Development of a System Identification Model for an Air Source Heat Pump

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ABSTRACT

Heat pump system can save energy and installation cost and reduces CO2 emissions. In this study, a system identification of an air source heat pump system using R410A with a variable speed compressor was experimentally investigated under various ambient and indoor temperature and cooling or heating capacity. The experimental study was also performed under cold and hot climate conditions as well as normal ambient temperature in cooling and heating mode. A heat pump system was installed and tested in an environment chamber, where temperature and humidity are precisely controlled. Experimental points were selected under 28 conditions for each cooling and heating mode using center composition design method. Developed steady-state model predicted the experimental values within 10% error. In this study, we carried out in-situ performance test of the air source heat pump system. Further study will be carried out for fault detection and diagnosis for a heat pump system using developed system identification model.

1. INTRODUCTION

Global warming and energy crisis became a major international issue. So, many studies are focused on the development of new energy source and energy conservation.

Heat pump system is a device reversing the natural heat flow from a lower to a higher temperature level. Recent advance in variable speed compressor technology has produced new approaches to improve performance and energy efficiency of a heat pump system.

The existing method for the energy consumption efficiency of a heat pump system is energy efficiency ratio (EER) or coefficient of performance (COP) that the ratio of cooling or heating capacity and power consumption are calculated at rated operating condition. Recently, however, there has been a lot of research to indicate actual performance of a heat pump using variable capacity type compressor with the inverter.

Park et al. (2001) studied the system performance of a multi-split VRF system having two indoor units based on the compressor frequency, total cooling load, and the cooling load fraction between two zones. (Park et al., 2001)

Choi and Kim (2003) studied the performance of a multi-split VRF system having two indoor units with individual EEVs by varying the indoor loads, the EEV opening and the compressor speed. (Choi and Kim, 2003)

Aynur et al. (2006) conducted a field-performance test with a multi-split VRF system in an actual office suite in order to provide real time operational characteristics of the system. (Aynur et al., 2006)

In this study, the performance characteristics of an air source heat pump system using R410A with variable speed compressor were experimentally investigated under the various ambient and indoor temperatures, cooling and heating loads in cooling and heating modes. Performance analysis model for the heat pump system was developed using the data obtained through experiments under various operating conditions.
In this study, we carried out in-situ performance test of developed performance model for the air source heat pump system.

2. PERFORMANCE ANALYSIS MODEL

In this study, we developed the performance model of the heat pump system using the DOE-2 chiller model. (DOE, 1980, Hydeman and Gillespie, 2002)
Performance of the heat pump system varies with operating conditions such as outdoor and indoor air temperature.
Cooling capacity at the indoor unit in a heat pump system was described using outdoor air dry bulb temperature($T_{db,OD}$) and indoor air wet bulb temperature($T_{wb,ID}$) as shown in Eq. (1) and (2).

$$CAP_{IDU,i} = CAP_{IDU,R,i} \times f_{CAP}(T_{wb,ID,i}, T_{db,OD}) \tag{1}$$

$$f_{CAP}(T_{wb,ID,i}, T_{db,OD}) = a_0 + a_1 \times T_{db,OD} + a_2 \times T_{db,OD}^2 + a_3 \times T_{wb,ID,i} + a_4 \times T_{wb,ID,i}^2 + a_5 \times T_{db,OD} \times T_{wb,ID,i} \tag{2}$$

$CAP_{IDU,R,i}$ means rated capacity of the indoor unit for the ith thermal zone in the standard cooling operating condition.
Load factor of the indoor unit for the ith thermal zone is calculated using capacity of the indoor unit(Eq. (1)) and the internal load of ith thermal zone($\dot{Q}_{zone,i}$) as shown in Eq. (3).

$$PLR_{IDU,i} = \frac{\dot{Q}_{zone,i}}{CAP_{IDU,i}} \tag{3}$$

Demanded capacity at the outdoor unit in a heat pump system can be described by summing cooling capacities and load factors of the indoor units as shown in Eq. (4).

$$CAP_{ODU,de} = \sum_{i=1}^{n} (CAP_{IDU,i} \times PLR_{IDU,i}) \tag{4}$$

When heat pump is installed in a building, the extension of pipe length and head difference between indoor and outdoor unit degrades the performance of the heat pump system. However, in this study, effect of pipe length and head difference was not considered for the performance model of the heat pump.
Capacity at the outdoor unit in a heat pump system is described using outdoor air dry bulb temperature and average temperature of indoor air wet bulb as shown in Eq. (5). The average temperature of indoor air web bulb is defined in Eq. (6).

$$CAP_{ODU} = CAP_{ODU,R} \times f_{CAP}(T_{wb,ID,ave}, T_{db,OD}) \tag{5}$$

$$T_{wb,ID,ave} = \frac{\sum_{i=1}^{n} (T_{wb,ID,i} \times \dot{Q}_{zone,i})}{\sum_{i=1}^{n} \dot{Q}_{zone,i}} \tag{6}$$

$CAP_{ODU,R}$ is rated capacity of the outdoor unit in standard cooling operating condition.

Load factor of the outdoor unit can be calculated using demanded capacity of the outdoor unit(Eq. (4)) and actual capacity of the outdoor unit(Eq. (5)) as shown in Eq. (7).

$$PLR_{ODU} = \frac{CAP_{ODU,de}}{CAP_{ODU}} \tag{7}$$

Power consumption and coefficient of performance for the heat pump system in cooling mode is calculated using Eq. (8) and Eq. (11).

$$W = W_R \times f_W(T_{wb,ID,ave}, T_{db,OD}) \times f_{PLR,ODU}(PLR_{ODU}) \tag{8}$$

$$f_W(T_{wb,ID,ave}, T_{db,OD}) = a_0 + a_1 \times T_{db,OD} + a_2 \times T_{db,OD}^2 + a_3 \times T_{wb,ID,ave} + a_4 \times T_{wb,ID,ave}^2 + a_5 \times T_{db,OD} \times T_{wb,ID,ave} \tag{9}$$

$$f_{PLR,ODU}(PLR_{ODU}) = a_0 + a_1 \times PLR_{ODU} + a_2 \times PLR_{ODU}^2 \tag{10}$$

$$COP = \frac{CAP_{ODU,de}}{W} \tag{11}$$

W_R is rated power consumption in standard cooling operating condition.

Performance model of the heat pump system in heating mode was constructed with the same model structure in cooling mode. But, the indoor dry bulb temperature was used instead of the indoor wet bulb temperature for performance model of the heat pump system in heating mode.

In this study, the effect of frosting at outdoor unit on the performance of the heat pump system was not considered.

3. PERFORMANCE EXPERIMENT

The studied system was an R410A 58kW VRF heat pump system with inverter compressor. In this study, the heat pump system has an outdoor unit and four indoor units with fin-tube type heat exchangers and variable speed type compressor. Test units were installed in environmental chambers and charged with R410A according to the manufacturer's specifications.

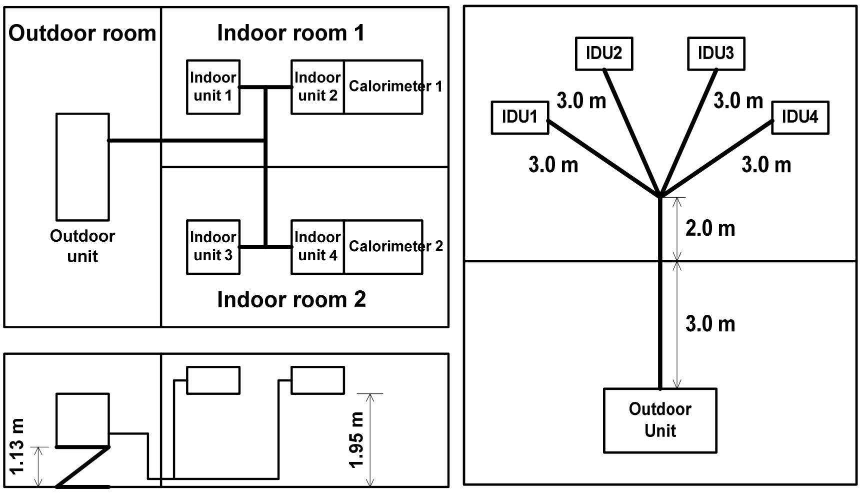


Figure 1: Schematic diagram of experimental setup

Table 1: Experimental condition for performance test of heat pump in cooling and heating mode

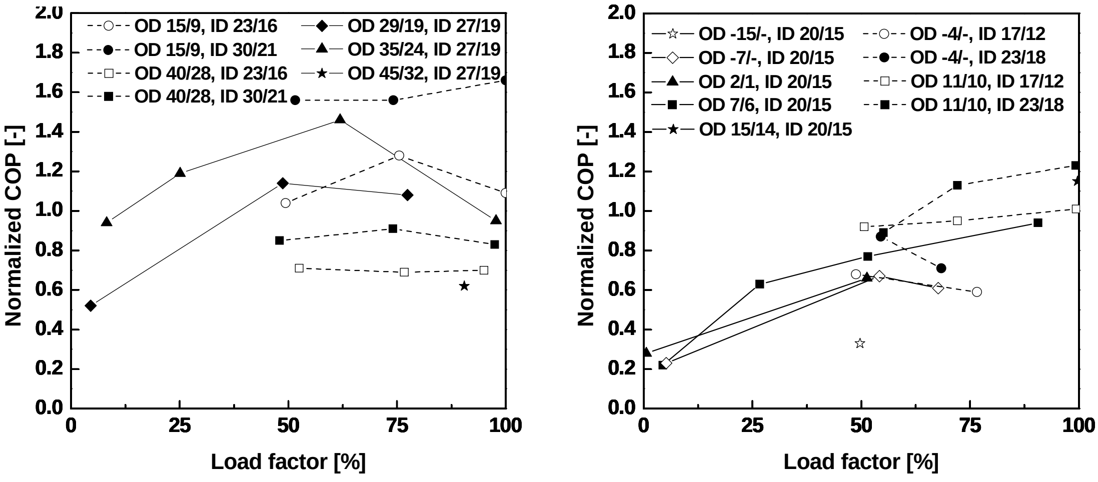
Design parameter		Unit	Conditions
Cooling mode	Indoor temperature(DB/WB)	°C	23/16~33/24
	Outdoor temperature(DB/WB)	°C	15/9~45/32
	Part load factor	%	Minimum~100
Heating mode	Indoor temperature(DB/WB)	°C	15/10~27/21
	Outdoor temperature(DB/WB)	°C	-15/-~15/14
	Part load factor	%	Minimum~100

Figure 1 shows schematic diagram of experimental setup including standard pipe length and head difference by AHRI 1230 for performance test of heat pump unit.(AHRI 1230, 2010) AHRI 1230 is a standard code of a performance test requirements for multi-split air conditioners and heat pump purposed by Air-Conditioning, Heating and Refrigeration Institute (AHRI).
Table 1 shows experimental conditions to develop the heat pump performance model in cooling and heating mode. In this study, experimental points were selected under 28 conditions for each cooling and heating mode using central composite design method.

4. RESULTS

4.1 COPs in cooling and heating mode

Figure 2 shows the variation of normalized COPs of a heat pump system according to indoor and outdoor temperature, and load factor. Normalized value means ratio of an actual measurement value to nominal value by manufacturer at rated operating condition.



(a) Cooling mode (b) Heating mode

Figure 2: Variance of normalized coefficient of performance of a heat pump system under cooling and heating operating conditions

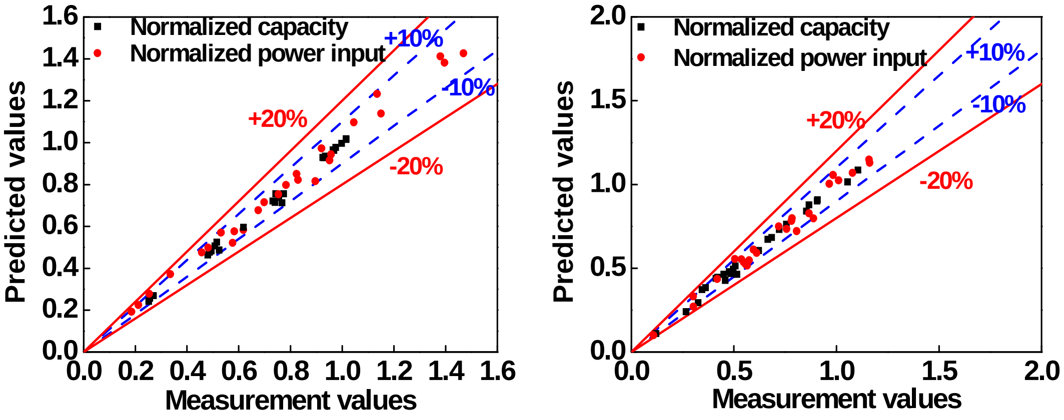
In cooling mode, the normalized COPc decreases with increase of outdoor air temperature and decrease of indoor air temperature as shown in Fig. 2 (a). Because cooling capacity decreases linearly and power consumption increases rapidly when outdoor air temperature increases. As indoor air temperature decreases, decreasing rate of power consumption is greater than decreasing rate of cooling capacity. As load fact increases, cooling capacity of heat pump linearly increases, but power consumption increases rapidly. Therefore, there is maximum COPc in each operating condition.
In heating mode, normalized COPh increases under full load operating with increase of outdoor air temperature and indoor air temperature as shown in Fig. 2 (b). As load factor increases, heating capacity of heat pump and power consumption increase. So, variation of COPh is similar with COPc.

4.2 Sensitivity analysis of the developed performance model

Table 2 shows coefficient of the performance model of the heat pump in cooling and heating modes. Figure 3 compares predicted and measurement values of normalized capacity and power input of heat pump system in cooling and heating mode. Developed performance model can predict the experimental values within 10% error.

Table 2: Coefficient of the performance model of the heat pump in cooling and heating mode

Operating mode		a0	a1	a2	a3	a4	a5
Cooling mode	$f_{CAP}(T_{wb,ID,i}, T_{db,OD})$	0.995e0	0.165e-2	-0.538e-4	-0.902e-2	0.484e-3	-0.405e-4
	$f_W(T_{wb,ID,ave}, T_{db,OD})$	0.217e1	-0.193e-1	0.484e-3	-0.714e-1	-0.308e-3	0.708e-3
	$f_{PLR,ODU}(PLR_{ODU})$	-0.241e0	0.222e1	-0.100e1			
Heating mode	$f_{CAP}(T_{wb,ID,i}, T_{db,OD})$	0.205e1	0.827e-2	-0.582e-3	-0.888e-1	0.144e-2	0.221e-3
	$f_W(T_{wb,ID,ave}, T_{db,OD})$	0.130e1	0.632e-1	-0.707e-3	-0.552e-2	-0.544e-4	-0.356e-2
	$f_{PLR,ODU}(PLR_{ODU})$	-0.263e-1	0.117e1	-0.240e0			



(a) Cooling mode (b) Heating mode

Figure 3: Comparison the predicted and measurement values of normalized capacity and consumption power of heat pump system

5. CONCLUSIONS

- In this study, Performance model for the heat pump system was developed using the data obtained through experiments under various operating conditions.
- Developed performance model predicted the experimental values within 10% error.

NOMENCLATURE

The nomenclature should be located at the end of the text using the following format:

CAP	capacity	(kW)	**Subscripts**	
PLR	part load ratio	(-)	db	dry bulb
T	temperature	(^oC)	wb	wet bulb
W	power consumption	(kW)		

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