Variations in the Thermal Performance of R22 and R410A Refrigeration Systems Depending on Operation Conditions

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ABSTRACT: Experiments have been conducted in order to make comparisons of characteristics of a R410a cycle with a R22 cycle in terms of cooling capacity and coefficient of performance (COP). The parameters examined in the present work include air flow rate, indoor and outdoor air temperatures, and indoor relative humidity. These two refrigeration cycles constructed for this study share all components except compressor, accumulator, oil separator, and piping. The measurements were made using a psychrometric calorimeter. The experimental results show that the R410A cycle has several advantages for indoor units while the R22 cycle yields better performance for outdoor units.

Nomenclature

C : specific heat [kJ/kg℃]
H : relative humidity [%]
k : thermal conductivity [w/m℃]
M : refrigerant charging amount [kg]
P : pressure [MPa]
Q : cooling capacity [W]
T : temperature [℃]
V : air volume flow [m^3/min]
v : specific volume [m^3/kg]

Greek symbols

μ : viscosity [kg/ms]
Φ : latent heat [kJ/kg]

Subscripts

con : condensation
crit : critical
di : dry, indoor
do : dry, outdoor
eva : evaporation
f : saturated liquid
g : saturated gas
i : indoor
o : outdoor
ref : reference
wi : wet, indoor
wo : wet, outdoor

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1. Introduction

A conventional refrigerant of R22 has long been used in a variety of air-conditioning and refrigeration applications. Due to the concerns on environmental problems such as ozone layer depletion, this hydrochloro-fluorocarbon (HCFC) refrigerant is being substituted by R407C (R-32/125/134a, 23/25/52 wt%) and R410A (R-32/125, 50/50 wt%) which belong to hydrofluorocarbon (HFC) group. The domestic HVAC equipment industry also has been developing R410A refrigeration cycles.

The R410A has a merit that the temperature glide is 0.1℃ so that it may be treated as a single component refrigerant. On the other hand, it has a demerit that the operating pressure of R410A refrigeration cycle is 50~60% higher than that of R22. This high operating pressure requires redesign of the system. The thermodynamic properties of R410A and R22 are listed in Table 1. The operating pressure, liquid specific volume, and latent heat of R410A are larger than those of R22 by 60%, 16%, and 6%, respectively, while critical pressure is lower. In general, refrigerants having low critical pressure show low coefficient of performance (COP) and large refrigeration capacity.[1] In addition, for the same volume flow rate of refrigerant, head loss R410A is smaller because of larger gas density.[2] Also, it is possible to reduce system size because the compressor displacement volume for the same capacity is smaller by 30%.[3] The R410A has merits that the ozone depletion parameter (ODP) is zero and global warming parameter (GWP) is lower than that of R22.

Table 1 Properties of R410A and R22

Refrigerant	R22	R410A
T_{crit} [℃]	96	72.5
P_{eva} (at 7℃) [MPa]	0.622	0.992
P_{con} (at 46℃) [MPa]	1.77	2.789
ϕ (at 7℃) [kJ/kg]	199.2	212.2
ϕ (at 46℃) [kJ/kg]	159.4	146.9
v_g (at 7℃) [m^3/kg]	0.03796	0.02629
v_f (at 46℃) [m^3/kg]	0.00091	0.00106
k_g (at 7℃) [W/m℃]	0.00991	0.0125
k_f (at 46℃) [W/m℃]	0.07413	0.08327
μ_g (at 7℃) [kg/ms]	11.82	12.72
μ_f (at 46℃) [kg/ms]	129.5	88.74
C_g (at 7℃) [kJ/kg℃]	0.771	1.175
C_f (at 46℃) [kJ/kg℃]	1.384	2.076
ODP	0.05	0

Refrigeration cycles using R410A and R22 have been investigated by many researchers. Kim and Lee[4] studied the performance variation in an air-conditioner using R410A depending on outdoor dry-bulb temperature change. This study showed that R410A cycle was inferior to a R22 cycle in cooling capacity and energy efficiency ratio (EER) when outdoor dry-bulb temperature was above 35℃ (relative humidity was constant at 50%) while the former was superior to the latter when the outdoor temperature was below 35℃. They claimed that the discharge pressure of the R410A cycle increased larger than the R22 cycle as the outdoor temperature increased and this caused a deterioration in performance of the R410A cycle. Since gas specific volume of R410A is smaller than that of R22, R410A cycle will experience less pressure drop in low pressure side piping. They claimed that this effect leaded to an increase in EER of the R410A cycle by 1% compared with the R22 cycle under test conditions of Korean Industrial Standards (KS C 9306). Murphy et al.[5] compared the performances of R410A, R407C, and R22 refrigeration cycles under performance testing conditions by Canadian Standards Association (CSA). They performed experiments under four test conditions. They reported that the R410A cycle showed equivalent or superior cooling capacity to the R22 cycle in all conditions, while the COP of the R410A cycle is inferior to the R22 cycle in three among the four test conditions. They also showed that COP and cooling capacity of the R410A cycle were reduced drastically while those of the R22 cycle were little affected when refrigerant was overcharged. Cho et al.'s experimental results[6] also showed that refrigerant overcharge in R410A cycle reduced cooling capacity and increased the consumption of electricity.

The previous studies on alternative refrigerant cycles have been focused on component technologies such as heat transfer coefficient, pressure drop, heat exchanger design, but less interested in the characteristics of cycle performance. In this regard, the present work aims to compare performance characteristics of R22 and R410A refrigeration cycles depending on indoor and outdoor temperatures, refrigerant charge, indoor humidity, and indoor air flow rate.

2. Description of experiments

2.1 Experimental apparatus

A schematic diagram of the refrigeration cycle is shown in Fig. 1. The condenser consists of 9.5 mm O.D. tube and super slit fins. Its capacity is 2 refrigeration ton (RT). The evaporator consists of 7.0 mm O.D. tube and louvered fins, and also has 2 RT capacity. The R22 cycle and the R410A cycle constructed for the present study share the condenser, evaporator, electronic expansion valve (EEV), receiver, filter dryer, and sight glass. However, different compressor, accumulator, and oil separator were used for each cycle. The R410A cycle makes use of a rotary type compressor of which capacity is 1.5 US RT, while R22 cycle uses reciprocating type compressor whose capacity is 2.25 US RT. The R410A compressor operates at constant speed while the R22 compressor has an inverter so that the speed is controllable.

Temperatures and pressures are monitored at five locations. The temperatures are measured using RTDs (PT 100 Ω) whose accuracy is ±0.1 ℃. The pressures are read using C206 pressure transducers from Setra Co., whose range is −0.1~6.9 MPa. Refrigerant flow rate is measured using a mass flowmeter whose range is 15~500 L/hr. The condenser's fan speed is controlled by a voltage transformer. The refrigeration cycle performance is precisely measured using a psychrometric calorimeter, which is well described in Lee et al.[7] The accuracy of temperature and pressure measurements are ±0.1℃ and ±0.1%, respectively.

2.2 Experimental methods

Experiments to decide optimal refrigerant charge are first carried out. Then, the standard condition experiment is performed in order to compare the performance between two cycles. After these two basic experiments are completed, main experiments are performed. The

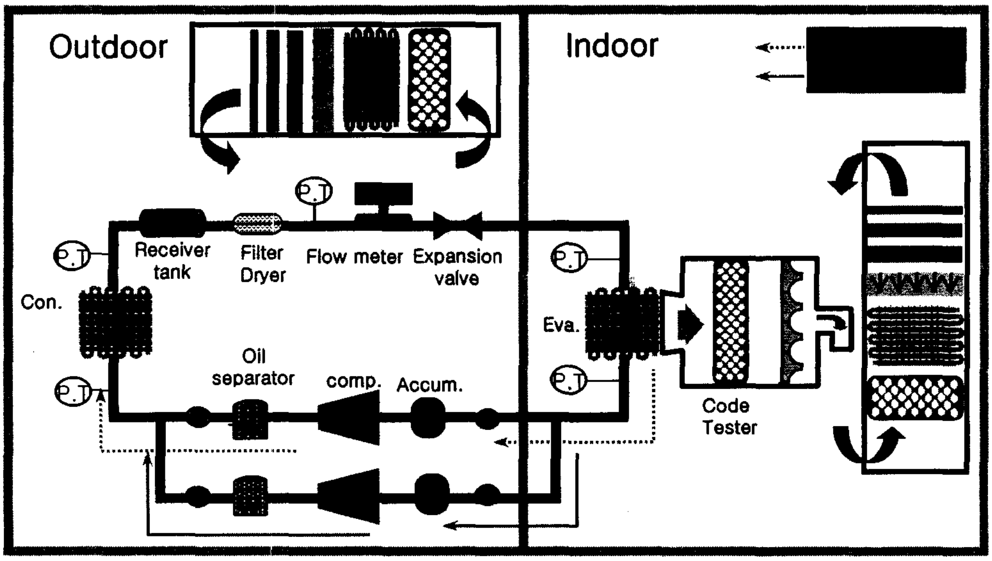


Fig. 1 Schematic diagram of refrigerant cycle.

Table 2 Experimental conditions for each test

Conditions	Refrigerant charge		Standard condition		Outdoor drybulb temperature		Indoor drybulb temperature		Indoor air flow		Indoor relative humidity	
	22	410A	22	410A	22	410A	22	410A	22	410A	22	410A
M(kg)	variable		3.95	4.27	3.95	4.27	3.95	4.27	3.95	4.27	3.95	4.27
V_{eva}(CMM)	14.7	12.3	12.3		14.7	12.3	14.7	12.3	variable		14.7	12.3
V_{con}(rpm)	797	788	788		797	788	797	788	797	788	797	788
T_{di}(℃)	27		27		27		variable		27		27	
T_{wi}(℃)	19.5		19.5		19.5		-		19.5		-	
H_i(%)	-		-		-		50		-		variable	
T_{do}(℃)	35		35		variable		35		35		35	
T_{wo}(℃)	24		24		-		24		24		24	
H_o(%)	-		-		50		-		-		-	

main experiments compare the cooling capacity and COP depending on variations in indoor and outdoor temperatures, indoor humidity, and condenser air flow rate.

Table 2 summarizes the experimental conditions. Indoor/outdoor temperature and humidity are determined based on Korean Industrial Standards (KS C 9306). Evaporator air flow rate is determined such that the R22 cycle shows the best performance under standard conditions for cooling in KS C 9306. It is a practical experience that the R22 cycle shows the highest efficiency when operated such that the evaporator exit is superheated by 5℃. This implies that evaporator exit temperature is 12℃ since the R22 cycle operates at 5.3 atmospheric pressure (7℃). In this regard, the evaporator air flow is selected such that the evaporator exit temperature maintains at 12℃. On the other hand, condenser air flow is decided such that condenser exit subcooling becomes 1℃. Refrigerants are charged with optimal amount of refrigerants through all experiments.

Thermo-physical properties of refrigerants are evaluated using REFPROP 6.01.[8] The heat transfer rate is measured from the refrigerant side as well as from the air side. When saturated refrigerant enters evaporator, enthalpy at evaporator inlet is substituted by the value at EEV inlet. Through the present experiments, the heat balance between air-side and refrigerant-side was maintained within 6% for the R22 cycle and 5% for the R410A cycle.

3. Results and discussions

The performance comparisons between the two refrigeration cycles are made in terms of cooling capacity (Q) and COP. Two types of COP are used in this work. The first one is calculated based on electricity consumption (coefficient of performance-electric power: COP_E), which is conventional. The second one is based on enthalpy gained from compressor (coefficient of performance-hydraulic power: COP_H). The R22 cycle and R410A cycle use different compressors as mentioned in the previous section. In general, different compressors have different characteristics in terms of efficiency and electricity consumption. These differences in com-

Table 3 Conditions for reference experiment

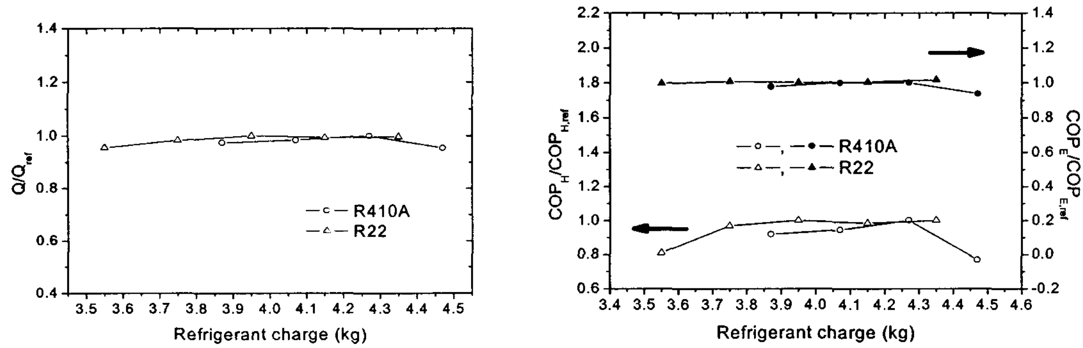
Parameter	Values
Outdoor drybulb temperature (℃)	35
Indoor relative humidity (%)	50
Indoor drybulb temperature (℃)	27
Indoor airflow (CMM)	8
Refrigerant charge (kg)	3.95 for R22 4.27 for R410A

pressors may make the cycle performance comparison unclear. In order to exclude the effect of using different types of compressor, COP_H is also used in the present work. In this study, all measurements are presented in terms of ratio to the reference values. These values are obtained from a reference experiment at conditions specified in Table 3.

3.1 Effect of refrigerant charge

The test conditions for finding optimal refrigerant charge are listed in the column titled "Refrigerant charge" in Table 2. The response of the two cycles depending on change of the refrigerant charge are plotted in Fig. 2. The Q_{ref} and the COP_{ref} correspond to the measurements made at the optimal refrigerant charging amount. R22 and R410A cycles show the highest performance when refrigerant is charged with 3.95 kg and 4.27 kg, respectively. If the charged mass is less or larger than this values, cooling capacity and COP tends to decrease. Based on these observation, these values are optimal in the present cycles. Optimal charging mass of R410A is larger than R22 charge by 320 g in spite of the fact that liquid density of R410A is larger than that of R22 by 16%. The cause is speculated to be that the system pressure and gas density of R410A cycle is larger than those of R22 cycle by 60% and 30%, respectively. Figure 2 also shows that the cooling capacity and the COP of R410A cycle drastically decrease if refrigerant charge exceeds an optimum value. This observation suggests that caution should be payed not to exceed optimal refrigerant charge.

It can be seen in Fig. 2 (b) that decrease in COP_H is larger than that in COP_E. Operation characteristics of compressors are believed to cause this trend. In this experiment, the compressor operated at constant speed so that variation in electricity consumption depending on load change was small. Since variations in cooling capacity and electricity consumption are small in the range of refrigerant charge variation, the change in COP_E appears to be small. On the other hand, the enthalpy of refrigerant obtained from a compressor is influenced by compressor efficiency. The enthalpy of refrigerant measured at entrance and exit of compressors showed that its increase rate in the R410A cycle was larger than that in the R22 cycle. This large enthalpy increase rate in R410A caused steeper decrease in COP_H for the R410A cycle. COP_E has been conventionally used as a performance index of refrigeration cycles. COP_H also needs to be used to compare different refrigeration cycles because



(a) Cooling capacity ratio (b) COP ratio

Fig. 2 Performance of R22 and R410A cycles versus refrigerant charge.

it is independent on the selection of compressor types. In this regards, both of COP_E and COP_H are used in the present work.

3.2 Standard condition experiment

A standard condition experiment based on KS C 9306 has been performed to compare two cycles using different types of compressor. In this experiment, evaporator air flow rate, condenser air flow rate, refrigerant temperature at evaporator exit, and inlet/outlet conditions of the air are set identical in both cycles. In order to make the refrigerant temperature at evaporator exit the same, the compressor speed and expansion valve opening are adjusted in R22 cycle while only expansion valve opening is adjusted in the R410A cycle. The test conditions for this experiment are listed in the column titled "Standard condition" of the Table 2. The evaluated cooling capacity and COP_H are summarized in Table 4. It is natural to get nearly the same cooling capacity from both cycles since inlet/outlet conditions for air are set to be identical. With the same cooling capacity, however, R410A cycle shows higher COP_H than R22 cycle by 6.6%. A smaller liquid viscosity of R410A causes a less frictional head loss, and speculated to lead to this higher COP. Based on this observation, it can be said

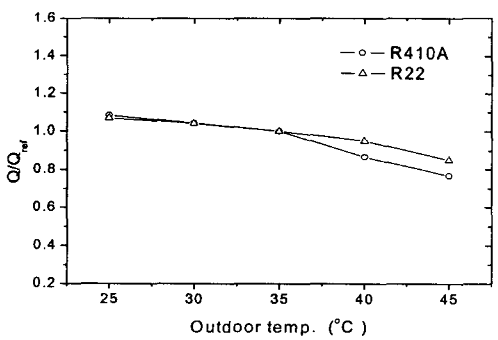
Table 4 Cooling capacity and COP_H under standard conditions

	R22	R410A
Cooling capacity (W)	5758.3	5727.4
COP_H (W/W)	3.84	4.1

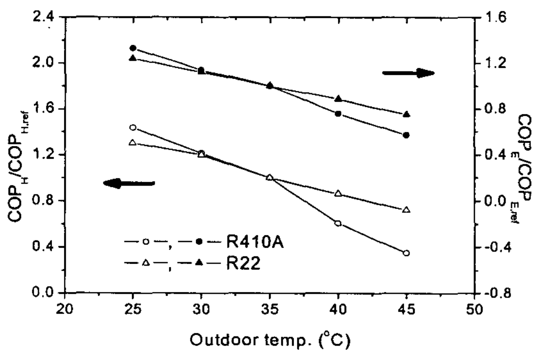
that R410A cycle will produce larger cooling capacity if a compressor of the same capacity is used.

3.3 Effects of outdoor dry-bulb temperature

Variations in performance of the both cycles depending on outdoor drybulb temperature change are examined. The test conditions for this experiment is listed in the column titled "Outdoor drybulb temperature" of Table 2 and the results are plotted in Fig. 3. Figure 3 (a) shows that the cooling capacity of the R410A cycle reduces more as the outdoor temperature increases over 35℃. The COP change shows the same trend as cooling capacity. This trend is the same as that reported by Kim and Lee.[4] Schematic P-h diagrams for the measurements in Fig. 3 at 25, 35, 45℃ are shown in Fig. 4. As the outdoor temperature increases, the evaporator pressures of both cycles experience mild increase while condenser pressures experience drastic increase. This increase in condenser pressure is larger in R410A than R22 so that the

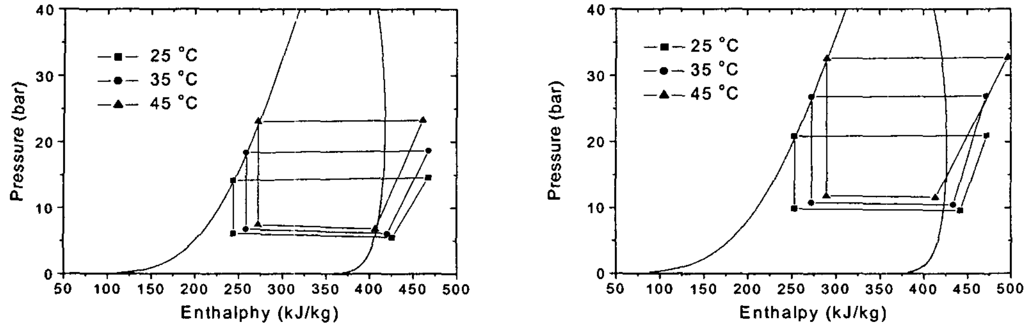


(a) Cooling capacity ratio



(b) COP ratio

Fig. 3 Performance of R-22 and R-410A cycles versus outdoor drybulb temperature.



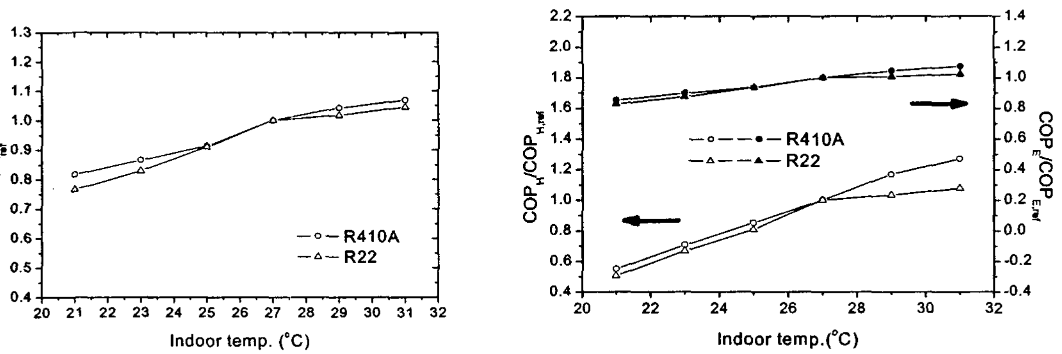
(a) R22 (b) R410A

Fig. 4 P-h diagrams for R22 and R410A cycles versus outdoor drybulb temperature.

pressure difference between condenser and evaporator increases further in the R410A cycle. In addition, Fig. 4 shows that the enthalpy difference between evaporator inlet and outlet of R410A reduces further than R22. These facts influence the R410A cycle to increased cooling capacity reduction ratio, increased compressor load, and eventually leads to a decrease in COP. Judged from these observations, the performance of the R410A cycle can be enhanced by using a larger condenser than the R22 cycle to augment condensing heat transfer capability.

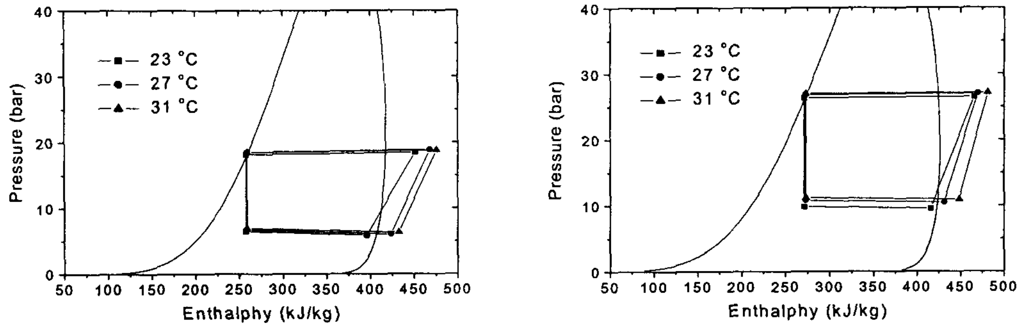
3.4 Effects of indoor dry-bulb temperature

Test conditions for the experiments of indoor temperature effects are listed in the column titled "Indoor drybulb temperature". The measured performance and P-h diagram are plotted in Fig. 5 and Fig. 6. The cooling capacity ratio and the COP of the R410A cycle appears to be slightly larger than those of the R22 cycle. When indoor temperature exceeds 27℃, the performance of R22 cycle shows a level-off while the performance of R410A cycle continues to linearly increase. This trend appears prominent in COP_H. This is because the evaporator exit enthalpy of R410A cycle linearly increases while that of R22 cycle levels off as the indoor temperature goes over 27℃. In addition, the evaporator pressure of R410A cycle increases while that of R22 cycle changes little with the increase in indoor temperature. Since there is little change in the condenser pressures of both cycles, reduction in the operation pressure difference between the condenser and the evapo-



(a) Cooling capacity ratio (b) COP ratio

Fig. 5 Performance of R22 and R410A cycles versus indoor drybulb temperature.



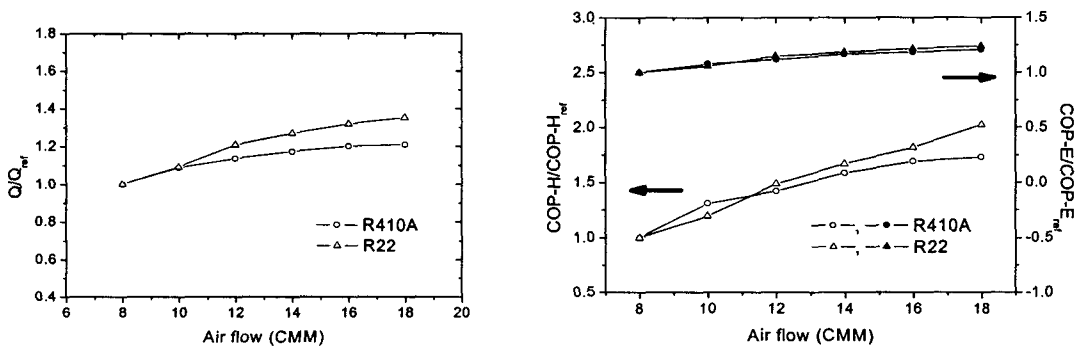
(a) R22 (b) R410A

Fig. 6 P-h diagrams for R22 and R410A cycles versus indoor temperature.

rator is apparent in R410A while little change is observed in R22 cycle. This means that the compressor load of R410A cycle is decreased as the temperature increases. This is why the COP of the R410A cycle linearly increases while the R22 cycle shows a level-off in COP as the temperature increases. Based on this observation, it is expected that the R410A cycle will show a better performance than the R22 cycle as the indoor temperature increases. Furthermore, it is believed that the R410A cycle with a smaller evaporator may accomplish the same performance as the R22 cycle.

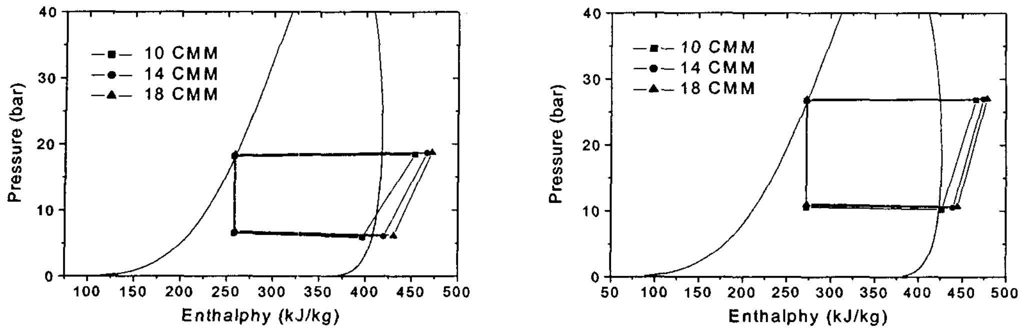
3.5 Effects of indoor air flow rate

Performance of both cycles with a variation in evaporator air flow rate is examined and the results are plotted in Fig. 7. Test conditions for this experiment are listed in the column titled "Indoor air flow" of Table 2. This figure shows that cooling capacity of the R22 cycle is more sensitive to the change of evaporator air flow rate. This is because the enthalpy of R22 at evaporator exit increases further than R410A with an increase in the air flow rate as can be seen in Fig. 8. The COP ratio shows a similar trend as the cooling capacity ratio except at 10 cubic meter per minute (CMM). This is because the evaporator exit condition is at saturated state for R22 while it moved to a superheated state for R410A when the air flow rate reaches 10 CMM as shown in Fig. 8. That is, saturated two-phase refrigerant burdens compressor with larger load than superheated vapour and leads to poor COP. This phenomenon is speculated



(a) Cooling capacity ratio (b) COP ratio

Fig. 7 Performance of R22 and R410A cycles versus indoor air flow.



(a) R22 (b) R410A

Fig. 8 P-h diagrams for R22 and R410A cycles versus indoor air flow.

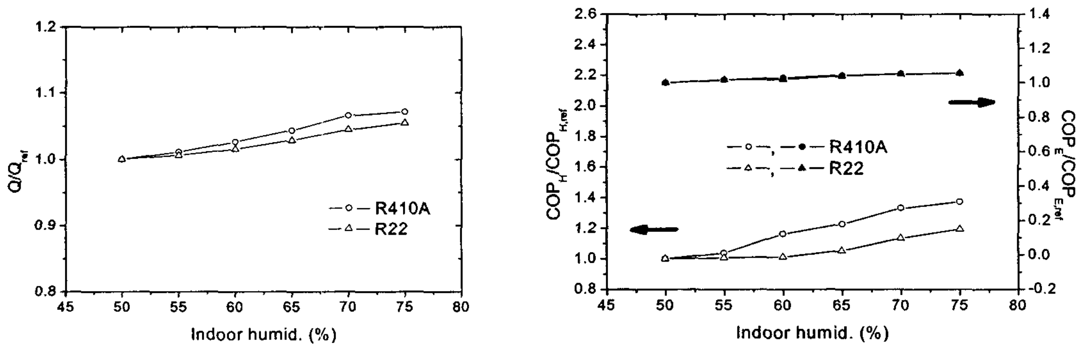
to be caused by the difference of compressor for the two cycles.

3.6 Effects of indoor relative humidity

Experimental conditions for the effects of indoor relative humidity variation is listed in the column titled "Indoor relative humidity" of Table 2. The measured performance variation depending on indoor relative humidity is plotted in Fig. 9. The cooling capacity ratio of the R410A cycle increases more than the R22 cycle as indoor relative humidity increases. Figure 9 (b) shows the variations in COP_E ratios of the two cycles are nearly the same while there is a difference in COP_H ratios. As mentioned in a previous section, it is because electric consumption in both compressors are quite close even if the added enthalpy of refrigerant from the two compressors are a little different from each other.

3.7 Tube-side pressure drop in heat exchangers

Pressure transducers are installed at inlet/outlet of evaporator and condenser to measure the tube-side pressure drop of the heat exchangers. Table 5 summarizes the pressure drop data measured during the experiments for indoor drybulb temperature effects in the section 3.4. This table shows that the pressure drop of R410A is less than that of R22. This trend is observed in all the other experiments. This is caused by the difference in thermodynamic properties of R22 and R410A. The vapour viscosity of R410A is larger than that of R22 by



(a) Cooling capacity ratio (b) COP ratio

Fig. 9 Performance of R22 and R410A cycles versus indoor relative humidity.

Table 5 Tube-side pressure drop with variation in indoor drybulb temperature

Indoor drybulb temperature (℃)	Evaporator (bar)		Condenser (bar)	
	R22	R410A	R22	R410A
21	0.6	0.28	0.41	0.32
23	0.61	0.3	0.42	0.33
25	0.62	0.29	0.43	0.34
27	0.62	0.29	0.44	0.36
29	0.63	0.3	0.43	0.37
31	0.63	0.29	0.42	0.37

7.6% while the vapour specific volume is smaller by 30.7% at evaporator pressure. This smaller vapour specific volume of R410A makes volume flow rate of R410A smaller and leads to reduced pressure drop. In the evaporator, this pressure drop reduction due to smaller volume flow rate is more than the pressure drop increase due to larger viscosity. On the other hand, liquid specific volume of R410A is larger than that of R22 by 16.5% while liquid viscosity is smaller by 31.5% at condenser pressure. In the condenser, pressure drop reduction due to smaller liquid viscosity of R410A is more than the pressure drop increase due to larger liquid flow rate.

These results imply that we can reduce the diameter of heat transfer tube in heat exchangers of the R410A cycle or fabricate grooves such as micro fins for heat transfer enhancement.

4. Concluding remarks

A series of experiments have been performed to compare the performance between the R22 and the R410A refrigeration cycles. The effects of refrigerant charge, indoor/outdoor temperature, humidity, and evaporator air flow rates were examined. The results can be briefed as follows:

(1) It is important to operate cycles with optimal refrigerant charge. In particular, a performance deterioration of the R410A cycle is prominent when overcharged. Therefore, special cautions are needed in charging the R410A cycle.

(2) When the R410A cycle is designed to have the same capacity as the R22 cycle, it is needed to adopt a condenser larger than that of the R22 cycle.

(3) The R410A cycle shows less performance degradation with a decrease in indoor temperature and larger performance improvement with an increase in indoor temperature than the R22 cycle. In this regard, the R410A cycle may adopt smaller evaporator compared to the R22 cycle having the same capacity.

(4) The R22 cycle is sensitive to the change of the evaporator air flow rate, and the R410A cycle outperforms the R22 cycle as indoor relative humidity increases.

(5) Tube-side pressure drops in condenser and evaporator of the R410A cycle appear to be less than those of the R22 cycle. Therefore, we can reduce the tube diameter of heat exchangers of the R410A cycle or fabricate grooves such as micro fins for heat transfer augmentation.

Acknowledgement

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