.3**). This is likely due to the increased thickness of the water film and the reduced surface temperature of the water film in the second or third defrosting cycles. The lower the surface temperature of the water film is, the less kinetic energy of the water molecules on the surface has. Therefore, less water is vaporized in the second and third defrosting cycles than it is in the first defrosting cycle.

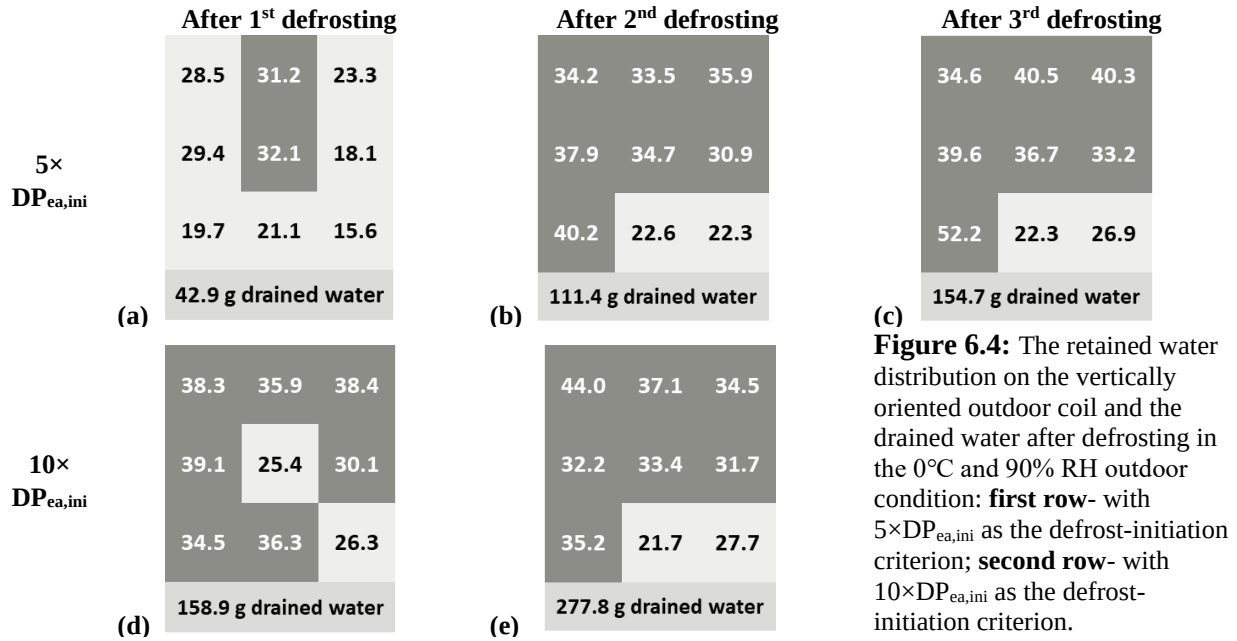


Figure 6.4 shows the distribution of retained water on the vertically oriented outdoor coil and the drained water after several continuous defrosting cycles with 5 or 10 times the initial

outdoor DP_{ea} as the defrost-initiation criterion. The dark grey background represents the block retaining more than 30 g of water after defrosting, and the light grey background represents less than 30 g of retained water. It is clear that the water distribution is relatively uneven though the evaporator is flooded in the HP operation.

First, the uneven distribution of retained water matches the frost distribution on the outdoor coil after the first short defrosting cycle, when there is limited water drainage, as shown in **Figure 6.4(a)** and the frost image in author's previous paper (Zhang and Hrnjak, 2022a). Due to the frictional pressure drop of the refrigerant, the evaporating temperature drops along the flow direction, and the frost accumulates most by the top of the first and the second pass. Also, more frost accumulates on the left half of the second pass than on the right half (**Figure 6.4(a)**) because of the uneven distribution of two-phase refrigerant: more liquid refrigerant enters the microchannel tubes in the left half of the second pass. In addition, the distribution of retained water becomes less even in the following defrosting cycles due to water drainage and vaporization. The superheated refrigerant enters from the right-bottom of the outdoor coil during the defrosting, and most of the evaporation occurs here. On the other hand, more water drains without vaporization from the top to the bottom in the first pass, as shown in **Figure 6.4**. The unevenness of the accumulated frost/retained water can increase heat loss (from the blocks with less frost/retained water) to the outdoor cold air and reduce the defrosting efficiency and the overall performance of the heat pump system.

6.4.2 The energy supply during continuous defrosting cycles

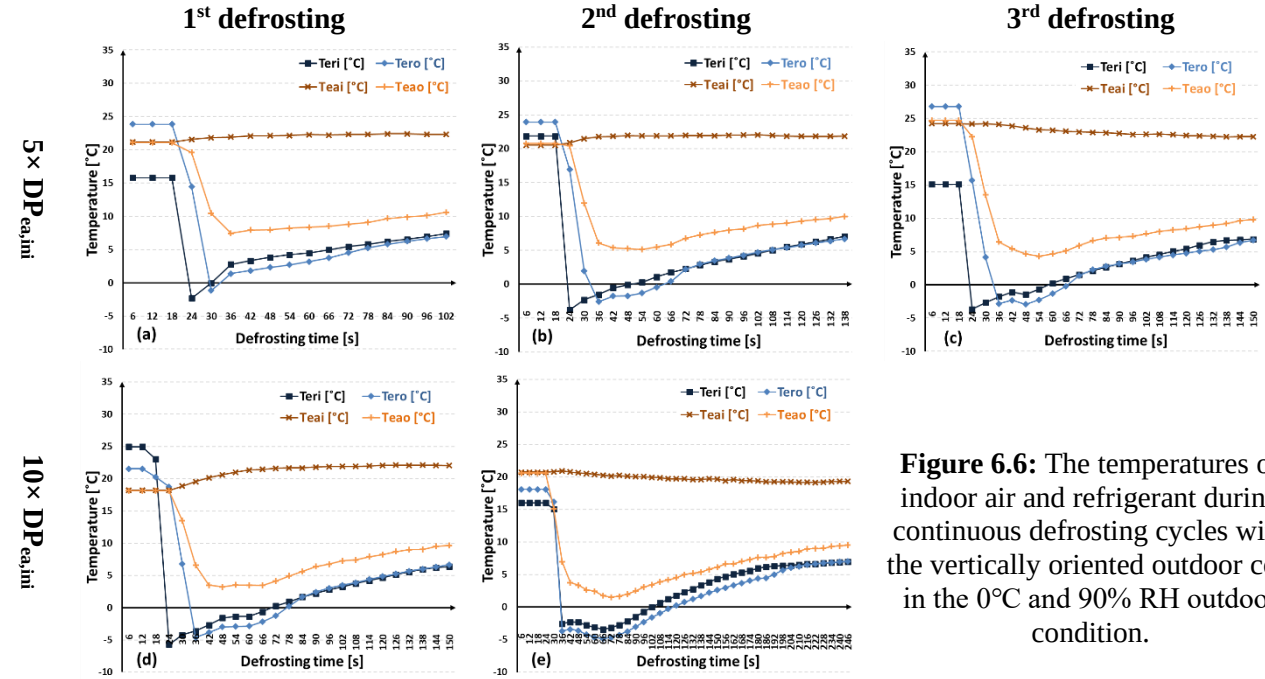
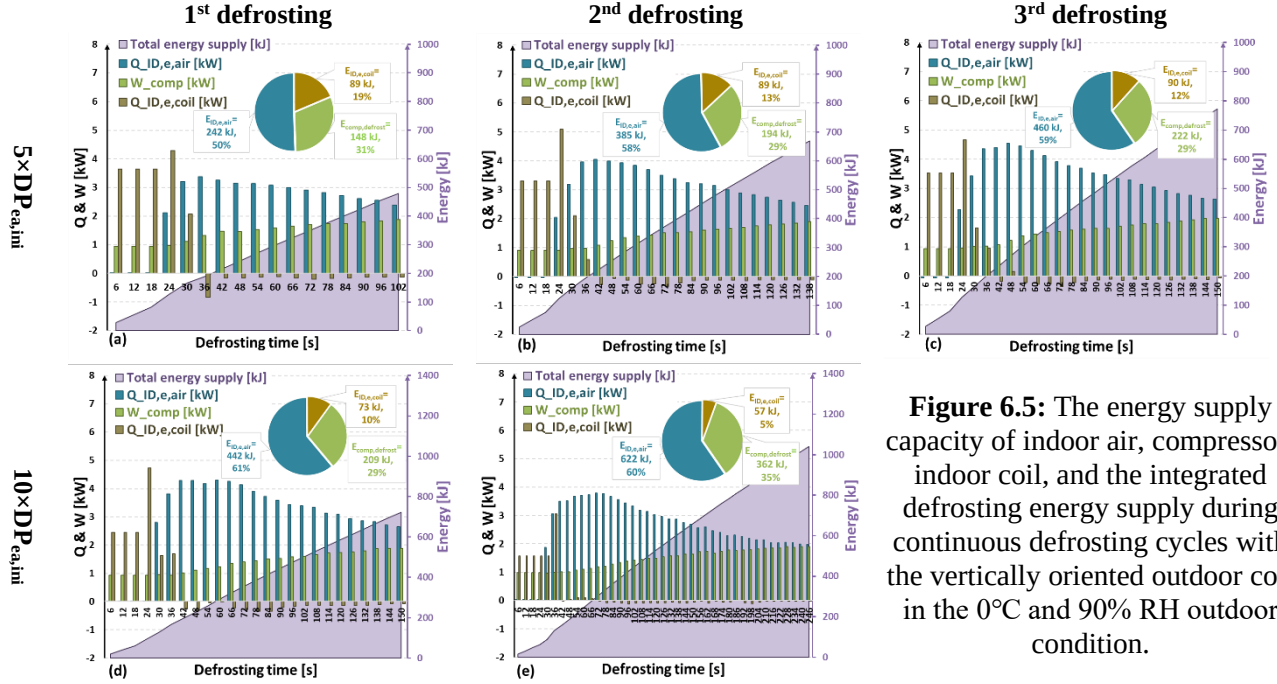
Figure 6.5 presents the transient energy supply capacities (bars in kW) of the indoor air $Q_{ID,e,air}$, compressor W_{comp} , indoor coil metal $Q_{ID,e,coil}$, and the integrated total energy supply

($E_{total,supply}$ in kJ) over the defrosting time with different defrost-initiation criteria in the outdoor condition of 0°C and 90% RH with vertically oriented outdoor coil for continuous defrosting cycles. The pie charts in **Figure 6.5** show the integrated energy of each capacity ($E_{ID,e,air}$, $E_{comp,defrost}$, and $E_{ID,e,coil}$ in kJ) and the percentage of each component out of the integrated total energy supply $E_{total,supply}$. **Figure 6.6** shows the temperatures of indoor air and refrigerant at the inlet and exits of the vertically installed indoor coil during the continuous defrosting cycles with different defrost-initiation criteria in the same condition.

a. The transient trends of the energy supply during one defrosting cycle

As shown by the brown bar in **Figure 6.5**, the indoor coil metal, which is warmer than the indoor air, provides most of the defrosting energy at the beginning of the defrosting process—approximately the first half to one minute for different defrosting cycles. But in the first 18 s to half a minute of one defrosting cycle, there is almost no difference between T_{eai} and T_{eao} (**Figure 6.6**), so the indoor air provides negligible heat for defrosting.

In the previous frosting cycle/HP operation, the indoor coil works as the indoor gas cooler. So, by the end of the previous frosting cycle, the average indoor coil temperatures were approximately 48°C and 35°C, respectively, for the cycles with 5 and 10 times the initial outdoor DP_{ea} as the defrost-initiation criterion. $Q_{ID,e,coil}$ firstly peaks due to the decreasing indoor evaporating temperature during the start-up of the compressor, as shown in **Figure 6.6**. Then, it reduces to zero as the temperature of the coil approaches the evaporating temperature, which can be as low as -5°C. Finally, $Q_{ID,e,coil}$ stays close but below zero Celsius degree because the temperatures of both refrigerant and indoor coil metal increase during this start-up process (as shown by T_{eri} and T_{ero} in **Figure 6.6**), so that the coil metal no longer provides energy for defrosting



in the latter half of one defrosting process. On the other hand, the indoor air supplies most of the energy after the coil temperature approaches the evaporating temperature. After reversing the

system and the restarting of the compressor, the evaporating temperature first hits the lowest point of approximately -5°C and then increases and stabilizes at 7°C , as shown in **Figure 6.6**. As a result, the temperature difference between T_{eai} and T_{eao} (**Figure 6.6**), the cooling capacity $Q_{ID,e,air}$ (**Figure 6.5**) increase to the peak values and then decrease. And the power consumption of the compressor W_{comp} increases gradually from about 1 to 2 kW during the defrosting, the mass flow rate of refrigerant increases from approximately 20 to 38 g/s. In the meanwhile, the evaporator exit quality drops significantly and the two-phase refrigerant enters the accumulator, so the liquid refrigerant migrates from the cold outdoor coil and liquid line (in previous HP mode) to the accumulator during the start-up of the compressor and the short defrosting operation.

b. The periodic defrosting performance

Figure 6.5(a) to (c) show the periodic defrosting performance of the CO₂ HP system with 5 times the initial DP_{ea} as the defrost-initiation criterion. Overall, the defrosting time is 102, 138, and 150 s, and the total defrosting energy supply $E_{total,supply}$ is 479, 669, and 773 kJ for the continuous three defrosting cycles. The total energy supply $E_{total,supply}$ is increasing linearly with the defrosting time $t_{defrosting}$.

For the three energy sources, the heat provided by the indoor coil $E_{ID,e,coil}$ is almost constant to be 90 kJ for the three defrosting cycles, as shown in **Figure 6.5(a) to (c)**. Because the temperature change of the indoor coil during the three continuous defrosting processes is the same with 5 times initial outdoor DP_{ea} as the defrost-initiation criterion. However, the heat provided by the indoor air $E_{ID,e,air}$ and the energy supplied by the compressor $E_{comp,defrost}$ are increasing proportionally with the defrosting time. $E_{ID,e,air}$ is 242, 385, and 460 kJ, and $E_{comp,defrost}$ is 148, 194, and 222 kJ for the continuous three defrosting cycles. Especially, $E_{ID,e,air}$ increases faster than

$E_{comp,defrost}$ with the defrosting time, so the percentage of $E_{ID,e,air}$ out of the total energy supply increases from 50% to 58% from the first to the third defrosting cycle. The higher percentage of $E_{ID,e,air}$ indicates more negative impacts on the indoor temperature: As shown in **Figure 6.6(a)** to **(c)**, the minimum air exit temperature T_{eao} is 7.4, 5.1, and 4.3°C for the continuous three defrosting processes.

c. The effects of the defrost-initiation criterion

By comparing the first and second rows of **Figure 6.5**, the major impacts of different defrost-initiation criteria are the defrosting time and the percentage of each component out of $E_{total,supply}$. The defrosting time $t_{defrosting}$ is 47.1% longer and $E_{total,supply}$ is 50.9% higher for the first defrosting cycle using 10 times than using 5 times of the initial outdoor DP_{ea} as the defrost-initiation criterion (**Figure 6.5(a)** and **(d)**). On the one hand, the mass of accumulated frost M_{frost} is higher with 10 times the initial outdoor DP_{ea} as the defrost-initiation criterion than it is with 5 times (**Figure 6.3**), so it takes a longer time and more energy to melt the frost and warm up the retained water. On the other hand, the heating capacity of the HP $Q_{heating}$ and the average temperature of the indoor coil by the end of the previous frosting cycle drop more with longer operation time using 10 times the initial outdoor DP_{ea} as the defrost-initiation criterion than they do with 5 times (Zhang and Hrnjak, 2022a). Therefore, the defrosting energy from the indoor coil metal $E_{ID,e,coil}$, and the percentage is lower with 10 times the initial outdoor DP_{ea} as the defrost-initiation criterion than what they are with 5 times: it is 73 (10%) and 89 kJ (19%), respectively. As a result of the longer $t_{defrosting}$, the indoor air and compressor power provides more defrosting energy with higher percentages in the experiments with 10 times the initial outdoor DP_{ea} as the defrost-initiation criterion than 5 times: indoor air provides 50% of $E_{total,supply}$ with 5 times the

initial outdoor DP_{ea} as the defrost-initiation criterion in the first defrosting cycle, and it is 61% for 10 times. Consequently, the lowest indoor air exit temperature T_{eao} is 7.4 and 3.2°C for 5 and 10 times the initial outdoor DP_{ea} as the defrost-initiation criterion in the first defrosting cycle (**Figure 6.6(a)** and **(d)**), which means more temperature fluctuations of the indoor air during the defrosting with 10 times the initial outdoor DP_{ea} as the defrost-initiation criterion. However, it is possible to defrost the outdoor coil when the vehicle is parking in the parking lot or charging center and minimize the negative effects on passengers' comfort. Considering the long operating time (68 minutes) in the most demanding outdoor condition (0°C and 90%RH), it is longer than the average commuting time of Americans- 50 minutes (Zhang and Hrnjak, 2022a).

In addition, the defrosting efficiency is higher with 10 times the initial outdoor DP_{ea} as the defrost-initiation criterion. First, the percentage-wise defrosting time is 7.0% and 5.7% for using 5 and 10 times the initial outdoor DP_{ea} as the defrost-initiation criterion in the experiments. Second, the sum of $E_{total,supply}$ in the three continuous defrosting cycles using 5 times the initial outdoor DP_{ea} as the defrost-initiation criterion can be calculated and divided by the total operating time of the HP system ($t_{frosting} + t_{defrosting}$): the operating-time-averaged energy supply for defrosting is 20.76 kJ/min. Similarly, it is 15.14 kJ/min for the two continuous defrosting cycles using 10 times the initial outdoor DP_{ea} as the defrost-initiation criterion. In conclusion, it is more time- and energy-efficiently to apply 10 times the initial outdoor DP_{ea} as the defrost-initiation criterion than 5 times.

6.4.3 The energy consumption during continuous defrosting cycles

Figure 6.7 shows the heating capacity of the vertically oriented outdoor coil based on refrigerant-side measurements $Q_{OD,c,ref}$ and the integrated total defrosting energy consumption

$E_{OD,c,ref}$. **Figure 6.8** presents the four calculated/estimated heating loads ($Q_{OD,c,frost}$, $Q_{OD,c,coil}$, $Q_{OD,c,water}$, $Q_{OD,c,loss}$) and the integrated energy consumption of those four loads. The pie charts in **Figure 6.8** show the integrated values of the four loads and the percentages of them out of the sum. **Figure 6.9** shows the inlet and exit refrigerant temperature T_{cri}/T_{cro} and the surface temperature of outdoor coil T_{surf} . All the figures include experimental data during continuous defrosting cycles in the outdoor condition of 0°C and 90% RH with different defrost-initiation criteria.

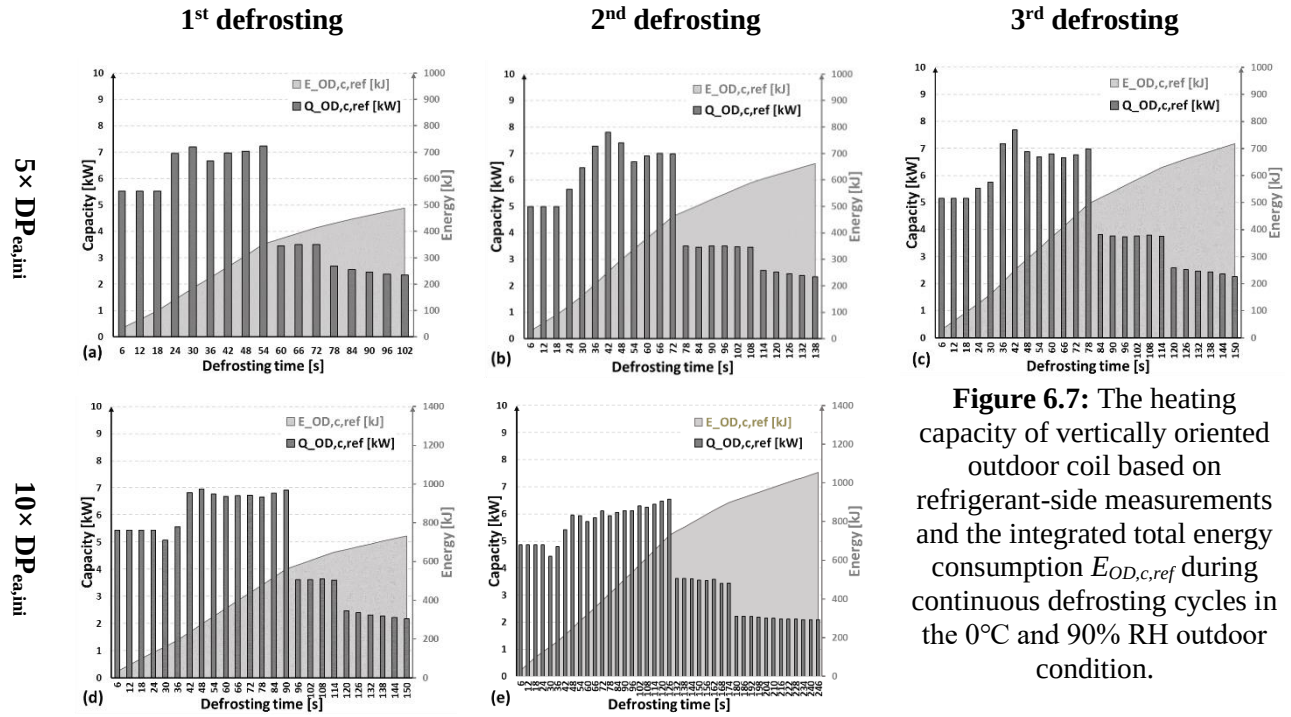


Figure 6.7: The heating capacity of vertically oriented outdoor coil based on refrigerant-side measurements and the integrated total energy consumption $E_{OD,c,ref}$ during continuous defrosting cycles in the 0°C and 90% RH outdoor condition.

For the accuracy of the energy consumption calculation, it is observed that the integrated total defrosting energy consumption calculated using refrigerant-side measurements $E_{OD,c,ref}$ in **Figure 6.7** agrees well with the integrated total energy supply $E_{total,supply}$ in **Figure 6.5** for all defrosting experiments (within -4% to 8%). This indicates a good energy balance between

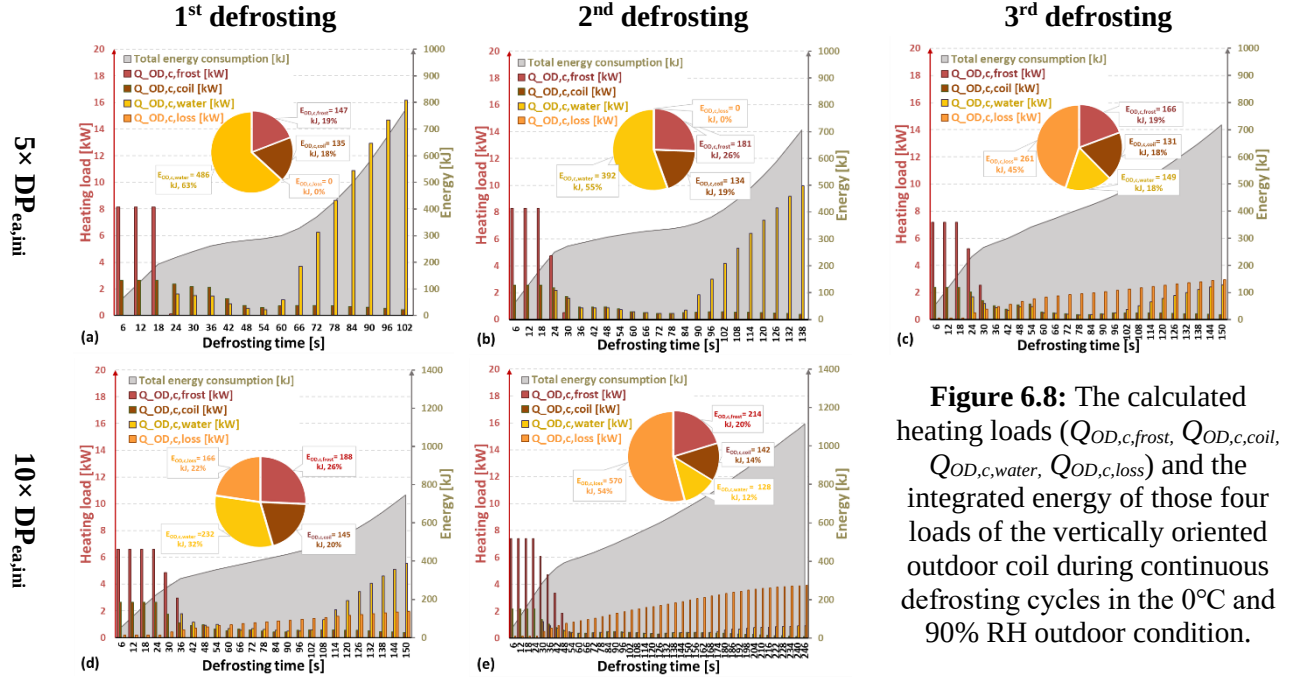


Figure 6.8: The calculated heating loads ($Q_{OD,c,frost}$, $Q_{OD,c,coil}$, $Q_{OD,c,water}$, $Q_{OD,c,loss}$) and the integrated energy of those four loads of the vertically oriented outdoor coil during continuous defrosting cycles in the 0°C and 90% RH outdoor condition.

independent measurements. However, $E_{OD,c,water}$ can be overestimated in **Figure 6.8(a)** and **(b)**, and the integrated total energy consumption of all four loads ($Q_{OD,c,frost}$, $Q_{OD,c,coil}$, $Q_{OD,c,water}$, $Q_{OD,c,loss}$) in **Figure 6.8(a)** and **(b)** is higher than $E_{OD,c,ref}$ by 57% and 7% for the first and second defrosting cycles using 5 times of the initial outdoor DP_{ea} as defrosting criterion. This is due to the imperfectness of water collecting methods and mass measurements in the first few defrosting cycles, as mentioned in the data reduction section. Therefore, there is a gap between the energy balances calculated using refrigerant measurements and using water mass measurements. In addition, the transient heating capacity $Q_{OD,c,ref}$ is not necessarily matching the sum of the four heating loads for each moment during the defrosting since there can be a time delay during the transfer of the energy.

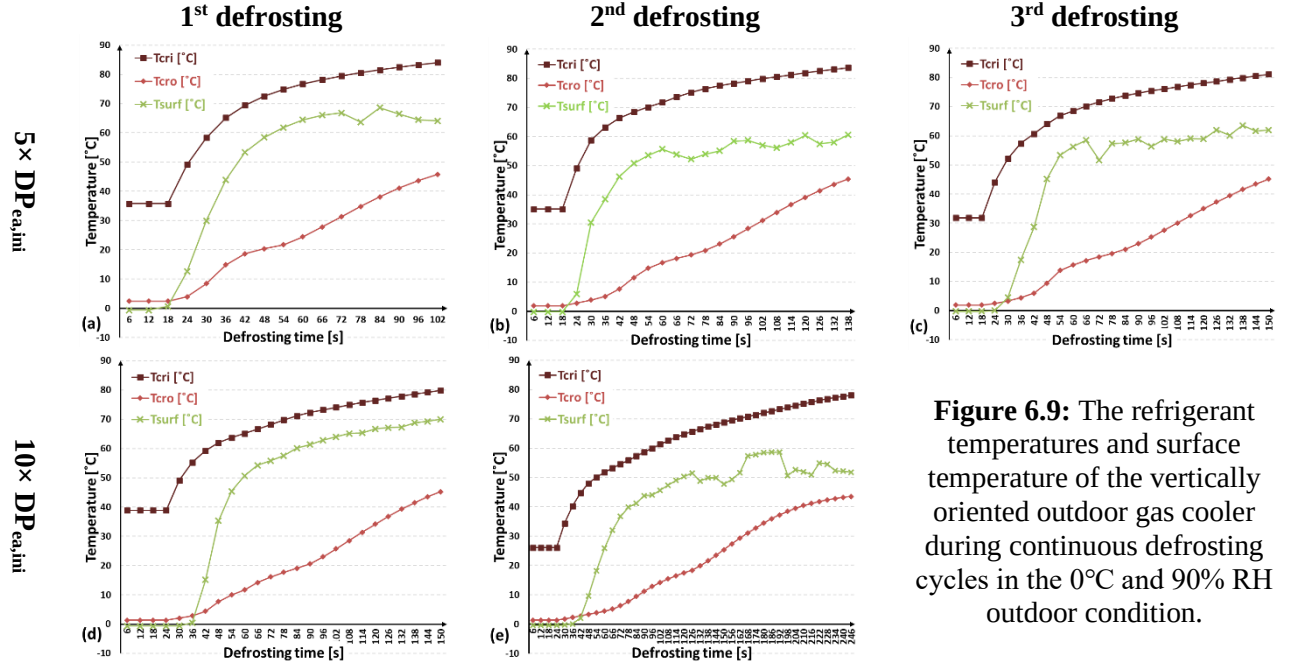


Figure 6.9: The refrigerant temperatures and surface temperature of the vertically oriented outdoor gas cooler during continuous defrosting cycles in the 0°C and 90% RH outdoor condition.

a. The transient trends of the energy supply during one defrosting cycle

In the first half of defrosting, the heating capacity of outdoor condenser $Q_{OD,c,ref}$ is significantly higher than it is in the latter half (**Figure 6.7**). This is due to the low temperature of the outdoor coil metal and the frost at the beginning of defrosting, which leads to a low refrigerant exit temperature T_{cro} and a high heating capacity $Q_{OD,c,ref}$. Also, most of the energy is used to warm the outdoor coil metal and to melt the frost on the outdoor coil in the first half of defrosting, as shown by the brown ($Q_{OD,c,coil}$) and red ($Q_{OD,c,frost}$) bars in **Figure 6.8**. This matches the observation that all frost melt in the first 20 to 40 seconds based on the images taken with the webcams.

However, as the temperature of the coil metal and the retained water are rising above the critical temperature of CO₂, refrigerant leaves the outdoor coil as superheated vapor or supercritical CO₂ (as shown by T_{cro} in **Figure 6.9**). Thus, the heating capacity $Q_{OD,c,ref}$ drops significantly from 7 to 2 kW by the end of each defrosting cycle (**Figure 6.7**) due to the reduced

LMTD, although the refrigerant mass flow rate increases slightly at the same time. In addition, heat loss $Q_{OD,c,loss}$ increases proportionally to the surface temperature during the defrosting process, and $Q_{OD,c,water}$ rises significantly after the surface temperature is above 50°C, as shown by the orange and yellow bars in **Figure 6.8**.

b. The periodic defrosting performance

The total energy consumption $E_{OD,c,ref}$ increases linearly with the defrosting time for the continuous defrosting cycles. It is 489, 662, and 718 kJ for the three defrosting cycles with 5 times of initial outdoor DP_{ea} as the defrost-initiation criterion (**Figure 6.7(a) to (c)**), and it is 731 and 1054 kJ for the two defrosting cycles with 10 times of initial outdoor DP_{ea} as defrost-initiation criterion (**Figure 6.7(d) and (e)**).

In **Figure 6.8**, the load $Q_{OD,c,water}$ calculated based on the mass measurements are clearly overestimated in the first two defrosting cycles (**Figure 6.7(a) to (b)**) with 5 times of initial outdoor DP_{ea} as the defrost-initiation criterion. However, as the coil surface gets saturated with retained water, the behaviors of water drainage become repeatable, and the accuracy of the mass measurements as well as the load $Q_{OD,c,water}$ are improved after longer operation in HP mode. As a result, the integrated total energy consumption of all four loads becomes closer to the most accurate value $E_{OD,c,ref}$ in the third/second defrosting cycle (**Figure 6.7(c) and (e)**) with 5/10 times of initial outdoor DP_{ea} as defrost-initiation criterion.

As shown in **Figure 6.9(a) to (c)**, the refrigerant temperatures and surface temperature of the outdoor coil are similar at the beginning and end for three continuous defrosting cycles using identical defrosting-initiation and termination criteria. There are two major differences in the temperatures of the outdoor coil: First, T_{surf} and T_{cro} stay at 0°C and 2°C for 18, 18, and 24 s for

the three continuous defrosting cycles with 5 times of initial outdoor DP_{ea} as defrost-initiation criterion. This indicates the time duration of melting the frost increases with the number of defrost cycles. Second, there are two-, three-, and four-times temperature fluctuations on the curves of T_{surf} (green) during the three continuous defrosting cycles with 5 times of initial outdoor DP_{ea} as the defrost-initiation criterion, which relates to the number of the falling water streams. It shows that more water drains as the number of defrosting cycles increases, and this conclusion agrees well with previous measurements of $M_{drained}$ (**Figure 6.3**).

c. The effects of the defrost-initiation criterion

Although the defrosting time is longer, and the total energy consumption is higher for each defrosting cycle using 10 times the initial outdoor DP_{ea} as the defrost-initiation criterion than they are for 5 times (e.g., comparing **Figure 6.7(a)** and **(d)**). But considering the longer operating time in HP mode and fewer defrosting cycles, the time-averaged energy consumption (the sum of $E_{OD,c,ref}$ divided by the total operating time of HP) is 20.21 and 15.32 kJ/min for using 5 times and 10 times of the initial outdoor DP_{ea} as the defrost-initiation criterion, respectively. Again, it is more energy-efficiently to apply 10 times the initial outdoor DP_{ea} as the defrost-initiation criterion.

6.5 Effects of the orientation of the outdoor coil: horizontal vs. vertical

In this section, the effects of the orientation of the outdoor coil on the defrosting capacity, efficiency, and meltwater distribution of the reversible R744 HP in continuous defrosting cycles under the outdoor condition of 0°C and 90% RH are discussed and analyzed.

6.5.1 The overall performance during continuous defrosting cycles

Figure 6.10 presents the accumulated mass of frost and meltwater on both the vertically and horizontally oriented outdoor coil during two continuous frosting/defrosting cycles with 10 times initial outdoor DP_{ea} as the defrost-initiation criterion in the outdoor condition of 0°C and 90% RH. The different rates of frost accumulation on the vertically and horizontally oriented outdoor evaporators during frosting cycles are discussed and linked with the refrigerant distribution and pressure drop in chapter 5. The operating time $t_{frosting}$, because of the different rates of frost accumulation, is 56 minutes and 17.6% shorter for the HP with the horizontally oriented outdoor coil than it is with the vertically oriented outdoor coil in the first frosting cycle. In addition, the defrosting time $t_{defrosting}$ is 2.6 and 5.2 minutes for the horizontally oriented outdoor coil, though it is 2.5 and 4.0 minutes for the vertically oriented outdoor coil. Percentagewise, it is 7.0% and 5.6% of the total operating time for the HP system with horizontally and the vertically oriented outdoor coil, respectively. On the one hand, this is due to the different surface temperatures and frost density: denser and more frost accumulates on the surface of the horizontally oriented outdoor coil by the ends of the frosting cycles (**Figure 6.10**). On the other hand, this is also because of the worse water drainage of the horizontally oriented outdoor coil. As shown by the $M_{retained}$ in **Figure 6.10**, it retains 567.0 g for the horizontally oriented outdoor coil after the second defrosting cycle, but it retains only 297.8 g of water for the vertically oriented outdoor coil. In the meanwhile, it drains 79.6 and 277.8 g for horizontally and vertically oriented outdoor coil. In conclusion, the vertical orientation of the outdoor coil reduces the defrosting time by increasing the surface temperature and improving the water drainage during defrosting.

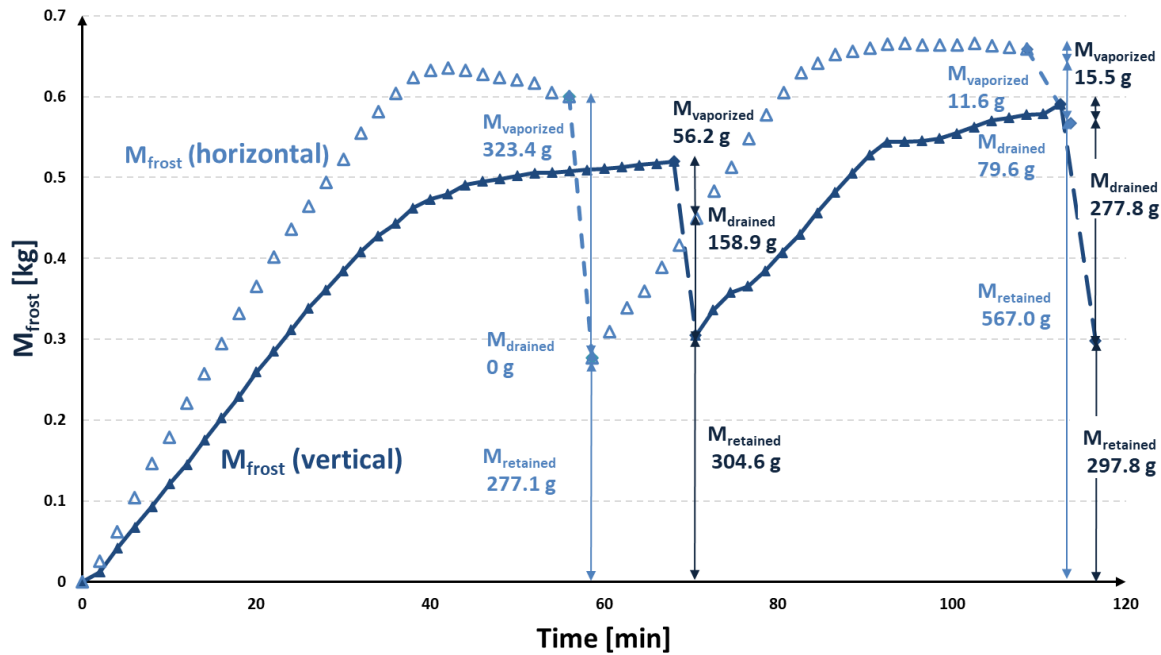


Figure 6.10: The mass of frost accumulated and the meltwater on the horizontally and vertically oriented outdoor coil during continuous frosting/defrosting cycles in the 0°C and 90% RH outdoor condition. The long blue dash lines represent the water removal, and the masses of the vaporized, drained, and retained water are labeled by the end of each defrosting cycle.

		After 1 st defrosting			After 2 nd defrosting		
Horizontal	(a)	24.6	26.2	22.6	37.1	29.4	28.2
		30.1	30.2	34.9	45.3	47.7	35.4
		30.7	40.1	37.4	101.2	131.4	111.1
		0 g drained water			79.6 g drained water		
Vertical	(c)	38.3	35.9	38.4	44.0	37.1	34.5
		39.1	25.4	30.1	32.2	33.4	31.7
		34.5	36.3	26.3	35.2	21.7	27.7
		158.9 g drained water			277.8 g drained water		
		(b)			(d)		

Figure 6.11: The retained water distribution and the drained water after defrosting of both horizontally and vertically oriented outdoor coil in the 0°C and 90% RH outdoor condition for two continuous defrosting cycles.

Figure 6.11 presents the distribution of the water retention and the water drainage for both horizontally and vertically oriented outdoor coils after two continuous defrosting cycles. It is clear that the water distribution is less even due to the drainage for the horizontally oriented outdoor coil: the water retained on the lowest rows almost doubles compared to the other blocks. The unevenness distribution can increase the heat loss to the ambient and reduce the defrosting efficiency.

6.5.2 The energy supply during continuous defrosting cycles

Figure 6.12 presents the transient energy supply capacities (bar charts, in kW) of the indoor air $Q_{ID,e,air}$, compressor W_{comp} , indoor coil metal $Q_{ID,e,coil}$, and the integrated total energy supply ($E_{total,supply}$ in kJ) over the defrosting time with horizontally and vertically oriented outdoor coil in the outdoor condition of 0°C and 90% RH with vertically oriented outdoor coil for continuous defrosting cycles. The pie charts in **Figure 6.12** show the integrated energy of each capacity ($E_{ID,e,air}$, $E_{comp,defrost}$, and $E_{ID,e,coil}$ in kJ) and the percentage of each component out of the integrated total energy supply $E_{total,supply}$. **Figure 6.13** shows the temperatures of indoor air and refrigerant at the inlet and exit of both horizontally and vertically installed indoor coil during two continuous defrosting cycles in the same condition.

By comparing the first and second row of **Figure 6.12**, the general trends of capacities and the percentage of each component out of $E_{total,supply}$ are similar for both orientations during the two defrosting cycles. The percentage of $E_{comp,defrost}$ increases from 29% to 37% and from 29% to 35% for horizontally and vertically oriented outdoor coil from the first to the second defrosting cycle, respectively. This indicates the longer defrosting time requires higher quality/temperature heat and

results in lower efficiency. Therefore, the optimized defrosting-termination criteria can be further explored.

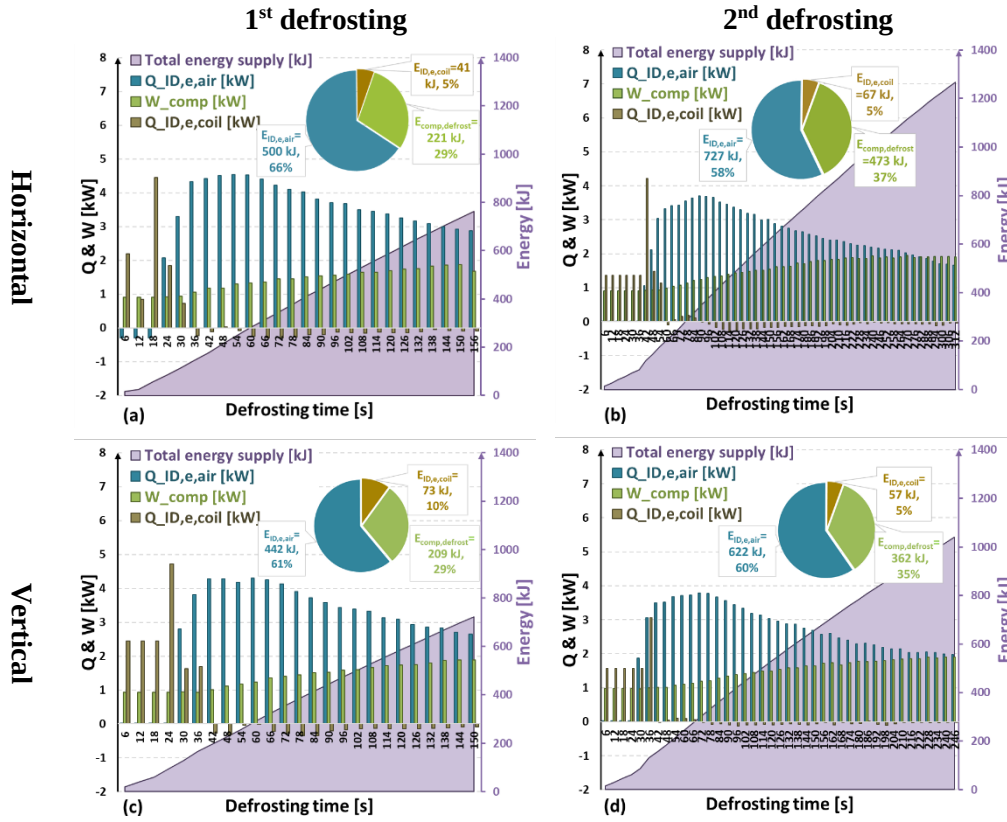


Figure 6.12: The energy supply capacity of the indoor air, compressor, indoor coil, and the integrated defrosting energy supply with both horizontally and vertically oriented outdoor coil in the 0°C and 90% RH outdoor condition for two continuous defrosting cycles.

The main differences between the two orientations are that $t_{defrosting}$ is 26.8% longer and $E_{total,supply}$ is 10.9% higher for the horizontally oriented outdoor coil in the second defrosting cycle, as shown in **Figure 6.12(b)** and **(d)**. The reasons are discussed in the last section: denser frost and worse water drainage for the horizontally oriented outdoor coil, so it takes longer time and more energy to defrost and warm up the outdoor coil as well as the retained water. The sum of $E_{total,supply}$ in the two continuous defrosting cycles can be calculated and divided by the total operating time

of the HP system ($t_{\text{freezing}} + t_{\text{defrosting}}$): it is 17.84 and 15.14 kJ/min for horizontally and vertically oriented outdoor coil.

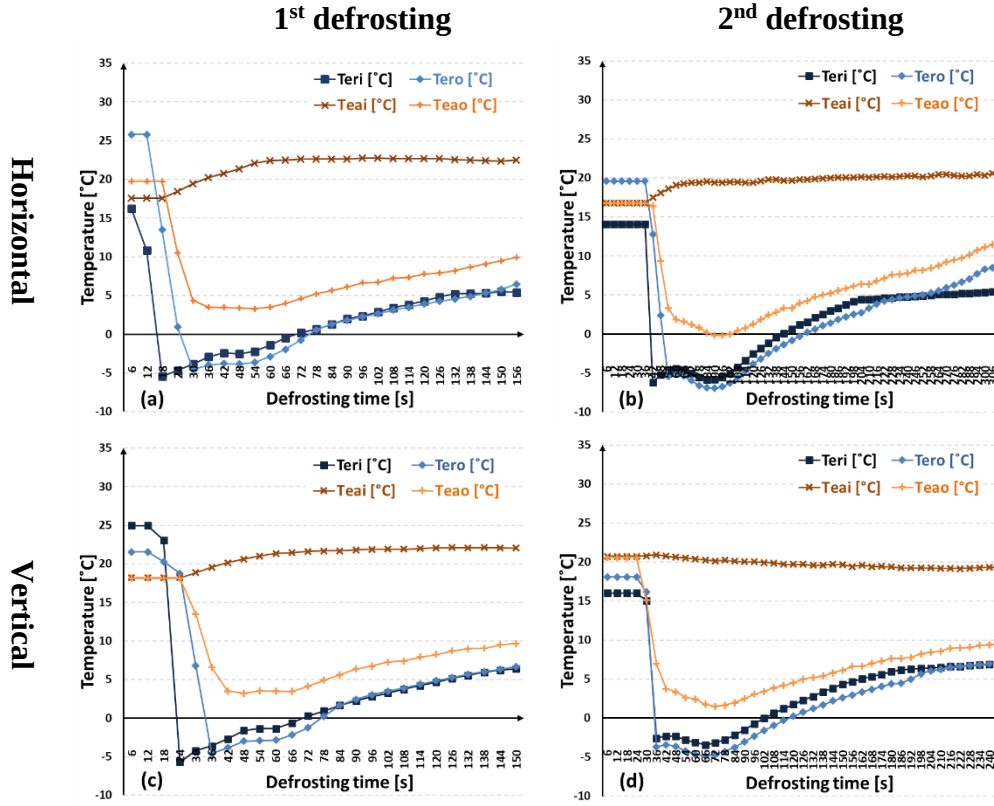


Figure 6.13:
The temperatures of indoor air and refrigerant with both horizontally and vertically oriented outdoor coil in the 0°C and 90% RH outdoor condition for two continuous defrosting cycles.

Also, the lowest air exit temperatures T_{eao} are identical for both orientations in the first defrosting cycle, but they are -0.2 and 1.5°C for horizontally and vertically oriented outdoor coil in the second defrosting cycle, as shown in **Figure 6.13**. Therefore, there are more negative temperature fluctuations of the indoor air in the continuous defrosting cycles with a horizontally oriented outdoor coil.

In conclusion, it is more time- and energy-efficient to defrost a vertically oriented outdoor coil in the two-hour operation of the air-source heat pump in the 0°C and 90%RH outdoor condition.

6.5.3 The energy consumption during continuous defrosting cycles

Figure 6.14 shows the heating capacity of both the horizontally and vertically oriented outdoor coil based on the refrigerant-side measurements $Q_{OD,c,ref}$ and the integrated total defrosting energy consumption $E_{OD,c,ref}$. **Figure 6.15** presents the four calculated/estimated heating loads ($Q_{OD,c,frost}$, $Q_{OD,c,coil}$, $Q_{OD,c,water}$, $Q_{OD,c,loss}$), the integrated energy of those four loads, as well as the percentages of them out of the sum. **Figure 6.16** shows the inlet and exit refrigerant temperature T_{cri}/T_{cro} and the surface temperature of the outdoor coil T_{surf} . All figures include the experimental data with both horizontally and vertically oriented outdoor coil during two continuous defrosting cycles in the outdoor condition of 0°C and 90%RH.

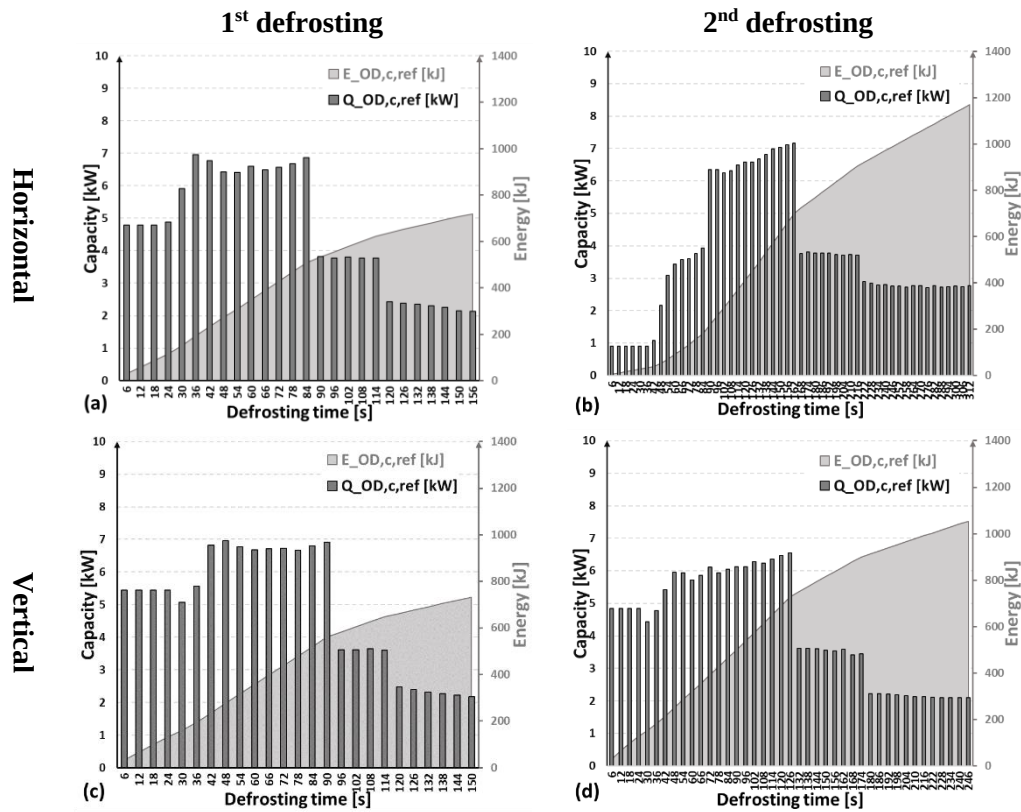


Figure 6.14: The heating capacity of both horizontally and vertically oriented outdoor coil based on refrigerant-side measurements and the integrated energy consumption $E_{OD,c,ref}$ in the 0°C and 90% RH outdoor condition for two continuous defrosting cycles.

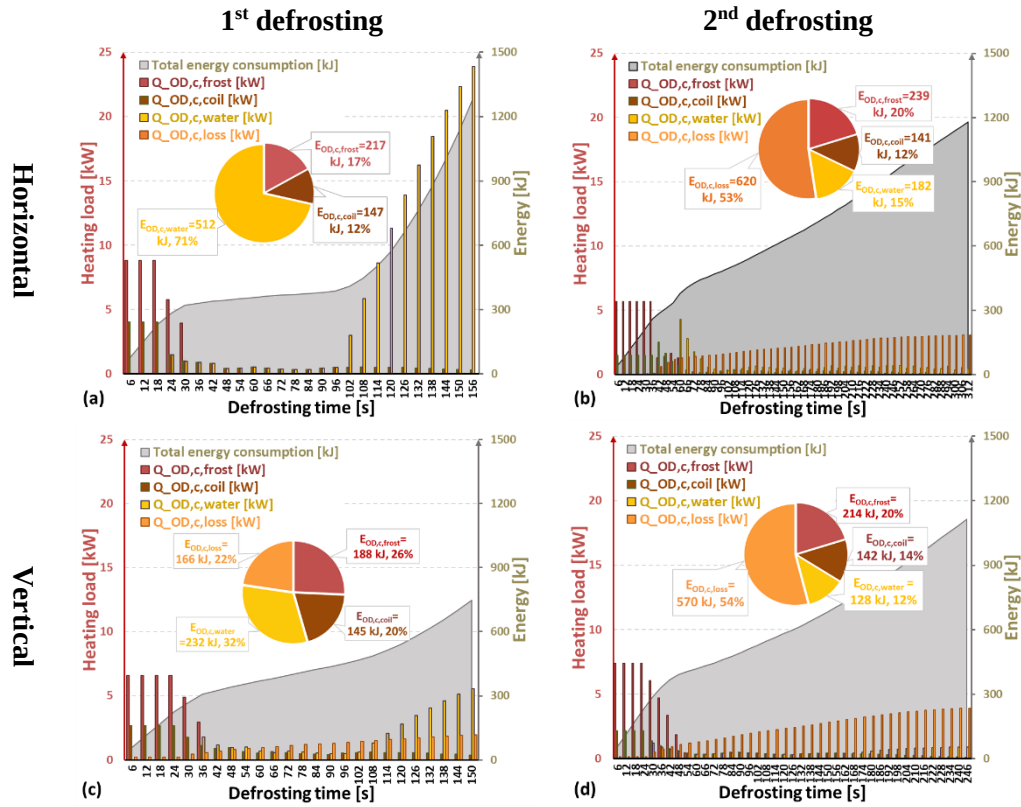


Figure 6.15: The calculated heating loads ($Q_{OD,c,frost}$, $Q_{OD,c,coil}$, $Q_{OD,c,water}$, $Q_{OD,c,loss}$) and the integrated energy of those four loads of both horizontally and vertically oriented outdoor coil in the 0°C and 90% RH outdoor condition for two continuous defrosting cycles.

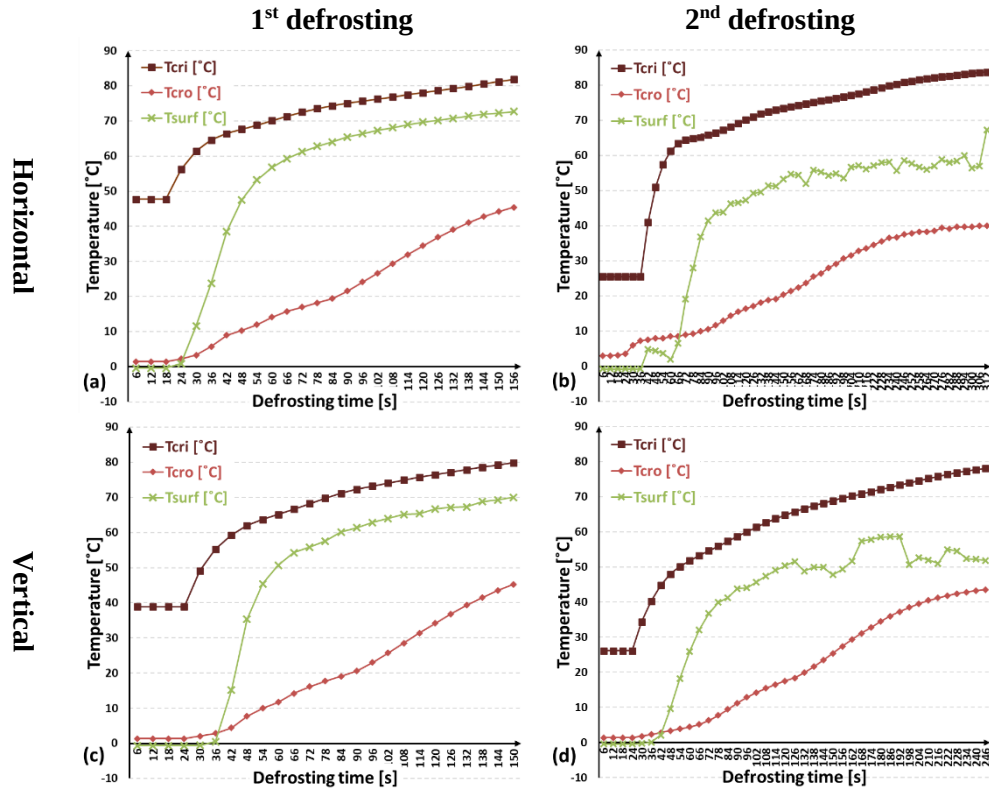


Figure 6.16: The refrigerant and surface temperatures of both horizontally and vertically oriented outdoor coil in the 0°C and 90% RH outdoor condition for two continuous defrosting cycles.

For the accuracy, $E_{OD,c,ref}$ in **Figure 6.14** agrees well (within -4% to 8%) with the total defrosting energy supply $E_{total,supply}$ in **Figure 6.12**. However, it is noticeable that the sum of the integrated four calculated loads in the first defrosting cycle (**Figure 6.15(a)**) with the horizontally oriented outdoor coil is significantly higher than the corresponding $E_{OD,c,ref}$ in **Figure 6.14(a)**. The imbalance between the energy calculation using refrigerant measurement and the water mass measurements might be due to the imperfect water collection methods, which should be improved in future studies. Also, it is believed that the accuracy of the results is improved in the second frosting/defrosting cycle.

As the trend of $Q_{total,supply}$, the trends of $Q_{OD,c,ref}$ are almost identical for both orientations in the first defrosting cycle (**Figure 6.14(a)** and **(c)**). But the duration $t_{defrosting}$ is longer and $E_{OD,c,ref}$ is higher for the horizontally oriented outdoor coil for the second defrosting cycle (**Figure 6.14(b)** and **(d)**) due to the more frost accumulation. The operating-time averaged energy consumption (the sum of $E_{OD,c,ref}$ of two defrosting cycles divided by the total operating time) is 16.59 and 15.32 kJ/min for horizontally and vertically oriented outdoor coil. Again, it requires more defrosting energy for the horizontally oriented outdoor coil. For different defrosting loads, the percentage of each component in the pie chart in **Figure 6.15(b)** and **(d)** are similar for both orientations, though the absolute values are higher for the horizontally oriented outdoor coil.

The amplitude of temperature fluctuations of surface temperature T_{surf} is smaller for the horizontally oriented outdoor coil (**Figure 6.16(a)** and **(b)**) than it is for the vertically oriented outdoor coil (**Figure 6.16(c)** and **(d)**). This indicates that the volume/mass flow rate of the downward flow of the water stream is less for the horizontally oriented outdoor coil than it is for

the vertically oriented outdoor coil, which agrees well with our measured water drainage in **Figure 6.10**.

In this chapter, the effects of different defrost-initiation criteria and the orientation of the outdoor coil on the defrosting performance and the energy flow during the defrosting are studied using a reversible R744 HP system in two continuous frosting/ defrosting cycles under the outdoor condition of 0°C and 90% relative humidity. The results show:

- The integrated total defrosting energy consumption calculated using refrigerant-side measurements $E_{OD,c,ref}$ agrees well with the integrated total energy supply $E_{total,supply}$ for all defrosting experiments (within -4% to 8%). This indicates a good energy balance between independent measurements. However, the measurement of the mass of water in the first one to two cycles and the calculation of four defrosting loads should be improved in future studies.
- The defrosting time, water retention, and water drainage of the outdoor coil increase gradually with the number of the frosting/defrosting cycles. The retained water gets saturated with approximately 300 g for the vertically installed outdoor coil, and it is more than 567 g for the horizontally oriented outdoor coil. But the mass of the vaporized water $M_{vaporized}$ reduces from hundreds of grams to tens of grams with the increased number of the frosting/defrosting cycles due to the increased thickness of the water film on the surface and the lower surface temperature of the water film.
- The major impacts of different defrost-initiation criteria are the defrosting time, defrosting energy, and the percentage of each component out of $E_{total,supply}$. First, the percentage-wise defrosting time is 7.0% and 5.7% for using 5 and 10 times the initial

outdoor DP_{ea} as the defrost-initiation criterion in the experiments. Second, the operating-time-averaged energy supply for defrosting is 20.76 kJ/min for the three defrosting cycles using 5 times the initial outdoor DP_{ea} as the defrost-initiation criterion, though it is only 15.14 kJ/min for the two continuous defrosting cycles using 10 times the initial outdoor DP_{ea} as the defrost-initiation criterion. Third, the indoor air provides 50% of $E_{total,supply}$ with 5 times the initial outdoor DP_{ea} as the defrost-initiation criterion in the first defrosting cycle, but it is 61% for 10 times. Consequently, the minimum indoor air exit temperature T_{eao} is 7.4 and 3.2°C for 5 and 10 times the initial outdoor DP_{ea} as the defrost-initiation criterion in the first defrosting cycle.

- The main differences between the horizontal and vertical orientation of the outdoor coil are the defrosting time and the total defrosting energy. First, the defrosting time $t_{defrosting}$ is 2.6 and 5.2 minutes for the horizontally oriented outdoor coil, though it is 2.5 and 4.0 minutes for the vertically oriented outdoor coil. Percentagewise, it is 7.0% and 5.6% of the total operating time for the HP system with horizontally and vertically oriented outdoor coils, respectively. Second, the sum of $E_{total,supply}$ in the two continuous defrosting cycles can be calculated and divided by the total operating time of the HP system ($t_{frosting} + t_{defrosting}$): it is 17.84 and 15.14 kJ/min for horizontally and vertically oriented outdoor coil. This is due to the lower surface temperature, denser frost, and worse water drainage of the horizontally oriented outdoor coil.

In conclusion, the results show that it is more time- and energy-efficiently to apply 10 times the initial outdoor DP_{ea} as the defrost-initiation criterion than 5 times at a cost of more temperature

fluctuations of the indoor air during the defrosting cycles. However, it is possible to defrost the outdoor coil when the vehicle is parking in the parking lot or charging center and minimize the negative effects on passengers' comfort, considering the 68-minute-long operating time. Also, the vertical orientation of the outdoor coil reduces the defrosting time and energy by increasing the surface temperature of the outdoor evaporator in HP mode and improving the water drainage during the defrosting. The conclusion of this work can be applied by the researchers and engineers working on designing and optimizing automotive heat pumps.

CHAPTER 7: SUMMARY AND CONCLUSION

7.1 Summary and conclusion

To study the performance of the reversible R744 HP system with components designed and optimized for AC mode only, especially in the practical and a wide range of the operating conditions including frosting conditions, a comprehensive study was conducted to examine the component- and system-level performance using a full reversible R744 HP system and a few numerical models.

7.1.1 The performance of the reversible R744 HP system in dry conditions- the role of IHX

After conducting a throughout literature review of the role of IHX in the R744 HP system, both numerical and experimental investigation of the integrated IHX/ Acc in the reversible R744 HP system has been conducted in dry conditions. The experimental results show that

- c. In AC mode
 - The IHX AC system requires less refrigerant charge amount than the basic AC system in the most demanding condition- when the outdoor air temperature is 40°C.
 - Introducing an IHX reduces the COP-maximizing high-side pressure for the R744 AC system by 0.24 to 0.67 MPa, and this reduction is more significant when the ambient temperature is increased.
 - IHX has a negligible effect on system efficiency (COP) when the R744 AC system is operating in subcritical mode, though it improves system efficiency by up to 13.8% when the AC system is operating in transcritical mode and with higher ambient temperatures.

- d. In HP mode

- The IHX HP system requires the same refrigerant charge amount as the basic HP system in the defined condition for the charging experiment: outdoor air temperature equals 0°C, indoor air temperature equals 10°C.
- Introducing an IHX does not change the HPF-maximizing high-side pressure much, and this is likely due to the relatively fixed inlet air temperature and the lower airflow rate for the gas cooler in HP mode than they are in the AC mode.
- IHX has a negligible effect on HPF when the HP system is operating with lower outdoor air temperature T_{eai} , due to the limitation of the compressor discharge temperature and the high-side pressure cannot reach the optimal value. However, IHX can improve the HPF by up to 10.5% when the HP system is operating in transcritical mode with higher ambient temperatures.
- Introducing an IHX potentially delays frost accumulating on the outdoor evaporator due to an increased evaporating temperature, which can be higher than it is for the basic HP system by 2.3°C when the outdoor air temperature is 0°C.

Though the cycle analysis shows that the impacts of IHX on COP are always positive or negligible in a transcritical cycle, and these benefits increase with refrigerant temperature at the gas cooler exit and the effectiveness of the IHX. However, only a high-effectiveness IHX can improve the COP of the CO₂ subcritical cycle by up to 5% when the difference between the evaporating temperature and condensing temperature is large. Because the COP_{basic} decreases as the refrigerant temperature at the gas cooler exit increases, the throttling loss is increasing with the pressure and temperature differences between the low- and high-pressure sides of the CO₂ cycle. Overall, IHX benefits the COP most when COP_{basic} is low and refrigerant temperature at the gas

cooler exit is high due to the environment temperature or space limitation based on the cycle analysis.

Overall, the results of cycle analysis underestimate the COP improvement of introducing an IHX into an R744 vapor compression system, and the deviation between the theoretical and experimental results increases with the refrigerant temperature at the gas cooler exit. Experimental data show a more than doubled COP improvement due to IHX when the refrigerant temperature at the gas cooler exit for both basic and IHX systems is 41.2°C. Because simplified theoretical analysis neglects the following factors: First, the optimal high-side pressure drops more by introducing IHX in experimental results. Second, the compressor isentropic efficiency increases from 0.47 to 0.59 as the high-side pressure and compression ratio decreases in experiments, though it is isentropic compression in the cycle analysis. Third, the refrigerant distribution is improved, and the pressure drop across the evaporator decreases from 0.21 to 0.12 MPa by adding IHX, since both the inlet and exit quality of the evaporator decrease by 0.21 and 0.10. However, the pressure drop of refrigerant is considered negligible in the cycle analysis. In general, it is difficult to accurately predict the role of the IHX in the R744 HP system for a wide range of operating conditions, especially when those aforementioned factors are neglected in numerical studies.

In addition, a 1-D EES model of the integrated IHX/Acc was developed and it predicts the heat transfer rate and pressure drop of the low-pressure side within $\pm 10\%$, by considering the effects of oil circulating rate on heat transfer and pressure drop. Also, the benefits and negative effects of introducing an IHX are analyzed in typical AC/ HP operating conditions. Results show that the integrated IHX/Acc provides higher benefits in AC mode, especially in high ambient temperature conditions. Finally, the modeling results of the accumulator behavior are presented.

To provide more detailed information on the accumulator's behavior, a 3-D model of the accumulator was developed. The flow characteristics of the two-phase CO₂ and POE mixture in the accumulator in a typical HP operating condition were studied, considering the effects of different sizes of the oil bleed hole.

7.1.2 The performance of the reversible R744 HP system in frosting conditions

For AC/HP systems, one of the major concerns in a cold climate is the frosting issue, so the performance of the reversible R744 HP system in frosting conditions was studied, considering the environmental factor, the defrost-initiation criteria, and the orientation of the outdoor coil. The heating capacity, efficiency, frost accumulation in HP operation, as well as the defrosting capacity, efficiency, energy flow, and water drainage were studied comprehensively. The results show:

- Although the evaporator is flooded during the HP operation, frost grows unevenly on the outdoor evaporator due to the maldistribution of two-phase refrigerant and the evaporation of liquid refrigerant along the microchannel tubes.
- As frost accumulates on the outdoor evaporator, the cooling capacity and the quality of refrigerant at the evaporator exit drop, and liquid refrigerant migrates into the accumulator. Therefore, the effective volume of the accumulator should be optimized to hold extra liquid refrigerant in the wide operating range of a reversible HP system.
- With fixed relative humidity of ambient air, the outdoor air temperature affects the capacity and efficiency of the HP system significantly, as well as the frost accumulating speed and the frosting time. When the outdoor air temperature drops from 0 to -10°C, the peak value of heating capacity reduces from 4.85 to 3.54 kW and HPF reduces from 1.61 to 1.47 in the first cycle. In the meanwhile, the frost

- accumulating speed decreases from 0.013 to 0.005 kg/min, and the frosting time increases from 32 to 108 minutes.
- However, the relative humidity of the outdoor air has a negligible impact on the capacities and efficiency of the HP system. Though it directly affects the humidity ratio and thus the frost accumulating speed and the frosting time, which increases significantly from 32 to 84 minutes when the outdoor air temperature is 0°C and the relative humidity decreases from 90% to 80%.
 - The heating capacity and the efficiency/ HPF of the HP system drop only by less than 9% during the frosting cycles with 5 times the initial air-side pressure drop as the defrost-initiation criterion in all experimental conditions. But heating capacity and HPF drop by about 30% and 15% during the frosting cycles with 10 times the initial air-side pressure drop as the defrost-initiation criterion in all experimental conditions. Considering the operating-time-averaged performance, the 30% and 15% performance reduction can be considered mild, which indicates a longer operating time in HP mode, a lower defrosting frequency, fewer impacts on passengers' comfort, and potentially a higher periodic performance. Although the optimal defrost-initiation and termination criteria are still open to discussion and require further investigation.
 - The vertical orientation of the outdoor coil prolongs the frosting time of the HP system from 56 to 68 minutes (+18.2%) in the 0°C and 90%RH outdoor condition for the first frosting cycle using 10 times the initial air-side pressure drop as the defrost-initiation criterion. Because the vertical orientation improves the refrigerant

distribution, reduces the refrigerant-side pressure drop, and increases the evaporating as well as the surface temperature of the outdoor coil.

- Also, the outdoor coil in the vertical orientation retains only 0.298 kg of water after the second defrosting, which is 58.7% less than it does in the horizontal orientation. As a result, the defrosting time for the vertical outdoor coil reduces by 1.3 minutes. Considering the total two-hour operation, the vertical orientation reduces the defrosting time from 7.0% to 5.6%.
- The integrated total defrosting energy consumption calculated using refrigerant-side measurements $E_{OD,c,ref}$ agrees well with the integrated total energy supply $E_{total,supply}$ for all defrosting experiments (within -4% to 8%). This indicates a good energy balance between independent measurements. However, the measurement of the mass of water in the first one to two cycles and the calculation of four defrosting loads should be improved in future studies.
- The major impacts of different defrost-initiation criteria are the defrosting time, defrosting energy, and the percentage of each component out of $E_{total,supply}$. First, the percentage-wise defrosting time is 7.0% and 5.7% for using 5 and 10 times the initial outdoor DP_{ea} as the defrost-initiation criterion in the experiments. Second, the operating-time-averaged energy supply for defrosting is 20.76 kJ/min for the three defrosting cycles using 5 times the initial outdoor DP_{ea} as the defrost-initiation criterion, though it is only 15.14 kJ/min for the two continuous defrosting cycles using 10 times the initial outdoor DP_{ea} as the defrost-initiation criterion. Third, the indoor air provides 50% of $E_{total,supply}$ with 5 times the initial outdoor DP_{ea} as the

defrost-initiation criterion in the first defrosting cycle, but it is 61% for 10 times.

Consequently, the minimum indoor air exit temperature T_{eao} is 7.4 and 3.2°C for 5 and 10 times the initial outdoor DP_{ea} as the defrost-initiation criterion in the first defrosting cycle.

- The main differences between the horizontal and vertical orientation of the outdoor coil are the defrosting time and the total defrosting energy. First, the defrosting time $t_{defrosting}$ is 2.6 and 5.2 minutes for the horizontally oriented outdoor coil, though it is 2.5 and 4.0 minutes for the vertically oriented outdoor coil. Percentagewise, it is 7.0% and 5.6% of the total operating time for the HP system with horizontally and vertically oriented outdoor coils, respectively. Second, the sum of $E_{total,supply}$ in the two continuous defrosting cycles can be calculated and divided by the total operating time of the HP system ($t_{frosting} + t_{defrosting}$): it is 17.84 and 15.14 kJ/min for horizontally and vertically oriented outdoor coil. This is due to the lower surface temperature, denser frost, and worse water drainage of the horizontally oriented outdoor coil.

In conclusion, the results of the frosting/defrosting study show that it is more time- and energy-efficiently to apply 10 times the initial outdoor DP_{ea} as the defrost-initiation criterion than 5 times at a cost of more temperature fluctuations of the indoor air during the defrosting cycles. However, it is possible to defrost the outdoor coil when the vehicle is parking in the parking lot or charging center and minimize the negative effects on passengers' comfort, considering the 68-minute-long operating time. Also, the vertical orientation of the outdoor coil reduces the defrosting time and energy by increasing the surface temperature of the outdoor evaporator in HP mode and

improving the water drainage during the defrosting. The conclusion of this work can be applied by the researchers and engineers working on designing and optimizing automotive heat pumps.

7.2 Contributions

7.2.1 The contributions to the field of IHX/Acc studies unique to this dissertation are:

- a. Evaluated the performance of components that are designed and optimized for AC mode in a wide range of the HP operation both numerically and experimentally.
- b. Considered the oil effects in the 1-D and CFD model of the IHX/Acc.
- c. Firstly studied and quantified the effects of the diameter of the oil-bleed hole on the liquid level and the local pressure drop inside the accumulator using CFD methods in the open literature.

7.2.2 The contributions to the field of frosting/ defrosting studies unique to this dissertation are:

- a. Evaluated the performance of the outdoor coil which is designed and optimized for AC mode in practical and a wide range of the frosting conditions experimentally.
- b. Linked the frost accumulation on the outdoor heat exchanger with the performance drop of the system.
- c. Compared two different defrost-initiation criteria and two different orientations of the outdoor coil in terms of periodic performance of the HP system.
- d. Firstly studied and quantified the energy flow during the defrosting process of an R744 heat pump system with reasonable accuracy.
- e. Firstly studied and quantified the meltwater distribution on the outdoor microchannel heat exchanger in the open literature.

7.3 Recommended future work

In this dissertation, the CFD simulation focused only on the effects of the diameter of the oil bleed hole, but the internal diameter, the height of the accumulator, and the dimension of the J-tube can also be optimized for the separation efficiency. Also, visualization experiments of the accumulator should be conducted to validate the CFD results and examine the effects of the bubbling phenomenon of the CO₂-oil mixture.

For the frosting and defrosting experiments, the water collection methods should be improved in the future study for better accuracy in the first one to two frosting cycles. Also, the calculation of the energy consumption during the defrosting process should be improved. One improved 2-D model of the defrosting process of the outdoor heat exchanger would be very helpful. In addition, only two defrost-initiation criteria were compared in the dissertation, but the optimized value is still open to discussion and requires further investigation.

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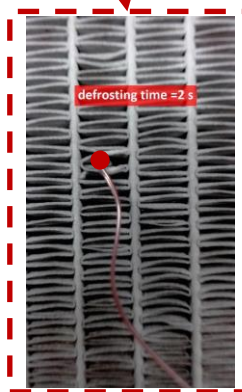
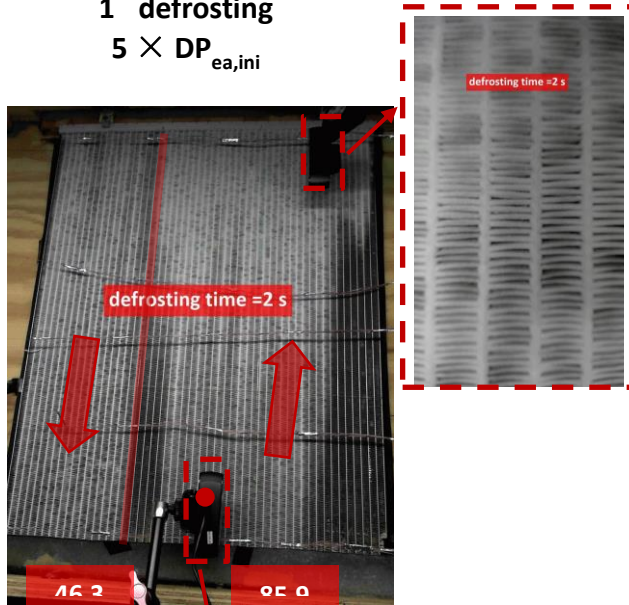
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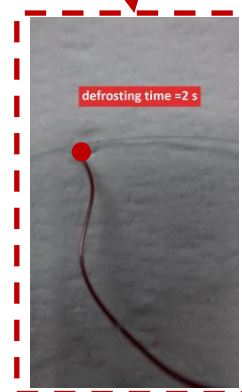
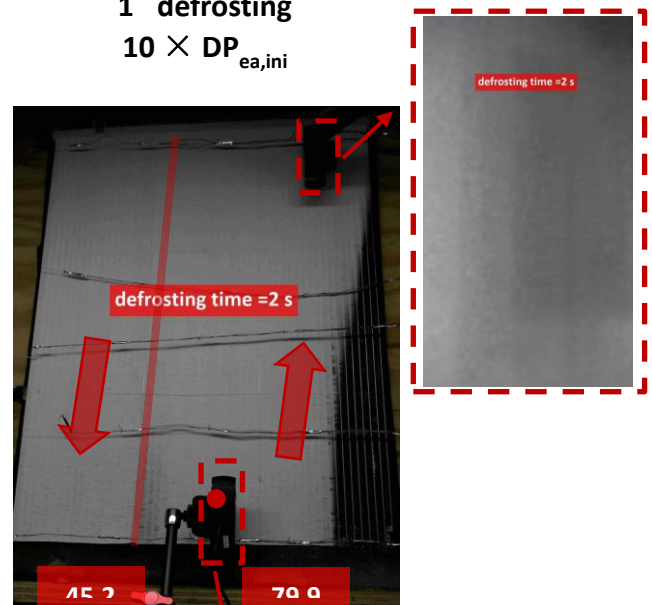
APPENDIX A: DEFROSTING IMAGES

Vertical OD coil
0°C 90%RH
1st defrosting
 $5 \times DP_{ea,ini}$



Tsurf

Vertical OD coil
0°C 90%RH
1st defrosting
 $10 \times DP_{ea,ini}$



Tsurf

**APPENDIX B: EXPERIMENTAL DATA OF THE OPERATING CONDITIONS FOR
THE 1-D MODELING OF THE INTEGRATED IHX/ ACC**

Table B.1: Operating conditions of the R744 air-conditioning system

	AC mode operating conditions			
	A1/A2/A3	B1/B2/B3	C1/C2/C3	D1/D2
Indoor air flow rate AFR_{ID} [kg/min]	6.5	6.5	9.0	9.0
Indoor air inlet temperature $T_{ID,ai}$ [°C]	15	25	35	35
Outdoor air flow rate AFR_{OD} [m/s]	1.5/2.0/3.0	1.5/2.0/3.0	1.5/2.0/3.0	1.5/4.0
Outdoor air inlet temperature $T_{OD,ai}$ [°C]	15	25	35	45
Outdoor air inlet relative humidity $RH_{OD,i}$ [%]	50%			
Reciprocating compressor A speed [%]	22.5%/ 45.0%/ 62.5%	22.5%/ 45.0%/ 62.5%	22.5%/ 45.0%/ 62.5%	22.5%/ 100%

Table B.2: Operating conditions of the R744 heat pump system

	HP operating conditions			
	E2	F1/F2	G1/G2	H1-H6 (warm-up)
Indoor air flow rate AFR_{ID} [kg/min]	6.5	3.5/6.5	3.5/6.5	3.5
Indoor air inlet temperature $T_{ID,ai}$ [°C]	0	-5	-10	-15
Outdoor air flow rate $AFR_{OD,i}$ [m/s]	3.0	3.0	3.0	3.0
Outdoor air inlet temperature $T_{OD,ai}$ [°C]	0	-5	-10	-15/-10/-5/0/10/20
Outdoor air inlet relative humidity $RH_{OD,i}$ [%]	80%			
Reciprocating compressor B speed [%]	100%			

Table B.3: Operating conditions for the integrated IHX/Acc in AC mode

Operating conditions		Mass flow rate $\dot{m}_r$ [g/s]	OCR	T_{hi} [°C]	P_{li} [kPa]	X_{li} [-]	T_{lo} [°C]	P_{lo} [kPa]	T_{hi} [°C]	P_{hi} [kPa]	T_{ho} [°C]	Q_{IHX} [kW]	Effective ness _{IHX} [-]	dP_l [kPa]
AC mode	A1	10.76	2%	2.6	3779	0.926	8.7	3760	16.5	6184	15.3	0.29	0.75	18
	A2	11.59	2%	2.6	3782	0.941	11.7	3760	19.3	5818	16.8	0.33	0.61	20
	A3	10.43	2%	2.4	3745	0.931	11.1	3740	18.5	5642	15.5	0.31	0.66	5
	B1	20.12	2%	1.6	3680	0.926	16.4	3643	28.2	8137	15.3	0.81	0.56	37
	B2	20.83	2%	1.7	3688	0.928	16.6	3648	28.3	7597	16.8	0.83	0.52	40
	B3	19.73	2%	1.5	3672	0.929	15.8	3638	27.0	7377	15.5	0.76	0.54	34
	C1	34.73	2%	10.1	4573	0.948	24.8	4518	40.1	10867	28.5	1.33	0.46	55
	C2	46.79	2%	0.4	3573	0.948	21.3	3377	41.7	10682	29.0	2.11	0.40	196
	C3	46.99	2%	-1.9	3355	0.940	18.0	3152	38.8	10756	24.3	2.12	0.43	203
	D1	29.36	2%	9.6	4503	0.942	30.8	4454	48.4	11463	36.1	1.45	0.42	49
	D2	53.33	2%	-2.0	3352	0.927	23.6	3078	47.6	10164	38.2	2.99	0.34	274

Table B.4: Operating conditions for the integrated IHX/Acc in HP mode

Operating conditions		Mass flow rate $\dot{m}_r$ [g/s]	OCR	T_{li} [°C]	P_{li} [kPa]	X_{li} [-]	T_{lo} [°C]	P_{lo} [kPa]	T_{hi} [°C]	P_{hi} [kPa]	T_{ho} [°C]	Q_{IHX} [kW]	Effectiveness _{essIHX} [-]	dP_l [kPa]
HP mode	E2	16.74	4%	-1.8	3128	0.876	3.8	2993	24.4	7408	9.2	0.72	0.63	136
	F1	11.38	4%	-6.9	2769	0.774	0.7	2667	29.9	7963	6.5	0.79	0.71	102
	F2	12.29	4%	-7.5	2697	0.879	-6.0	2587	13.0	7443	-3.0	0.46	0.80	110
	G1	13.03	4%	-11.2	2451	0.621	-11.1	2340	29.9	7575	-5.6	1.31	0.90	109
	G2	11.89	4%	-11.5	2373	0.871	-10.4	2260	9.8	7398	-8.1	0.47	0.85	113
	H1	10.51	4%	-16.3	2111	0.677	-18.2	2012	21.1	6617	-15.1	0.92	0.97	98
	H2	11.38	4%	-16.4	2132	0.588	-18.3	2036	26.7	6831	-14.6	1.26	0.97	97
	H3	10.63	4%	-16.4	2145	0.572	-18.0	2051	28.0	7085	-14.4	1.22	0.97	94
	H4	9.89	4%	-16.1	2155	0.547	-17.7	2068	29.8	7407	-14.2	1.20	0.97	87
	H5	8.15	4%	-15.8	2190	0.515	-16.7	2110	33.2	8142	-13.3	1.05	0.96	80
	H6	6.20	4%	-16.0	2215	0.426	-16.0	2142	35.3	8208	-12.6	0.95	0.96	72

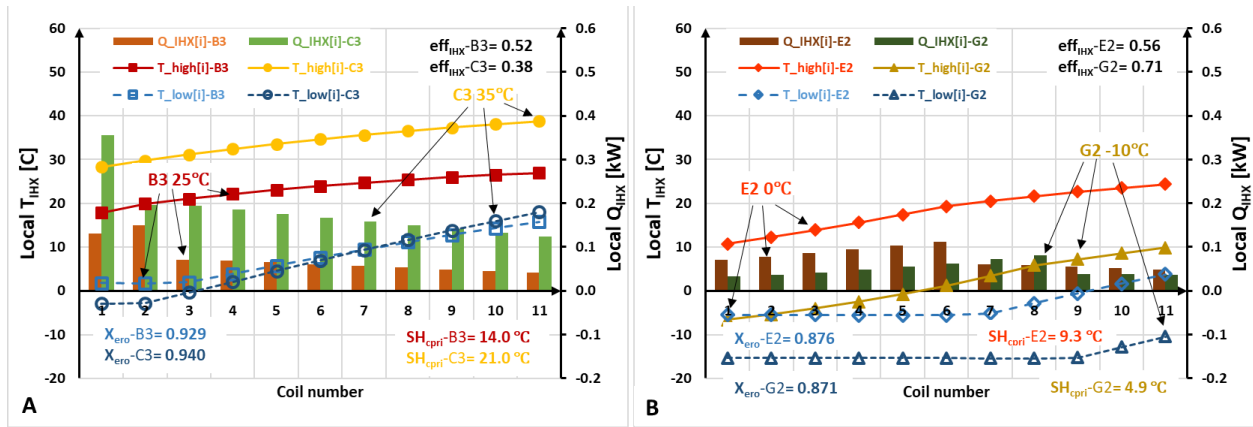


Figure B.1: Local temperatures and the heat transfer rates in each section of the IHX in **A:** AC mode and **B:** HP mode with unity mass flow rate

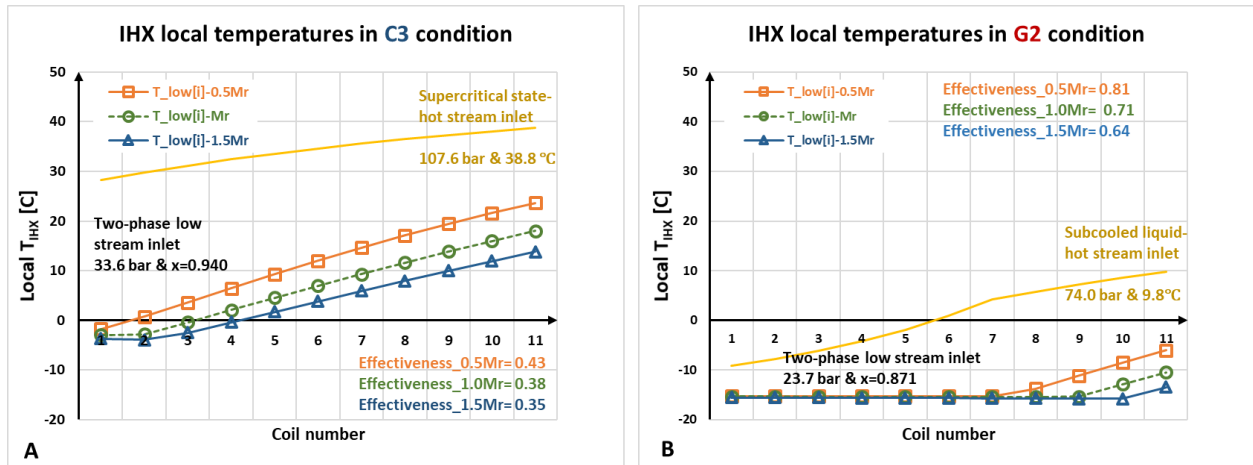


Figure B.2: Local low-pressure side temperatures in each section of the IHX in **A:** C3 condition in AC mode and **B:** G2 condition in HP mode with different relative mass flow rates