Full and Partial Load Performances of RAC and CAC Heat Pump using R-32

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Abstract: Refrigerant with high GWP such as R-410A for residential and commercial heat pump systemsmight bereplaced with alternative refrigerants with low GWP. R-32 with the GWP of 675 might be one of strong candidates. Present study investigated the performances of a residential and commercial heat pump system using R-32 and R-410A under full and partial load conditions.Optimum refrigerant charge amounts for both RAC and CAC systems using R-32were less by 20~30% than those for R-410A system. EER of R-32 was higher by approximately 3.8%~9.9% for RAC system and 3.3%~13.3 for CAC system than those of R-410A RAC and CAC system, respectively, when the capacities of the system using three different refrigerants were set the same.SEERusing R-32 was larger by 6 -15% than that using R-410A, while SCOP using R-32 was larger by 2 - 12% for the RAC system. SEER using R-32 was larger by 1 -13% than that using R-410A, while SCOP using R-32 was larger by 1 -4% for the CAC system.

Key Words: Heat pump, R-32, Partial load, SEER, SCOP

1 INTRODUCTION

Residential heat pump and air-conditioner (RAC) using R-410A has been investigated alternative refrigerant with low Global Warming Potential (GWP) since R-410A has high GWP of 2088. Alternative refrigerants with low GWP might be R-32 as pure refrigerant since it has the GWP of 675.R-32 shows lower total equivalent warning impact (TEWI) than R-410A (Kagawa 2012). TEWI might be calculated with the total effect of direct emissions and indirect effect of energy used forspecified period of time.
There are lots of literatures on the heat pump system using R-32. Performances of air-water heat pump system using R-410A with cooling capacity of 1.8~2.4kW and heating capacity of 1.6~2.4kW were compared with those using R-32(Koyama et al 2010). Heat pump using R-32 showed optimal refrigerant charge less by 20% than that using R-410A. Heating and cooling performances using R-32 were larger by 8~11% than those using R-410A, while compressor discharge temperature of heat pump using R-32 was higher by 20~30℃ than that using R-410A. Performance of air-water heat pump system with the capacity of 4.5kW system by using R-410A and R-32 was also compared(Huang et al 2011). Optimal refrigerant charge of the heat pump using R-32 was less by 35% than that using R-410A. Cooling and heating capacities of the heat pump using R-32 were larger by 2.5% and 14.5% than those using R-410A respectively, while compressor discharge temperature of the heat pump using R-32 was larger by 15 ~ 20℃ than that using R-410A. Drop-in test on the performance of a residential air conditioner using R-410A was performed by substituting with

R-32 (Zhuang et al 2011). Optimal refrigerant charge of the heat pump using R-32 was lower by 25% than that using R-410A. Cooling capacity of the system using R-32 was larger by 1.2% than that using R-410A, while heating capacity of the heat pump using R-32 was larger by 2.7% than that using R-410A. Compressor discharge temperature of the system using R-32 was higher by 15~35℃ than that using R-410A. Heat pump using R-32 with cooling capacity of 4kW and heating capacity of 5kW was tested (Hara et al 2012). Optimal refrigerant charge of the heat pump using R-32 was lower by 25% than that using R-410A. Cooling capacity of the heat pump using R-32 was larger by 2% than that using R-410A, while heating capacity of the heat pump using R-32 was larger by 5% than that using R-410A. Performance of the heat pump system with the cooling capacity of 17.6kW using R-32 was tested(Barve et al 2012). Tests at high outdoor temperature of 43℃ as well as cooling and heating standard modes were performed. Power at cooling and heating standard modes by using R-32 were lower by 7%~9% than those by using R-410A, while compressor discharge temperature of the system using R-32 was higher by 20~30℃ than that using R-410A.

Capacities of R-32 system were generally larger by 2.5~11.5% than those of R-410A system, while refrigerant charge of R-32 system was ranged within 75~80% of the R-410A system. Discharge temperature of compressor of the R-32 system was larger by 20~25% than that of the R-410A system. Since R-32 system showed high discharge temperature of compressor, change of oil, increase of compressor discharge volume might be suggested to decrease the discharge temperature of the compressor.

As literature on commercial air conditioning system(CAC system), heat pump using R-32 with capacity 10.6kW was tested (Palmiter et al 2011). Performance of R-32 was higher by 3~4% under cooling mode, higher by 1~3% under heating mode than those of R-410A. Air-cooled water chiller using R-32 with cooling capacity 10kW was tested (Schultz et al 2012). Cooling capacity of R-32 was higher by 2% than that of R-410A. Heat pump using R-32 with cooling capacity of 12.5kW and heating capacity of 13.3kW was also tested (Crawford et al 2012). Performance of R-32 system was higher by 1~2% than that of R-410A system.

Most of literatures reported the data by drop-in test at full load condition. Partial load performances of the heat pump using low GWP refrigerants might be investigated. Present study aimed to investigate the partial load performance of the residential and commercial heat pump using R-32.

2 EXPERIMENTAL APPARATUS AND PROCEDURE

Residential and commercial heat pump system with cooling capacity of 3.5kW and 10kW respectively were utilized for drop-in test. Figure 1 shows the schematic diagram of the experimental apparatusas an air-to-air heat pump system. Refrigerant flow loop was reversed by a 4-way valve under cooling and heating conditions. Refrigerant loop was consisted of an inverter type rotary compressor, a 4-way valve, a condenser, a solenoid expansion valve, a mass flow meter, and an evaporator. Mass flow meter was installed at the outlet of the condenser to measure the refrigerant mass flow rate under heating and cooling modes. Test was performed for R-410A, R-32. Table 1 shows thermo-physical properties of R-410A and R-32 at both 47℃ and 7℃. Table 2 shows the partial load test conditions under heating and cooling modes according to EN 14825.

Cooling and heating capacities of the residential and commercial heat pump were measured by air enthalpy method applied for the indoor and outdoor units of psychrometric calorimeter. Opening level of electric expansion valve(EEV) and inverter frequency of the rotary compressor was controlled to keep the same capacity for three different refrigerants. Fan speed of indoor unit was kept constant regardless of partial load test condition, while fan speed of outdoor unit was updated according to SEER and SCOP conditions. Compressor frequency was controlled to keep pre-determined capacity through the drop-in test.

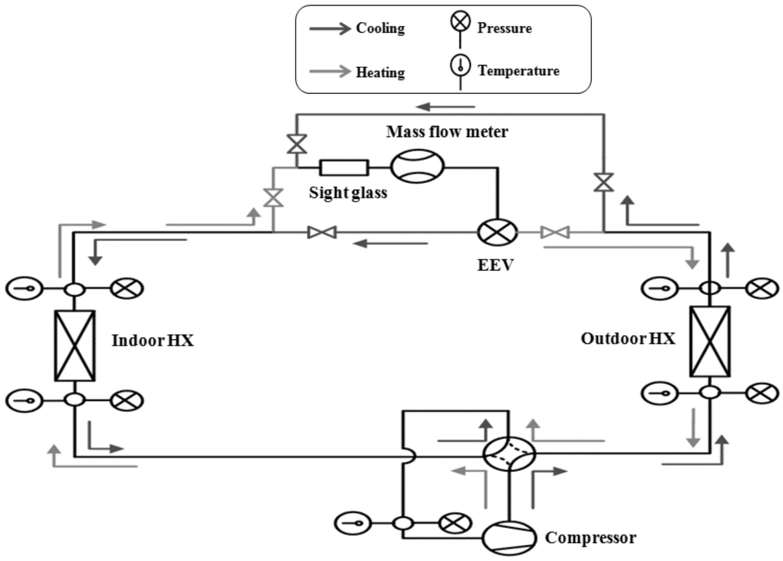


Figure 1: Schematic diagram of the experimental apparatus

Compressor displacement was adjusted due to comparably large latent heat of R-32. Compressor displacement was decreased by approximately 3~4% for R-32 RAC and CAC system in order to maintain the same capacity with R-410A system. Data were gathered every 2 seconds until it reached steady condition. Uncertainty of the measured data was estimated according to the uncertainty analysis (Moffat 1988). Detailed specifications of measuring instruments and uncertainties are shown in Table 3.

Table 1: Thermo-physical properties of test refrigerants

	R-410A		R-32	
T (℃)	47	7	47	7
ρ_f (kg/m^3)	928.9	1141.2	856.4	1030.6
ρ_v (kg/m^3)	126.4	37.51	90.05	27.56
h_{fg}(kJ/kg)	146.6	217.3	218.4	304.0
$C_{p,f}$(kJ/kgK)	2.134	1.5590	2.338	1.786
$C_{p,v}$(kJ/kgK)	2.201	1.213	2.304	1.329
k_f (mW/m□K)	77.23	99.08	109.3	139.7
k_v (mW/m□K)	23.04	12.84	21.38	12.45
μ_f(μPa.s)	86.01	148.2	86.72	139.2
μ_v(μPa.s)	15.66	12.57	14.43	11.85
Pr	2.377	2.331	1.855	1.779

3 DATA REDUCTION

Heat transfer rates of condenser and evaporator, Q_c and Q_e, were estimated by using equations (1) and (2) by putting measured mass flow rate (of refrigerant and estimated enthalpy . Enthalpies of refrigerant were found by applying measured data of temperature and pressure for Refprop (version 9.0) developed by the NIST. Air-side heat transfer rate ($\dot{Q}_a$) was obtained by equation (3). Mass flow rate of air ($\dot{m}_a$) was obtained by using density of air and air flow rate, while enthalpies of air were obtained by using temperature of air and relative humidity at both inlet and outlet. Compressor power was obtained by using equation (4). EER was obtained by substituting heat transfer rate in equations (1) and (2) and compressor power in equation (4) into equation (5).

$$\dot{Q}_c = \dot{m}_r\left(h_{r,in} - h_{r,out}\right) \tag{1}$$

$$\dot{Q}_e = \dot{m}_r(h_{r,out} - h_{r,in}) \tag{2}$$

$$\dot{Q}_a = \dot{m}_a(h_{a,in} - h_{a,out}) \tag{3}$$

$$\dot{W}_{com} = \dot{m}_r(h_{r,out} - h_{r,in}) \tag{4}$$

$$\mathrm{EER} = \frac{\dot{Q}_e}{\dot{W}_{com}+\dot{W}_{fan}} \tag{5}$$

Table 2: Test Conditions

		DB(℃)	WB(℃)	DB(℃)	WB(℃)
Cooing	SEER A (Std.)	27	19	35	24
	SEER B			30	24
	SEER C			25	24
	SEER D			20	-
Heating	Standard	20	15	7	6
	SCOP T			-10	-11
	SCOP A			-7	-8
	SCOP B			2	1
	SCOP C			7	6
	SCOP D			12	11

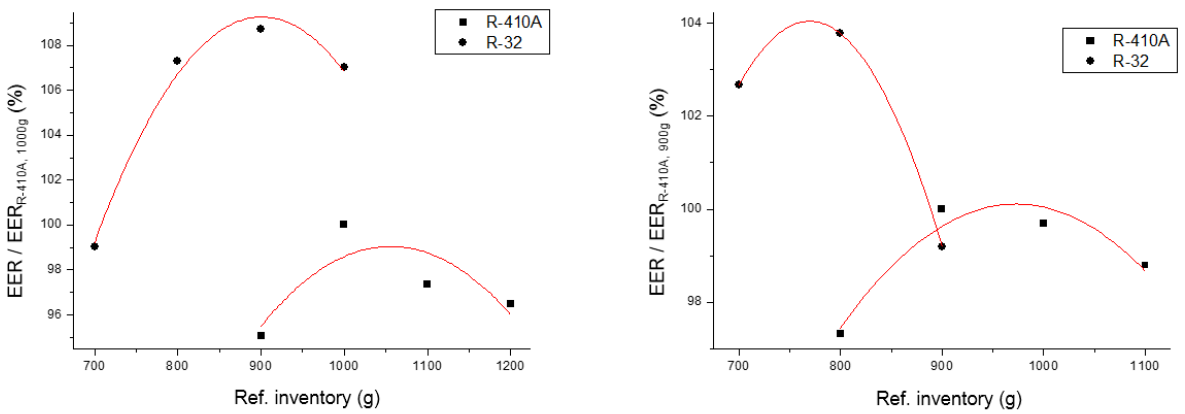
4 RESULTS AND DISCUSSION

Figure 2 showed RAC system performance according to refrigerant inventory under cooling and heating standard modes. R-410A showed the best EER at the refrigerant charge of 900g under heating mode and 1000g under cooling mode. Ratio of EER for R-32to the best EER for R-410A for the RAC systemshowed the best EER at the refrigerant charge of 800g under heating mode and 900g under cooling mode. Optimal charge of R-32 RAC system was less by approximately 20%~30% than that of R-410A RAC system.

Figure 3 showed CAC system performance according to refrigerant inventory under cooling and heating standard modes. R-410A showed the best EER at the refrigerant charge of 3000g under heating mode and 3100g under cooling mode. Ratio of EER for R-32 to the best EER for R-410A for the RAC system showed the best EER at the refrigerant charge of 2100g under heating mode and 2100g under cooling mode. Optimal charge of R-32 CAC system was also less by approximately 20%~30% than that of R-410A CAC system.

Table 3: Specification of measuring instrument

	Instrument	Measurement		Uncertainty
refrigerant side	thermocouple (T-type)	T_r(℃)	-200 ~ 300	± 0.15
	pressure gauge	P_r (Bar)	0~35	±0.04%
	mass flow meter	$\square_r$(kg/min)	0 ~ 5	± 0.1%
air side	dew point meter	T_a(℃)	-10~60	± 0.2℃
	hygrometer	Φ_a (%)	5~98	± 0.8%
	air flow meter	E_a (m^3/min)	5.5~60	± 2%

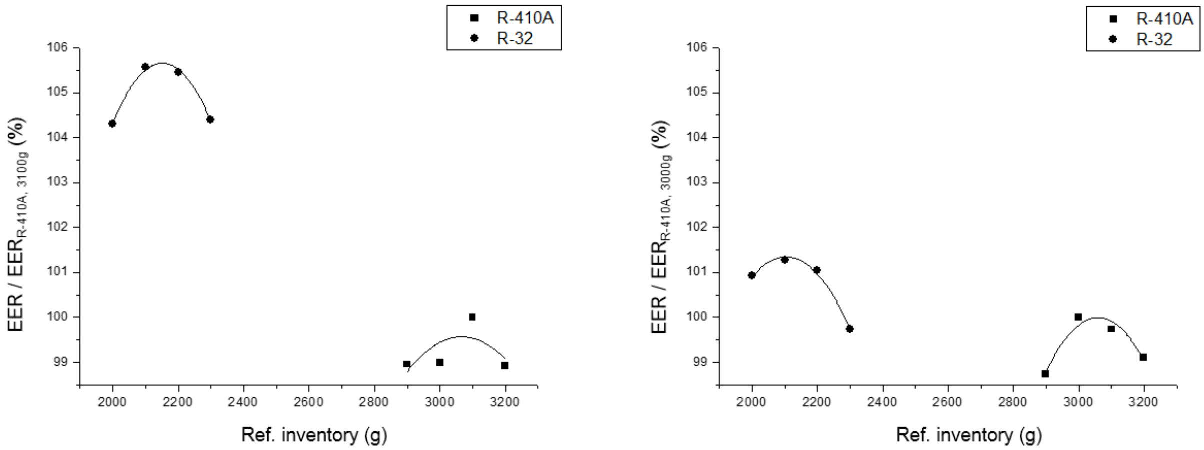


(a) Cooling standard mode (b) Heating standard mode

Figure 2: RAC system performance according to refrigerant inventory under cooling and heating standard modes

Reduction of refrigerant charge of R-32 was due to the difference of thermo-physical properties of R-410A and R-32. Latent heat and specific volume of R-32 were higher by 43% and 40% than those of R-410A, respectively.

Figures 4, 5, 6 and 7showed comparison of $\dot{Q}$, EER, G and compressor discharge temperature for RAC and CAC systems under cooling and heating standard modes by setting equalized capacity for R-410A and R-32.



(a) Cooling standard mode (b) Heating standard mode

Figure 3: CAC system performance according to refrigerant inventory under cooling and heating standard modes

EER of R-32 RAC system was higher by approximately 3.8~9.9% than that of R-410A RAC system.EER of R-32 CAC system was higher by approximately 3.3%~13.3 than that of R-410A CAC system. Since compressor input powers for R-32 RAC and CAC systems were decreased to keep the same capacity with R-410A RAC and CAC systems, its EER was increased.

Compressor discharge temperatures of R-32RAC and CAC systems were higher than those of R-410A RAC and CAC systems, because of different compressor suction specific volume and specific heat ratio. They went up to 82.7℃ and 81.5℃ for RAC system, 84.2℃ and 73.6℃ for CAC system under cooling and heating standard modes, respectively.

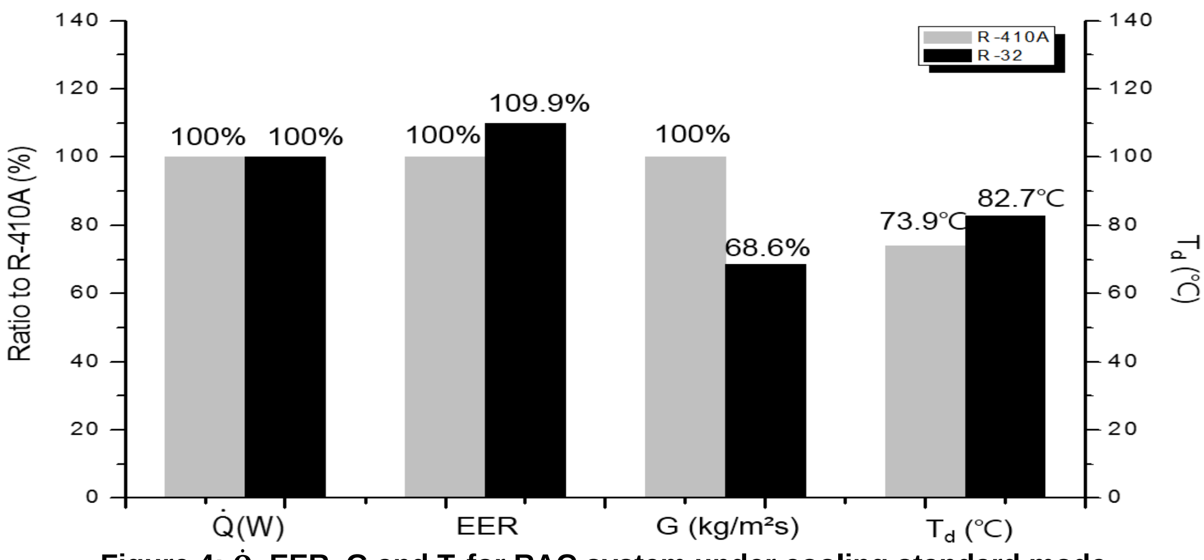


Figure 4: $\dot{Q}$, EER, G and T_d for RAC system under cooling standard mode

Table 4 and 5 showed partial-load performancesof RAC and CAC systemsaccording to EN14825.

Partial load cooling and heating performances except D condition of R-32 RAC system were higher by 6~15% and 2~12% than those of R-410A RAC system, respectively. Partial load cooling and heating performances except D condition of R-32 CAC system were higher by 1~13% and 1~4% than those of R-410A CAC system, respectively.

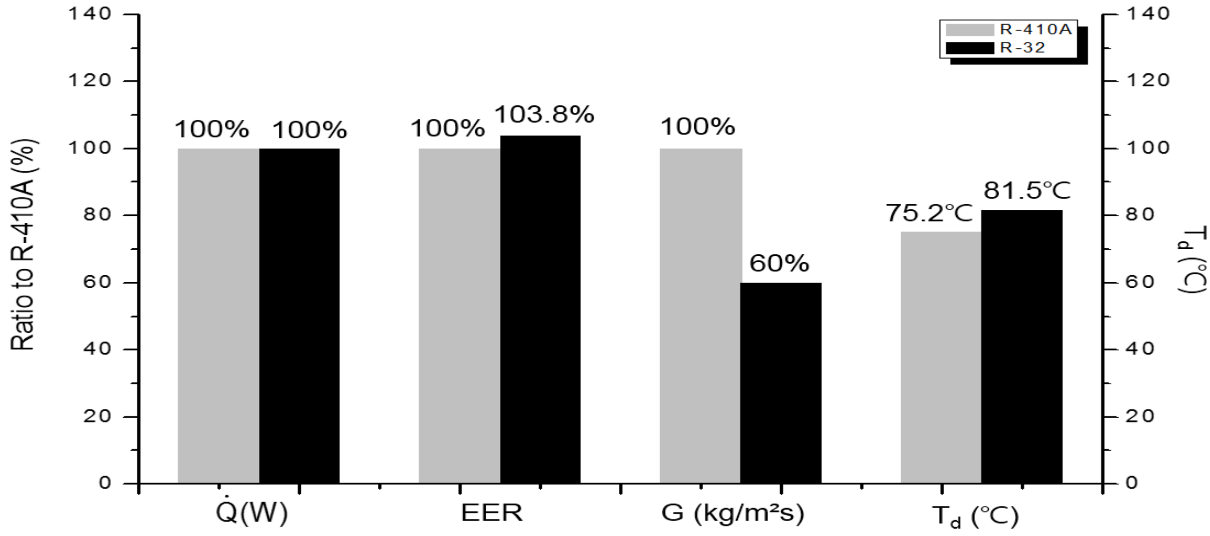


Figure 5: Q̇, EER, G and T_d for RAC system under heating standard mode

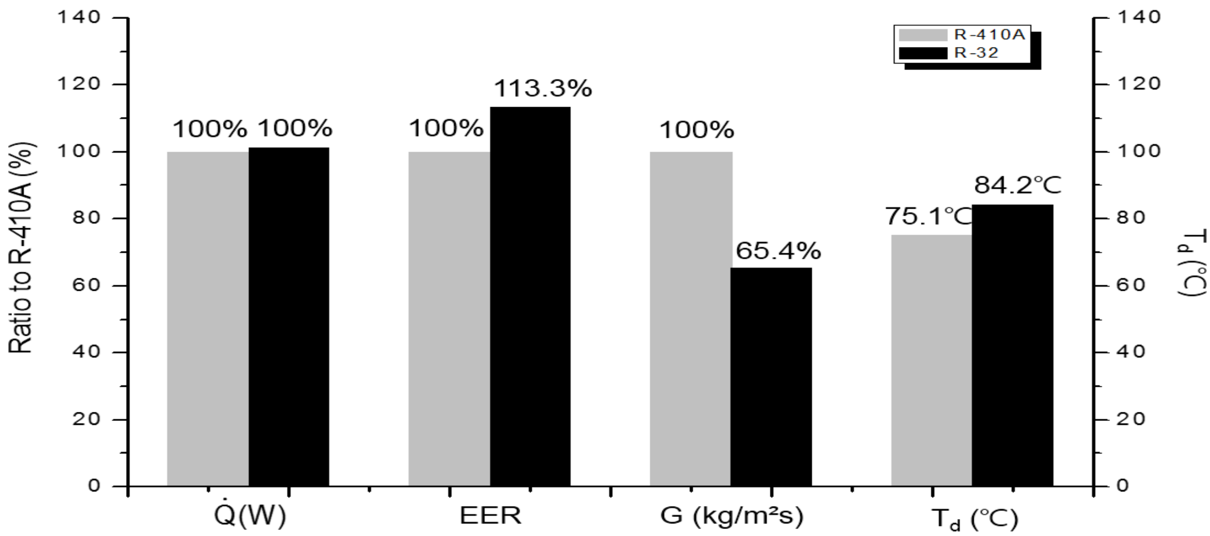


Figure 6: Q̇, EER, G and T_d for CAC system under cooling standard mode

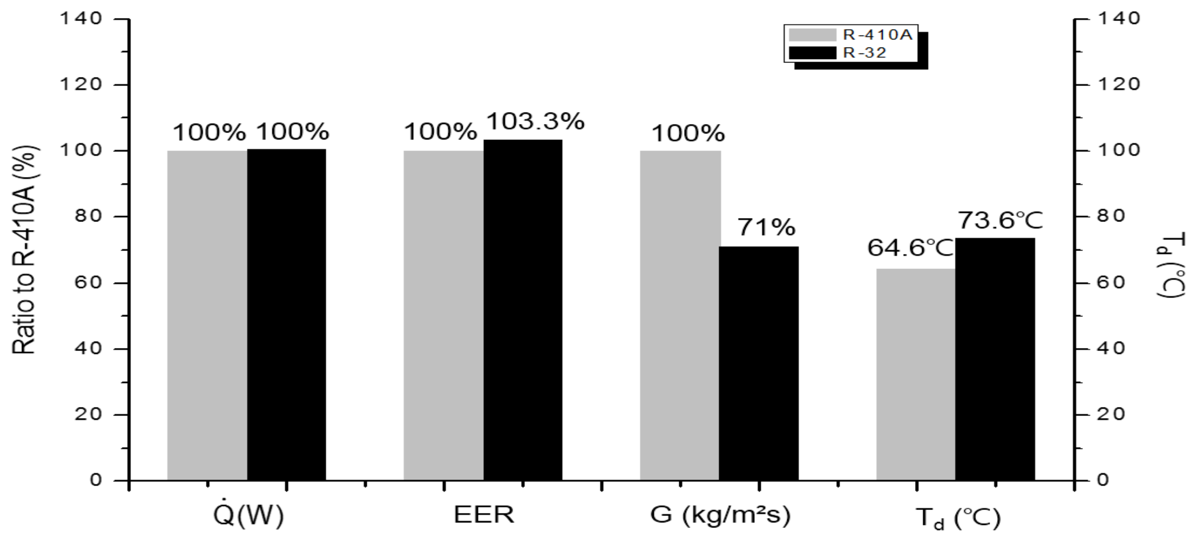


Figure 7: Q̇, EER, G and T_d for CAC system under heating standard mode

Partial load performances of R-32 system were higher than those of R-410A system. It was due to the high latent heat of R-32. High latent heat caused high capacity. When capacities of R-32 RAC and CAC systems were similar than those of R-410A RAC and CAC systems, input power of R-32 RAC and CAC systems were decreased by 11%, and thus EER was higher than R-410A RAC and CAC systems.

Table 4: Partial load performances of RAC system according to EN14825

	R-32 / R-410A	
SEER Condition (Cooling)	$\dot{Q}$	EER
A	100%	115%
B	100%	106%
C	100%	108%
D	100%	111%

	R-32 / R-410A	
SCOP Condition (Heating)	$\dot{Q}$	EER
T	96%	102%
A	98.5%	105%
B	100%	105%
C	99%	112%
D	123%	131%

Table 5: Partial load performances of CAC system according to EN14825

	R-32 / R-410A	
SEER Condition (Cooling)	$\dot{Q}$	EER
A	101.2%	113.3%
B	100.7%	104.8%
C	100.5%	101.7%
D	96.9%	100.3%

	R-32 / R-410A	
SCOP Condition (Heating)	$\dot{Q}$	EER
T	99%	99.3%
A	101.1%	98.1%
B	102.4%	103.7%
C	101.3%	100.9%
D	96.6%	104.5%

4 CONCLUSIONS

The present study might be summarized as follows.

(1) Optimal charge of R-32 RAC and CAC systems was less by approximately 20%~30%than that of R-410A RAC and CAC systems.

(2) EER of R-32 was higher by approximately 3.8%~9.9% for RAC system and 3.3%~13.3for CAC system than those of R-410A RAC and CAC system, respectively, when the capacities of the system using three different refrigerants were set the same.

(3) Discharge temperatures of R-32 were higher by the maximum of 8.8℃forRAC systems, 9.1℃for CAC system than those of R-410A RAC and CAC system, respectively.

(4) SEERusing R-32 was larger by 6 -15% than that using R-410A, while SCOP using R-32was larger by 2 - 12% for the RAC system. SEER using R-32 was larger by 1 -13% than that using R-410A, while SCOP using R-32 was larger by 1 -4% for the CAC system.
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NOMENCLATURE

C_p	Specific heat (kJ/kg-K)
G	Mass flux (kg/m^{2}s)
h	Enthalpy (kJ/kg)
k	Thermal conductivity (mW/m□K)
$\dot{m}$	Mass flow rate (kg/s)
P	Pressure (Pa)
Pr	Prantl number
$\dot{Q}$	Heat transfer rate (W)
T	Temperature (℃)
W	Power(W)

Greek symbols

μ	Viscosity (μpa.s)
ρ	Density (kg/m^3)

Subscripts

A	Air
c	Condenser
com	Compressor
d	Discharge
e	Evaporator
f	Liquid
in	Inlet
out	Outlet
r	Refrigerant
v	Vapor

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