공학석사학위논문

이론해석을 통한 히트펌프 운전조건에 따른 최적 냉매 충전량 변화 예측방법

Prediction of Optimal Refrigerant Charge Variation according to Operating Conditions of Heat Pump by Theoretical Analysis

2019년 2월

서울대학교 대학원
기계항공공학부
이 강 록

이론해석을 통한 히트펌프 운전조건에 따른 최적 냉매 충전량 변화 예측방법

Prediction of Optimal Refrigerant Charge Variation according to Operating Conditions of Heat Pump by Theoretical Analysis

지도교수 김 민 수

이 논문을 공학석사 학위논문으로 제출함

2018 년 10 월

서울대학교 대학원

기계항공공학부

이 강 록

이강록의 공학석사 학위논문을 인준함

2018년 12월

위 원 장 ____________________________

부위원장 ____________________________

위　　원 ____________________________

Abstract

Prediction of Optimal Refrigerant Charge Variation according to Operating Conditions of Heat Pump by Theoretical Analysis

Lee Kang Rog

Mechanical and Aerospace Engineering

The Graduate School

Seoul National University

The proper refrigerant charge in the heat pump is important for the system operation for performance and reliability. However, it takes a lot of time and money to accurately

determine the charge amount now. This paper presents a method for predicting optimal refrigerant charge variation according to operating conditions of the heat pump by theoretical analysis. This method can be used as part of a permanently installed control or monitoring system to detect and diagnose optimal refrigerant charge according to various operating conditions. The technician can also use it as a stand-alone tool in determining the existing charge and adjusting the charge of the refrigerant.

The optimal refrigerant charge can be predicted under a wide range of operating conditions by changing variables such as system outdoor temperature, compressor speed and so on. It verifies that it can be easily implemented and installed in terms of hardware and software. In addition, comparing the actual optimum charge (by experimental value) with the predicted optimum charge (through theoretical analysis) is about 7.2%

error. Although the average error occurred comparing to experimental verification, this study was meaningful in that it had theoretical advantages.

Keywords : heat pump, optimal refrigerant charge, prediction method, theoretical analysis, charge variation

Student Identification Number : 2017-27561

Contents

List of Tables

List of Figures

Nomenclatures

A	area (m2)
Cp	specific heat (kJ/kg·K)
COP	coefficient of performance
DSC	degree of subcooling (K)
DSH	degree of superheat (K)
EEV	electronic expansion valve
h	specific enthalpy (kJ/kg)
$\dot{m}$	mass flow rate (kg/s)
P	pressure (kPa)
PD	piston displacement per second (m/s)
q	specific capacity (kJ/kg)
Qh	heating capacity (kW)
Qc	cooling capacity (kW)
T	temperature(K)
U	overall heat transfer coefficient (kW/m2·K)
V	Volume
v	specific volume (m3/kg)

W	Compressor work (kW)
ρ	density (kg/m3)
η_V	volumetric efficiency
ω	

Subscript

cond	condenser
comp	compressor
eva	evaporator
in	inlet
liq	liquid
out	outlet
ref	refrigerant side
sat	saturation
vap	vapor

Chapter 1. Introduction

1.1 Background of the study

Heating, ventilation and air conditioning (HVAC) is a technology used to comfort the indoor environment, providing thermal comfort and providing improved indoor air quality. So HVAC has increasingly used in many places as the quality of life improves. According to the building energy data book published by the US Department of Energy, HVAC accounts for 41% of total energy use during residential, building and commercial energy usage, as shown in Figure 1.1.

The heat pump is a combined cooling and heating device that absorbs heat from a low-temperature heat source using a renewable heat source and releases heat with a high-temperature heat source. Therefore, it is very energy efficient and essential for HVAC. Environmental problems such as global warming and climate change are increasing with the use of fossil fuels. In order to reduce fossil fuels, which are the cause of environmental pollution, it is important to use energy efficiently. For this reason, heat pumps are now attracting much attention in terms of energy savings, high efficiency and environmental protection.

The penetration rate of the heat pump in the United States is shown in Figure 1.2 over time. Although home heat pumps initially led the market in the form of air conditioners, commercial heat pumps are gaining popularity these days. Both residential and commercial heat pump systems continue to increase in market and maintenance of heat pumps is important because of their high penetration rate. All heat pump systems have an optimal refrigerant charge to maximize the coefficient of performance (COP). Figure 1.3 shows the COP change for the refrigerant charge of the water heat pump, and the maximum point of the COP is the optimal charge amount.(Corberán et al., 2008). The charge amount is very important because of the system performance and reliability. If the refrigerant leaks or overcharges, the COP, cooling, and heating capacity will decrease. The performance of the heat pump is remarkably reduced according to the charge amount which is not optimum. Similar results were obtained in previous studies (Corberán et al., 2011; Grill and Singh, 2017; Kim et al., 2007; Kim et al., 2014; Kim, 2017; Yoo et al., 2017).

However, the optimal refrigerant charge varies depending on various conditions of the system. According to various conditions, not only the COP but the charge amount is changed. Therefore, the optimal charge amount in a heat pump system can be determined by considering various conditions.

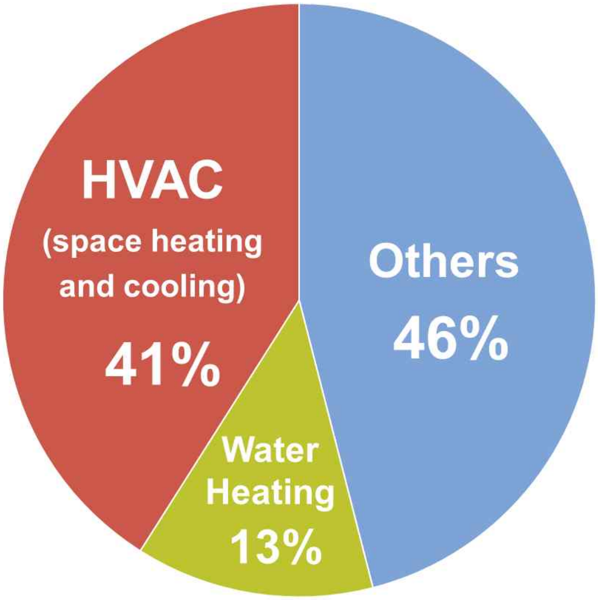


Fig. 1.1 Residential/Building/Commercial energy usage (USA, 2015)

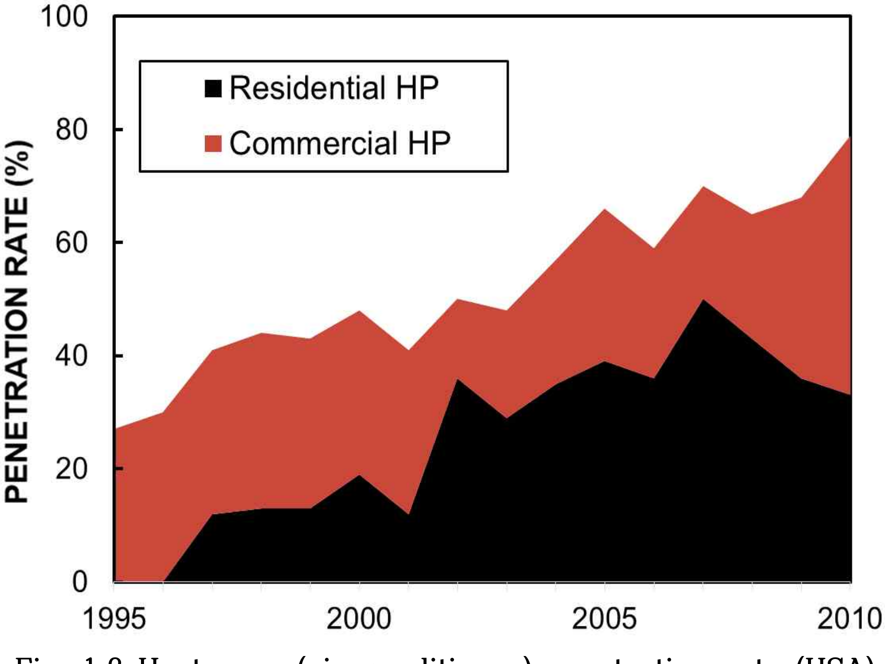


Fig. 1.2 Heat pump(air conditioner) penetration rate (USA)

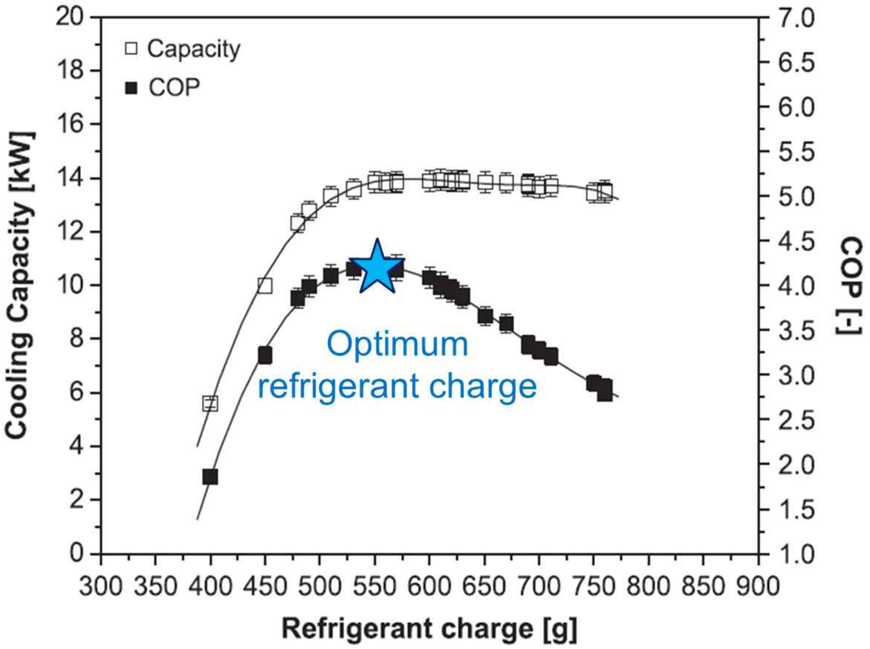


Fig. 1.3 COP and cooling capacity according to refrigerant charge (Corberán et al., 2008)

1.2 Literature survey

In order to remove various cooling loads that raise the indoor temperature, the heat pump in the cooling mode installs an evaporator in the room, and the heat of the inside is moved to the refrigerant by the evaporation of the refrigerant. The refrigerant discharged from the evaporator is compressed using a compressor to raise the temperature of the refrigerant to higher than the condensation temperature. In a condenser installed outside, the heat of the refrigerant is released to the outdoor, and the expansion device lowers the high pressure refrigerant at the outlet of the condenser to the evaporation pressure, and the system enables high efficiency operation. If the refrigerant is charged to less than the proper amount, the compressor driving current decreases but the refrigerant vapor at the inlet of the compressor is overheated and the discharge gas temperature rises to deteriorate/carbonize the oil in the compressor. On the contrary, when the refrigerant is over charged in the system, the liquid refrigerant flows into the compressor, which causes liquid back and causes the compressor to be damaged. In addition, the compressor power increases, which causes the COP to decrease. As a result, the appropriate refrigerant charge is important because it is directly linked to its system performance, stability, and durability.

The use of optimized refrigerants is an important issue in the development of future sustainable refrigeration and heat pump. The charging of the system is directly related to the emissions of high GWP materials when using synthetic refrigerants and the discharges of potentially harmful substances when natural refrigerants such as ammonia or hydrocarbons are used. On the other hand, the decrease in efficiency means an increase in power consumption and carbon dioxide emissions, so that the minimization of the charge amount cannot be realized unless the COP of the system is reduced. Therefore, to minimize CO_2 emissions, an integrated optimization of equipment design and charging is required.

Although the optimal refrigerant charge the heat pump has been determined by repeated refrigerant charging and performance testing, it would be possible to design a more sophisticated and highly efficient system if the charge quantity was determined considering the various surroundings of the system. The effect of refrigerant charge on system performance is mostly in steady-state studies. Some of the representative studies are as follows.

1.2.1 Optimal refrigerant charge

Domingorena et al. (1980) conducted an experimental study on the effect of inaccurate charge on heating performance using an air-to-air heat pump. As a result, COP and capacity degradation were small when overcharged, The COP and ability were decreased rapidly when charged. As a result, COP and ability decrease were small when overcharged, while COP and ability decreased sharply when charged less than prescribed amount. Houcek et al. (1984) conducted an experiment using a detached system of 1.5 ton size based on five outdoor temperatures

(21.1, 23.9, 26.7, 35, 37.8 ℃) and one indoor condition (26.7 ℃ DB, 19.4 ℃ WB) , and the performance of the refrigerant was evaluated with respect to the case where the refrigerant was optimal charge, over/undercharged by ±23%. Experiments were carried out only at the outdoor temperature of 27.8 ℃ and at 35 ℃, when the refrigerant was over/undercharged. When the refrigerant was undercharged, the performance at the outside temperature was decreased by 23% at 27.8 ℃ and by 38% at 35 ℃. In other words, if the refrigerant is undercharged, the performance degradation increases as the outdoor temperature increases. In the case of 23% undercharging the steady-state energy efficiency ratio (EER) is decreased by 34% at the outdoor temperature of 35 ℃. Therefore, when the refrigerants are undercharged from their optimal charge, their COP decreased significantly compared to overcharging. Farzad et al. (1991) conducted an experimental study on system performance characteristics of an air conditioner over a range of charging conditions by R-22 and capillary tube. Experiments were carried out on cooling capacity, flow rate, superheat of evaporator outlet, compressor power consumption, and seasonal energy efficiency ratio (SEER) according to charge amount.

When the refrigerant was undercharged, its performance was much lower than overcharged, and if it was under/overcharged by 10%, its cooling capacity decreased by 13.6%/3.6% respectively. The SEER decreased from 9.44 to 7.5/8.47 when 20% undercharge/overcharge respectively so it showed similar results to the above study. A special point here is that when the refrigerant is charged below the rated amount, the cooling capacity increases as the outside air temperature increases than overcharging. A possible reason for this is a change in the flow rate in the capillary depending on the refrigerant charge amount. When the refrigerant charge amount is small, the refrigerant at the inlet of the condenser is significantly overheated, and the condensation pressure rises due to the high outside air temperature, so that the pressure drop at the capillary inlet / outlet increases and the flow rate of the refrigerant increases. In Korea, Choi et al. (1998) conducted an experimental study on the effect of refrigerant charge on the performance of residential air conditioners. Experiments were carried out with increasing amount of refrigerant in the initial vacuum state for the domestic separate air conditioner, and the experiment was conducted while the amount of refrigerant was gradually decreased. As a result, when the charge amount is smaller than the rated amount, the cooling capacity, the COP and the compressor's work increase as the refrigerant charge increases, and the cooling capacity and COP decrease sharply when the refrigerant amount is larger than the rated amount. The refrigerant vapor temperature at the inlet of the compressor also decreased sharply above the rated amount. On the other hand, the pressure in the pipe does not vary greatly regardless of the charge amount, and it is not appropriate to judge the adequacy of the refrigerant charge by the pressure in the pipe when the installer further charges the refrigerant at the actual site. Therefore, it has been suggested that it is useful to judge the proper charge amount based on the variables sensitive to the amount of refrigerant such as the temperature difference of the compressor / evaporator. Coberán et al. (2007) conducted a study on the optimal charge amount using a water-to-water heat pump using propane as a refrigerant. They analyzed the effect of condensation pressure, supercooling, system capacity and COP through the refrigerant charge.

However, previous studies tend to depend on experimental information to find various variables for optimal COP. Because of the limit of experimental data amount, there are low accuracy in the actual field. In addition, experimental data is different for each system so experiments must be performed according to each system. Finally, it is necessary to monitor the charge amount because the operating condition is continuously changed even while the heat pump is operating.

1.3 Objectives and scopes

The optimum refrigerant charge varies depending on various operating conditions. In this study, it is important to find the optimal refrigerant charge considering various operating conditions. The purpose of this study is to predict the change of the optimal refrigerant charge according to operating conditions of the heat pump by theoretical analysis. Experiments are conducted to propose and validate these methods.

In chapter 2, theoretical analysis of the optimal refrigerant charge is presented. Before the theoretical analysis, the basic theory is explained, and the optimal charge variation according to the change of the outdoor condition and the compressor speed is introduced.

In chapter 3, the theoretical analysis of the optimal charge amount is verified by the experimental results. The experimental equipment and conditions were described and the results were verified.

In the last chapter, a brief summary of the study and conclusion are given.

Chap. 2 Theoretical Analysis of the Optimal Refrigerant Charge

2.1 Introduction

This chapter treats the theoretical analysis for the prediction of optimal refrigerant charge in heat pump system. From the previous research papers, there are variable factors that affect the optimal charge, such as temperature, pipe, compressor speed, heat exchanger, expansion valve, and so on. However, that researches depend on the experimental data and one variables for optimal COP. In this study, various variables use for optimal charge amount by theoretical analysis.

2.2 Concept of the prediction method

Figure 2.1 is the basic p-h diagram. Generally, there are DSH (degree of superheat) and DSC (degree of subcool) for compressor and expansion valve. In this diagram COP can be expressed as followed :

$$\mathrm{COP} = \frac{Q_L}{W} \tag{2.1}$$

The idea of predicting the optimal refrigerant charge amount usually starts by considering a very small increase in the condensation pressure, which is achieved by slightly charging the refrigerant. As the condensation pressure rises, compression work and cooling capacity for the specific amount of refrigerant mass flow is also increased (Figure 2.2). In this situation, the ratio of the expected specific cooling capacity increment to the expected specific compression work increment determines whether COP will increase or decrease. The simultaneous equations of the equation and inequality (2.4) show that COP increases if the ratio is greater than the COP of the current system. If the pressure is increased by ΔPh due to the increase in outside temperature as shown in Figure 2.2, the existing COP changes to the new COP.

$$\text{COP} = \frac{Q_L}{W} \ \text{(not optimal)} \tag{2.2}$$

$$\text{COP} \xrightarrow{\pm\, charge\, amount} \text{COP}_{\text{new}} = \frac{Q_L + \Delta Q_L}{W + \Delta W} \ \text{(optimal)} \tag{2.3}$$

$$\therefore \ \text{COP}_{\text{new}} > \text{COP when } \frac{\Delta Q_L}{\Delta W} > \frac{Q}{W} \tag{2.4}$$

To apply the above inequality, the current COP, expected compression work increment (△W) and cooling capacity increment ($\triangle Q$) should be calculated based on the measured data. Power meters, temperature sensors, pressure sensors and mass flow meters are available. The actual 'increased compression' and 'increased cooling capacity' are determined by the complex interactions between operating conditions, system specifications and components. Simulation programs with many iterative loops are currently in use, but we will make predictions through theoretical analysis for a wider application. For the actual use of the prediction method, appropriate assumptions must be made to make the calculation process simpler and easier. Assumptions are as follows.

(1) The mass flow rate ($\dot{m}$) will remain at its present value in same operating conditions.

(2) If the condenser capacity is large sufficient, the refrigerant temperature at the condenser outlet is constant.

Assumption (1) is related to the compressor flow rate (2.5). The compressor volume and speed are fixed values and efficiency is normally constant. Since the compressor inlet pressure is fixed on the assumption, the compressor inlet density is also unchanged.

$$\dot{m}_{comp} = \rho_{in} V_{disp} \omega_{comp} \eta_{V} \tag{2.5}$$

Assumption (2) requires many constraints that conflict with the actual behavior of the system. However, this assumption is necessary to implement the prediction method and does not reflect the actual situation, but the prediction performance is quite acceptable. In addition, it reflects reality and other situations.

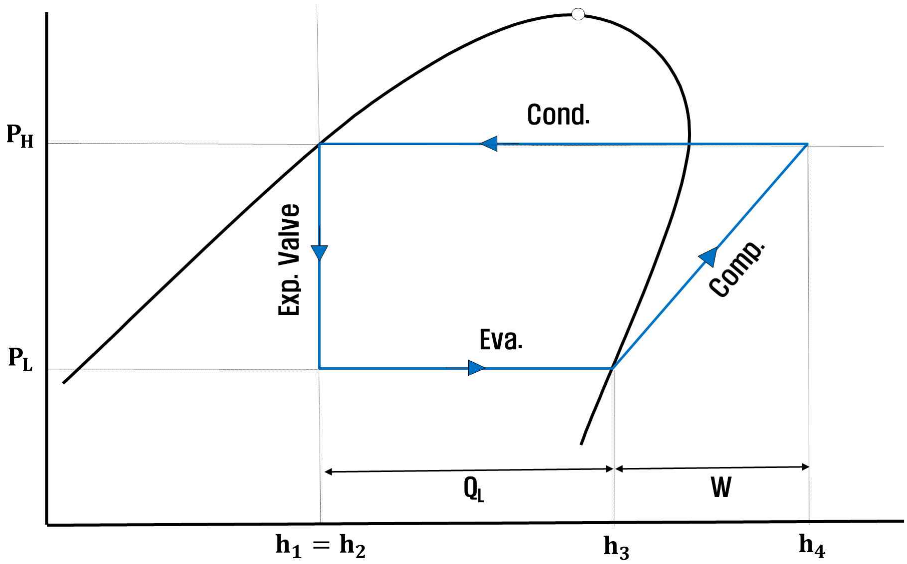


Fig. 2.1 Basic p-h diagram

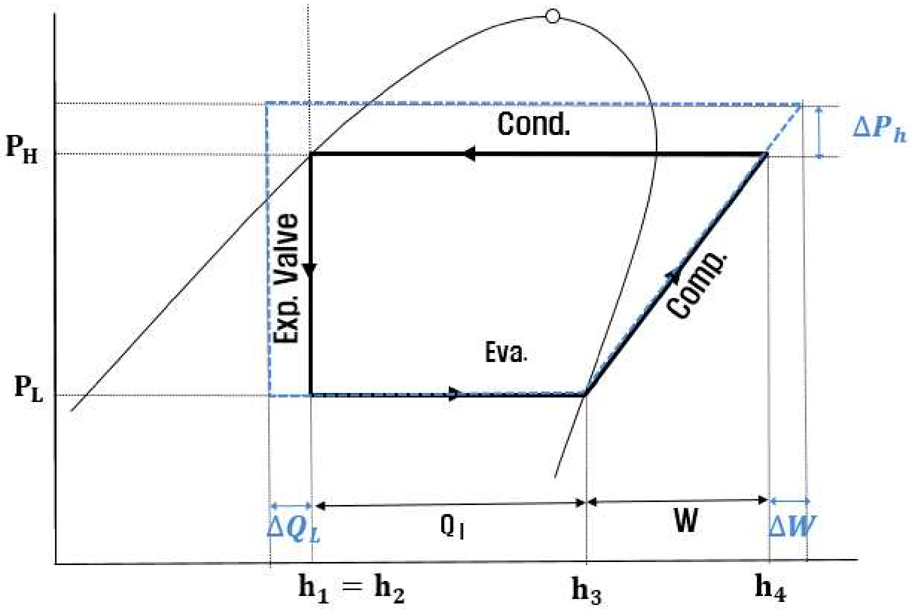


Fig. 2.2 COP according to the outdoor conditions

2.3 Optimal charge amount prediction method

Figure 2.3 is a basic diagram showing the temperature change according to condenser length. The superheated refrigerant from the compressor exits the condenser after entering the supercooled section through two-phase state. When the refrigerant is further charged in the present state, the amount of the liquid refrigerant in the condenser is increased, so that the refrigerant is liquefied quickly and more heat exchange occurs. This results in a higher temperature difference between the outside temperature and the condenser resulting in a higher condensation pressure (Figure 2.4). However, when the refrigerant charge is insufficient, the refrigerant exits the two-phase state from the condenser before entering the subcooled state (Figure

2.5). In this case, the influence of the two-phase state is applied rather than the normal state DSC. Looking at the change in the cooling capacity ($\triangle Q_{subcool}$) according to the DSC variation ($\triangle DSC$), when the DSC is insufficient and falls below 0, the $\triangle Q_{subcool}$ decreases sharply. On the contrary, it is gradual increase in the section of 0 or more (Figure 2.6).

$$Q_{total} = Q_{superheat} + Q_{two-phase} + Q_{subcool} \tag{2.6}$$

- $Q_{superheat} = \dot{m} C_{p,v} \Delta T$ (2.7)

  $\rightarrow \ \Delta Q_{superheat} = \dot{m} C_{p,v} \Delta \mathrm{DSH}$

- $Q_{two-phase} = \dot{m} h_{fg}$ (2.8)

  $\rightarrow \Delta Q_{superheat} = \Delta \dot{m} h_{fg}$

- $Q_{subcool,0} = \Delta T_0 \dot{m} C_p$ (2.9)

  $Q_{subcool} = (\Delta T_o + x) \dot{m} C_p = Q_{subcool,0} + \Delta \mathrm{DSC}\dot{\mathrm{m}}\mathrm{C_p}$

  $(where, x = \Delta \mathrm{DSC})$

  $\rightarrow \Delta Q_{subcool} = \dot{m} C_{p,l} \Delta DSC$

$\triangle Q_{subcool}$ is affected by the 2-phase section in not enough DSC, so the slope of $Q_{subcool}$ is obtained by applying the proportional expression of Eq. (2.10). Therefore, $\triangle Q_{subcool}$ is as shown in Eq. (2.11), and the overall $\triangle Q_{total}$ of the condenser is shown in Eq. (2.12).

$$h_{fg}\dot{m} : \triangle T_{cond} = Q_{subcool} : \triangle T \tag{2.10}$$
$$Q_{subcool} = h_{fg}\dot{m}\frac{\triangle T}{\triangle T_{cond}}$$
$$Slope\,of\,Q_{subcool} = \frac{\dot{m}h_{fg}}{\triangle T_{cond}}$$

$$\triangle Q_{subcool} \tag{2.11}$$
$$= \left(\frac{\dot{m}h_{fg}}{\triangle T_{cond}}\left(1-\frac{\mathrm{DSC}}{\triangle T_{cond}}\right)+k\frac{\mathrm{DSC}}{\triangle T_{cond}}\right)(\triangle T_2 - \triangle T_1)$$
$$\left(where,\, k = \dot{m}C_{p,l},\, \triangle\mathrm{DSC} = \frac{\mathrm{Q}-\mathrm{Q}_0}{\mathrm{k}}\right)$$

$$\Delta Q = \dot{m}C_{p,v}\Delta\mathrm{DSH} + \Delta\dot{\mathrm{m}}\mathrm{h}_{\mathrm{fg}} + \tag{2.12}$$
$$\left(\frac{\dot{m}h_{fg}}{\Delta T_{cond}}\left(1-\frac{\mathrm{DSC}}{\Delta T_{cond}}\right)+\mathrm{k}\frac{\mathrm{DSC}}{\Delta \mathrm{T}_{\mathrm{cond}}}\right)(\Delta \mathrm{T}_2 - \Delta \mathrm{T}_1)$$
$$\left(where,\, k = \dot{m}C_{p,l},\, \Delta\mathrm{DSC} = \frac{\mathrm{Q}-\mathrm{Q}_{\mathrm{o}}}{\mathrm{k}}\right)$$

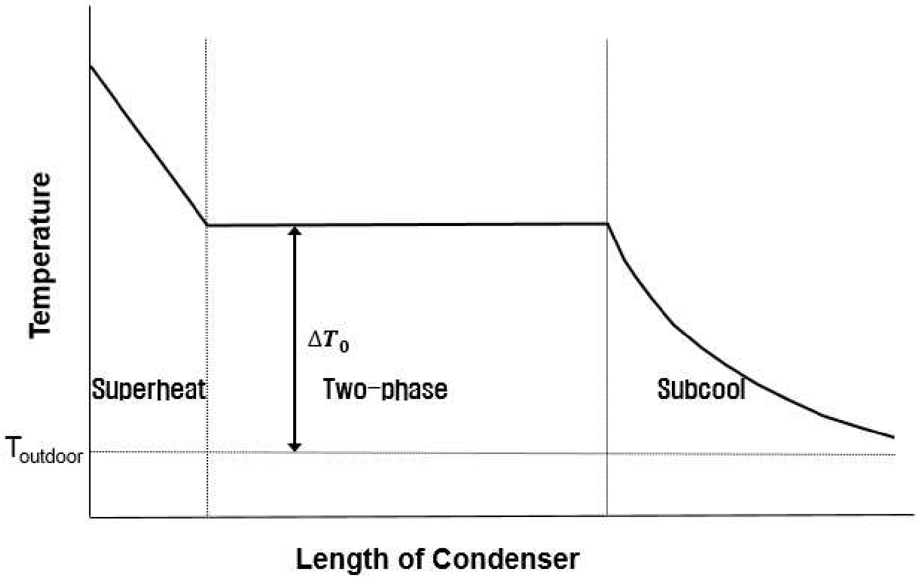


Fig. 2.3 Temperature according to length of condenser with different phase

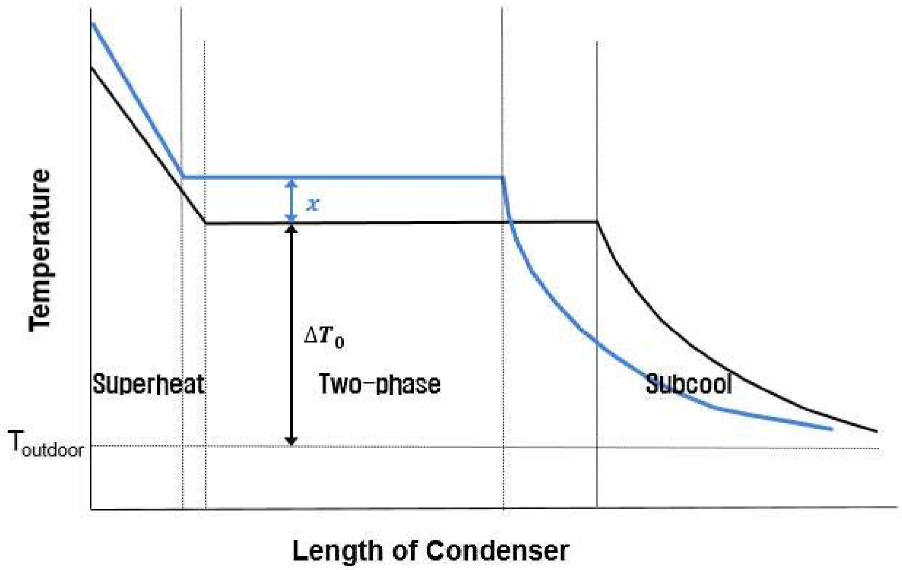


Fig. 2.4 Temperature according to length of condenser with different phase (charge refrigerant)

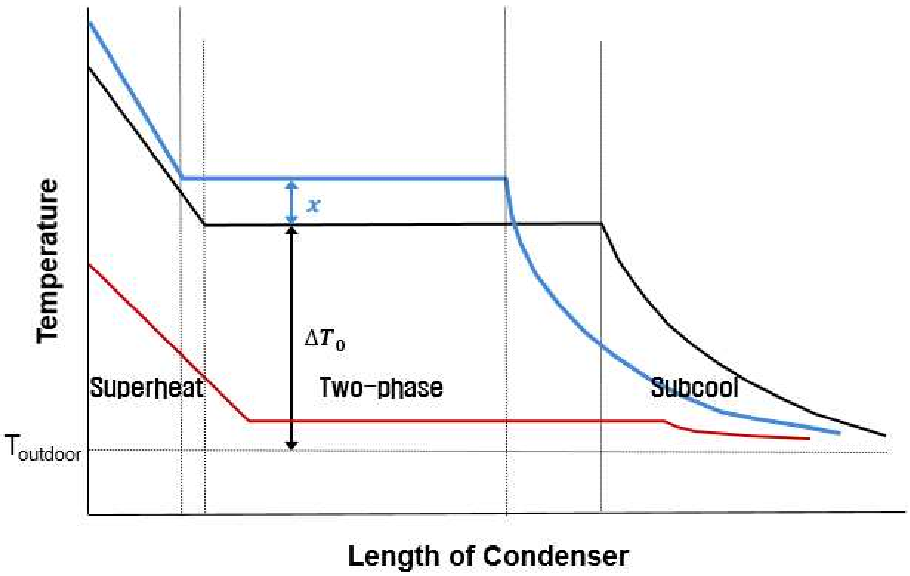


Fig. 2.5 Temperature according to length of condenser with different phase (insufficient DSC)

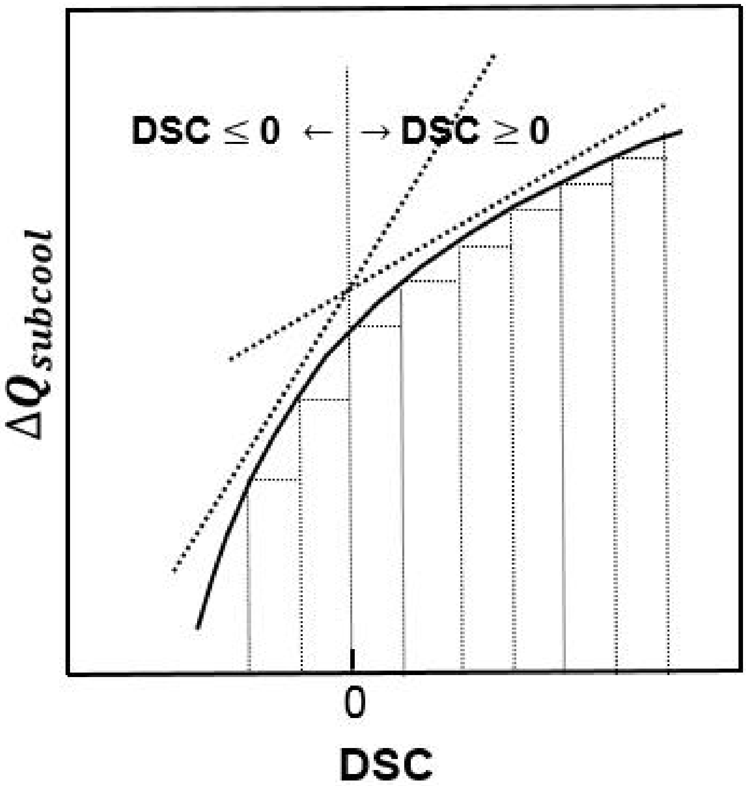


Fig. 2.6 $\triangle Q_{subcool}$ according to DSC

Chap. 3 Experimental Verification

3.1 Introduction

This chapter deals with empirical verification of the results of previous theoretical formulas. From the previous chapter, it was found that the optimal charge amount changes in accordance with the refrigerant charge change. The equipment used for the experiment is a commercial heat pump, which has a lot of sensors rather than a residential heat pump. Previous researchers on refrigerant leakage have conducted research using near superheat and subcooling. The most accurate method to date is Kim and Braun (2013). They showed the utility of a hypothetical refrigerant charge sensor using DSH and DSC, and the overall root mean square error (RMS) was 3.8%. However, this operation is limited to systems with thermal expansion valves or fixed orifices. Also, without any tuned parameters, the RMS error increases to 13.9%.

The main purpose of this study is to make an equation that predicts the optimal refrigerant charge according to various variables and to verify it by experiment. Therefore, the prediction analysis and the experimental results are compared by applying various variables in changing amounts of the refrigerant.

3.2 Experimental method

3.2.1 Experimental equipment

One of the commercial heat pump systems with a nominal capacity of 30 kW was chosen for the experiment. Figure 3.1 shows the images of the indoor and outdoor units of the selected heat pump. Two indoor units and one outdoor unit were installed to accommodate the capacity between the evaporator and the condenser. Similar to residential heat pumps, there are scroll-type cylinders and variable speed compressors with EEVs (eletronic expansion valve). Figure 3.2 is a photograph of an environmental chamber for the experiment, and Figure 3.3 shows a schematic of the experimental setup. The outdoor unit has an accumulator, compressor, oil separator and condenser. The evaporator and EEV are in the indoor unit. In the experiment, the heat pump system has a relatively large number of sensors compared to residential. The temperature sensors are installed at the indoor and outdoor air sides, the accumulator inlet, the compressor outlet, the condenser inlet and outlet, the liquid line and the inlet and outlet of each evaporator. Also, the pressure at the accumulator inlet and at the compressor outlet is measured, and the sensor installed is marked with a black circle. Since the flow meter is more expensive than the pressure and temperature sensors, it is installed separately and marked with a gray circle. The specifications of the sensor are summarized in Table 3.1.

(a) Indoor unit

(b) Outdoor unit

Fig. 3.1 Commercial air heat pump (target model)

Fig. 3.2 Environmental chamber

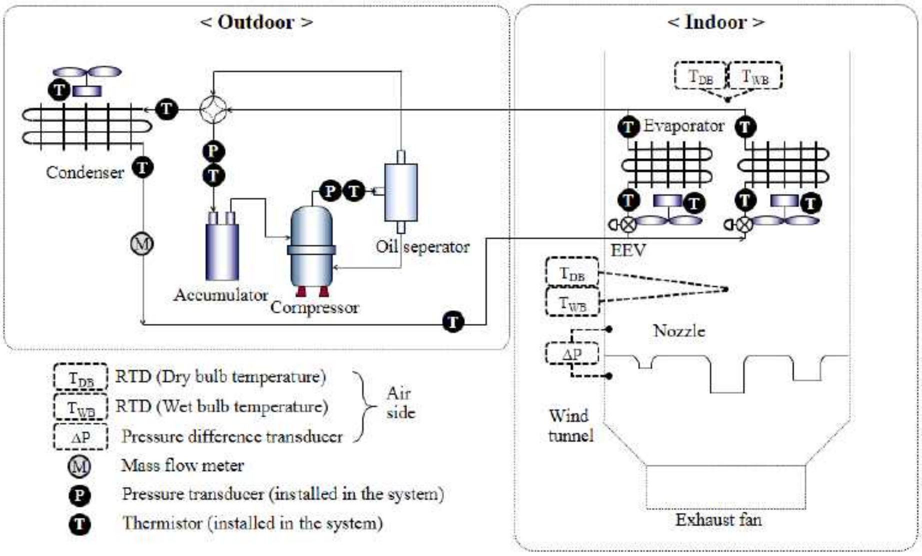


Fig. 3.3 Schematic of experiment setup

Table 3.1 Specifications of measurement instruments

Model Flow Meter (Refrigernat side)	
Model	Oval, CN015C-SS-200K
Type	Coriolis
Range	0~200 g/s
Accuracy	±0.1% F.S.
Thermistor(installed in the system - Refrigerant side)	
Model	MITSUBISHI Materaials, ACD-45
Range	-30 ~ 130℃
Accuracy	±0.7℃
Resistance Temperature Detector (Air side)	
Model	Netsushin, NR-351(PT 100Ω)
Range	-200 ~ 250℃
Accuracy	(0.15+0.002|T|)℃

Table 3.1 Specifications of measurement instruments (continued)

Absolute pressure transducer (Refrigerant side)	
Model	KELLER PA-21Y
Range	0~5000 kPa, 0~6000 kPa
Accuracy	±0.25% F.S.
Differential pressure transducer (Air side)	
Model	Yokogawa, EJA-110A
Range	0~100 mmAq
Accuracy	±0.2% F.S.
Power meter	
Model	Yokogawa, WT1600
Range	0~12,000 W
Accuracy	±0.1% of reading +0.05% of range

3.2.2 Experimental conditions

The information and experimental conditions for each component are shown in Table 3.2. The refrigerant charge ranged from 6 kg to 11 kg by 1 kg unit. DSH was fixed at 5, 10, and 15 K for various conditions. Four different conditions were selected and applied according to ANSI / AHRI Standard 1230 (2010). The compressor speed was operating at 60 Hz or 100 Hz. Data was monitored with Labview and recorded every 2 seconds when steady-state was reached.

Table 3.2 Specifications of each component

Part	Specifications	Value
Comopressor	Displacement voloume ($cm^{3)}$	62.1
	Sump diameter / height (mm)	200 / 20
Accumulator	Diameter / height (mm)	150 / 1100
Oil Separator	Diameter / height (mm)	100 / 250
Condenser	Width / length / height (mm)	1650 / 60/ 1220
	Tube inner / outer diameter (mm)	6.78 / 7.34
	Tube transverse / longitudinal pitch (mm)	20 / 21
	Fin pitch (mm)	1.7
	Flow path row / step	3 / 58
Evaporator	Width / length / height (mm)	3000 / 40 / 200
	Tube inner / outer diameter (mm)	6.78 / 7.34
	Tube transverse / longitudinal pitch (mm)	20 / 20
	Fin pitch (mm)	1.7
	Flow distribution row / step	2 / 10
EEV	Maximum area : $A_{EEV,max}$ (mm^2)	10

Table 3.3 Calculation and experiment conditions for cooling mode (Commercial heat pump)

Variables	Value			
Refrigerant	R410a			
Refrigerant charge amount (kg)	6 ~ 11 (△=1)			
Lubricant	Polyvinyl ether(PVE) type			
Lubricant charge amount (kg)	3			
DSH (K)	5, 10, 15 (or maximum EEV opening)			
Air condition (ANSI/AHRI 1230)	Rating	Maximum	Minimum	Condensate
ID inlet DB (℃)	26.7	26.7	19.4	26.7
ID inlet WB (℃)	19.4	19.4	13.9	23.9
OD inlet DB (℃)	35.0	46.1	19.4	26.7
OD inlet WB (℃)	23.9	23.9	13.9	23.9
Compressor speed (Hz)	60, 100	60, 100	60	60, 100

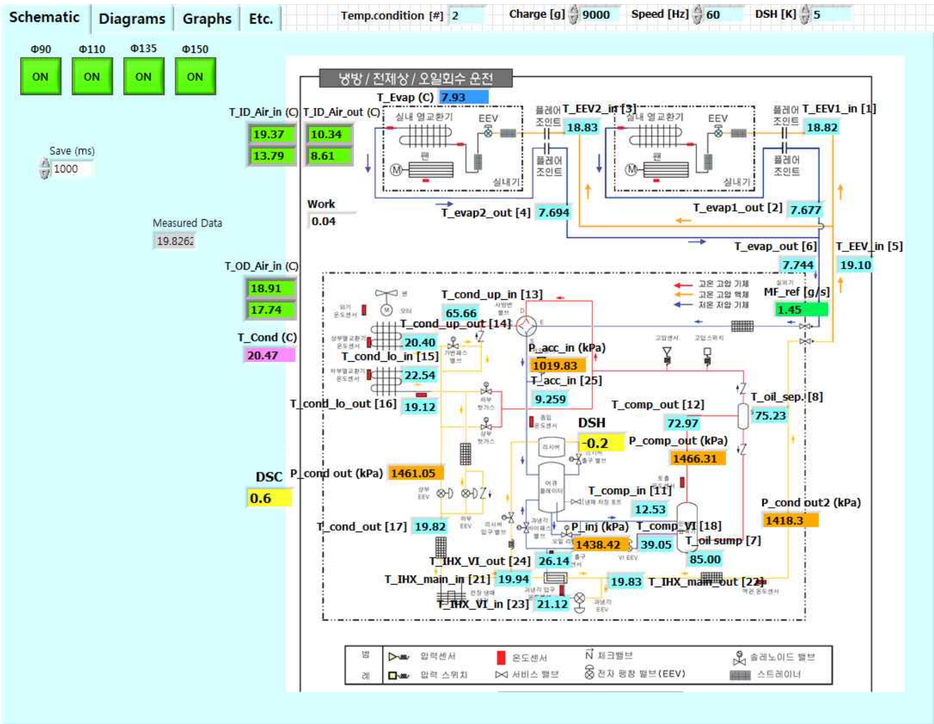


Fig. 3.4 Real time estimation of variables in LABVIEW program

Fig. 3.5 Control panel of LABVIEW program

3.3 Experimental result

3.3.1 Effect of various conditions on cycle performance

The experiment was carried out using these variables: outside temperature, compressor speed, and DSH. In Figure 3.6, 3.7, 3.8 the COP changes according to the outdoor temperature. The inverter frequency is fixed at 60 / 100 Hz, and the test results show that the higher the outdoor temperature, the lower the COP as the DSH is changed by 5, 10, and 15 degrees.

The maximum COP charge amount was 8 to 10kg, and the refrigerant undercharged section had a small amount of refrigerant flow, and the cooling capacity was not influenced by the change of outdoor air temperature.

However, the refrigerant overcharged area has a large change in the cooling capacity due to the change in the flow rate due to the pressure difference between the inlet and outlet of the expansion valve according to the change in the outdoor temperature. Figure 3.9, 3.10, 3.11 is the experimental results according to the outside temperature of the compressor at 100 Hz. It can be seen also the COP decreases as the outside temperature increases. From the 9 kg of refrigerant under the third temperature condition, the data could not be further tested due to the compressor discharge temperature high. As a result, it can be seen that as the compressor speed increases, the overall COP decreases. As the outdoor temperature increases, there is no change in the evaporator pressure, so the refrigerant mass flow rate is constant but the compression ratio is increased, which means that the consumption power of the compressor increases and the COP decreases. As the DSH increases, the compressor efficiency and the refrigerant circulation amount decrease due to the rise of the compressor discharge temperature, and the COP decreases. When the compressor speed increases (Figure 3.12), the COP decreases as the compressor discharge pressure rises (Figure 3.13.).

The optimum refrigerant amount was found to be 9 kg in most outdoor conditions, except for the section with superheat of 5 K and the low temperature section (19 ˚C / 19 ˚C). However, there are the result of increasing the refrigerant charge by 1 kg, Therefore it is a need to further subdivide. The maximum COP point and the under / over charged ± 1 kg interval were set and the maximum point was predicted assuming the graph as a quadratic function. As a result, it was found that the optimal charge amount could be predicted by 100 g unit. The condition of No. 3 at 100 Hz does not predict the optimal refrigerant charge because the number of samples is not sufficient (Table 3.4).

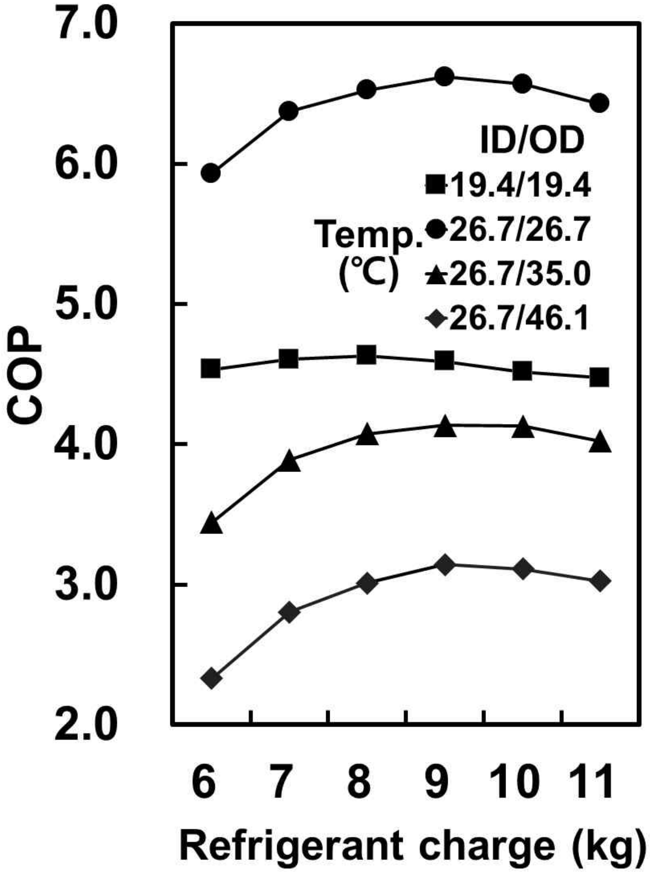


Fig. 3.6 Experimental results at 60 Hz (DSH 5 K)

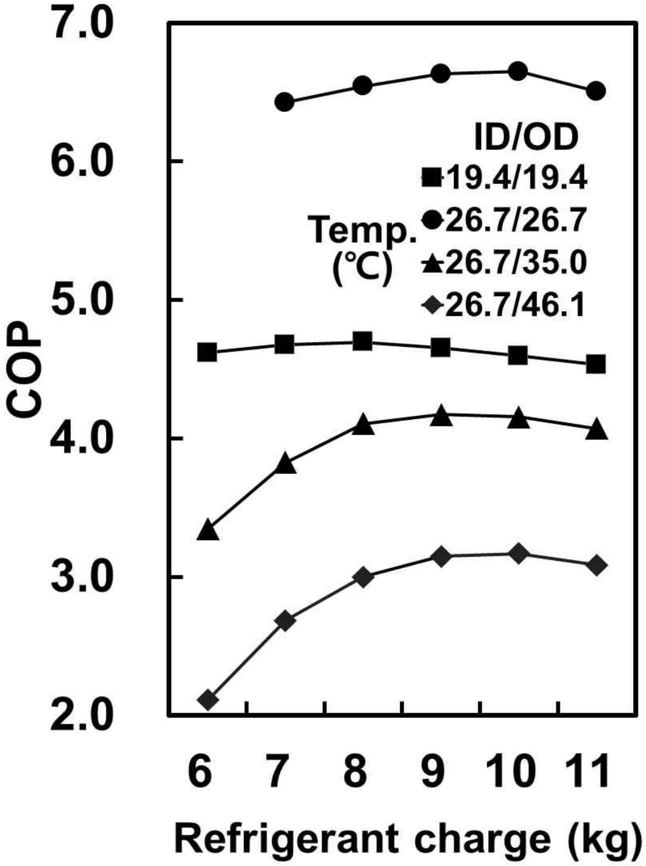


Fig. 3.7 Experimental results at 60 Hz (DSH 10 K)

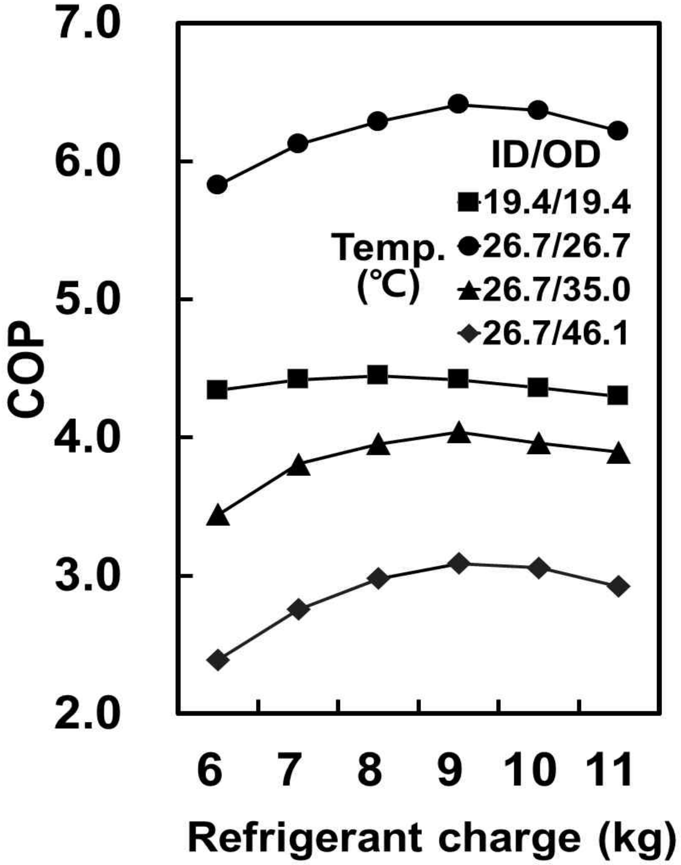


Fig. 3.8 Experimental results at 60 Hz (DSH 15 K)

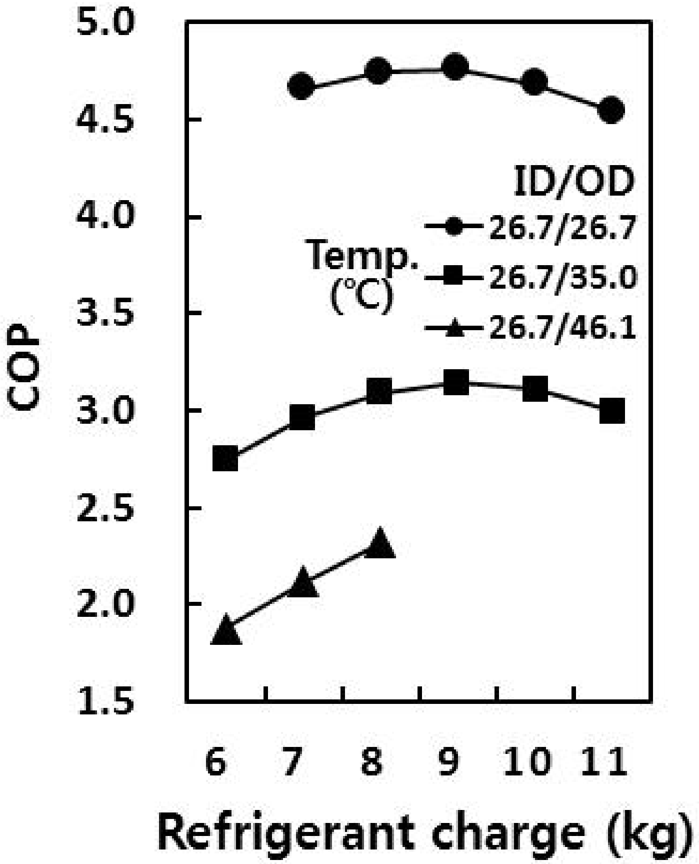


Fig. 3.9 Experimental results at 100 Hz (DSH 5 K)

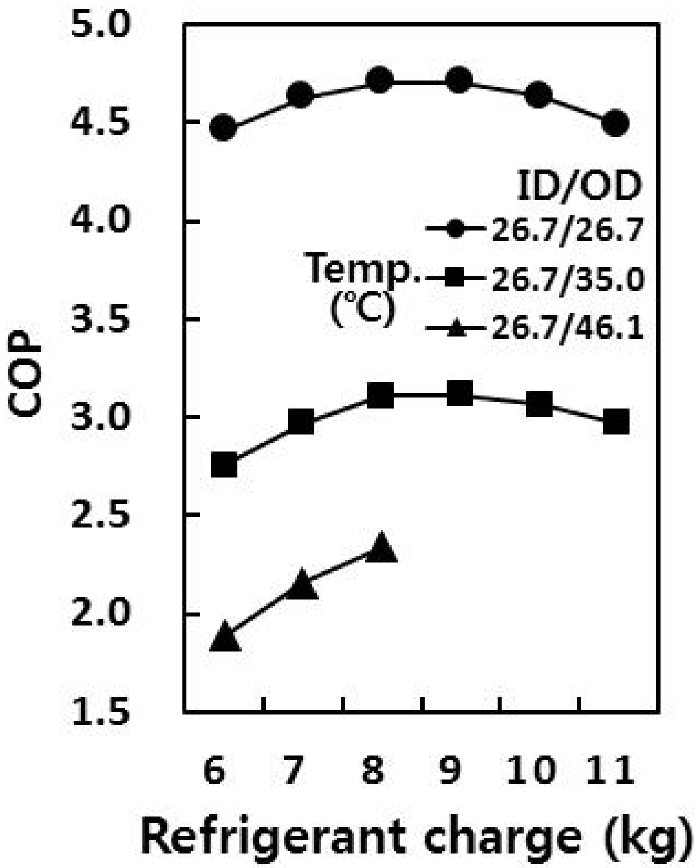


Fig. 3.10 Experimental results at 100 Hz (DSH 10 K)

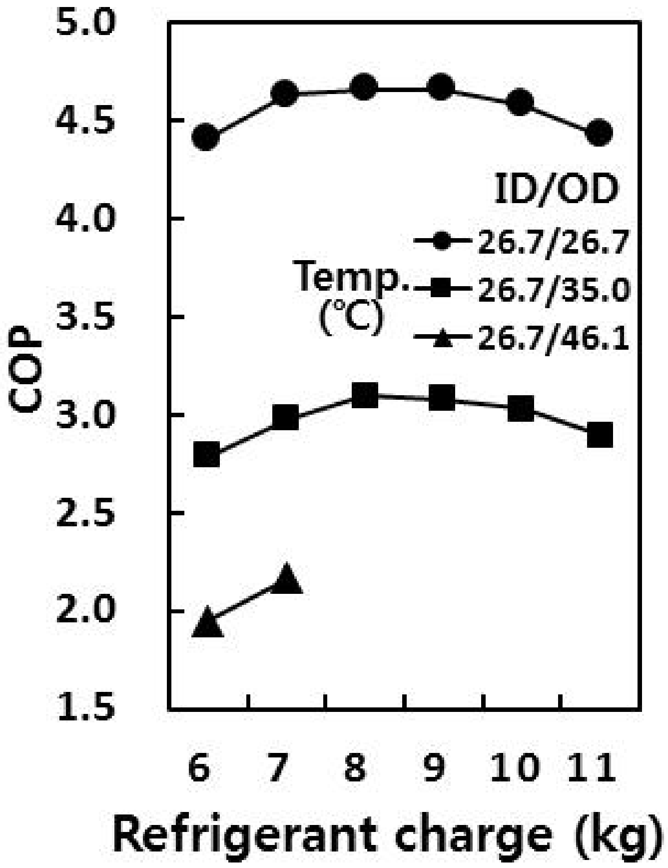


Fig. 3.11 Experimental results at 100 Hz (DSH 15 K)

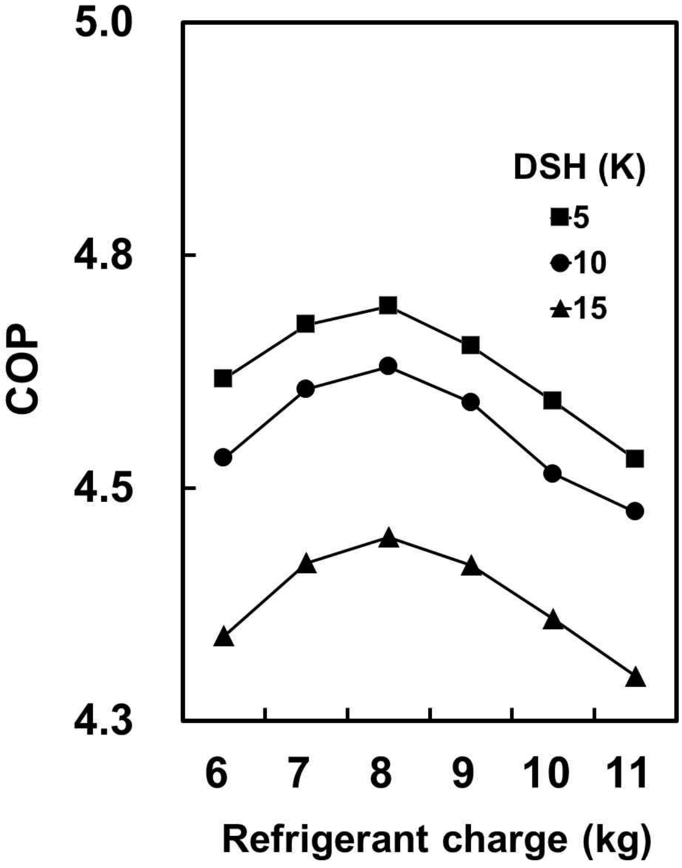


Fig. 3.12 Experimental results at diverse DSH

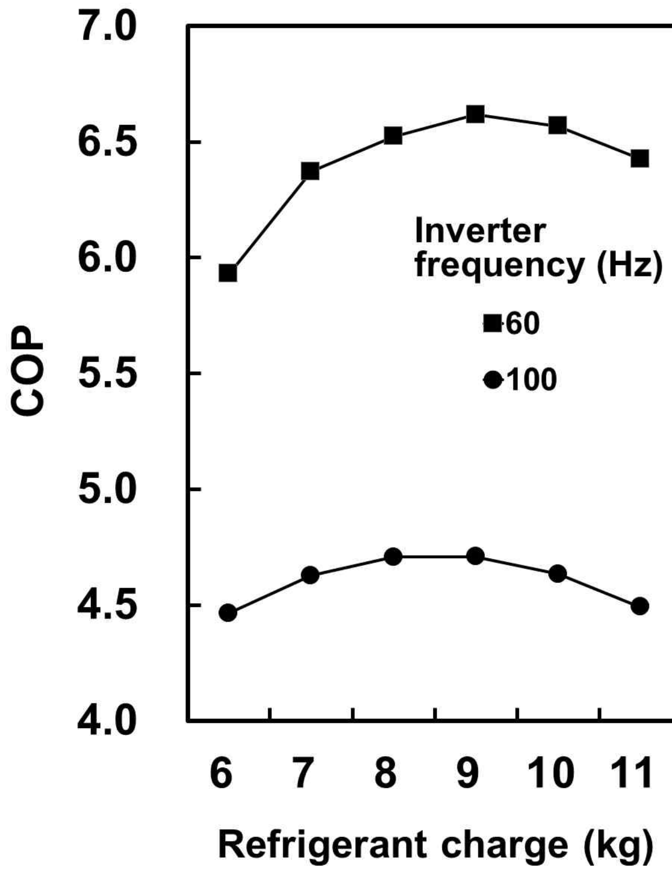


Fig. 3.13 Experimental results at diverse inverter frequency

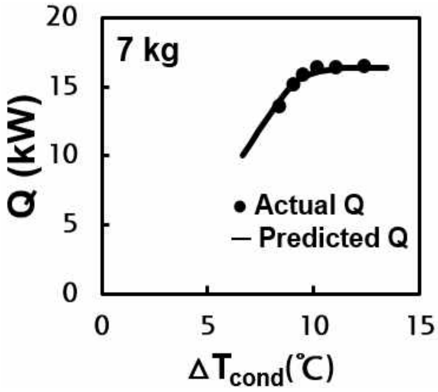
Table 3.4 Variation of optimal refrigerant charge

Comp. inverter frequency	DSH	ID temp.	OD temp.	Optimal refrigerant charge
60 Hz	5 K	26.7°C	26.7 °C	9.6 kg
		26.7°C	35.0 °C	9.3 kg
		26.7°C	46.1 °C	9.7 kg
	10 K	26.7 °C	26.7 °C	9.2 kg
		26.7 °C	35.0 °C	9.4 kg
		26.7 °C	46.1 °C	9.3 kg
	15 K	26.7 °C	26.7 °C	9.2 kg
		26.7 °C	35.0 °C	9.0 kg
		26.7 °C	46.1 °C	9.3 kg
100 Hz	5 K	26.7 °C	26.7 °C	8.6 kg
		26.7 °C	35.0 °C	9.1 kg
		26.7 °C	46.1 °C	8.5 kg
	10 K	26.7 °C	26.7 °C	8.6 kg
		26.7 °C	35.0 °C	8.5 kg
		26.7 °C	46.1 °C	8.4 kg

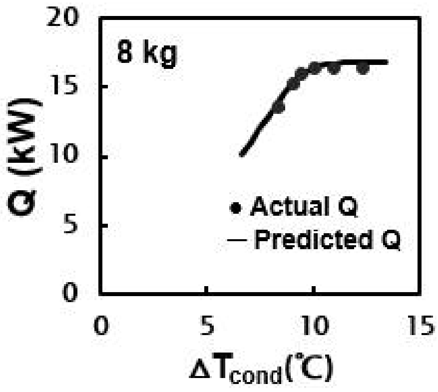
3.3.2 Comparison of theoretical analysis and experimental results

Figure 3.14 compares the $\triangle Q$ (cooling capacity change) analysis of Chapter 2 and the experimental results of Chapter 3. As a result of comparing the △Q according to the change of the condenser temperature, the actual Q and the predicted Q are substantially equal to each other. For example, each dot is the actual value by the experiment. 7, 8, 9, 10. 11 kg. Figure 3.14 (a) is the expected cooling capacity variation when 7 kg of refrigerant is injected. From the following figure (b), (c), and (d), the refrigerant is gradually increased by 1 kg. Figure 3.15 compares the cooling capacity according to the charge amount based on △Q verification. The initial point was 6kg and then increased by 1kg to 11kg. 6 kg was excluded from comparison with the expected cooling capacity because the amount of refrigerant was too insufficient to obtain an objective indicator from the condenser in a two-phase state.

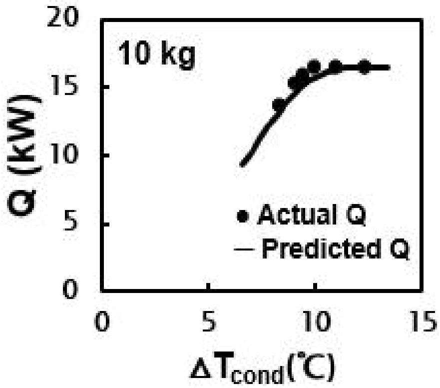
Figure 3.16 presents the variation of compressor work according to refrigerant charge. The predicted compressor work for the 6 kg point was also excluded as cooling capacity. It can be confirmed that the actual work and the predicted work are substantially identical. In Figure 3.17, the COP is obtained from the compressor work (W) and cooling capacity (Q) as shown in Eq. (2.2). Comparing the actual optimum charge (by experimental value) with the predicted optimum charge (through theoretical analysis) is about 7.2% error.



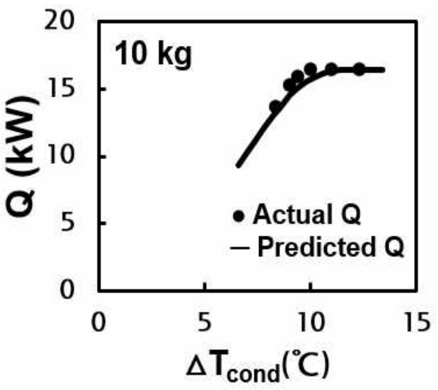
(a)



(b)



(c)



(d)

Fig. 3.14 Actual and predicted Q (cooling capacity) according to $\triangle T_{cond}$

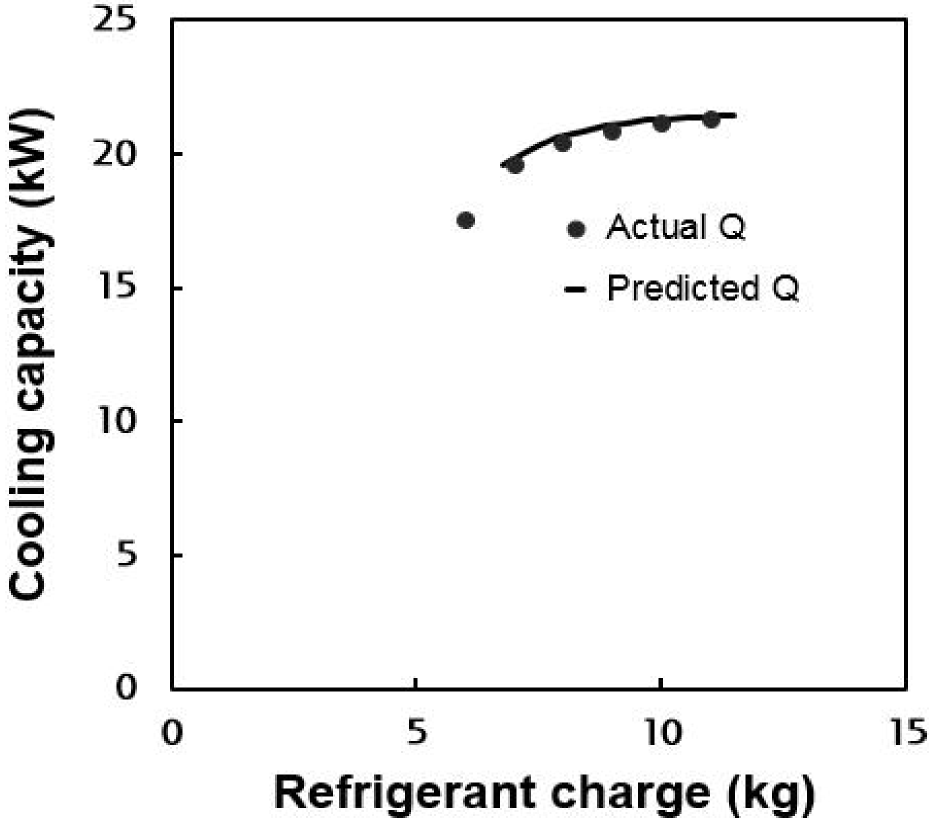


Fig. 3.15 Actual and predicted Q (cooling capacity) according to charge amount

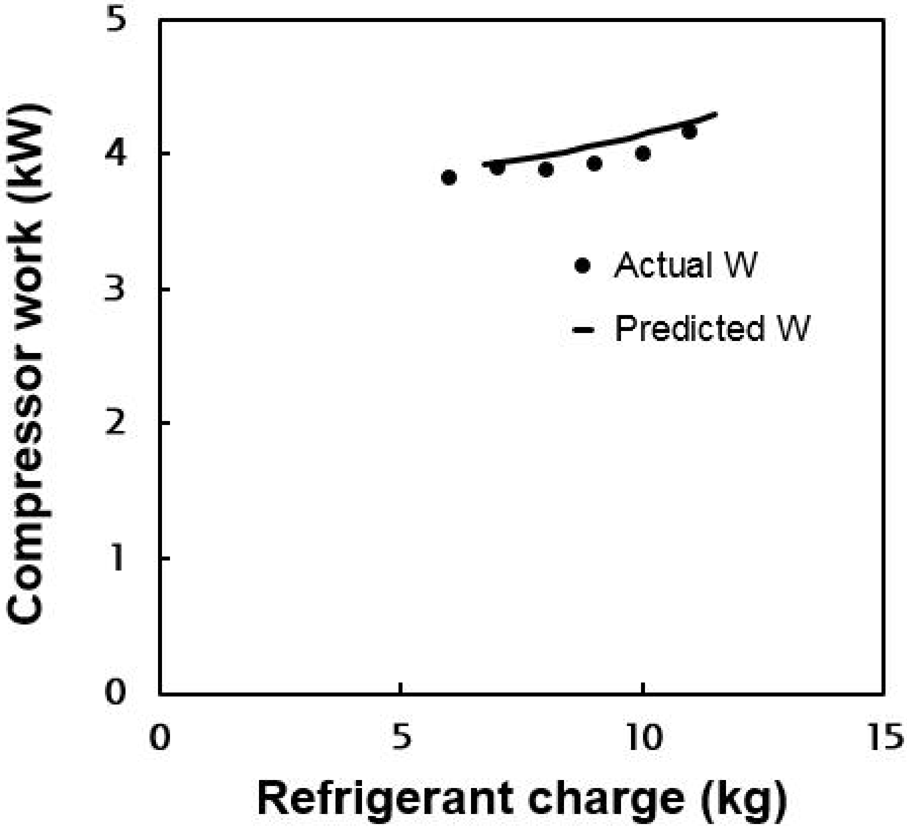


Fig. 3.16 Actual and predicted W (compressor work) according to charge amount

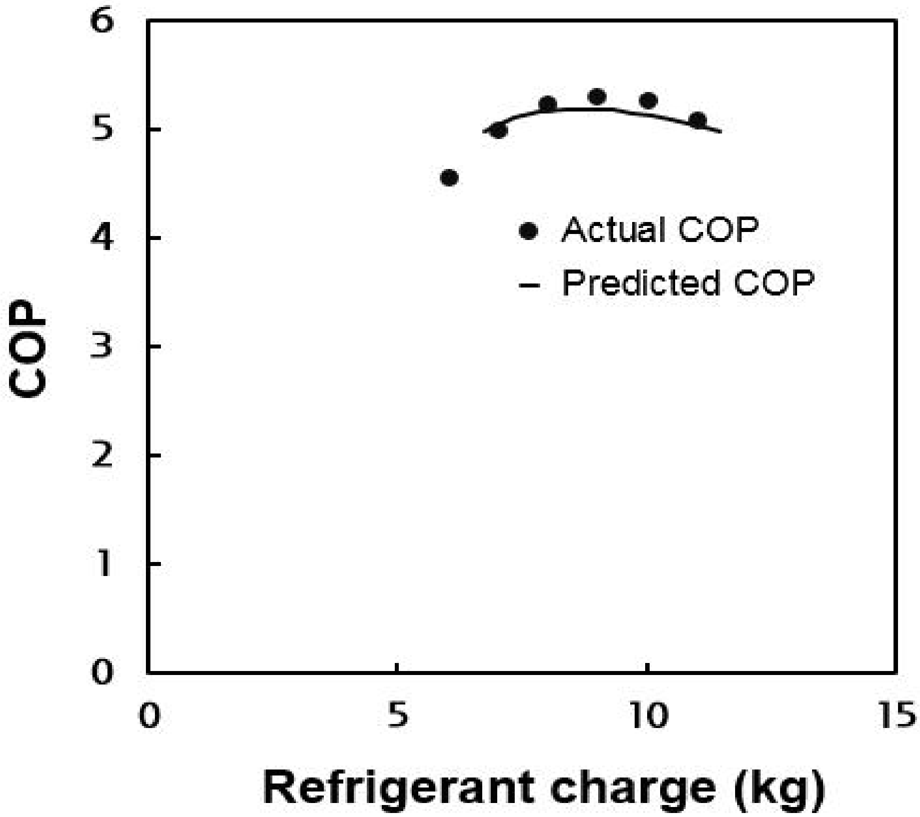


Fig. 3.17 Actual and predicted COP (coefficient of performance) according to charge amount

3.3.3 Suggestion for applying Theoretical Analysis

The average error is 7.2%, which is expected to be due to the assumption that $\dot{m}$ remain its present value in the same operating conditions. In actual situations, as the charge amount varies, the evaporator pressure also changes. Therefore considering that, it will have a lower error.

3.4 Summary

In this study, we analyzed quantitatively the optimal COP changes according to various operating conditions and refrigerant charge in a heat pump system and tried to confirm the necessity of controlling the optimal charge amount accordingly. The results showed that the optimal charge amount exists depending on the outdoor temperature, compressor speed and superheat degree.

In addition, the optimal charge amount by experimental results was predicted precisely through analyzing of tendency with the maximum COP center. While the refrigerant charge has changed by up to 6.6% for outdoor temperature and superheat, compressor speed is shown that with respect to up to 12.4%, the compressor inverter frequency has a significant factor compared to other conditions.

In the prediction of the optimal refrigerant charge variation according to the refrigerant charge amount, the changing of cooling capacity, compressor work, and COP is coincided with actuality and prediction. In addition, comparing the actual optimum charge (by experimental value) with the predicted optimum charge (through theoretical analysis) is about 7.2% error.

Therefore, in order to maximize the efficiency of the heat pump, an efficient control algorithm of the refrigerant charge is required according to various operating conditions.

Chapter 4. Conclusions

The energy use through heating, ventilation, and air conditioning (HVAC) takes 41% of residential, building and commercial in the United States. Heat pumps are the essential of HVAC systems, and maintenance is also important because the penetration rate is increasing. All heat pump systems have an optimal refrigerant charge to maximize the coefficient of performance (COP). The charge amount is very important because of the system performance and reliability. If the refrigerant leaks or overcharges, the COP, cooling, and heating capacity will decrease. The performance of the heat pump is remarkably reduced according to the charge amount which is not optimum. However, the optimal refrigerant charge varies depending on various conditions of the system. According to various conditions, not only the COP but the charge amount is changed. Therefore, the optimal charge amount in a heat pump system can be determined by considering various conditions.

In chapter 2, theoretical analysis of the optimal refrigerant charge is presented using theoretical analysis. Two hypotheses have been used for theoretical prediction and an equation has been derived to predict the optimal refrigerant charge in the condenser. In addition, DSC changes were considered when there was not enough refrigerant.

Chapter 3 deals with empirical verification of the results of previous theoretical formulas. In conventional cooling mode, it was validated that the results of theoretical analysis was not much difference with experimental results. Although the prediction in low refrigerant charge section was different with the result of the experiment, the overall results presented a similar tendency compared with the theoretical analysis. By the results of the theoretical prediction, the optimal refrigerant charge could be anticipated ahead of experiment.

Through this study, prediction of optimal refrigerant charge variation is suggested by theoretical analysis. This study is very useful because it is possible to predict the optimal charge even under various heat pump operating conditions.

References

ANSI/AMCA 210, 2007. Laboratory methods of testing fans for certified aerodynamic performance rating. Air Movement and Control Association International, Inc., Arlington, IL, U.S.A.

ANSI/AHRI Standard 1230, 2010. Performance Rating of Variable Refrigerant Flow (VRF) Multi-Split Air-Conditioning and Heat Pump Equipment. Arlington, VA, U.S.A.

ASHRAE, 2009. ASHRAE Handbook Fundamentals, Chapter 1. Psychrometrics, Atlanta, GA, U.S.A.

ASHRAE Guideline 2, 2010. Engineering analysis of experimental data. ASHRAE, Atlanta, GA, U.S.A.

Buildings Energy Data Book, 2011. Department Of Energy : https://openei.org/doe-opendata/dataset/buildings-energy-data-book; 2017

Choi, Jongmin, and Yongchan Kim. 2004. 'Influence of the expansion device on the performance of a heat pump using R407C under a range of charging conditions', International Journal of Refrigeration, 27: 378-84.

Choi, J. M., and Y. C. Kim. 2002. 'The effects of improper refrigerant charge on the performance of a heat pump with an electronic expansion valve and capillary tube', Energy, 27: 391-404.

Domingorena, AA, and SJ Ball. 1980. "Performance evaluation of a selected three-ton air-to-air heat pump in the heating mode." In.: Oak Ridge National Lab., TN (USA).

Energy, U.S. Department of. 2012. '2011 Buildings Energy Data Book'.

Eun Soo, Choi, In Gyu, Park. 1998. 'Effects of charged mass of refrigerant on the performance of a home air-conditioner', The Society Of Air-Conditioning And Refrigerating Engineers Of Korea. : 204-208.

Farzad, Mohsen, and Dennis L O'neal. 1991. 'System performance characteristics of an air conditioner over a range of charging conditions', International Journal of Refrigeration, 14: 321-28.

Houcek, John, and Marvin Thedford. 1984. 'A research into a new method of refrigeration charging and the effects of improper charging'.

Kang Rog, Lee, Hong Sung Bin, and Kim Min Soo. 2018. "A Study on the Change of Optimal Refrigerant Charge According to Cooling Operating Conditions." In, 13-16. The Society Of Air-Conditioning And Refrigerating Engineers Of Korea.

Kim, Mo Se, Dae Hoon Kang, Min Soo Kim, and Minsung Kim.

2017. 'Investigation on the optimal control of gas cooler pressure for a CO2 refrigeration system with an internal heat exchanger', International Journal of Refrigeration, 77: 48-59.

Kim, Mo Se, Chang Soo Shin, and Min Soo Kim. 2014. 'A study on the real time optimal control method for heat rejection pressure of a CO2 refrigeration system with an internal heat exchanger', International Journal of Refrigeration, 48: 87-99.

Sung Bin, Hong, Yoo Jin Woo, Lee Kang Rog, and Kim Min Soo. 2017. "A Method for Estimating the Amount of Refrigerant Charge in a VRF Heat Pump System Using Sensor Information." In, 25-28. The Society Of Air-Conditioning And Refrigerating Engineers Of Korea.

Zhang, Ze, Xiaojun Dong, Zheng Ren, Tianwei Lai, and Yu Hou. 2017. 'Influence of Refrigerant Charge Amount and EEV Opening on the Performance of a Transcritical CO2 Heat Pump Water Heater', Energies, 10: 1521.

Yoo Jin Woo. 2018. 'Study on the Detection Method of Refrigerant Leakage Amount in Air Heat Pump System', Seoul National University

국문초록

히트 펌프에서 적절한 냉매의 충전은 시스템이 효율적이고 안정적으로 작동하는데 중요하다. 그러나 충전량을 정확하게 판단하기 위해 현재는 매우 많은 시간과 비용이 소요된다. 이 논문은 이론 해석을 통해 히트펌프의 운전조건에 따른 최적 냉매 충전량 변화 예측방법을 제시한다. 이 방법은 여러 가지 운전조건에 따른 최적 냉매 충전량을 감지하고 진단하기 위해 영구적으로 설치된 제어 또는 모니터링 시스템의 일부로 사용될 수 있다. 또한 기술자가 기존 충전량을 결정하고 냉매 충전을 조정하는 과정에서 독립된 도구(stand-alone tool)로 사용할 수도 있다.

최적 냉매 충전은 시스템의 외기조건, 과열도, 압축기 속도 등 다양한 변수들을 복합적으로 변화시켜 광범위한 작동 조건에서 예측이 가능하다. 실험 데이터와 비교 검증하여 하드웨어와

소프트 웨어면에서 쉽게 구현되고 설치 될 수 있음을 검증한다. 실험을 통한 최적 충전량과 이론 해석을 통해 예측된 최적 충전량을 비교한 결과 약 7.2%의 오차를 보여 유용한 의미가 있다.