Testing and analysis of retrofitting R-22 air conditioning unit with alternatives R-407C and R-410A

Mousa M. Mohamed and Ahmed E. Hanafy
Mechanical Power Engineering Department, Faculty of Engineering, Minufiya University, Shebin El-Kom, Egypt
E-mail: mousa_4m@hotmail.com

Retrofitting R-22 air conditioning unit with alternatives R-407C and R-410A was studied to investigate the cooling coil capacity, compressor-consumed power, condenser load, and system coefficient of performance at the same operating conditions. Cooling coil airflow rates of 0.1989, 0.2203 and 0.2344 m^3/s, and controlled air temperatures of 25, 28 and 30 ^oC, were circulated to simulate the room cooling capacity. The condenser coolant airflow rates of 0.2118, 0.2286 and 0.2345 m^3/s and controlled air temperatures of 28, 30, 35 and 38 ^oC were adapted to investigate the effect of various environmental conditions. The results showed that the cooling capacity and the system performance increased with increasing the room temperatures and airflow rates, and the increase was significantly higher for R-22 than that for R-407C and R-410A. The cooling capacity and system performance were systematically decreased with increasing the environmental air temperatures for R-407C and R-410A, but the decrease was little for R-22. The cooling capacities were decreased by 4.9 ~ 36.66 % for R-407C and 5.69 ~ 34.55 % for R-410A, while the consumed power increased to about 6.82 % for the two alternatives compared to R-22. Consequently, the system performance was decreased by 5.92 ~ 39.11 % for R-407C and 6.71 ~ 36.87 % for R-410A. The condenser load decreased with increasing the environmental temperatures by ± 9.97 %. The results showed that the refrigerant R-410A works well than R-407C, so we recommend the R-410A for retrofitting R-22 air conditioning units.

تم دراسة أستبدال فريون ٢٢ لوحدة تكييف هواء بفريون ٤٠٧ جـ وفريون ٤١٠ أ لتوضيح السعة الحرارية لملف التبريد، القدرة المستهلكة، الحمل الحرارى للمكثف، ومعامل آداء النظام عند نفس ظروف التشغيل. تم إمرار معدل هواء على ملف التبريد من ٠,١٩٨٩ ، ٠,٢٢٠٣ ، ٠,٢٣٤٤ م^3/ث ودرجة حرارة من ٢٥ ، ٢٨ ، ٣٠ م^o لمحاكاة سعة التبريد للغرفة. معدل مرور هواء على المكثف من ٠,٢١١٨ ، ٠,٢٢٨٦ ، ٠,٢٣٤٨ م^3/ث ودرجة حرارة ٢٨، ٣٠، ٣٥، ٣٨ م^o تم ضبطهم لإستقصاء تأثير الظروف الخارجية. بينت النتائج أن سعة التبريد لملف التبريد ومعامل آداء النظام تزيد مع زيادة درجة حرارة الغرفة ومعدل مرور الهواء ولكن هذه الزيادة كانت كبيرة بوضوح مع فريون ٢٢ بخلاف فريون ٤٠٧ جـ ، ٤١٠ أ. سعة التبريد لملف التبريد وآداء النظام تناقصت بأنتظام مع زيادة درجة الحرارة المحيطة مع فريون ٤٠٧ جـ ، ٤١٠ أ ولكن التناقص كان قليل مع فريون ٢٢. سعة التبريد لملف التبريد تناقصت من ٤,٩ حتى ٣٦,٧ % مع فريون ٤٠٧ جـ ، ومن ٥,٧ حتى ٣٤,٨ % مع فريون ٤١٠ أ، بينما القدرة المستهلكة زادت حتى ٦,٨ % مع فريون ٤٠٧ جـ ، ٤١٠ أ مقارنة بفريون ٢٢. وبالتالى معامل آداء النظام تناقص من ٥,٩ حتى ٣٩,١ % مع فريون ٤٠٧ جـ ، ومن ٦,٧ حتى ٣٦,٩ % مع فريون ٤١٠ أ. الحمل الحرارى للمكثف تناقص مع زيادة درجة حرارة الهواء المار فى المكثف من –٩,٩ حتى ٩,٥ %. النتائج بينت أن فريون ٤١٠ أ يعمل أفضل من فريون ٤٠٧ جـ مقارنة بفريون ٢٢ ولذلك نوصى بإستبدال فريون ٤١٠ أ محل فريون ٢٢ فى وحدات تكييف الهواء.

Keywords: Retrofitting R-22, Alternative refrigerants R-407C and R-410A, Cooling capacity, Refrigeration system performance, Ozone depletion potential

1. Introduction

Refrigerants ChloroFluoroCarbon (CFC), and Hydro- ChloroFluoroCarbon (HCFC), are widely used in the refrigeration and air conditioning applications. CFCs as R-11, R-12 and R-115 containing chlorine atoms are known to damage the ozone layer. The HCFCs as R-22, R-123, R-124, R-141b and R-142b are chemically similar to CFCs with one or more hydrogen atoms that are less damaging to the ozone layer. Chlorine has been identified, as the destructive element that reacts with ozone, O_3, to convert it to normal oxygen, O_2. Refrigerants hydro-fluorocarbon, HFC, as R-23, R-125, R-32, R-134a, R-143a and R-152a not contain any chlorine and consequently have no ozone depletion effect [1]. Under the mandate of the Montreal Protocol, the use of CFCs had been phased out and the use of HCFCs will also be phased out in a short period of time. HFC refrigerants have recently been developed to replace the traditional CFC refrigerants. The non-azeotropic refrigerant mixtures, NARM, R-407C and near-azeotropic refrigerant mixtures, R-410A are new alternative refrigerants for R-22. The composition by weight of R-407C is (25 % R-125, 52 % R-134a, 23 % R-32,) and of R-410A is (50 % R-32, 50 % R-125).

For non-azeotropic refrigerant mixtures, the composition of the saturated vapor is slightly different from that of the saturated liquid. This means that during evaporation process, the vapor quality increases in the cooling coil, and the composition of the boiling liquid changes. This change in composition is associated with an increase in boiling point temperature at the given pressure. Conversely, during condensation, the vapor quality decreases and there is a fall in the equilibrium condensing temperature at constant condensing pressure. The saturation property tables of the non-azeotropic refrigerants are slightly different from those for single component, or azeotropic refrigerants. The non-azeotropic refrigerants have two different saturation pressure-temperature curves; one for the saturated liquid, known as the *Bubble Point*, and the other for the saturation vapor, known as the *Dew Point*. This compares to the one curve representing saturated liquid and vapor for single refrigerants or true azeotropic. In the property tables for non-azeotropic refrigerants, the different pressure-temperature values for both saturated liquid and saturated vapor are given as shown in fig. 1 [1]. The pressure-enthalpy charts for non-azeotropic refrigerant blends are slightly different from those for single-component refrigerants. The isotherms, or lines of constant temperature, in the wet region are not parallel to lines of constant pressure. In order to properly use the alternative refrigerants of R-22, we need to know their thermodynamic, thermo-physical flow heat transfer properties, and the system performance.

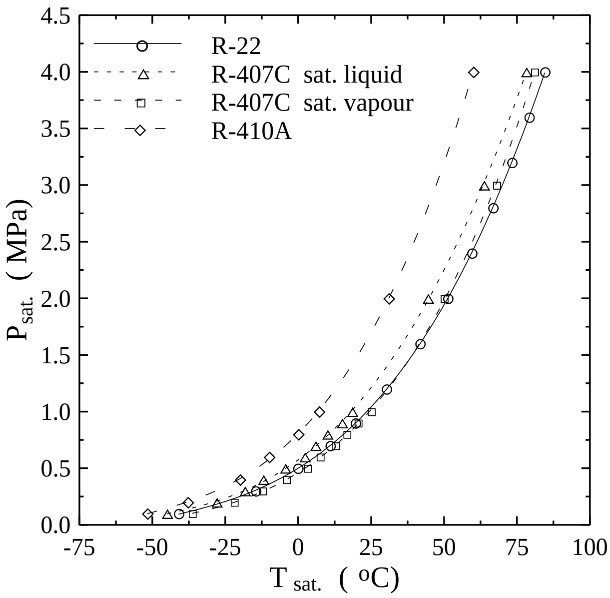

Fig. 1. Saturation pressure-temperature carves for R-22 and alternatives [1].

To evaluate the alternative refrigerants, the operating characteristics of individual components of the refrigeration cycle should be clarified. The heat transfer coefficient and pressure drop for the condensation of R-134A, flowing in a small pipe and a plate heat exchanger, were experimentally investigated and the data correlated [2, 3]. An experimental investigation of retrofitting R-22 with one of three mixtures R-407C, R-404A and R-410B was made to investigate the overall heat transfer coefficient on a full-scale test plant consisting of a horizontal shell-side condenser [4]. According to the measurements, the decrease in condenser overall heat transfer coefficient for non-azeotropic mixture R407C was 43 ~ 70 % compared to R-22, while for the near-azeotropic mixture R-404A was less than 15 %, but for R-410B the decrease was up to 20 %. Eleven alternatives for R-22, R-12 and R-114 under controlled conditions for a heat pump system was performed and tested [5]. The results showed that R-152A has the best performance except at lowest evaporating temperature, while R-143A was the best performing alternative at the lowest evaporating temperature. Comparative performance was studied experimentally between R-12 and R-413A [6]. The results indicated that using R-413A is a good alternative for R-12 with respect to its thermodynamic and actual performance, but the specific consumed power increased by 10 %, while the system coefficient of performance decreased by 12.2 % as compared to R-12. Theoretical performance of three alternative refrigerants, R-22, R-134A and R-717 compared to R-12 are conducted [7]. It was concluded that the use of R-22 increased the cooling and heating capacity by 58.8 % and 65.1 %, respectively and caused a decrease in the system performance of 1.6 %, while using of R-717 caused an increase in cooling and heating capacities and the system performance of 78.9 %, 84.1 % and 8.6 %, respectively compared to R-12. On the other hand, the study predicted that the use of R-134A results in negligible changes in the cooling and heating capacities, but the system performance was decreased by about 6 %. A car air conditioned by R-12 was retrofitted to R-134A and the results showed better performance [8]. The performance of capillary tube in a refrigeration system for R-22 and its alternatives, R-407C and R-410A was investigated experimentally and a dimensionless correlation for adiabatic capillary tubes was developed [9,10]. Performance of nineteen alternative refrigerant mixtures of R-22, were experimentally investigated [11]. The system performance of all mixtures was 10.59 ~ 21.67 smaller than that of R-22. The cooling capacity of R-407C was smaller than R-22 by 5 %, and other mixtures 23.17 % larger than that of R-22.

Retrofitting of R-22 air conditioners with alternative refrigerants of R-407C and R-410A at the same manufacture design and operating conditions is not sufficiently covered in the previous studies. Therefore, the major objective of the present study is to investigate experimentally the performance of R-22 air conditioning unit with alternatives R-407C and R-410A. The experiments were conducted at steady state by controlling the air temperatures circulating through the cooling coil to simulate the room heat capacity at various airflow rates. The coolant air temperatures and airflow rates entering the condenser were adapted to simulate the environmental conditions. The effects of air room temperature and environmental conditions on the system performance and power consumption were investigated. Comparisons between cooling coil capacity, consumed power and system performance for R-22 and its alternatives of R-407C and R-410A at the same conditions have been aimed.

2. Experimental apparatus

The experimental apparatus is shown in fig. 2. The apparatus consists of the main parts of refrigeration cycle: condenser, cooling coil, liquid receiver, expansion valve and hermetic reciprocating compressor. All parts of the experimental apparatus are the components of room window type air conditioner Carrier's Company trademark of model No. H24B28QABCB with cooling capacity of 22000 Btu/hr, and the specifications of the unit is shown in table 1. The condenser and cooling coil are reconstructed in an insulated case to control the volume airflow rate and temperatures. The electric heaters were installed in the vicinity of airflow. The air entrance of both condenser and cooling coil has rectangular shape as the electric heaters, but the air exit was round shape with diameter of 0.2 m for condenser and 0.152 m for cooling coil to measure the volume airflow rates. Both condenser and cooling coil are finned tubes heat exchangers. The entering air temperatures to both condenser and cooling coil were adjusted at appointed value by using electric heaters and voltage transformer, the potential volt on the electric heater terminal was regulated to get a desire temperature value. The airflow rate was adapted by air fan with variable speed motor. Turbine air meter was used to measure the volume airflow rates at three speeds for both condenser and cooling coil. Airflow rates of 0.1989, 0.2203 and 0.2344 m^3/s, with controlled air temperatures of 25, 28 and 30 ^oC, were circulated through the cooling coil to simulate the room cooling capacity. The condenser coolant airflow rates of 0.2118, 0.2286 and 0.2345 m^3/s and controlled air temperatures of 28, 30, 35 and 38 ^oC were adapted to investigate the effect of various environmental conditions. Two pressure gauges were used for measuring the refrigerant pressure in suction and delivery

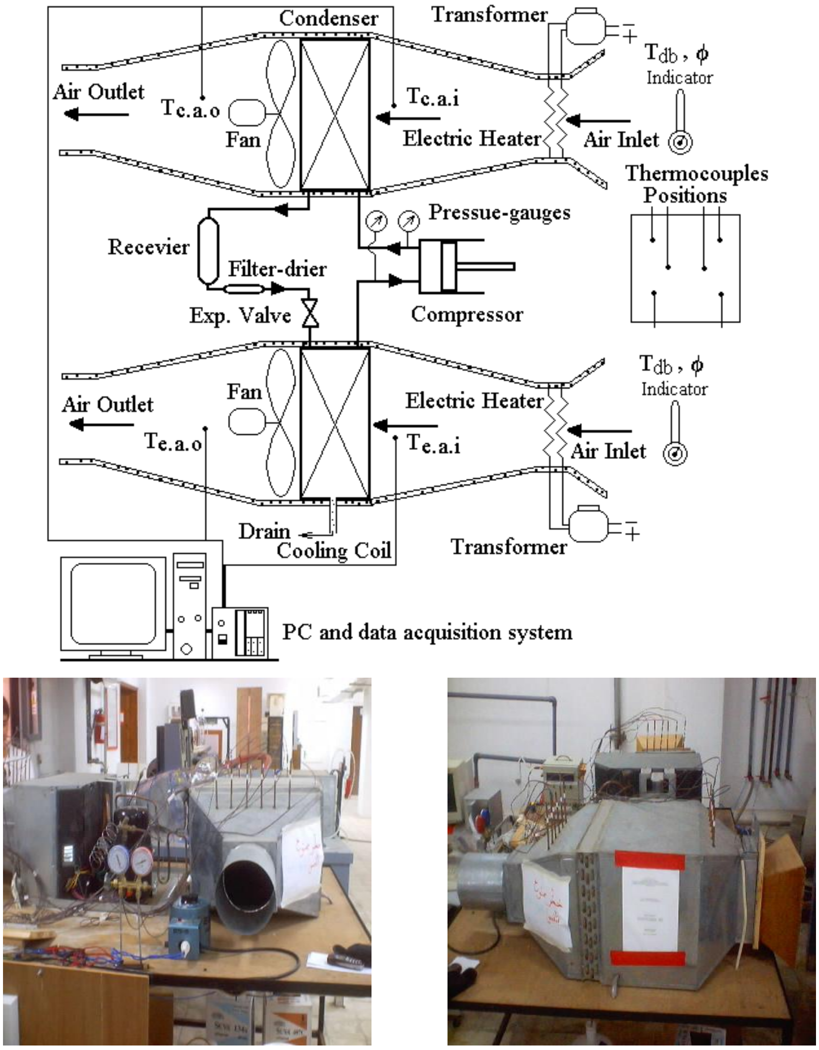

Fig. 2. Experimental apparatus and layout of components.

Table 1
The specifications of window type air conditioner as carrier's company data table

Part No. 701497-7570-01	Model No. H24B28QABCB
Cooling capacity 22000 Btu/Hr	240 V, 12 A, pH 1, 60 Hz, BRISTOL
3-Speed fan motor	Rated at (26.7/19 DB/WB) and Outdoor (35 DB)
Weight: 70 kg.	Dimensions: W = 660, D = 758, H = 450 mm.

lines. The compressor-consumed power was estimated from the volt and current of the power supply at each set of experiment.

The air temperatures at inlet and outlet of the condenser and cooling coil were measured with thermocouples type J and data acquisition system. Six thermocouples in four groups were used to estimate the average temperature at inlet and outlet of the condenser and cooling coil as shown in fig. 2 and the average values were used to estimate the cooling coil capacity and condenser load. Two thermocouples were used to record the refrigerant temperatures of suction and delivery lines at outer surface of the tube. All temperatures were measured in steady state and recorded instantaneously by using a data acquisition system. National Instrument Multifunction *DAQ Board*, *Lab-PC+* of 83.3 *kS/s* sampling rate and *LabView* software version 5 were used. The *DAQ Board* is managed through a personal computer Pentium I. Chassis *SCXI* 1000 for instrumentation and *SCXI* 1100 signal conditioning modules with *SCXI* 1300 terminal blocks were used for processing data. The temperatures reading were recorded at a rate of 4 loops per sec. The error in temperature measurements was within ± 0.1 ^oC. A computational program with *LabView* software was developed to calculate the cooling capacity, condenser load and the system performance instantaneously.

2.1. Data acquisition and procedures

The components of experimental apparatus and instrumentation were fixed and tested. Thereafter, the refrigeration cycle was evacuated by using vacuum pump to near zero absolute pressure. The system was checked after twenty-four hours to make sure that the vacuum pressure never changed. Then, the system was gradually charged by about 900 grams of R-22. After the optimum charge was passed to the refrigeration system, it was left to operate at steady state. The steps of evacuation and charging the system were repeated for alternatives R-407C and R-410A. The suction pressure obtained was about 4.8 bar ± 0.5 and the delivery pressure was 24.5 ~ 27.5 bar according to the air temperatures and flow rates passing through the condenser. For each experiment, the compressor was switched on and the air temperatures passing through cooling coil and condenser were adjusted at a desired value by controlling the voltage of electric heaters. The electric current and volt of compressor power supply were recorded and inserted in the measuring program to calculate the consumed power. After twenty minutes at least, the previous parameters had nearly attained constant values. At this moment, the data was recorded and used to estimate the system performance and make a comparison between R-22 and the alternatives R-407C and R-410A.

2.2. Data reduction

In particular, if Q_e is the total heat transfer rate between the refrigerant inside the tubes of cooling coil and the air passing outside the coil, the cooling capacity can be calculated from the airside by,

$$Q_e = \dot{m}_{e.a}(h_{e.a.i} - h_{e.a.o}) . \tag{1}$$

Indicators at inlet of both cooling coil and condenser measured the relative humidity and dry bulb temperature. The specific enthalpy of air at inlet and outlet of cooling coil were estimated as the procedure shown in fig. 3. The Apparatus Dew Point, (ADP), was taken equal to the saturation temperature of refrigerant at the pressure measured at suction line. The inlet air with known relative humidity and dry bulb temperature was heated and passed through the cooling coil. The average air temperatures at inlet and outlet of cooling coil were measured with the data acquisition system and recorded. In this study, the thermodynamic properties of moist air and refrigerants were obtained by using *Cool Pack* software version 1.46, [12]. The cooling capacity, Q_e, at each experimental set was calculated from the cools tool auxiliary program using Engineering Equation Solver (EES), with knowing the humidity ratio and dry bulb temperature of air at inlet of cooling coil, and the ADP and the dry bulb temperature at outlet of cooling coil.

The condenser coolant heat load, Q_c, expressed from the airside as,

$$Q_c = \dot{m}_{c.a}(h_{c.a.o} - h_{c.a.i}) . \tag{2}$$

Also, the procedure just explained above used to calculate the specific enthalpy of air at inlet and outlet of condenser, and the condenser load, Q_c, was obtained.

The compressor-consumed power was calculated from the measured data of current and volt of the power supply as,

$$P_{comp} = I.V.\cos\theta . \tag{3}$$

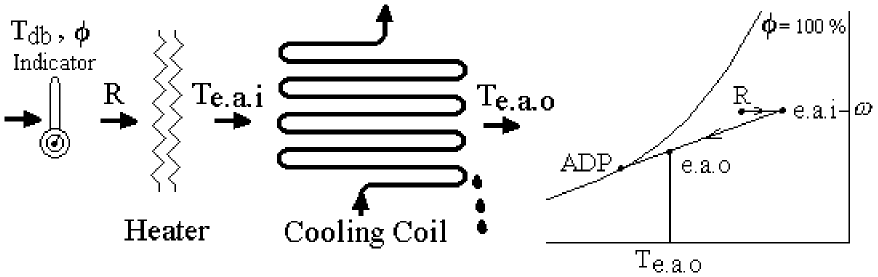

Fig. 3. Cooling and dehumidification process through cooling coil.

Where, $\cos\theta$ is the power factor and a value of unity was used as the data table of the maker. The system coefficient of performance was estimated from the cooling capacity and compressor-consumed power as follows,

$$COP = Q_e / P_{comp}. \tag{4}$$

The above calculation procedure was repeated for each experiment of R-22, R-407C and R-410A at the same operating conditions of airflow rates and temperatures.

3. Results and discussion

The system performance in the term of cooling capacity, compressor-consumed power and condenser load at various airflow rates and temperatures for R-22 and alternatives were illustrated in fig. 4. For each set of experimental conditions, the cooling capacity increases with increasing airflow rates for both cooling coil and condenser, and the increases are pronounced with the room temperatures. The range of air temperatures inlet to the cooling coil were set to achieve a temperature change as shown in table 2. The measured data showed that the difference between inlet and outlet temperature through the cooling coil was higher for R-22 than R-410A and R-407C. Obviously, a systematic increase of cooling capacity with increasing of airflow rates and room temperatures for R-22, but a little increase for R-407C and R-410A was observed. It may be attributed to the increase in temperature difference between liquid refrigerant inside tubes and air outside it and the enhancement of heat transfer coefficient at air-side according to the increase of airflow rates. The cooling capacity decreases with increasing the condenser air temperatures. The effect of temperatures on the cooling capacity for R-22 were small as compared to the airflow rates, the temperatures caused a decrease in the refrigerating effect of cooling coil. The general trend of fig. 4 showed a decrease in the cooling coil capacities for R-407C and R-410A as compared to R-22.

Table 2
Inlet and outlet air temperature through cooling coil

Te.a.i (°C)	Average Te.a.o (°C) ± 1.5		
	R-22	R-407C	R-410A
25	12.65	15.75	15.55
28	13.90	15.89	15.65
30	14.85	16.82	16.55

The compressor-consumed power and condenser load at various airflow and temperatures were illustrated in figs. 5 and 6. The consumed power for R-407C and R-410A was higher than that for R-22 as shown in fig. 5. For alternatives, non-azeotropic refrigerant, R-407C, and near azeotropic refrigerant, R-410A, with increasing the vapor quality of refrigerant during the evaporation process, the equilibrium evaporating temperature at constant pressure increases causing the increase of superheating at exit of the cooling coil and suction line. The condenser load decreased with increasing the airflow temperatures for R-22 and alternatives with the same behavior as shown in fig. 6. The system coefficient of performance was similar to the curves trend of cooling coil capacity for R-22 and alternatives as shown in fig. 7. It is observed that the system coefficient of performance decreases with increasing the airflow temperatures of condenser especially for R-407C and R-410A, this may due to the decrease in cooling capacity

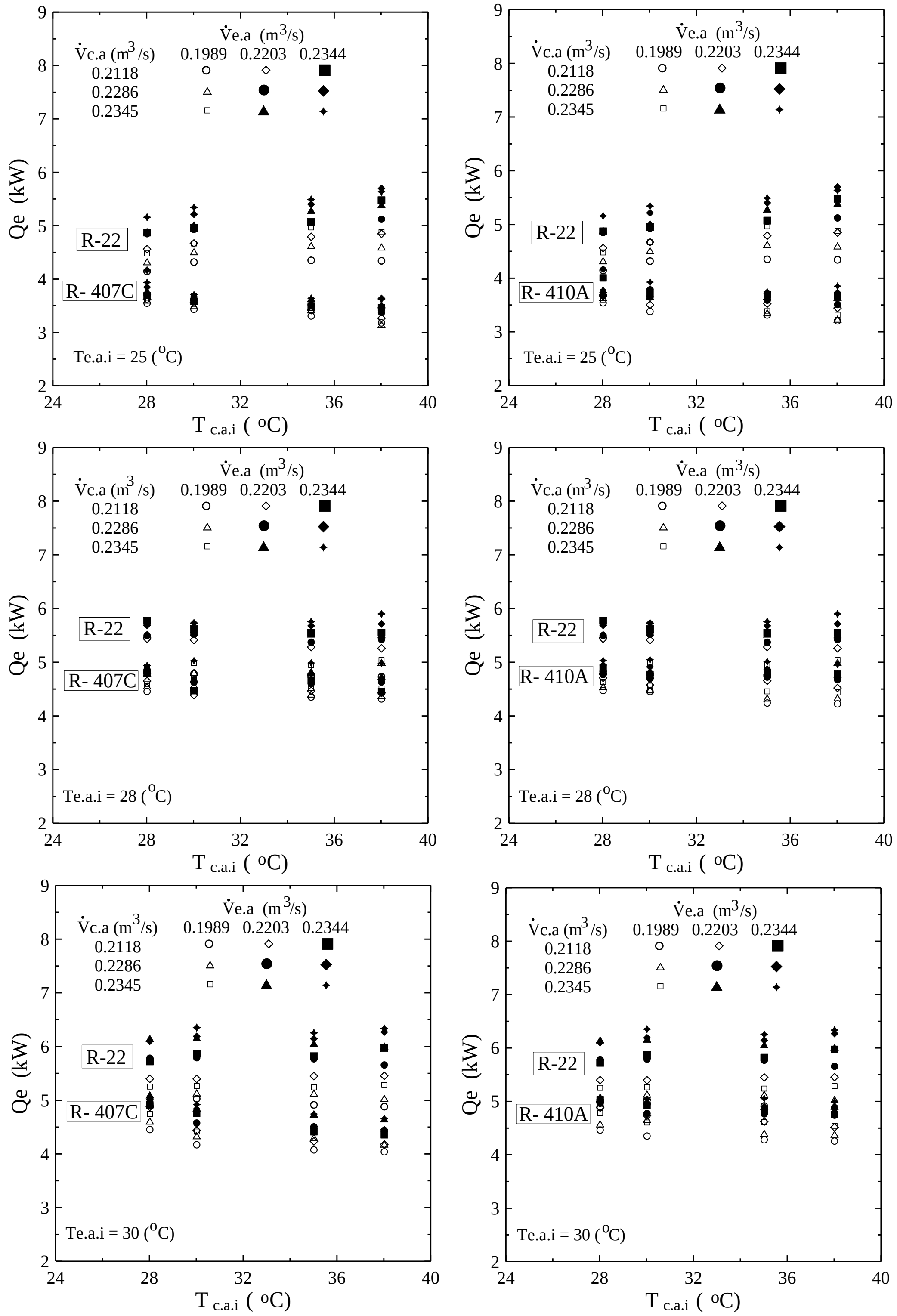

Fig. 4. Cooling coil capacity for R-22 and alternatives.

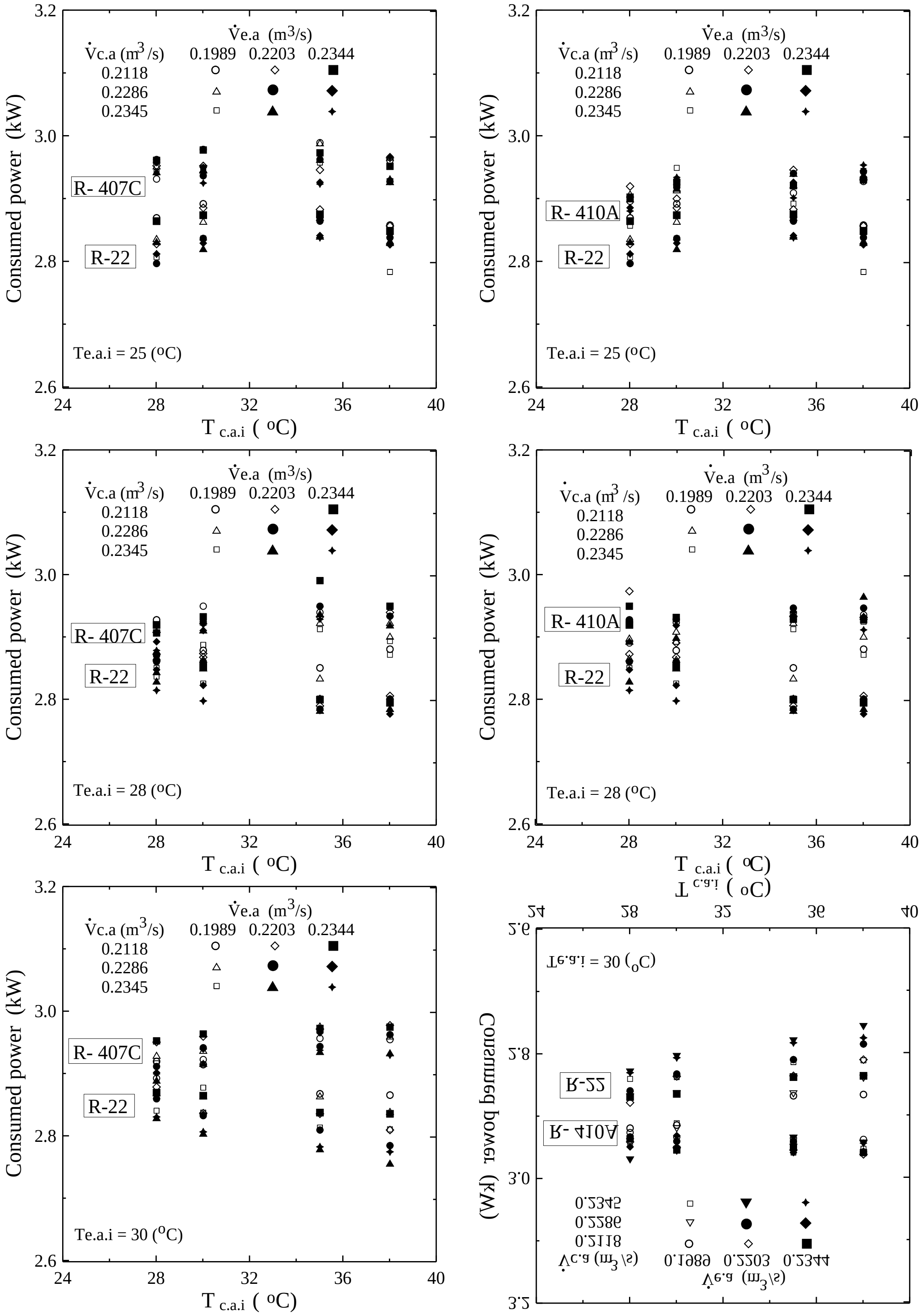

Fig. 5. Compressor-consumed power for R-22 and alternatives.

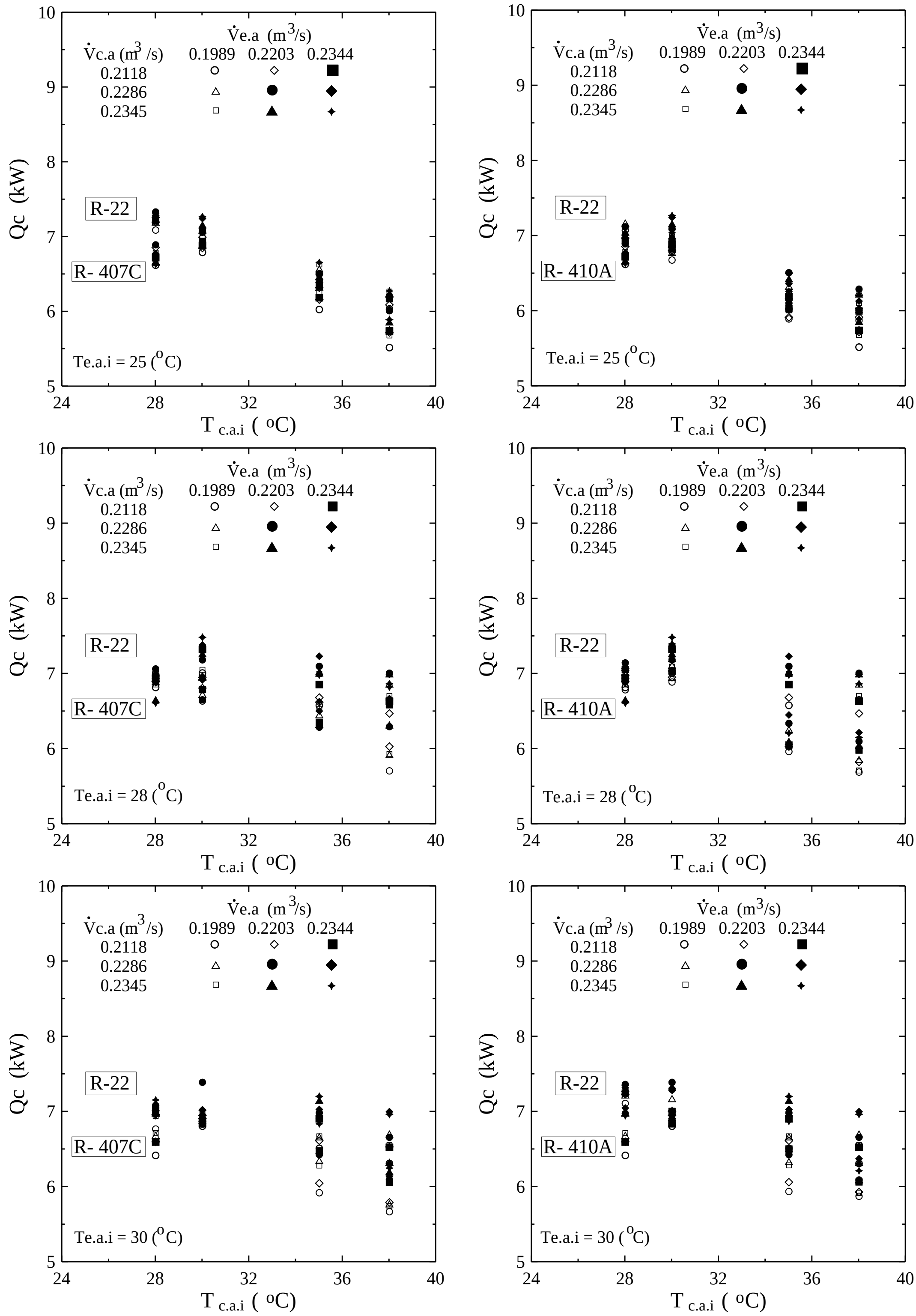

Fig. 6. Condenser load for R-22 and alternatives.

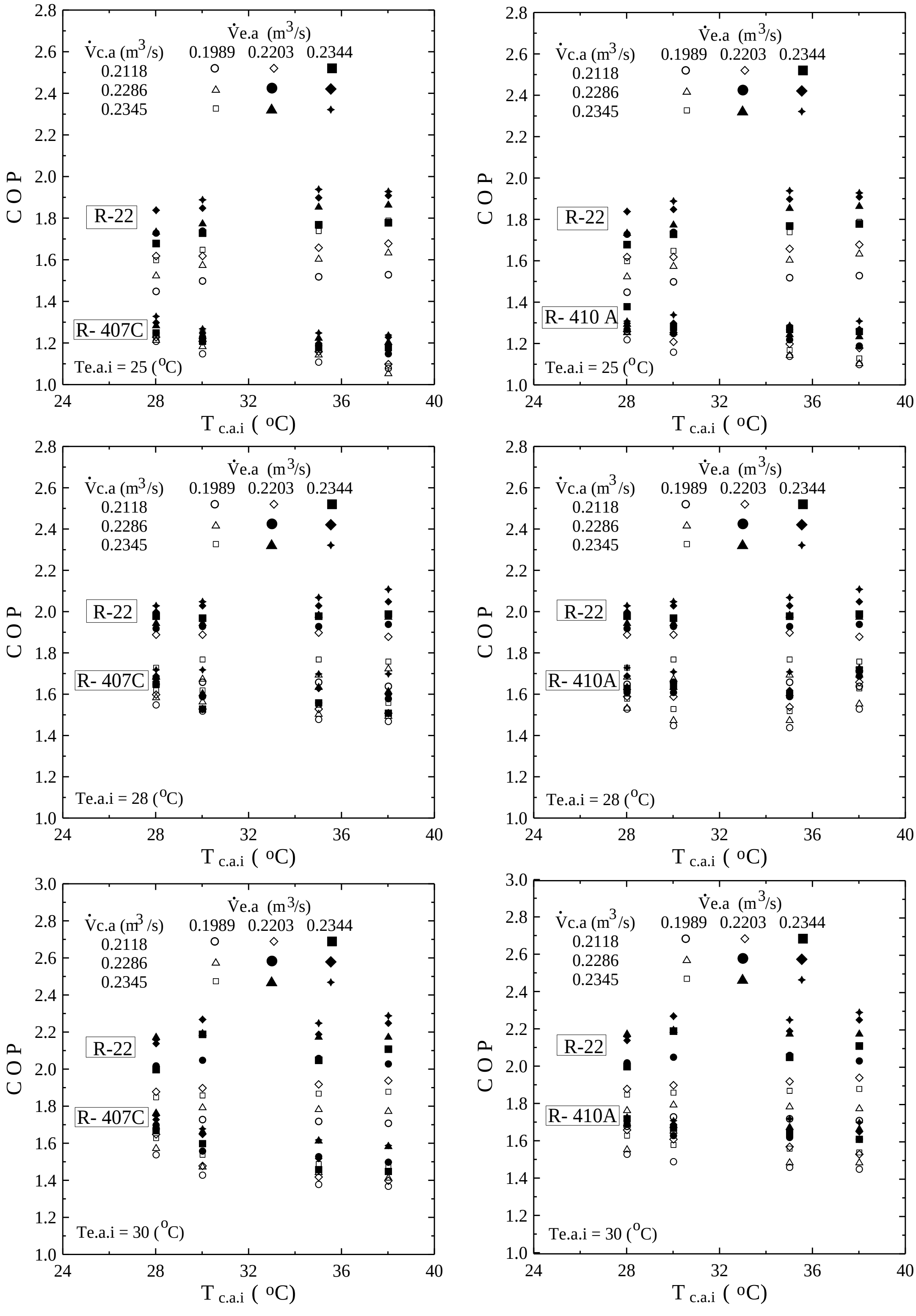

Fig. 7. System coefficient of performance for R-22 and alternatives.

and increase in compressor-consumed power.

3.1. Experimental data statistical analysis

The percentage deviation φ^+ of some parameter φ, which denote the cooling capacity, compressor-consumed power, condenser load and system coefficient of performance of a given refrigerant of R-407C or R-410A relative to the same parameter of R- 22 can be expressed by the following relation,

$$\varphi^+ = (\varphi_{Alternative} - \varphi_{R-22})/\varphi_{R-22} \,. \tag{5}$$

The standard deviation or the root mean square deviation was employed to estimate the relative error of φ^+ as the following,

$$SD = \sqrt{\sum_{i=1}^{i=n} (\varphi_i^+)^2 \Big/ n} \;. \tag{6}$$

Where, n is the total number of experimental data.

The percentage deviation of cooling capacity for R-407C and R-410A relative to R-22 was illustrated in fig. 8, which was decreased by 4.9 ~ 36.7 % for R-407C and 5.7 ~ 34.6 % for R-410A as compared to R-22 with root mean square of 2.9 ~ 6.7 %. The compressor-consumed power was increased to about 6.8 % for the two alternatives compared to R-22 as shown in fig. 9. Consecutively, the system coefficient of performance was decreased by 5.9 ~ 39.1 % for R-407C and 6.7 ~ 36.9 % for R-410A similar to the cooling coil capacity with root mean square of 3.6 ~ 6.2 % as shown in fig. 10. The condenser load decreased with percentage deviation of – 9.9 % ~ 9.5 % with root mean square of 3.4 ~ 5 % as shown in fig. 11.

3.2. Simulation of refrigeration cycle

In refrigeration cycles, the pressure-enthalpy diagram was frequently used in the analysis of vapor-compression machine. A simple thermodynamic cycle was performed to evaluate the cycle performance of R-22 and alternatives of R-407C and R-410A. The recorded pressures in the suction and delivery lines were used to simulate the pressure in the evaporator and condenser. The sub-cooling and superheating temperatures were assumed equal to 5 °C, and the pressure drop in suction and delivery lines assumed equal to 0.2 and 0.4 bar as shown in fig. 12. This analysis includes isentropic compression efficiency equal to 0.85. The evaporating and condensing pressures, and other data were

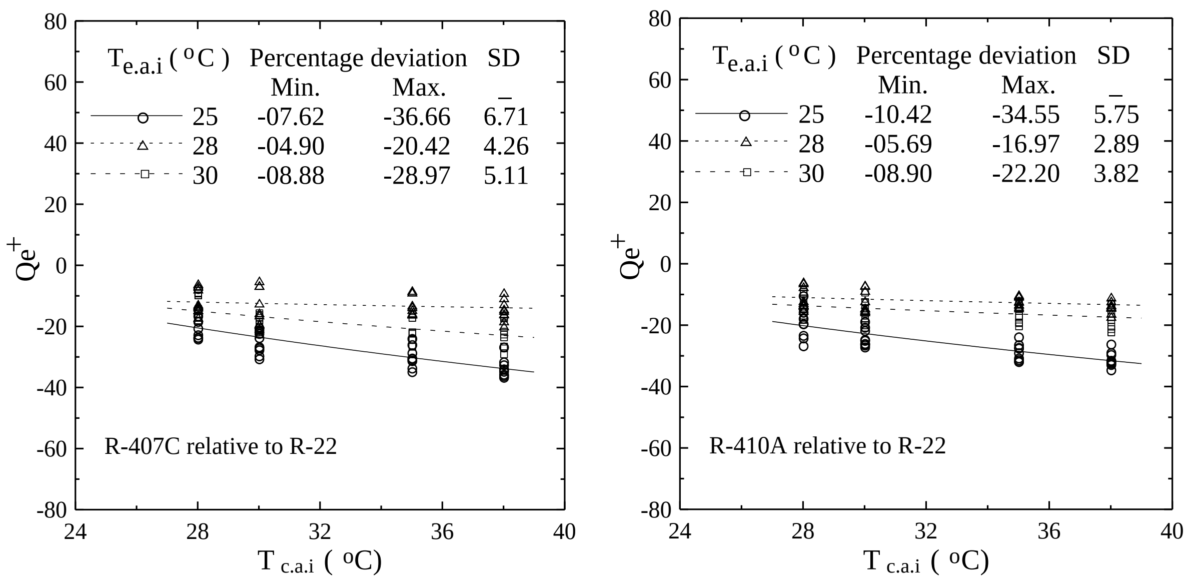

Fig. 8. Percentage deviation of cooling capacity.

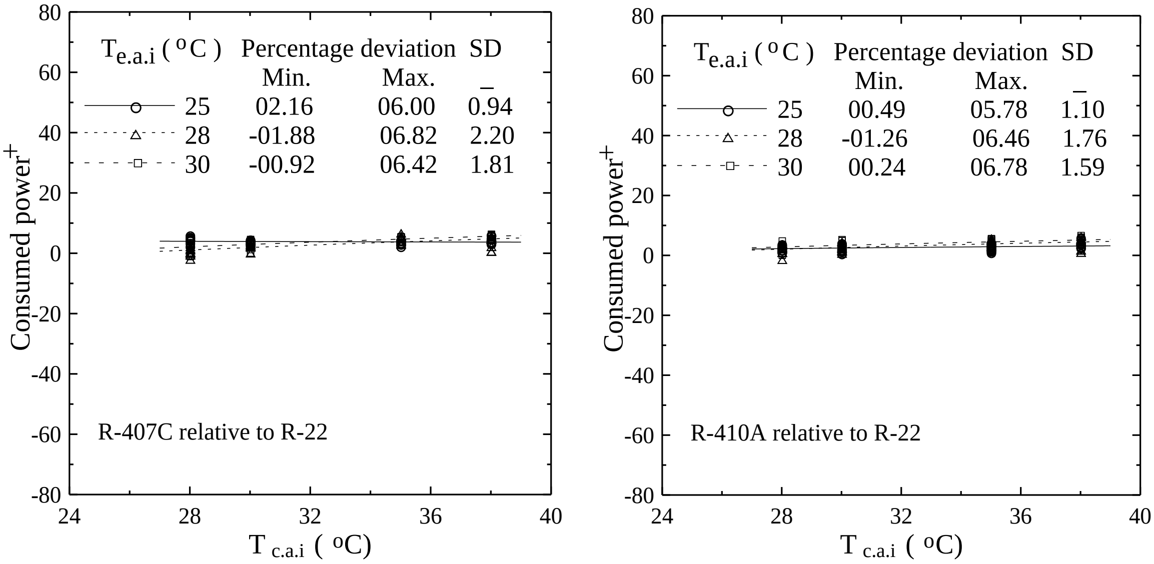

Fig. 9. Percentage deviation of compressor-consumed power.

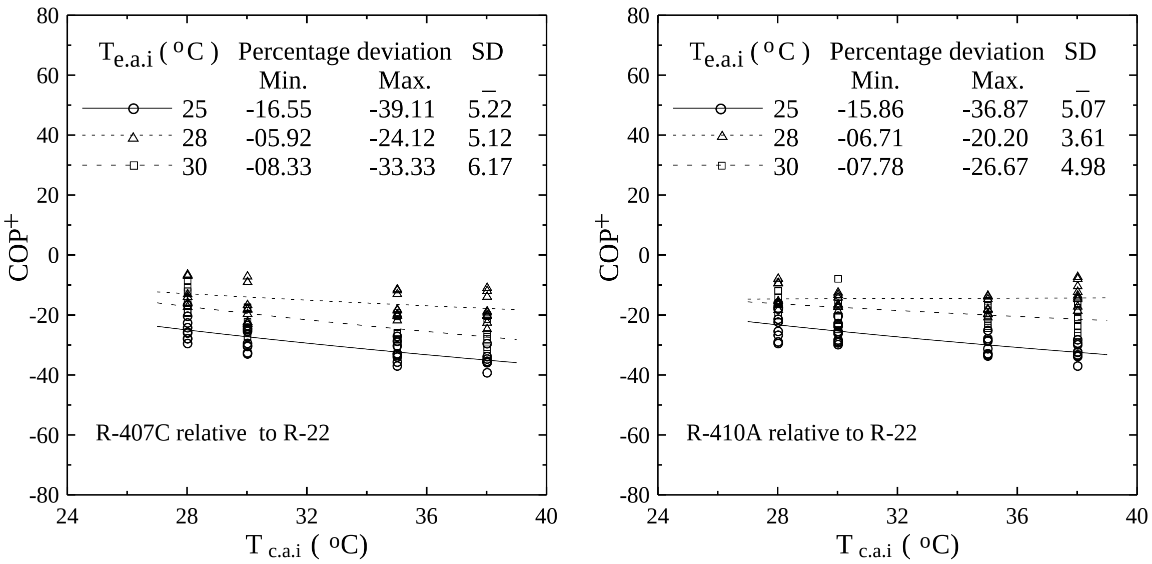

Fig. 10. Percentage deviation of system coefficient of performance.

specified as inputs to the cools tool auxiliary program of *Cool Pack* software to estimate the cycle performance.

$$COP_{cyc} = (h_1 - h_4)/(h_{2d} - h_{1s}) \,. \tag{7}$$

According to the results, the cycle coefficient of performance for R-407C was smaller than that of R-22 by 6 %, but for R-410A was higher than that of R-22 by 4 % as shown in fig. 13. Also, the measured data of system performance were illustrated in fig. 13 and showed that the system performance for R-410A was better than that of R-407C compared to R-22. Also, all the data of system performance were smaller than that of the

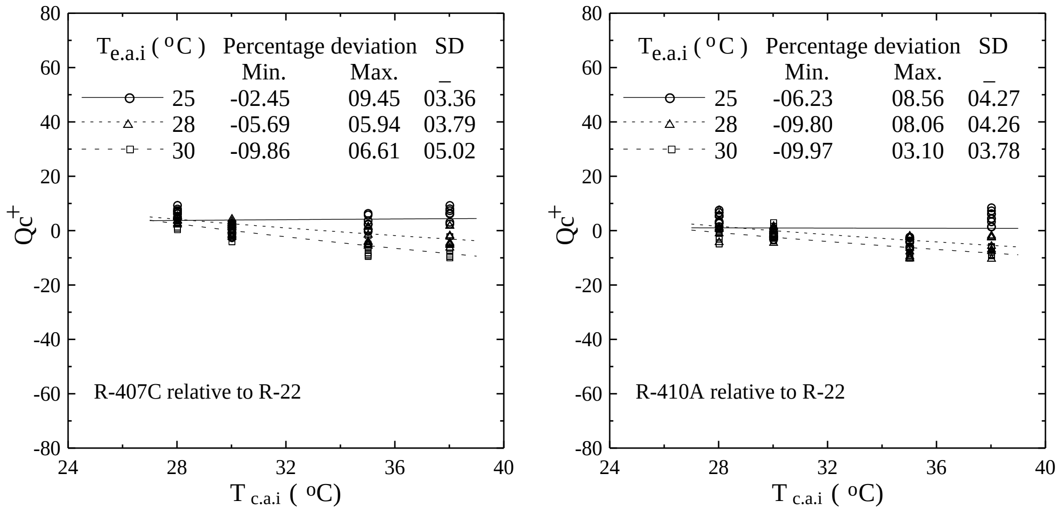

Fig. 11. Percentage deviation of condenser load.

cycle performance by 20 ~ 35 %. A comparison between the system performance of the present data for R-22 and alternatives and data in ref. [11] were illustrated in fig. 14. It was found that the data in ref. [11] for liquid cooler was higher than that of air condition by 20 % for the present data.

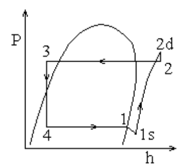

Fig. 12 Pressure –enthalpy diagram for simulation.

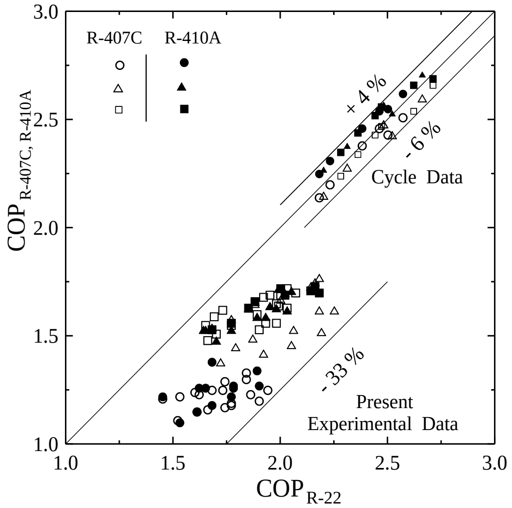

Fig. 13. Comparison between cycle performance for R-22 and alternatives and system.

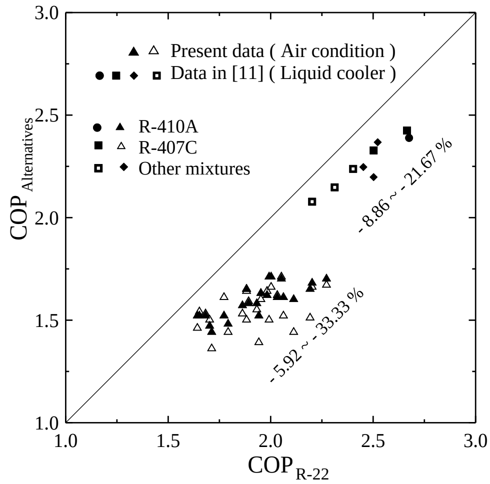

Fig. 14. Comparison between present data and data in ref. [11].

4. Conclusions

An experimental study was performed to investigate the system performance of R-22 air conditioning unit with alternatives R-407C and R-410A. The cooling capacity, consumed power, condenser load, and system performance were examined at the same operating conditions of various airflow rates and temperatures. The results are summarized as the followings:

1. The cooling capacity and the system performance were increased with increasing the airflow rates and temperatures passing

through the cooling coil, but the increase was significantly higher for R-22 than alternatives.
2. The cooling coil capacity and system performance were systematically decreased with increasing the environmental temperatures for the two alternatives, but the decrease was little for R-22.
3. The cooling capacity was decreased by 4.9 ~ 36.4 % for R-407C and 5.7 ~ 34.6 % for R-410A, while the consumed power increased to about 6.8 % for the two alternatives as compared to R-22.
4. The system performance was decreased by 5.9 ~ 39.1 % for R-407C and 6.7 ~ 36.9 % for R-410A similar to the cooling capacity.
5. The condenser load decreased with increasing the temperatures passing through the condenser for R-22 and alternatives by – 9.9 % ~ 9.5 %.
6. For all the data of system performance were smaller than that of the cycle performance by 20 ~ 35 %.
7. The system performance of R-410A was better than that of R-407C compared to R-22, so we recommend the refrigerant R-410A for retrofitting R-22 air conditioning units.

Nomenclature

COP	is the coefficient of performance,
h	is the specific enthalpy J/kg,
I	is the electric current A,
$\dot{m}$	is the mass flow rate kg/s,
P	is the power kW,
p	is the pressure MPa,
Q	is the heat transfer rate kW,
T	is the temperature °C,
V	is the electric volt V,
$\dot{V}$	is the volume flow rate m^3/s,
ϕ	is the relative humidity, and
ω	is the humidity ratio $kg/kg_{d.a}$.

Subscript

ADP	is the apparatus dew point,
a	is the air,
db	is the dry bulb,
c	is the condenser,
comp	is the compressor,
cond	is the condensate,
cyc	is the cycle,
e	is the evaporator,
i	is the inlet,
o	is the outlet,
sat	is the saturation condition,
SD	is the standard deviation,
φ	is the specified parameter, and
ϕ	is the phase angle.

References

[1] DuPont SUVA® Refrigerants Service Manual, UK (2003).

[2] Yi-Yi Yan and Tsing-Fa Lin, "Condensation Heat Transfer and Pressure Drop of Refrigerant R-134a in a Small Pipe," Int. J. of Heat and Mass Transfer, Vol. 42, pp. 697-708 (1999).

[3] Yi-Yi Yan, Hsiang-Chao Lio and Tsing-Fa Lin, "Condensation Heat Transfer and Pressure Drop of Refrigerant R-134a in a Plate Heat Exchanger," Int. J. of Heat and Mass Transfer, Vol. 42, pp. 993-1006 (1999).

[4] Cecilia Gabrielii and Lennart Vamling, "Replacement of R-22 in Tube-and Shell Condensers: Experiments and Simulations," Int. J. of Refrigeration, Vol. 20, pp. 165-178 (1997).

[5] J.R. Sand E.A.Vineyard and R.J. Nowak "Experimental Performance of Safe Alternative Refrigerants," ASHRAE Transactions, Vol. 96, part I, pp. 173-182 (1990).

[6] M.M. Magdy et al., "Comparative Performance of R-413A and Some Refrigerants as Alternatives to R-12," ERJ, Helwan University, Faculty of Engineering, Mataria, Cairo, Egypt, Vol. 75, pp. 35-50 (2001).

[7] O. Al-Rabghi "Review and Performance Comparison Among Some R-12 Candidate Substitutes," Proceedings of first UAE Conference on Air Conditioning in the Gulf, UAE University pp. 60-73 (1996).

[8] M. Omar Al-Rabghi and A. Abdulkareem Niyaz, "Retrofitting R-12 car Air Conditioner with R-134A Refrigerant," Int. J. of Energy Research, Vol. 24, pp. 467-474 (2000).

[9] S.G. Kim M.S. Kim and S.T. Ro "Experimental investigation of the performance of R-22, R-407C and R-410A in Several Capillary Tubes for Air-

Conditioners," Int. J. of Refrigeration, Vol. 25, pp. 521-531 (2002).
[10] J.D. Douglas et al., "A Cost-Based Method for Comparing Alternative Refrigerants Applied to R-22 Systems," Int. J. of Refrigeration, Vol. 22, pp. 107-125 (1999).
[11] B.G. Lee et al., "A Study on the Performance of Alternative Refrigerant Mixtures for HCFC-22," CFC Alternatives Research Center, Korea institute of Science and Technology (KIST), Cheongryang, Seoul 130-650, South Korea (2002).
[12] Department of Mechanical Engineering, Technical University of Denmark, www.et.dtu.dk/CoolPack (2003).

Received March 16, 2004
Accepted May 29, 2004