A Study on the Capacity Control of a Variable Speed Vapor Compression System Using Superheat Information at Compressor Discharge

Doo Soo Yang
Hyundai Motor Company & Kia Motors Corporation

Gil Bong Lee
Seoul National University

Min Soo Kim
Seoul National University

Young Man Cho
Seoul National University

Yoon Jei Hwang
LG Electronics Co.

See next page for additional authors

Authors
Doo Soo Yang, Gil Bong Lee, Min Soo Kim, Young Man Cho, Yoon Jei Hwang, and Baik Young Chung

A STUDY ON THE CAPACITY CONTROL OF A VARIABLE SPEED VAPOR COMPRESSION SYSTEM USING SUPERHEAT INFORMATION AT COMPRESSOR DISCHARGE

Dong Soo YANG[1], Gilbong LEE[2], Min Soo KIM[2]†, Young Man CHO[2], Yoon Jei HWANG[3], and Baik Young CHUNG[3]

[1] Advanced Packaging Development Team, Research & Development Division for Hyundai Motor Company & Kia Motors Corporation, 772-1, Jangduk-Dong, Whasung-Si, Gyunggi-do, 445-706, Korea

[2] School of Mechanical and Aerospace Engineering, Seoul National University, Seoul 151-742, Korea

[3] Air Conditioning Group, Digital Appliance Research Laboratory, 327-23, Seoul 153-802, Korea

† Corresponding author. Tel.: +82-2-880-8362; Fax.: +82-2-883-0179
E-mail address: minskim@snu.ac.kr

ABSTRACT

The capacity control of a variable speed vapor compression system using superheat information at the compressor discharge was investigated. The conventional control method of a vapor compression system is normally based on the superheat information at the compressor suction. However, the superheat at the compressor suction cannot be used as a control variable at over-duty operating conditions where the superheat at the suction becomes zero or negative. In this paper, the physical relationship between the superheat at the compressor discharge and the system variables such as inverter frequency, expansion valve opening and temperatures was devised. The results show that the superheat at the discharge was highly dependent on the cooling or heating capacity at optimal operating conditions. In order to verify that the superheat at the compressor discharge is reliable as a control variable, step changes of inverter output frequency and expansion valve opening were imposed to the system to investigate the dynamic response. During the step change operation, the superheat at the compressor discharge was stable while the superheat at the compressor suction was not always reliable.
As a control experiment, classical PID control method was used, and it was proved that the capacity control of a variable speed vapor compression system was successful in case when the outdoor temperature and the indoor setting temperature were varied.

1. INTRODUCTION

Various thermal systems developed so far have made human life more comfortable and convenient than ever. Among them, heat pump systems have been widely used for its outstanding performance. A heat pump is a thermal system that carries heat from a heat source at a low temperature to a high temperature heat sink. In general, the amount of heat carried by a heat pump can be several times larger than the electric power consumption in the compressor.
However, one of the weak points of a heat pump system is that the performance is dependent on the environmental conditions. Generally heat pump uses air as a heat source, and in this case, heat pump shows reasonable performance when the winter temperature is maintained above zero degree Celsius (Nishimura, 2002). But in the northeast Asian countries like South Korea, the winter temperature drops below the zero degree. When the temperature falls, the capacity of the heat pump system decreases while the demand of heating increases. Therefore, to cover the demand a

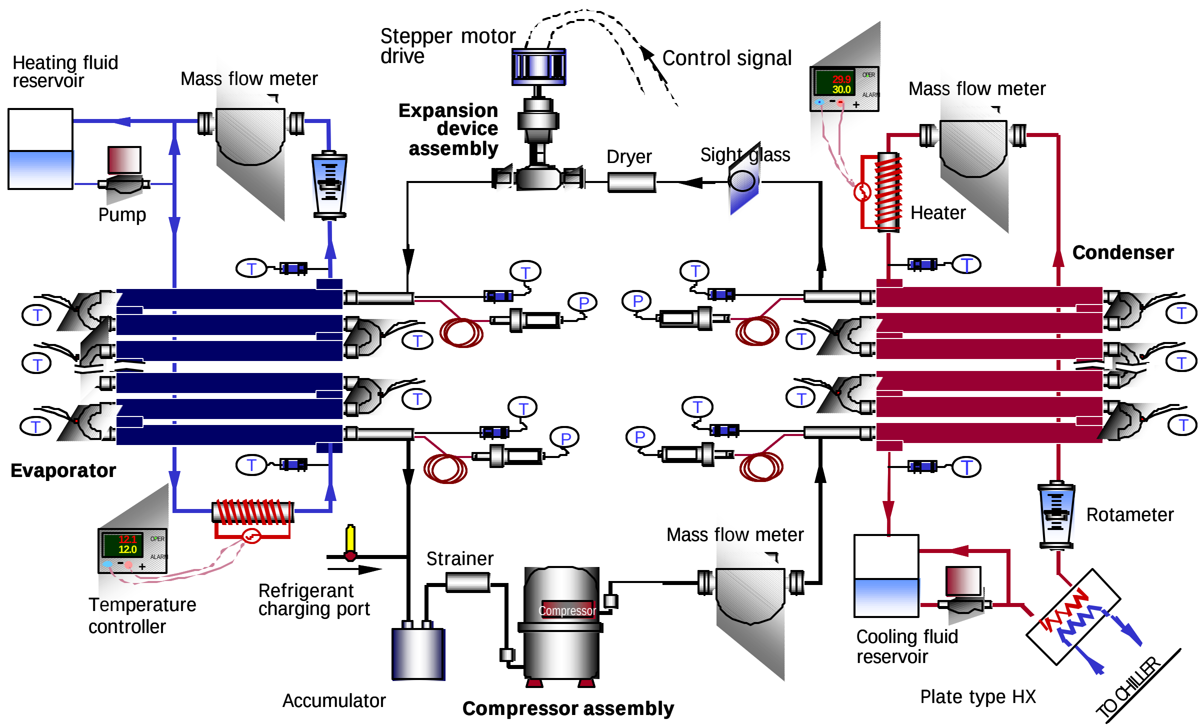


Figure 1: Schematic diagram of experimental apparatus

large compressor is required. However, on the other hand, there will be a surplus of capacity in normal operation condition with a large compressor. Therefore, it is required that the capacity of a heat pump system be modulated by an adequate control under various climate conditions.

For a capacity control, variable speed compressor driven by an inverter is usually used instead of a constant speed compressor since a variable-speed refrigeration system can provide higher seasonal efficiencies than a fixed-speed refrigeration systems (Tassou *et al*., 1998). Also, electronic expansion valve is installed instead of a capillary tube because this valve can actively control pressure drop or flow rate of refrigerant (Choi *et al*., 2001). Conventionally the superheat at the compressor suction is used as a control variable. At the very high heating demand, the inverter frequency increases to its maximum, and one possible change is negative superheat at compressor suction. This phenomenon makes it very hard to use suction superheat as a control variable. In addition, the constant value of the superheat at the compressor suction may cause the decrease in efficiency. Also, the nonlinearity of the suction superheat on the expansion valve opening makes it unsuitable to be used as a control variable (Chia *et al*., 1997).

In this study, superheat information at compressor discharge was selected as a control variable, and other system performance parameters such as heating capacity and COP are investigated in relation with this superheat at compressor discharge. One advantage in using the superheat at compressor discharge is that its value is always positive even in case that the superheat at compressor suction becomes negative. Experimental results will be provided in this paper.

2. EXPERIMENTAL SETUP

The experimental apparatus is shown in Figure 1 which is a typical heat pump system. The refrigerant was R410A and the compressor was a scroll type. To modulate the speed of the compressor, BLDC inverter was used. As a secondary fluid, water was used for cooling mode and a solution of ethylene-glycol and water mixture was used for heating mode to simulate low temperature condition during winter. Thermo couples were installed at the inlet and outlet of each section. Instead of measuring the saturation temperature of condenser and evaporator, pressures were measured to calculate the saturation temperature. The superheat is defined as follows.

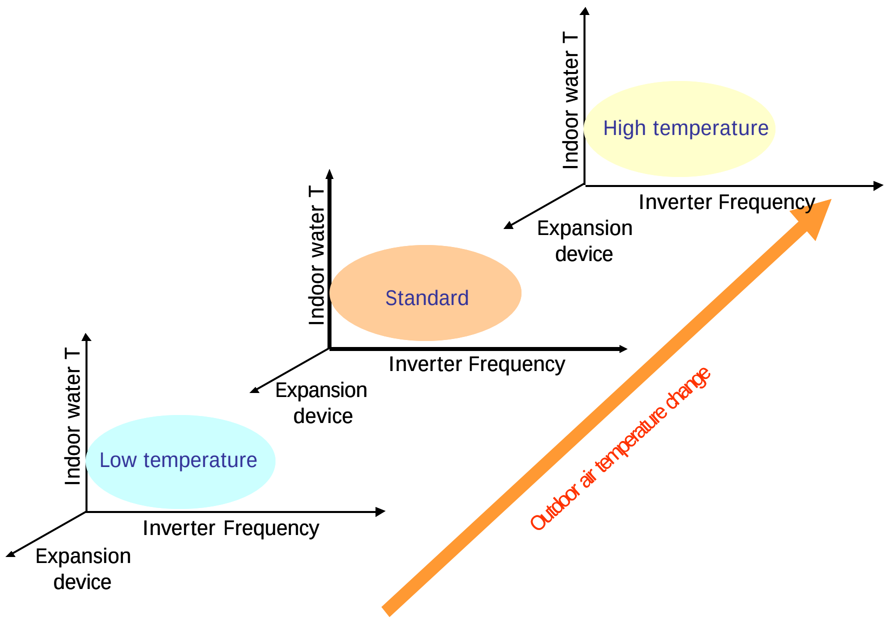


Figure 2: Test condition ranges in the cooling mode operation of a heat pump

$$\mathrm{DSH}_{dis} = T_{comp,dis} - T_{sat}(P_{cond}) \tag{1}$$
$$\mathrm{DSH}_{suc} = T_{comp,suc} - T_{sat}(P_{evap}) \tag{2}$$

DSH_{dis} means superheat at the compressor discharge and DSH_{suc} represents superheat at the compressor suction. The condenser and the evaporator were manufactured with the heat exchanger area of 0.226 m^2 and designed as a concentric counter flow heat exchanger. The secondary fluids circulate through the annulus of heat exchangers. Energy transfer was calculated based on the difference between inlet and outlet temperatures, and mass flow rate of the fluid. The secondary fluid inlet temperature was controlled by Proportional-Integral-Derivative controllers within an error of 0.1^{o}C from the setting temperature. A metering valve was used as an expansion device to control superheat at the compressor discharge.

3. TEST CONDITONS FOR STEADY EXPERIMENT

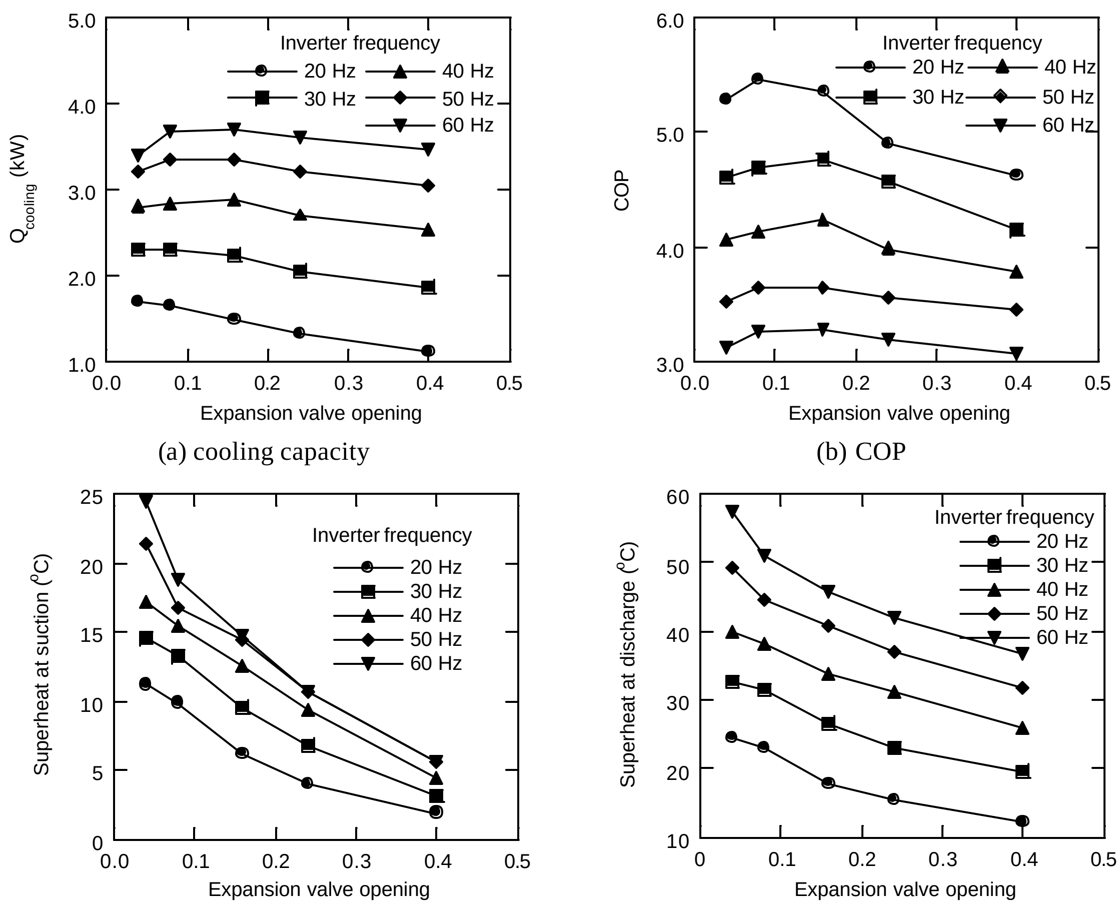
In cooling operation, test conditions are divided into 3 cases of standard, high temperature, and low temperature conditions according to the secondary fluid inlet temperature of outdoor unit, that is 35^{o}C, 43^{o}C and 27^{o}C. Experiments were conducted under various inverter frequencies, expansion valve openings and the secondary fluid inlet temperatures of indoor unit. Figure 2 shows the test ranges in cooling mode operation. In heating mode, experiments were conducted with outdoor temperatures of -10^{o}C and 0^{o}C to simulate low temperature operation.

4. RESULTS AND DISCUSSION

The main purpose of this experiment is to see whether the superheat at the compressor discharge is suitable as a control variable for best performance. The relation between performance and DSH_{dis} was investigated in cooling and heating mode. Additionally, capacity control experiment was conducted in cooling mode. Superheat at the compressor discharge was used as a control variable for an expansion valve opening, which eventually changes the system performance.

4.1 cooling mode test

The temperature of outdoor unit was fixed at 35^{o}C and the temperature of indoor unit was 27^{o}C. Figure 3 shows cooling capacity, COP, DSH_{suc} and DSH_{dis} variation with respect to the inverter frequency and expansion valve opening. As the inverter frequency increases, the cooling capacity increases and the COP decreases. The increase in cooling capacity is mainly due to the increased mass flow rate to yield greater capacity. The increase in COP is



(a) cooling capacity (b) COP

(c) superheat at the compressor suction (d) superheat at the compressor discharge

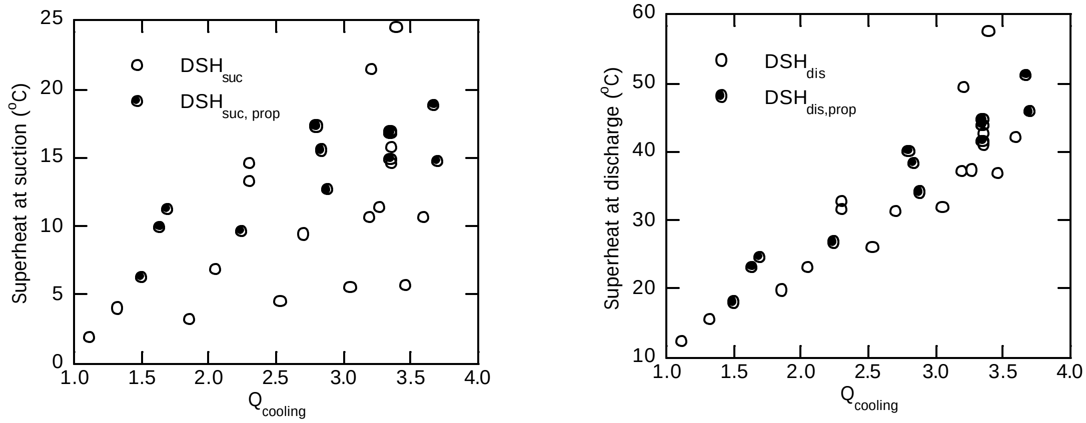
Figure 3: Cooling capacity, COP and superheat change with various compressor speed and expansion valve opening when the inlet temperature of secondary fluid of indoor unit was 27^{o}C.

attributed to the increased temperature difference between the condenser and the evaporator. As the opening of expansion valve increases, there are maximum points for capacity and COP. DSH_{dis} shows separate operation curves with respect to the inverter frequency and expansion valve opening while DSH_{suc} overlaps in high frequency regime.

When the system produces a certain amount of fixed cooling capacity, it should move to an operation point where the COP is high. We let this operation point be a proper working point. DSH_{suc} and DSH_{dis} of Figure 3 were rearranged according to cooling capacity in Figure 4. In Figure 4, there are several points near a designated cooling capacity. For example, there are about 3 points that have a cooling capacity of approximately 2.3 kW. Among these points, one point has higher COP than other points. These high COP points were selected and marked black in Figure 4. They are proper working points in a cooling mode operation. In Figure 4, DSH_{dis} are less scattered than DSH_{suc} in the proper working ranges. The experimental results of DSH_{suc} and DSH_{dis} at proper working points with various indoor unit temperatures are shown in Figure 5. Figure 5 indicates that indoor unit temperature has less effect on DSH_{dis} than DSH_{suc}. Therefore, with outdoor unit temperature fixed, DSH_{dis} is more dependent on the cooling capacity than indoor unit temperature. Figure 6 is DSH_{suc} and DSH_{dis} profiles when the temperature of outdoor unit changed. In Figure 6, DSH_{dis} has separate operation curves depending on the outdoor temperatures while DSH_{suc} overlaps.

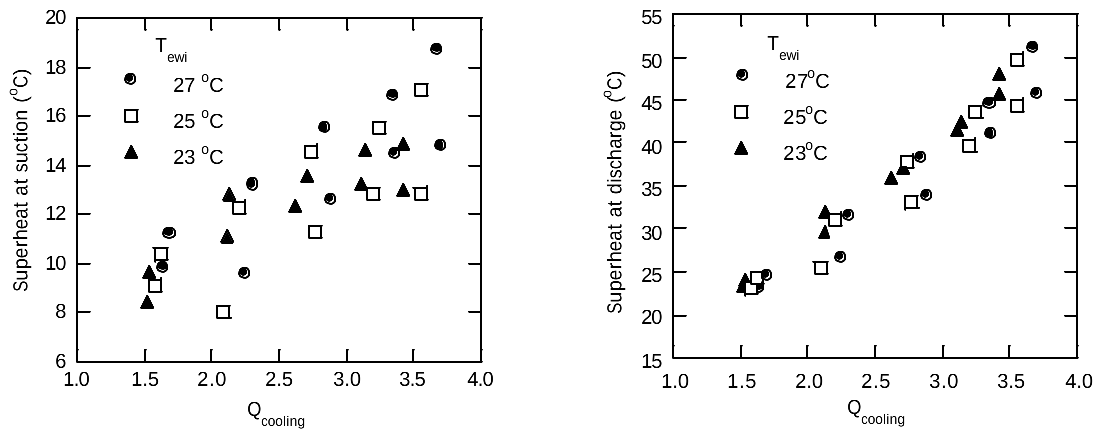
4.2 Heating mode test

With low outdoor (evaporator) temperature, there exists zero or negative value of superheat at the compressor suction side. Heating mode experiments were conducted when the secondary fluid inlet temperature of outdoor unit was 0^{o}C and -10^{o}C and Figure 7 is the results. Figure 7 shows that superheat at suction became negative at low



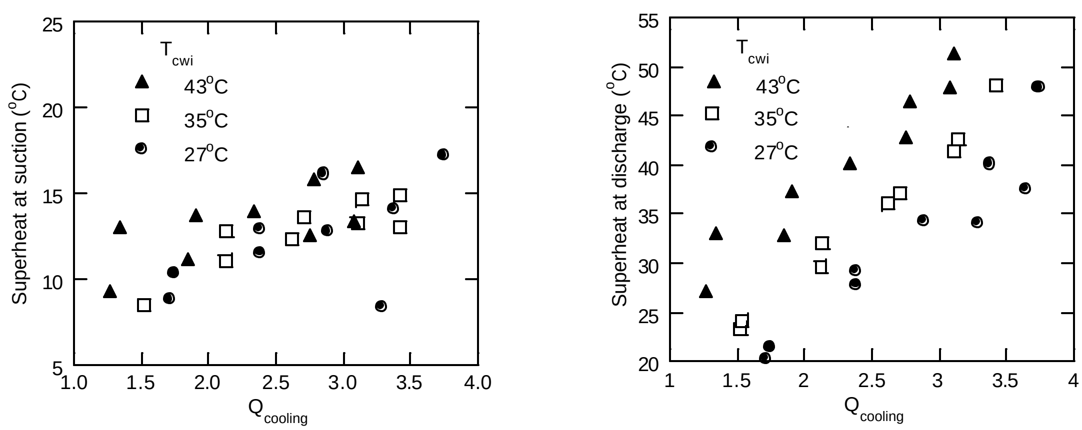
(a) superheat at the compressor suction (b) superheat at the compressor discharge

Figure 4: Superheat versus cooling capacity with various test conditions



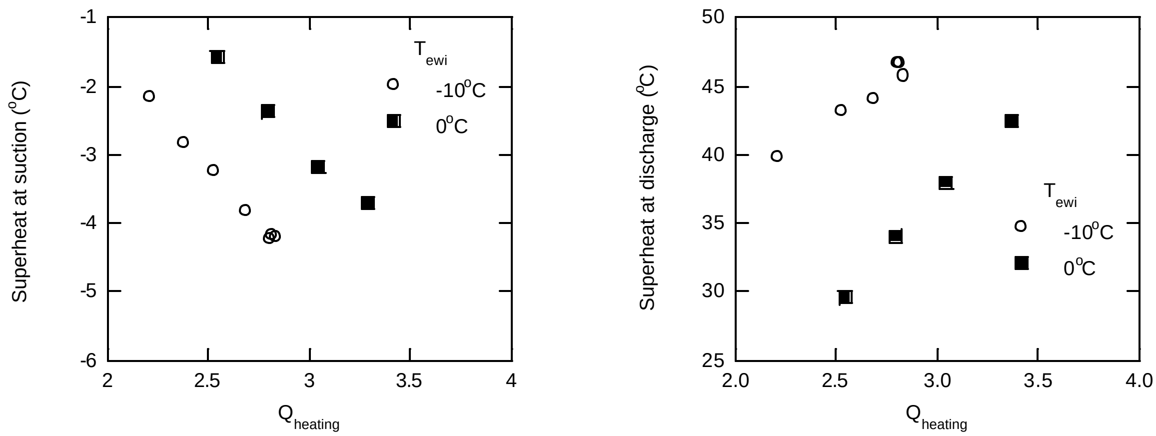
(a) superheat at the compressor suction (b) superheat at the compressor discharge

Figure 5: Superheat at proper working condition with various indoor unit temperature



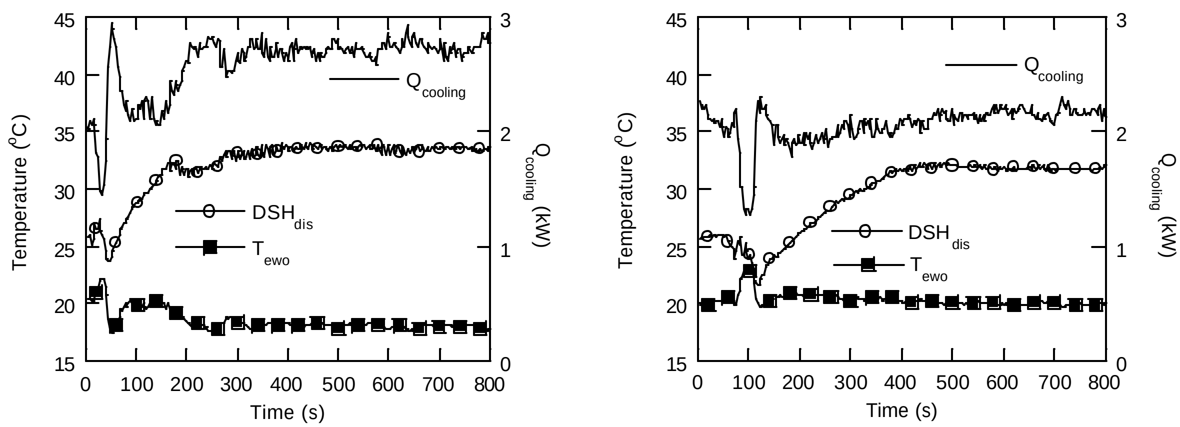
(a) superheat at the compressor suction (b) superheat at the compressor discharge

Figure 6: Superheat at proper working condition with various outdoor unit temperature



(a) superheat at the compressor suction (b) superheat at the compressor discharge

Figure 7: Superheat at proper working condition with various outdoor unit temperature in heating mode



(a) PID control when secondary fluid outlet setting temperature of indoor unit decreased from 20°C to 18°C in cooling mode

(b) PID control when secondary fluid inlet temperature of outdoor unit increased from 35°C to 40°C in cooling mode

Figure 8: PID control experiments using superheat at the compressor discharge as a control variable in cooling mode operation

temperature or low load condition, which represents it is not suitable as a control variable. On the contrary, the superheat at discharge is positive and shows much desirable relation to the system performance; almost linear to the heating capacity.

4.3 Capacity control experiments using superheat at the compressor discharge

A heat pump system varies compressor speed and expansion valve opening to control its capacity. As a control variable for expansion valve opening, the superheat at the compressor discharge was selected in the experiment. From the results of the steady state experiments in cooling mode, regression models for the inverter frequency and the superheat at the compressor discharge were developed. The exact forms of the equations are as follows.

$$F(Hz) = a + bQ_{req} + cT_{cwi} + dT_{ewi}T_{cwi} + fQ_{req}T_{cwi} + gQ_{req}T_{ewi} + hT_{cwi}^2 + iT_{ewi}^2 + jQ_{req}^2 \quad (3)$$

$$\mathrm{DSH}_{dis} = a + bF + cT_{cwi} + dT_{ewi} + eFT_{cwi} + fFT_{ewi} + gT_{ewi}T_{ciw} + hF^2 + iT_{cwi}^2 + jT_{ewi}^2 \quad (4)$$

The coefficients for equations (3) and (4) are shown in Table 1. The basic PID control was performed and the PID coefficients were calculated using Ziegler-Nichols second tuning method (Ogata, 1997). Figure 8(a) is the result when the designated secondary fluid outlet temperature of indoor unit decreased by 2°C. The required cooling

Table 1: Coefficients for equation (3) and equation (4)

Coefficients in equation (3)				Coefficients in equation (4)			
Coefficients	Value	Coefficients	Value	Coefficients	Value	Coefficients	Value
a	46.5675	f	0.3930	a	-52.9194	f	-0.2075
b	3.31479	g	-0.6593	b	0.6220	g	0.3471
c	-1.0667	h	0.0206	c	-0.9329	h	0.0106
d	-1.8032	i	0.0695	d	6.0616	i	-0.1777
e	-0.0269	j	3.7743	e	0.0464	j	-0.1385

capacity was increased to 2.71 kW. The experiment showed that the cooling capacity was maintained within an error of 6.64% and the secondary fluid outlet temperature of the indoor unit was maintained within an error of 0.2^{o}C. The DSH_{dis} should be increased from 25.8^{o}C to 33.6^{o}C. Figure 8(b) is the result when the secondary fluid inlet temperature of outdoor unit increased from 35^{o}C to 40^{o}C. The cooling capacity was maintained within an error of 5.54%. In this case, the DSH_{dis} should be increased from 25.8^{o}C to 31.8^{o}C.

5. CONCLUSIONS

This paper's main purpose is to investigate the possibility of using superheat at discharge as a control variable. Experimental results showed that the superheat at the compressor discharge was more closely related to the proper working points than the superheat at the compressor suction.
In cooling mode, the superheat at the discharge is more reliable as a variable for a capacity control than superheat at suction. In heating mode, when outdoor temperature is so low that superheat at suction becomes negative, superheat at discharge is still a good parameter that ensures proper operation and control.
Superheat at discharge is more qualitatively connected to optimum operation than the superheat at the suction based on the results of this paper.

NOMENCLATURE

DSH	degree of superheat	(^{o}C)
F	inverter frequency	(Hz)
P	pressure	(kPa)
Q	cooling/heating capacity	(kW)
T	temperature	(^{o}C)

Subscripts

c	condenser	o	outlet
dis	discharge	suc	suction
e	evaporator	w	secondary fluid
i	inlet		

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