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Characteristics of R134a/R410A Cascade Heat Pump and Optimization

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ABSTRACT

Single stage air source heat pump has been widely used around the world. However, it has some drawback, for example, deteriorated heating capacity and COP in cold ambient temperature. To overcome these disadvantages, cascade cycle has been suggested and shows better performance at low ambient temperature. Cascade cycle has smaller compression ratio at each cycle and shows higher compression efficiency compared to that of single stage cycle. Intermediate pressure is most critical parameter which decides the system performance. There have been many research studies about cascade heat pump system using NH_3/CO_2 and C_3H_8/CO_2 as refrigerants at high and low stage, respectively. Most studies conducted on system optimization, finding the optimal intermediate pressure. However, little information has given about the experimental optimization results.

In this study, AWHP (air to water heat pump), which is composed by two single stage cycle, was studied experimentally. AWHP high and low cycle adopts R134a and R410A as refrigerants, respectively, because critical temperature of R134a is higher than R410A. Since there are two single stage cycles, each cycle has a different compression ratio. In this research, by introducing numerical analysis, system optimization was studied. Characteristics of cascade cycle with respect to each compressor have been also studied. Experiment was conducted with various operating conditions; ambient temperature, water inlet temperature.

1. INTRODUCTION

A single stage heat pump has been widely used around the world due to its low cost and easy installation. Nevertheless, most single stage heat pump has a limitation on performance and this requires other concepts of heat pump to get over their performance disadvantages, for example, the deteriorated heating capacity and the low COP at very low ambient temperature. Bertsch and Groll (2008) stated four main problems for single stage air-source heat pump. Cascade or multi-stage heat pump system has been developed to overcome these weaknesses of single stage heat pump. The cascade cycle has a smaller compression ratio at each cycle and shows a better compression efficiency compared to single stage cycle.

There have been many studies on a two-stage cascade cycle with a different refrigerant. Lee et al. (2006), Alberto Dopazo et al. (2009) and Bingming et al. (2009) studied the cascade refrigeration system with NH_3 and CO_2 as working fluids in the high and low stages, respectively. Lee et al. (2006) determined the optimal condensing temperature of a cascade system and made correlations with several temperature parameters. Alberto Dopazo et al.

(2009) also showed the COP and exergetic efficiency with various operating conditions, and conducted the optimization study numerically. Bingming et al. (2009) analyzed the performance of a system experimentally and concluded that cascade system was very competitive in a low temperature applications. There also have been cascade system studies using CO_2 and C_3H_8 as refrigerants for refrigeration. Bhattacharyya et al. (2008) conducted the mathematical modeling of system and developed guidelines to the user for selecting intermediate temperature. Intermediate temperature is the most critical parameter for a multistage or a cascade heat pump system which affects the system efficiency since the intermediate pressure determines the compression ratio and compressor isentropic efficiency. Most studies focused on finding optimized intermediate temperature numerically and suggested the correlations in terms of several temperature related parameters. However, the suggested correlations can't be adopted to other researches and do not provide a general meaning.
Moreover, little information has given about the experimental optimization results since the cascade test setup is quite complex and hard to control. Bingming et al. (2009) and Dopazo and Fernández-Seara (2011) provided experimental optimum intermediate temperature in the NH_3/CO_2 cascade heat pump system. Nevertheless, these studies tended to focus on finding maximum COP with different heating capacity. Since heating capacity can affect the COP directly, the optimized intermediate temperature should be obtained at a fixed heating capacity.
The main purpose of this research is to find out the optimum temperature of R134a/R410A cascade heat pump system with numerical analysis. Experiments were conducted at a given operating condition (e.g. water inlet temperature, ambient temperature) and fixed heating capacity. The optimum intermediate temperature is experimentally obtained and compared with the numerical analysis results.

2. EXPERIMENTAL SETUP

2.1 System Description

In this study, the performance of AWHP (air to water heat pump), which is constituted by two single cycles, has been studied numerically and experimentally. The high stage cycle with R134a as a refrigerant is used to make hot water and low stage cycle with R410A as a refrigerant is used to absorb heat from ambient. Schematic diagram of the R134a/R410A cascade system is shown in Fig. 1. Two single cycles are connected with cascade heat exchanger, where R134a evaporates and R410A condenses. The refrigerant in the low cycle stage evaporates by absorbing heat from ambient, compressed and condensed at cascade heat exchanger. The refrigerant in the high cycle evaporates while the refrigerant in the low stage cycle condenses, then is compressed and condenses during the water heating.

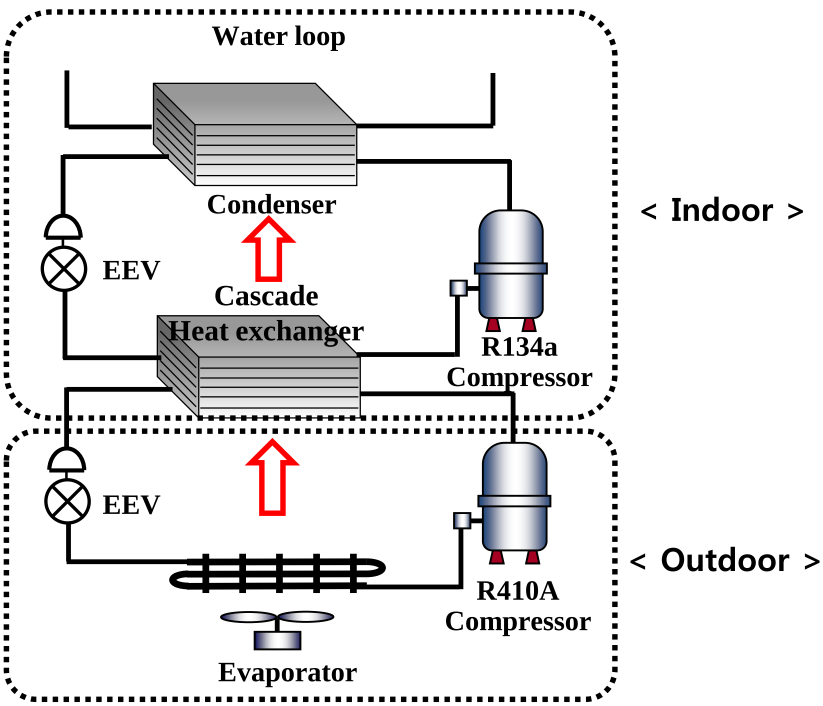


Figure 1: Schematic diagram of cascade system test setup

Table 1: Test conditions

Parameters	Value
Ambient temperature (^oC)	-7, -2, 7
Condenser water inlet temperature (^oC)	25 ~ 55
Degree of superheat (^oC)	~5

Each circuit adopts a different refrigerant suitable for each temperature level. This means the lower temperature unit use a refrigerant with lower boiling point temperature, which has a higher saturation pressure at low temperature (Bansal and Jain (2007)). High pressure at low temperature means the higher density of suction line, which requires a smaller compressor. Moreover, as the critical point of R410A (344.5 K) is lower than that of R134a (374.2 K), low stage cycle adopts R410A. Fig. 2 shows a typical P-h and T-s diagram of the cascade system. The difference between low stage condensing temperature and high stage evaporating temperature is an important parameter. Proper temperature difference in the cascade heat exchanger is required for proper operation of the system, and temperature difference can affect the heat pump performance strongly. In this study, temperature difference between the condensation of R410A and the evaporation of R134a is kept by 8~10 K.

2.2 Experimental Apparatus and Test conditions

As shown in Fig. 1, there are 3 enclosed loops including R134a, R410A and water loops and these three loops are totally separated. Outdoor unit (R410A) is placed in an environmental chamber to keep the ambient temperature constant. R410A evaporator whose nominal cooling capacity is 14 kW is a fin tube heat exchanger with 2 rows and 64 steps. Hermetic twin rotary compressor with inverter is used in each cycle to control the compressor speed. R134a condenser and cascade heat exchanger are plate heat exchangers. Each single stage cycle chooses EEV as an expansion valve, which controls the degree of superheat by varying the valve opening. Hot water from R134a condenser enters the water reservoir, and water chiller cools the hot water which makes the system stay in a steady-state.

Indeed, the main objective of this research is to find the highest COP for a given heating capacity and operation condition. In order to keep the heating capacity, compressor speed in each stage were precisely controlled. By increasing the compressor speed in the high stage cycle while decreasing the compressor speed in the low cycle, the heating capacity can be fixed at a desired value. The DSH (degree of superheat) is an important parameter since the large DSH deteriorates the heating capacity and small DSH may cause the durability problem on compressor. In order to exclude the effect of DSH on system performance, each cycle DSH was kept under 5 K by controlling the opening of expansion valve.

The operating test conditions are listed in Table 1. The ambient temperature and inlet water temperature are the most important parameters which determine the performance of AWHP. The ambient temperature is controlled by environmental chamber and the water inlet temperature is adjusted by controlling the water chiller.

2.3 Data Reduction

The evaluation of heat pump performance such as heating capacity and COP is carried out with measured data. The heating capacity of the cascade system, which actually is the target of heat pump system is calculated by equation (1) using the mass flow rate of secondary fluid.

$$Q_H = \dot{m}_w C_w (T_{w,out} - T_{w,in}) \tag{1}$$

The overall COP (coefficient of performance) of this system is defined as equation (2), which is the ratio of heating capacity to total power consumption including fan power.

$$\mathrm{COP} = \frac{Q_H}{(W_H + W_L)} \tag{2}$$

The intermediate temperature is calculated as equation (3), that is the average of the evaporating temperature of R134a and the condensing temperature of R410A. The saturation temperature was calculated by measured pressure data using REFPROP (Lemmon (2007)).

$$T_{INT} = \frac{\left(T_{E,134} + T_{C,410}\right)}{2} \tag{3}$$

3. NUMERICAL ANALYSIS

Numerical analysis was conducted to find the optimum intermediate temperature, and this is so called implicit solution. Since the useful heat from this system is only Q_H , the total system COP is represented as follows.

$$\mathrm{COP} = \frac{Q_H}{(W_H + W_L)} = \frac{Q_H}{(Q_H - Q_{INT}) + (Q_{INT} - Q_L)} = \frac{Q_H}{(Q_H - Q_L)} = \frac{1}{1 - Q_L/Q_H} \tag{4}$$

Minimizing the Q_H/Q_L gives the maximum COP, which means minimized compressor work considering the first law of thermodynamics. Q_H/Q_L term can be rearranged as a simple expression with several assumptions, isothermal heat transfer at all heat exchanger and zero superheating and zero subcooling temperatures.

$$Q_H/Q_L = \frac{T_{C,134}\Delta S_{C,134}}{T_{E,410}\Delta S_{E,410}} = \frac{T_{C,134}\dot{m}_H \Delta s_{C,134}}{T_{E,410}\dot{m}_L \Delta s_{E,410}} \tag{5}$$

In cascade heat exchanger,

$$T_{E,134} \cdot \dot{m}_H \Delta s_{E,134} = T_{C,410} \cdot \dot{m}_L \Delta s_{C,410}$$
$$\frac{\dot{m}_H}{\dot{m}_L} = \frac{T_{C,410}\Delta s_{C,410}}{T_{E,134}\Delta s_{E,134}} \tag{6}$$

By Substituting the equation (6) into equation (5), Q_H/Q_L is expressed as a product of temperature ratio and entropy change ratio. First two terms are the temperature ratio of high stage and low stages, last two terms are the entropy change ratio of each stage.

$$Q_H/Q_L = \frac{T_{C,134}}{T_{E,134}} \frac{T_{C,410}}{T_{E,410}} \frac{\Delta s_{C,134}}{\Delta s_{E,134}} \frac{\Delta s_{C,410}}{\Delta s_{E,410}} \tag{7}$$

The process of finding minimum value cannot be conducted explicitly, since the entropy change terms are quite complex to calculate. The minimum Q_H/Q_L value is obtained by varying $T_{C,410}, T_{E,410}$.

4. TEST RESULTS AND DISCUSSIONS

4.1 Characteristics of Cascade System

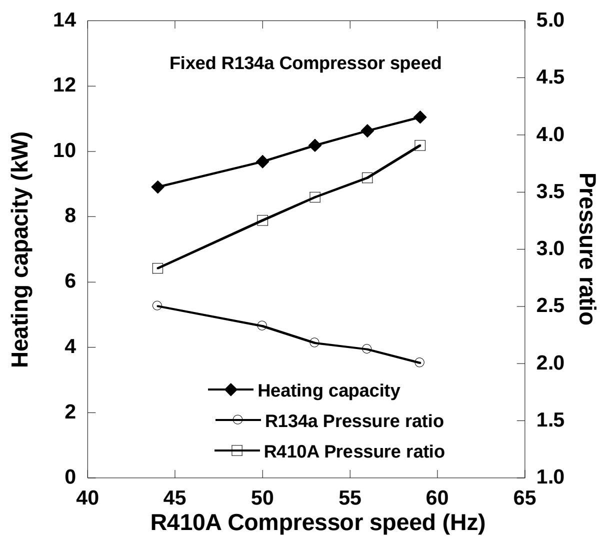


Figure 2: System variation with respect to compressor (R134a compressor speed : 50 Hz)

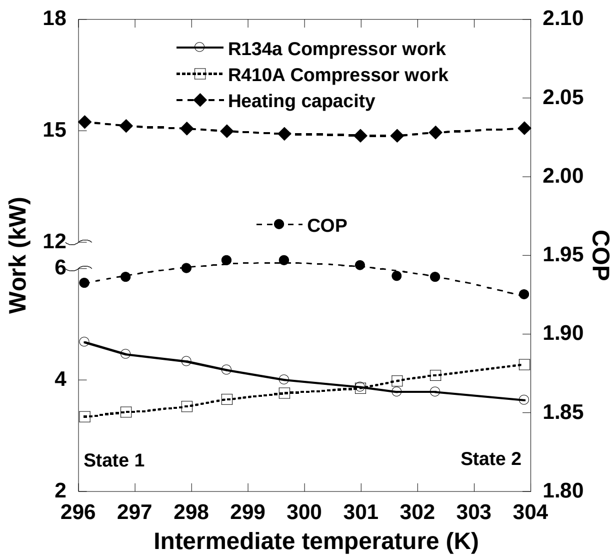


Figure 3: Optimal intermediate temperature and system variation
(Ambient temperature : -7^{o}C, Water inlet temperature : 55^{o}C, Heating capacity : 15kW)

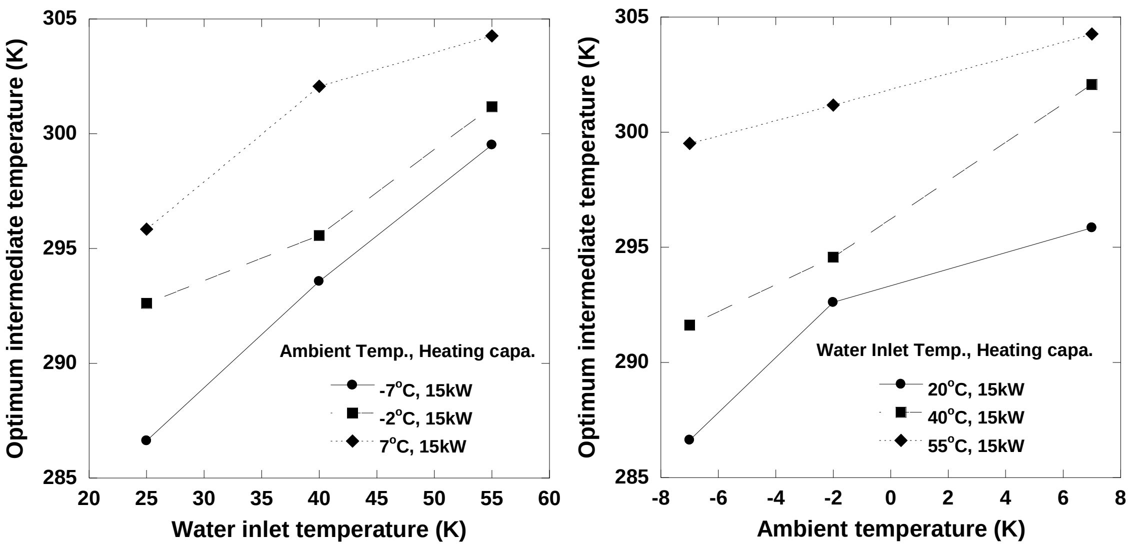
In this section, the characteristics of cascade heat pump system are depicted especially with respect to each compressor. Fig. 2 shows pressure ratio variation of each cycle and heating capacity with respect to the R410A compressor frequency. R134a condensing pressure and R410A evaporating pressure is kept almost constant during the variation of R134a compressor frequency whereas both R134a evaporating and R410A condensing pressure decrease. So, the pressure ratio of R410A increases whereas that of R134a decreases. High pressure ratio requires high power consumption on compressor, the power consumption of R134a compressor decreases whereas R410A compressor increases. Fig. 2 also shows variation of heating capacity. Heating capacity at high stage condenser is slightly enhanced because the mass flow rate of the low stage cycle was increased.

Similar tendency was shown with respect to R134a compressor frequency. There is little pressure change at high stage cycle condenser and low stage cycle evaporator as mentioned before. Heating capacity is also enhanced with the increase of R410A compressor frequency and the degree of enhancement is smaller than that for R410A compressor frequency. Heating capacity can be controlled and can be fixed by increasing one compressor frequency and decreasing the other compressor frequency.

Fig. 3 shows the different system state at given operating conditions for the same heating capacity. State 1 is the state that high stage cycle pressure ratio is much larger than that of low stage cycle, which means the high stage compressor requires more work than low stage compressor. State 2 is the state that low stage compressor requires more work than high stage compressor. The state 1 and state 2 are so called high stage overworking and low stage overworking, respectively. As system state changes from state 1 to 2, intermediate temperature increases and between these two states, there is a minimum point for the total compressor work near 300 K that will maximize the COP at a given heating capacity.

4.2 Experimental Results for Water Inlet Temperature and Ambient Temperature Change

Fig. 4 (a) shows COP variation with respect to water inlet temperature at a given heating capacity and ambient temperature. As water inlet temperature increase, optimal intermediate temperature also increases. Since high water inlet temperature affects the condensing temperature of R134a, whereas R410A evaporating temperature is almost constant since ambient temperature is kept constant. Since water inlet temperature strongly affects the high stage condensing temperature, higher water temperature leads to the increased intermediate temperature. That is, the increment of high stage condensing temperature is much larger than that of heating capacity variation. This is quite important information because water inlet temperature varies with time in actual situation. Fig. 4 (b) shows COP variation with respect to ambient temperature at a given water inlet temperature and heating capacity. As shown in Fig. 4 (b), optimal temperature increases with ambient temperature increase. Ambient temperature affects the evaporating temperature of low stage cycle that induces the increase of optimal temperature as ambient temperature increases. This is consistent that optimal temperature is related with the evaporating temperature of low stage cycle and the condensing temperature of high stage cycle. The effect of ambient temperature on optimal temperature is



(a) (b)

Figure 4: Optimal intermediate temperature variation with respect to water inlet and ambient temperature

shown in Fig 4(a). There was a significant deterioration of COP with water inlet temperature and ambient temperature. Since the high water inlet temperature and low ambient temperature requires more compression ratio of high and low cycle, COP becomes worse at high water inlet temperature condition and low ambient temperature condition. This is a common tendency of air to water heat pump system.

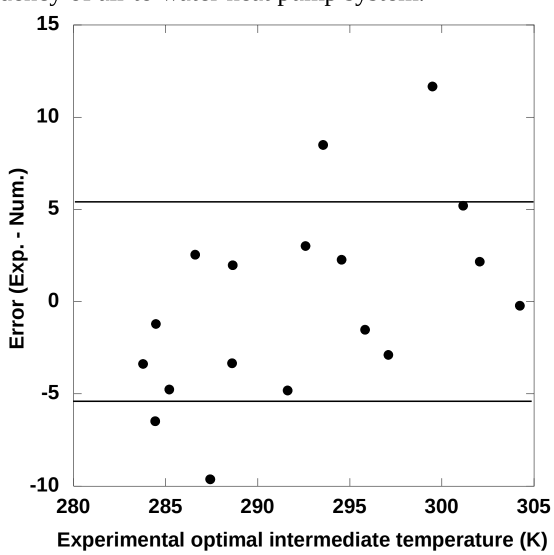


Figure 5: Comparison with numerical analysis

4.3 Comparison with Numerical Results

The results of numerical analysis and experiment are compared. Fig. 5 shows the difference between experiment results and numerical analysis. Implicit solutions well predicted the experimental optimum intermediate temperature and the RMSE (root mean squared error) of implicit solution was 5.162.

5. CONCLUSIONS

Optimal intermediate temperature on AWHP cascade system has been studied experimentally and numerically. Numerical analysis has been conducted by thermodynamic approaches with several assumptions. It is normally known that AWHP performance is worse at high water inlet temperature and low ambient temperature. These parameters affect not only optimal intermediate temperature but also COP. In other words, AWHP performances are deteriorated at high water inlet temperature and low ambient temperature. The optimal intermediate temperature is increased as ambient temperature increases and water inlet temperature increases. These phenomena are explained the by the differences between high stage condenser temperature and low stage evaporator temperature. Numerical analyses show good prediction of experimental results and the RMSE was about 5 K.

NOMENCLATURE

AWHP	air to water heat pump	
COP	coefficient of performance	
RMSE	root mean squared error	
Q	capacity	(kW)
Δs	entropy change	(kJ/kg.K)
T	temperature	(K)
W	compressor power consumption	(kW)

Subscripts

C	condenser
E	evaporator
H	high stage
L	low stage
INT	intermediate
134	R134a
401	R410A

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