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Analytical Study on the Performance Characteristics of a Liquid Injection Refrigeration Cycle

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ABSTRACT

A residential heat pump is faced with reliability problems due to a higher compressor discharge temperature and lower performance at lower ambient temperatures. Many researchers have been trying to overcome this problem by applying vapor injection techniques. However, the application of the vapor injection cycle caused higher cost due to the increased components such as internal heat exchanger or flash tank. Liquid injection technique has merits of cost and cycle reliability because the refrigerant discharged from the condenser is expanded and injected directly into the compressor. In this study, a simulation program for a liquid injection heat pump cycle was developed. The cycle performance was simulated using mass and energy balances. The model was validated by comparing the predictions with measured data at various operating conditions. Based on the simulation results, cycle performance characteristics were discussed at various operating and compressor design conditions.

1. INTRODUCTION

In recent years, a vapor injection technique has been used to enhance the performance of a heat pump system for tropical and cold regions. The vapor injection cycle has been rapidly developed to ensure reliability and improve the performance of the heat pump cycle by decreasing the discharge gas temperature and quickly responding to outside air temperature conditions. The vapor injection cycle enables wide application of heat pump system especially in severe condition. However, the application of the vapor injection cycle caused higher cost and reliability problems due to the increased components such as an internal heat exchanger or flash tank. Liquid injection technique has merits in the view point of cost and cycle reliability.

Liquid injection is a good option for high compression ratios. Operating the compressor at high compression ratios can cause excessively high discharge temperatures. This can chemically degrade oil and cause mechanical failure. Sami and Tulej (2001) and Sami and Aucion (2003) conducted a series of liquid injection tests with different refrigerant mixtures and liquid injection was effective in reducing the compressor frequencies. Dutta et al. (2001) investigated the influence of liquid refrigerant injection on the performance of a scroll compressor both experimentally and analytically. In this study, a simulation model of a heat pump cycle with liquid injection technique was newly developed. The model was validated by comparing the predictions with measured data at various operating conditions. In addition, the model was used to analyze the performance characteristics according to the design conditions for the injection hole.

2. SIMULATION MODEL AND VALIDATION

In this study, a simulation model for analyzing liquid injection cycle was developed. The model was developed based on the ORNL model. The liquid-injection refrigeration cycle consisted of a twin rotary compressor, a condenser, an evaporator and an expansion device. The simulation convergence of the model was checked by three balances of mass and energy. In the mass balance, the condensing pressure was adjusted until the mass flow rate through the compressor and expansion device were the same. In the energy balance, the evaporating pressure was adjusted until the calculated enthalpy at the evaporator exit and the compressor inlet were the same. As the evaporator exit pressure decreases, the enthalpy at the evaporator exit increases due to the increase in the temperature difference between the air and the refrigerant. In addition, the calculation was iterated by changing superheat until satisfying balance in the charge amount.
The performance of the twin rotary compressor was simulated using a loss and efficiency model by Fischer and rice (1983). The loss and efficiency model is derived from the internal energy balance in the compressor using the compressor design, internal efficiency, and heat-loss values. The mass flow rate (m), compressor speed (Nrps), displacement (D), volumetric efficiency (η_{vol}) was calculated by Eqs. (1)-(4), respectively and the equations were presented Davis and scott (1976).

$$\dot{m} = \eta_{vol} N_{rps} D \rho_{compin} \tag{1}$$

$$N_{rps} = \frac{N_{rpm}}{60} \tag{2}$$

$$D = \frac{\pi}{4} L_c \left(D_c^2 - D_r^2 \right) \tag{3}$$

$$\eta_{vol} = 1 + 0.0933 \left(\frac{\gamma - 1}{\gamma} \right) \frac{P_{inj}}{P_{sat.eva}} - 0.733 \tag{4}$$

The heat exchangers were modeled based on the tube-by-tube method. The heat exchanger is divided into several segments. In each segment heat transfer rate is calculated by Eq. (5).

$$q_{seg} = \dot{m}(h_{out} - h_{in}) \tag{5}$$

For both condenser and evaporator, the heat transfer rate was calculated by the ε-NTU method. Using the ε-NTU method with inlet conditions of each section, exit conditions of each section were calculated. In this study, on electronic expansion valve (EEV) was used as the expansion device. The mass flow rate through the EEV was calculated by the correlation proposed by Park et al. (2007). In order to develop experience flow model that can predict the mass flow rate of refrigerant passing through the EEV, developed an experimental correlation using dimensionless variables. The six dimensionless π-groups were derived by combining the variables based on the Buckingham π –theorem.
The simulation model was validated by comparing the measured data with the predictions. Fig. 1 and Table 1 show the schematic diagram and specifications of the experimental setup used to measure the performance of liquid injection cycle. The test setup using R-410a consists of a twin rotary compressor, two air-cooled heat exchangers and two EEVs. The twin rotary compressor with a cylinder volume of 26cc was varied from 50 Hz to 100 Hz by a commercial BLDC motor driver. The EEV opening is adjusted by varying the flow area through an orifice with needle. The liquid injection cycle was tested in an air-enthalpy type psychrometric calorimeter including indoor and outdoor chambers. Each chamber was designed for controlling air temperature form -20°C to 50°C and relative humidity from 0% to 100%. The cooling capacity was measured as given in Eq. 6 by utilizing the air-enthalpy method based on both the airflow rate and the enthalpy difference across the evaporator.

$$q_{act} = \frac{Q_a}{v}(h_{ao} - h_{ai}) \tag{6}$$

The compressor power was measured by a power meter. The refrigerant flow rate was measured by a coriolis-effect flow meter. The refrigerant pressures were measured by pressure transducers. The refrigerant temperatures were measured by T-type thermocouples and resistance temperature sensor. Mass flow rate of injection was determined by degree of opening the EEV toward injection hole. Injection ratio was calculated by Eq. 7.

$$InjectionRatio = \frac{\dot{m}_{inj}}{\dot{m}_{total}} \tag{7}$$

The cycle performance was measured for the dry bulb and wet bulb temperature of condenser was fixed at 54°C and 24°C respectively. The dry bulb and wet bulb temperature of evaporator was fixed 29°C and 19°C. The compressor frequency was fixed 100Hz. Air flow rate of condenser was 33 m^3min^{-1} and air flow rate of evaporator was $13m^3min^{-1}$.
Using the experimental setup, the experiments were carried out with the injection ratio. As shown in Fig. 2, the predicted total mass flow rate, injection mass flow rate, input power, cooling capacity, and COP showed good agreement with the measured data within ±10% deviation.

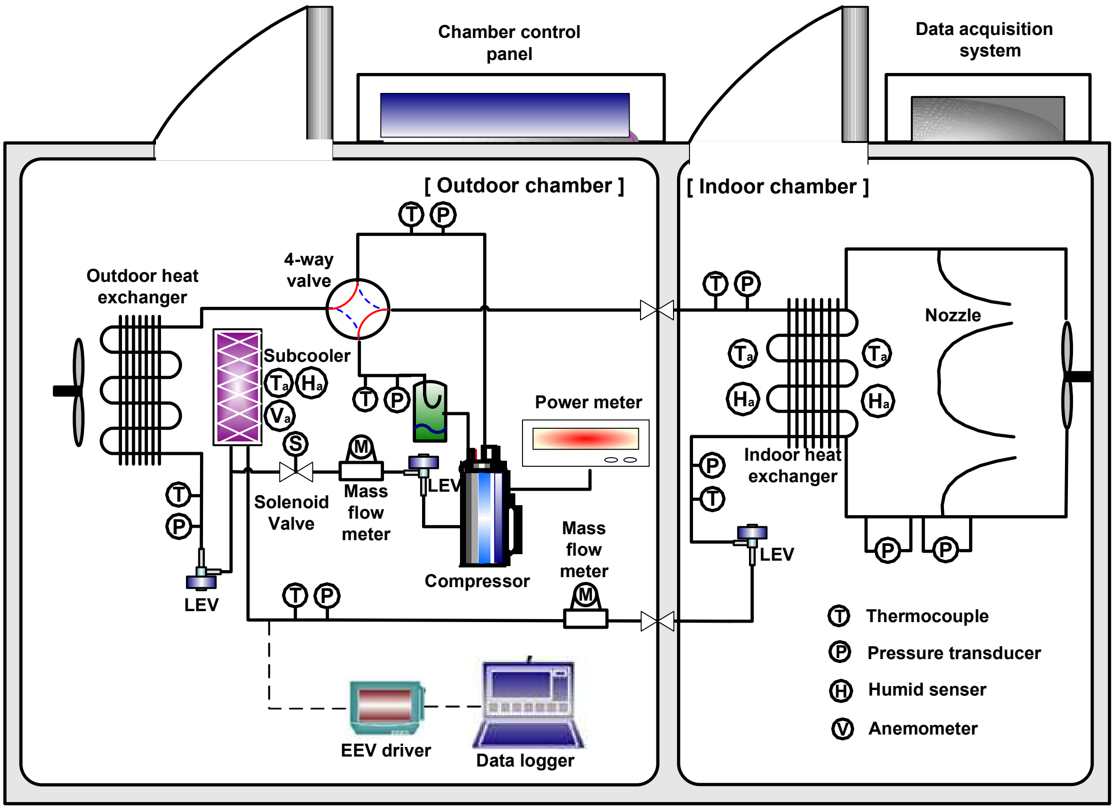

Figure 1: Schematic diagram of the experimental setup.

Table 1: Specification of the experimental setup.

Item	Specifications		Unit
Compressor	Type	Twin rotary	
	Cylinder diameter	48	mm
	Cylinder length	17	mm
	Roller diameter	42.6	mm
Evaporator	Fan type	Axial flow	
	Type	2 Rows / 4 Paths / 16 Steps	
	Tube length	0.68	m
Condenser	Fan type	Cross flow	
	Type	2 Rows / 4 Paths / 24 Steps	
	Tube length	0.814	m
EEV	Pulse rate	500 steps	
	Orifice diameter	1.2	mm

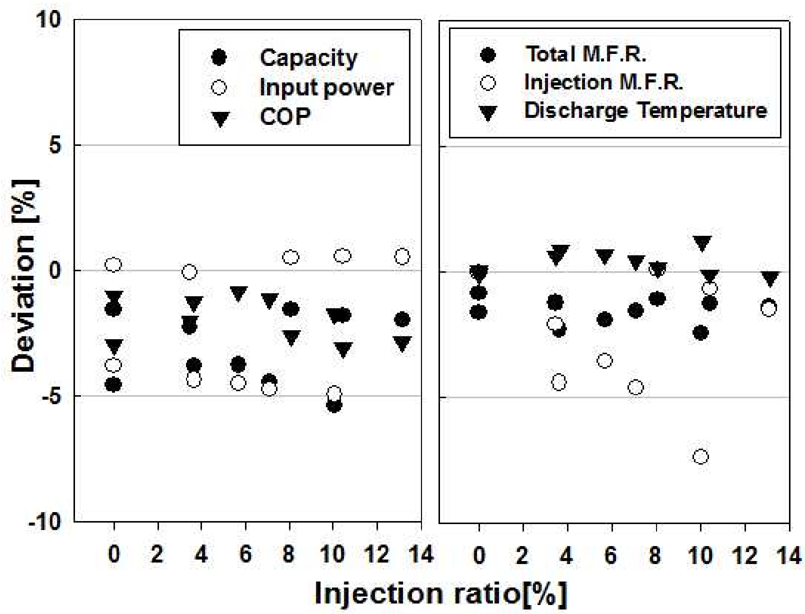

Figure 2: Comparison of the predicted data with measured data.

3. RESULTS AND DISCUSSION

The liquid injection cycle was analyzed according to the injection ratio by the simulation model. Fig. 3 shows the variations of the mass flow rate, cooling capacity, input power and COP according to the injection ratio. The total mass flow rate increased with the injection ratio. The cooling capacity decreased slightly according to the injection ratio due to a decrease in the mass flow rate through the evaporator. The COP also decreased slightly with the injection ratio. Fig. 4 shows the variations of the intermediate pressure and discharge temperature with the injection ratio. The intermediate pressure increased due to the decrease in the pressure drop according to the EEV opening. The discharge temperature decreased effectively with the increase in the injection ratio.

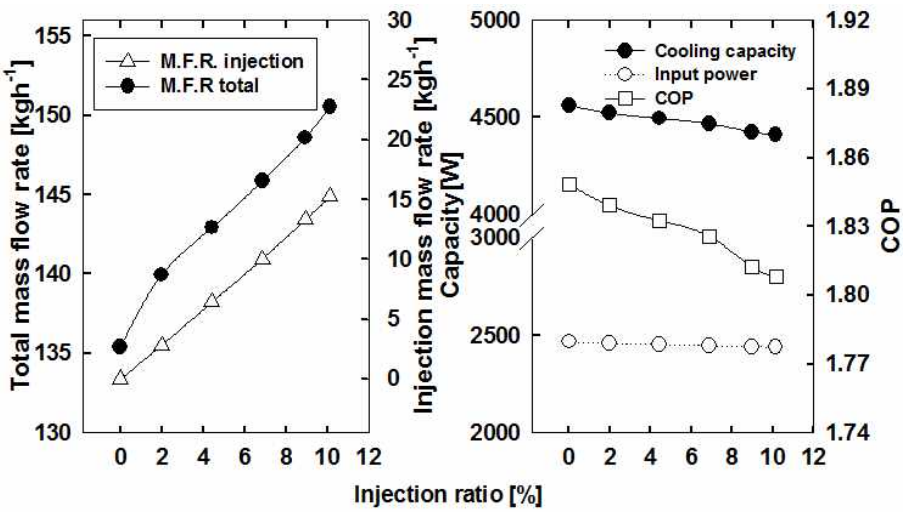

Figure 3: Mass flow rate, capacity, input power, and COP with injection ratio by simulation model.

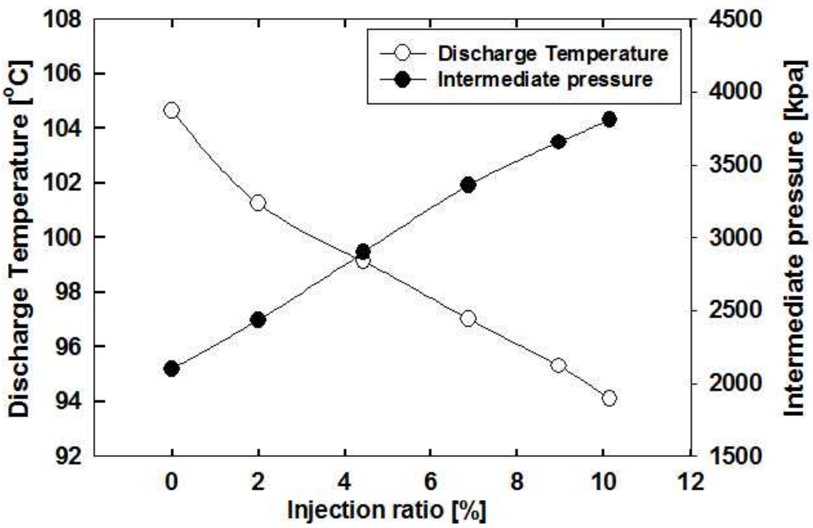

Figure 4: Discharge temperature and intermediate pressure with injection ratio by simulation model.

4. CONCLUSIONS

A simulation model for the liquid injection cycle with a twin rotary compressor was developed and validated. The performance of the twin rotary compressor with liquid injection was estimated using the loss and efficiency model. The convergence of the model was checked by three balances of mass and energy. The simulation model was validated by the experimental data with satisfactory agreement. In addition, the performance characteristics of the liquid injection were analyzed according to the injection ratio by the simulation model. The COP decreased slightly and the discharge temperature decreased effectively with the injection ratio.

NOMENCLATURE

h	enthalpy	(J/kg)
$\dot{m}$	mass flow rate	(kg/s)
P	pressure	(kPa)
q	heat transfer rate	(W)
Q	air flow rate (m^3/min)	

Subscripts

Seg	segment
In	inlet
Out	outlet
vol	volumetric
rpm	revolution per minute
rps	revolution per second
compin	compressor inlet
inj	injection
sat. eva	saturation evaporation

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