Journal of Engineering and Sustainable Development

Vol. 22, No.05, September 2018
ISSN 2520-0917
https://doi.org/10.31272/jeasd.2018.5.15

EFFECT OF USING R-22, R404 AND R-407C ON PERFORMANCE OF AN AIR-CONDITIONING SYSTEM

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Abstract: Due to the environment depletion, replacing chlorofluorocarbons in refrigeration industry was an important problem. In the experimental study a (1.0 TR) vapor compression system were used with original working fluid of R-22 and compared to alternatives refrigerant R-407C and R-22. The effects of the main parameters of performance analysis such as refrigerant type, compressor pressure ratio, outlet air evaporative temperature, coefficient of performance were investigated for various ambient temperatures above 50 ºC. The results showed that the compressor ratio of system using R404 and R-407C is more than the compressor ratio of system using R22 by 2% and 4% respectively. Also, the outlet air evaporative temperature of system using the alternative refrigerant R-22 is more than the outlet air evaporative temperature with system using R-407C and R-404 by 8% and 5% respectively. The coefficient of performance of R-407C was measured with the same reciprocating compressor that was supplied with the R-22 system. It could be concluded that there is a decrease of 8% in system COP as friendly environment R407C compared with system using R-22 and a decrease of 12% in system COP when using R-404. However; the result showed that R407C have thermodynamic performance similar to R-22. Engineering Equation Solver (EES) software is used for draw (p-h) diagram for the cycle.

Keywords: *Alternatives, Comparison, Refrigerant Replacement COP, R-22, R-407C*

تاثير استخدام R-22, R404 AND R-407Cعلى اداء منظومة التثليج

الخلاصة: يعتبر استبدال غاز الكلوروفلوروكربون المستخدم في صناعة التبريد من اهم المشاكل بسبب الاستنفاذ البيئي. تم استخدام (1 طن تثليج) في منظومة ضغط البخار استخدم فيها المائع الاصلي **R-22**وتمت مقارنته مع احد بدائل التثليج **R-407C** تم التحقق من تاثيرات العوامل الرئيسية لتحليل اداء المنظومه مثل نوع غاز التثليج، نسبة انضغاط الضاغط، درجة الحرارة للهواء الخارج من المبخر ومعامل الاداء، لدرجات حرارة مختلفه للهواء الخارجي. اظهرت النتائج ان نسبة الانضغاط للضاغط في منظومة تستخدم **(R-407C , R404)** اكبر من نسبة الانضغاط للضاغط في منظومه تستخدم **R-22** بنسبة 2% و 4% على التوالي. كذلك فان درجه حرارة الهواء الخارج من المبخر لمنظومة تستخدم **R-22** اقل بنسبة **8**% من درجه حرارة الهواء الخارج من المبخر لمنظومة تستخدم**R-407C** واقل بنسبة 5% لمنظومة تستخدم **R-404**. معامل الاداء للمنظومه تستخدم **R-407C** تم قياسه مع نفس الضاغط الترددي المجهز لمنظومه تستخدم**R-22**استنتج هنالك نقصان بنسبة **8**% في اداء المنظومة تستخدم البدائل الصديقة للبيئة مثل**R-407C** ونقصان بنسبة 12% في معامل اداء المنظومة التي تستخدم الغاز **R-404** . واظهرت النتائج ان لغاز التثليج **R407C** له اداء ثرموديناميكي مماثل ل **R-22** . استخدم برنامج (**Engineering Equation Solver –EES**) لرسم مخططات (**p-h**) الخاصة لدورة التبريد.

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1. Introduction

According to the Montreal Protocol the phase out of R-22 will commence in 1996 with a consumption cap, followed by a 35% reduction in consumption starting in 2004, and a complete phase-out is slated for 2020 in Canada. The phase out for R-22 has already been advanced in Europe, with Germany having a phase-out for new equipment starting in 2000.

When using refrigerant alternatives, it has an advantage that it needs no large change of the compressor dimension for R22 use. It is important to compare the performance of different mechanism compressors to supply most appropriate type according to cooling capacity range after R22 is replaced with alternatives.

By considering current compressor losses and its alternation by changing refrigerant, it should be studied technique about compressor dimensional optimization, and estimated efficiency of optimized compressor in the case that R22 is replaced with R407C or R410A, but in this paper there is no any change in the compressor mechanism.

Replacing chlorofluorocarbons in refrigeration industry is an important problem because of environment depletion. There are many types are suitable refrigerant blends as a replacement of R-22. R407 represented in this paper: R-407C, a zeotrope of HFC-32/HFC-125/HFC-134a (23/25/52 wt.%) . R407C is a mixture refrigerant and its thermodynamic and physical properties are like as R22. This refrigerant has been introduced as an alternative for R22 with zero ozone depletion potential and less greenhouse warming potential than R22. A comparison of performance of different refrigerants in the vapor compression cycle can be carried out on the basis of the internal temperatures of the refrigerant or on the basis of the external temperatures of the heat transfer fluids which acts as a heat source and a heat sink.

Herbert et. al [1], present the experimental performance study of an air-conditioning unit with M20 (a refrigerant mixture of R407C, R600a and R290 in the mass fraction 80:10.96:9.04 respectively). Experiments were conducted in a 5.25 kW window air-conditioner to assess the potential of M20 as a retrofit refrigerant without making any system alteration other than changing the capillary. The results presents that the system had 4.18% to 7.47% lower cooling capacity with respect to that of R22 when using M20. The COP for the M20 refrigerant was lower, in the range of 0.58% to 7.78% than that of R22. So, M20 can be used as a drop- in-substitute for R22 in window air-conditioners.

Saber et. al [2], investigated advantages and disadvantages of R407C as an alternative refrigerant for replacing R22 in refrigeration systems. Also, studied the effect of various parameters on cycle performance by modeled refrigerants thermodynamic properties, condenser and evaporator secondary fluid using artificial neural network, then reducing complicated computation and without using state equations. Also, a comparison of refrigeration cycle using R22 and R407C refrigerants with Wilson-Plot method.

Vaisman [3], presents a comparison of Computational performance of a rotary vane compressor air conditioning unit operates with R22 and R407C is produced taking into account actual system configurations. It is appeared that the cooling capacity of an air-conditioning unit using R407C is higher than system operating with R22 by factor of 1.038. also, the system COP of an air-conditioning unit operates with R-407C is lower than system operates with R-22 by factor of 0.984, for outdoor temperature of 95° *F* and indoor temperature of 80.6 $^{\circ}$ *F*, while the relative humidity is about 47%.

Anders and Lundqvist [4], present a comparison of COP, compressor efficiency and cooling capacity of the air-conditioning system when using the refrigerants (R-407C, R-404a, R-417A, R-134a and R290 with respect to R-22). "Fig. 1" shows the relationship between the relative COP's with relative capacity.

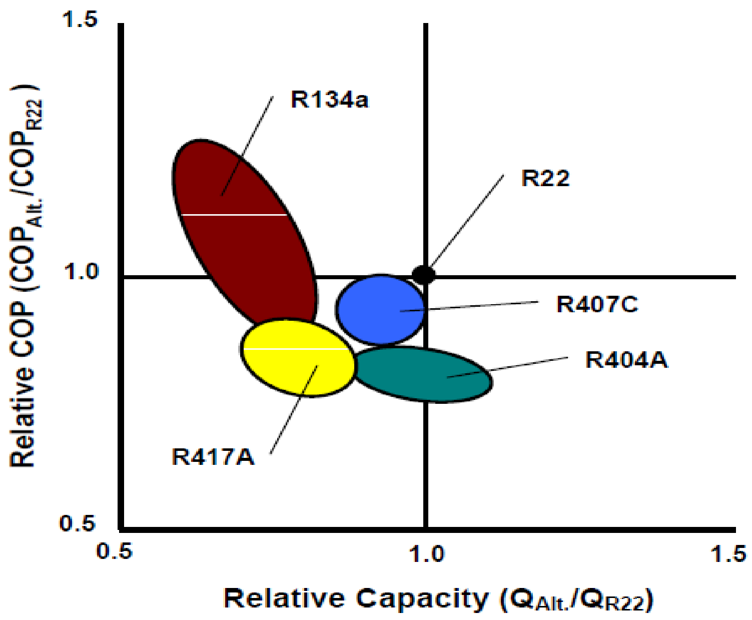

Figure 1. Relative COP with relative capacity [4].

Kato et. al [5], show an experimental and theoretical comparison of compressor efficiency between rotary and scroll type using R-22 and its alternative refrigerants, R407C and R410A. it is showed that the upper limit of cooling capacity range, in which efficiency of rotary type is higher than that of scroll type, is about 8,000 Btu/h with R22 and R407C. In the case with R410A, rotary type is superior to scroll type in the range approximately up to 24,000 Btu/h. it shows that the range in which rotary type operates more efficiently than scroll type will expand up to small capacity range of unitary air conditioners in the case with R410A.

Linton et. al [6], made a comparison between performance of two long term replacements R-407C (HFC-32/125/134a (23%/25%/52%)) and R-410A (HFC-32/125 (50%/50%)) with R-22 using 10.5 kW (3.0 TR) residential central heat pump. The performance evaluations were carried out in a psychometric calorimeter test facility located at NRC using the Canadian Standards Association (GSA)/Air-Conditioning and Refrigeration Institute (ARI) rating conditions. The performance of R-407C was measured with the same reciprocating compressor that was supplied with the R-22 system. Performance characteristics were measured including compressor power, cooling and heating capacity, mass flow rate of a refrigerant, cooling energy efficiency ratio (EER) and heating coefficient of performance (COP). The optimization included

Changing the direction of the air flow (inverting the evaporator coil) to get an approximation of a counter-flow heat exchanger to take advantage of R-407C's temperature glide.

Yunting and Cropper [7], developed a simulation model of steady-state case of a finned-tube air-cooled condenser by using the distributed modelling method. The model can be used to predict 3-D variations of parameters for both air and refrigerant sides. The model has been validated by comparing outputs from the model with test data, given in reference papers, derived from air-cooled condensers with differing dimensions and refrigerants. Application of the model is concentrated on optimization and exploration of existing and new control strategies for controlling refrigerant discharge-pressures in retail refrigeration systems, where air-cooled condensers are being utilized. Meaningful results are obtained for both the optimal design and development of the new control strategies and for performance comparisons when different refrigerants are employed in the condensers.

The aim of current work is comparing COP and evaporative temperature between the two types of alternatives (R-407C & R-404) with R-22 experimentally.

2. General Equipment Layout

In this work, a (1TR) air-conditioning split unit was used with three type of refrigerants, R-22, R-404 and R407C with same type of compressor and same amount of lubricating oil, the air condition unit consist of a compressor, condenser, capillary tube and evaporator are shown in "Fig. 2".

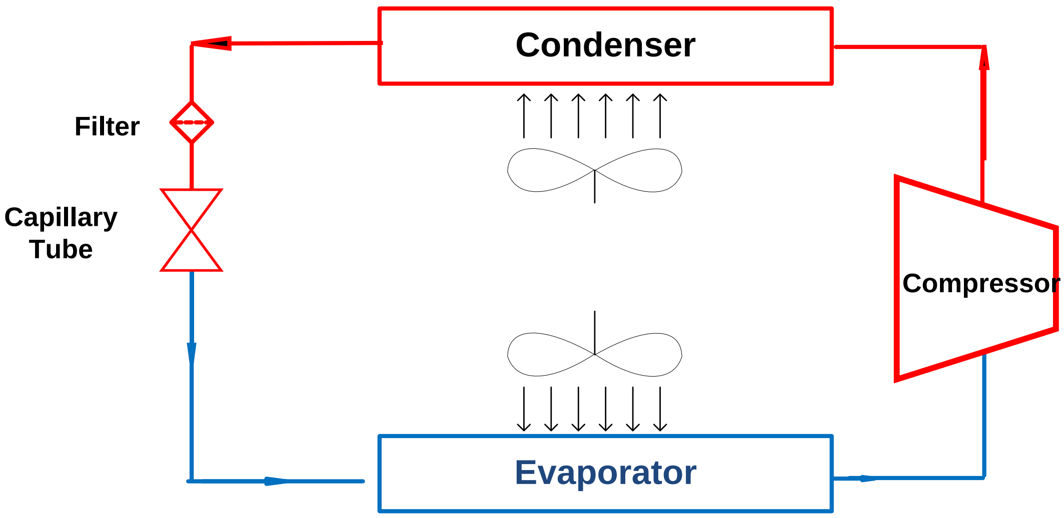

Figure 2. General Equipment Layout

Pair of high pressure gauges (type PM-500psi) are fixed after the compressor and condenser, while the others of low pressure gauges (type PM-250psi) are fixed after capillary tube and evaporator. Calibrated thermocouples type (pt100 with multi-reader and type k with selector switch) are used and shown in "Fig. 3", for measuring pressure and temperature around the cycle. "Fig. 4", shows the parameters used in the system.

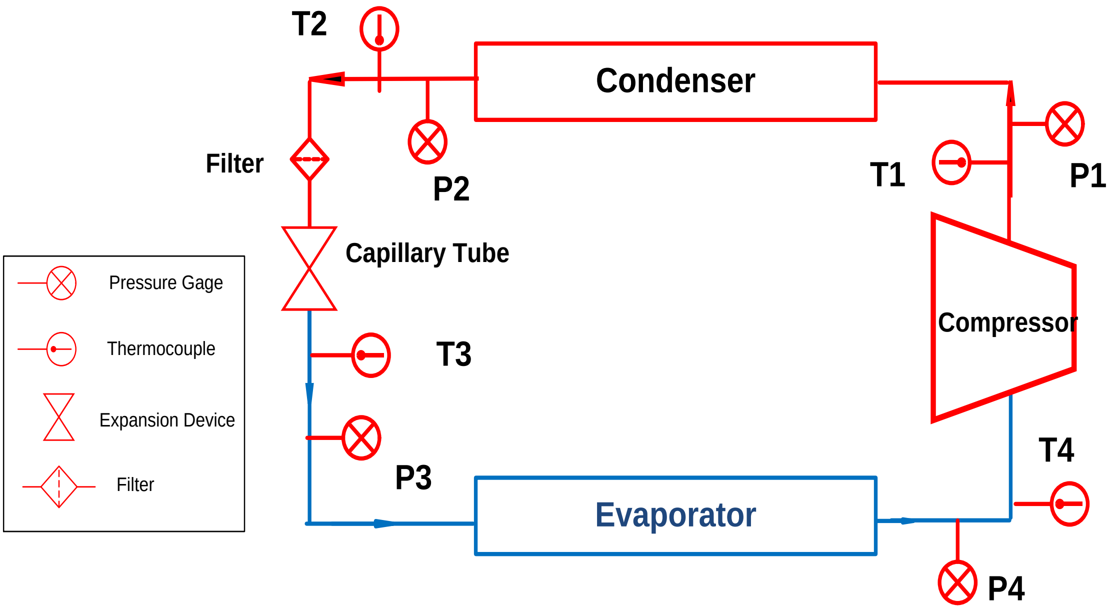

Figure 3. Schematic Diagram of system equipment with measurement devices

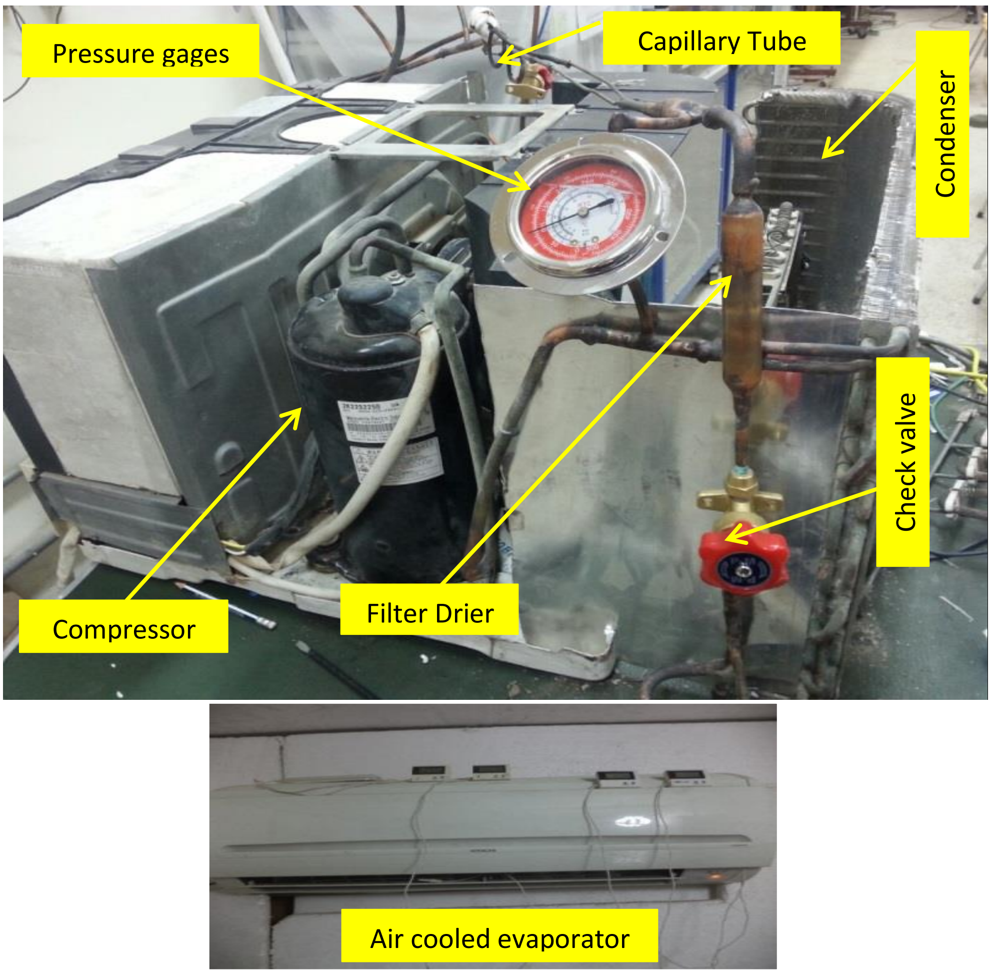

Figure 4. Air-conditioning unit details.

3. Theoretical Model

Rotary type, positive displacement compressor, has been used in current study. The amount of specific, isentropic work done by an ideal compressor can be found by the energy equation:

$$w_{s,com} = \dot{m}_r(h_{2s} - h_1) \qquad (1)$$

Since the finned-tube heat exchangers are most commonly used for residential air conditioning applications, a plate finned-tube type has been used in this work. "Fig. 5", shows the Z-type, finned – tube, staggered condenser used in this work.

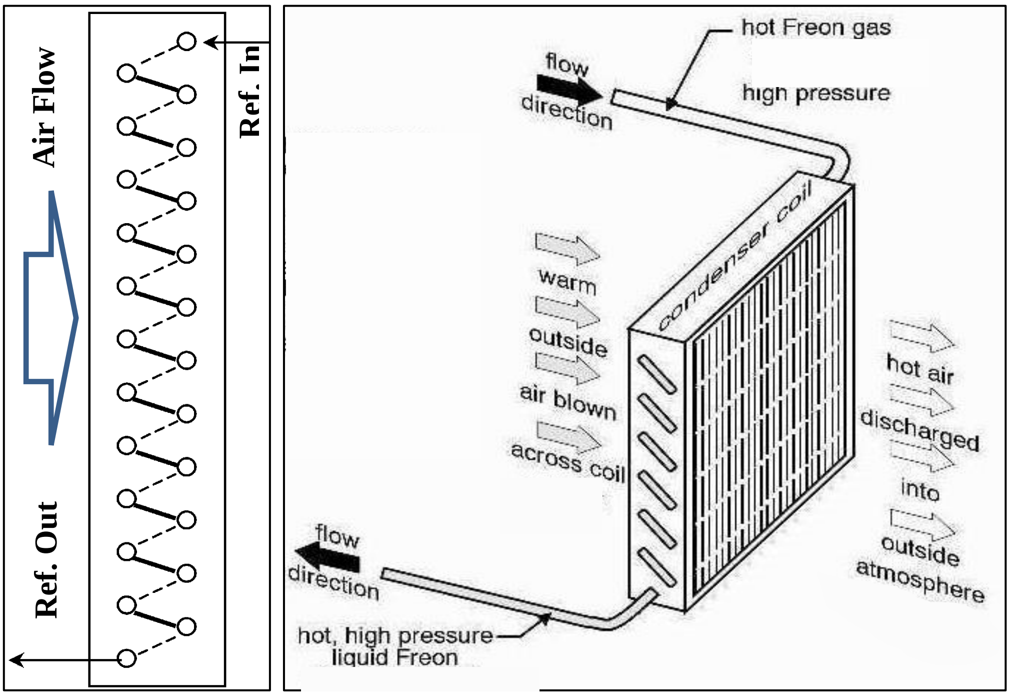

Figure 5. Z-Type finned-Tube staggered heat exchanger

The condenser can be separated into three sections: superheated, saturated, and sub-cooled shown in "Fig. 6". The specific heat rejected from each section can be found by evaluating the refrigerant enthalpies at the inlets and outlets, so the equations (2 to 4) presented the heat rejection.

$$q_{c,sh}=h_2-h_{2a} \qquad (2)$$

$$q_{c,sat}=h_{2a}-h_{2b} \qquad (3)$$

$$q_{c,sc}=h_{2b}-h_3 \qquad (4)$$

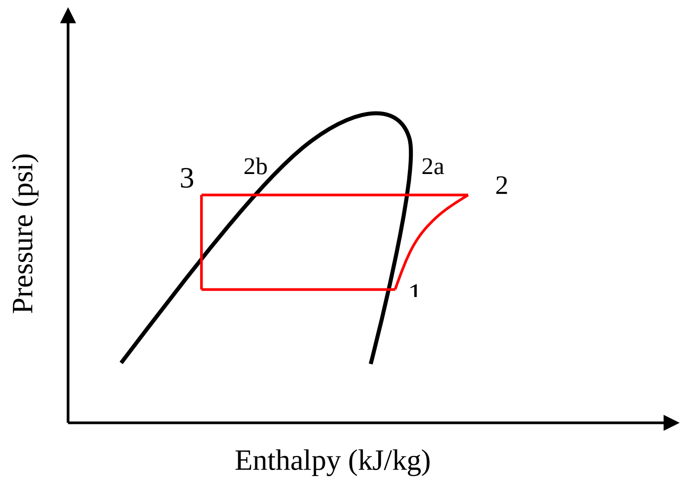

Figure 6. P-h Diagram of the cycle

Then the heat transfer of the air across the condenser can be calculated according to the difference bulb temperature between the air inlet and air outlet temperatures was given in the following equation [8].

$$\dot{Q}_a = \dot{m}_a Cp_a \, \Delta Tb_\infty \tag{7}$$

Where,

$$\Delta Tb_\infty = (T_{\infty 2} - T_{\infty 1}) \tag{8}$$

Also the air mass flow rate that effect on the condenser coil can be calculated from the following expression:

$$\dot{m}_a = \rho_a V_\infty A_c \tag{9}$$

While the surface temperature (T_s) is constant across the condenser, then the heat balance can be expressed by [8].

$$\dot{Q}_a = \dot{m}_a Cp_a \, \Delta Tb_\infty = \mathrm{h}_a \, A_{s,c}(T_s - Tb_m) \tag{10}$$

Where,

$$A_{s,c} = N_c \pi D_c L_c \tag{11}$$

Where, N_c = Number of refrigerant pipe in condenser
All the air properties can be taken at $Tb_m$of the inlet and outlet of an air duct.

$$Tb_m = \frac{(T_{\infty 1} + T_{\infty 2})}{2} \tag{12}$$

The heat transfer coefficient for the air side of the condenser can be calculated from the following equation (16) [9].

$$Q_a = h_{c,} N_c \pi D_c L_c (T_s - Tb_m) = \dot{m}_a cp \Delta Tb_\infty \quad (16)$$

Also $\dot{m}_a$: is the mass flow rate of the air, could be calculated from the following equation:

$$\dot{m}_a = \rho V_a N_c S_n \quad (17)$$

Also the heat transfer coefficient can be calculated using equation (18).

$$h_a = \frac{Nu.K}{D_c} \quad (18)$$

If the number of rows less than 10, so the following equation can be calculates the Nusselt number [8]:

$$Nu|_{(N_L<10)} = C_2 . Nu|_{(N_L\geq 10)} \quad (19)$$

The constants (C_1, n) are depends on the ratio of (S_p/D_c) & (S_n/D_c), that's shown in "Table 1", and (C_2) depends on the number of horizontal rows, as shown in "Table 2" below [10]:

Table 4.1Constant (C_1, n) [10]

$b=S_p/D_c$	$a=S_n/D_c$							
	1.25		1.5		2.0		3.0	
	C_1	n	C_1	n	C_1	n	C_1	n
in-line								
1.25	0.386	0.592	0.305	0.608	0.111	0.704	0.703	0.752
1.5	0.407	0.586	0.278	0.620	0.112	0.702	0.0753	0.744
2.0	0.464	0.570	0.332	0.602	0.254	0.632	0.220	0.648
3.0	0.322	0.601	0.396	0.584	0.415	0.581	0.317	0.608
Staggered								
0.6	-	-	-	-	-	-	0.236	0.636
0.9	-	-	-	-	0.495	0.571	0.445	0.581
1.0	-	-	0.552	0.558	-	-	-	-
1.125	-	-	-	-	0.531	0.565	0.575	0.560
1.25	0.575	0.556	0.561	0.554	0.576	0.556	0.579	0.562
1.5	0.501	0.568	0.511	0.562	0.502	0.568	0.542	0.568
2.0	0.448	0.572	0.462	0.568	0.535	0.556	0.498	0.570
3.0	0.344	0.592	0.395	0.580	0.448	0.562	0.467	0.574

Table 2 Constant (C_2, n) [10]

n	1	2	3	4	5	6	7	8	9	10
In-line	0.64	0.80	0.87	0.90	0.92	0.94	0.96	0.98	0.99	1.0
staggered	0.68	0.75	0.83	0.89	0.92	0.95	0.97	0.98	0.99	1.0

Reynolds and Prandtle numbers can be calculated from equation (20 & 23), as the following:

$$\mathrm{Re} = \frac{\rho V_{\max} D_c}{\mu} \tag{20}$$

The maximum velocity can be found by the following equation (21) according to the dimensions shown in "Fig. 7".

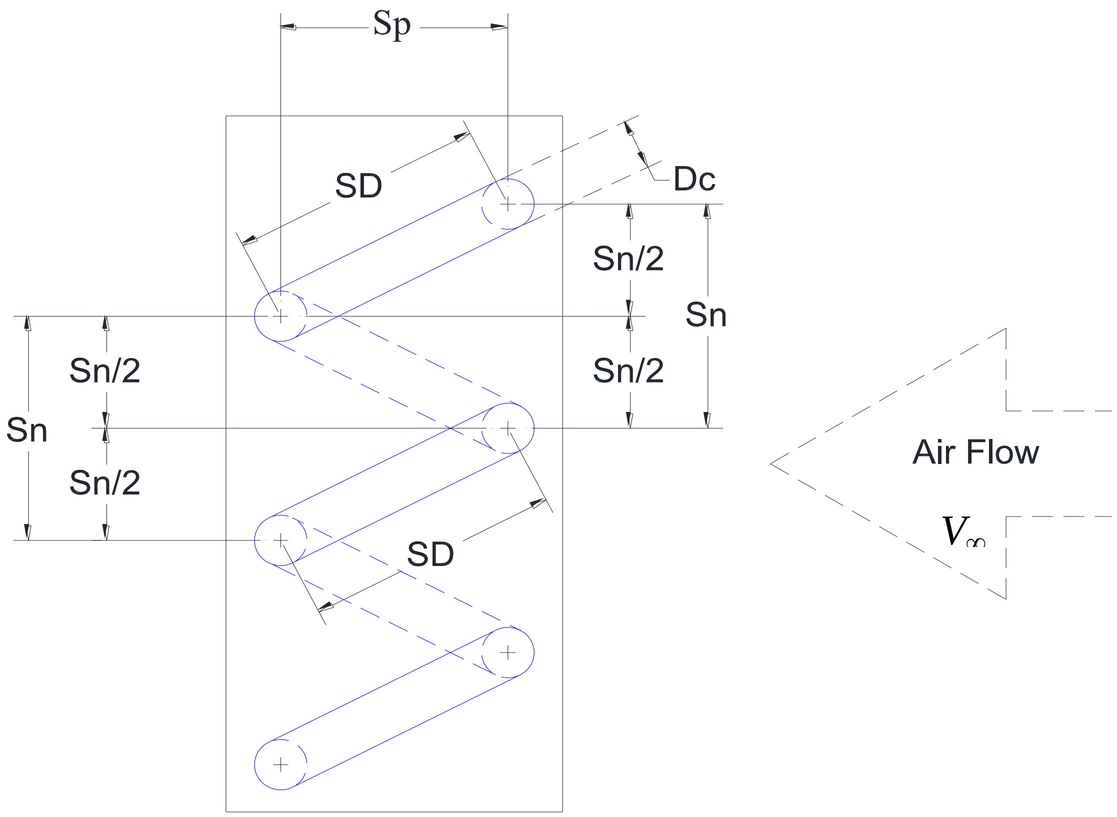

Figure 7. Z-type finned-tube dimensions

$$V_{\max} = V_\infty \frac{S_n}{2(S_D - D_c)} \tag{21}$$

$$S_D = \left[S_p^2 + \left(\frac{S_n}{2}\right)^2\right]^{1/2} \tag{22}$$

$$\mathrm{Pr} = \frac{\mu . C_p}{k} \tag{23}$$

Mass flow rate of the air to be assumed as a uniform distributed over the whole coil face regardless of the coil and fan respective locations. So each coil was associated with the same air mass flow rate. Also it was assumed that the air stream passing through the coil with a sufficient turbulence status so that the air with uniform properties was enters each tube bank.

For Turbulence flow, Dittus-Boelter correlation [9] has been used to calculate heat transfer coefficient for refrigerant side. The heat transfer coefficient is represented as:

$$h_r = 0.023\mathrm{Re}_f^{0.8}\ \mathrm{Pr}_f^{0.3}\ \left(\frac{k_f}{d_f}\right) \tag{23}$$

The heat transfer rate over the condenser of refrigerant side can be calculated from the following equation:

$$\dot{Q}_r = \dot{m}_r\,(h_2 - h_3) \tag{34}$$

3.1 Capillary Tube

Under normal operating conditions, there is a thermostat connected with capillary tube to maintain a fixed superheat exiting the evaporator.
The energy equation assumes that the enthalpy is constant across the expansion valve.

$$h_3 = h_4 \tag{42}$$

3.2 Evaporator

In this study an air-cooled evaporator was used. To keep the evaporator model simple, the coil is assumed to be dry, so the air-side heat transfer coefficient is not affected, but the specific heat is corrected to account for condensation. Because the air flowing over the evaporator is cooled below the wet bulb temperature, some of the heat rejected by the air results in condensing water out of the air rather than lowering the temperature. The total enthalpy change of the air is the sum of the enthalpy change due to temperature drop, or sensible heat, and the enthalpy change due to condensation, or latent heat.
The coefficient of performance (COP) is the ratio of the heat absorbed by the evaporator to the amount of compressor work as shown in the following equations:

$$COP = \frac{\dot{Q}_e}{\dot{W}_{com}} \tag{47}$$

$$\dot{W}_{com} = (h_2 - h_1) \tag{48}$$

$$\dot{Q}_e = (h_1 - h_4) \tag{49}$$

4. Results and Discussion

"Fig. 8", shows comparison between compression pressure ratio for the three of refrigerants (R-22 , R-407C and R404). It is clear that when using R-404 and R-407C gives satisfied compressor pressure ratio compared with R-22, the compression ratio increases when the ambient temperature increased, this due to the ambient temperature leads to raise the compressor pressure ratio. However, the compressor ratio of system using R404 and R407C is more than the compressor ratio of system with R22 by 2% and 4% respectively.

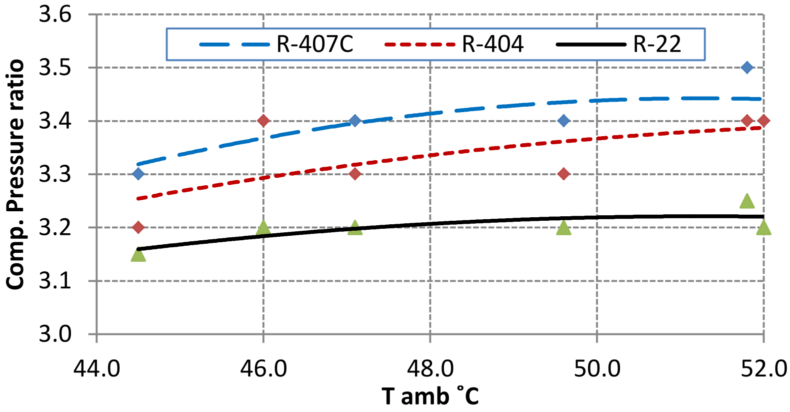

Figure 8. Compressor Pressure ratio for R-22 , R-407C and R404

"Fig. 9", shows the evaporator temperature comparison between the two type of refrigerants R-407C and R-404 compared with R-22. It is clear that there is an increase in outlet air evaporative temperature of system using R-22 compared to outlet air evaporative temperature of system using R-407C and R-404 by a percentage of 8% and 5% respectively. In this figure, a convergence in data measured from the unit that leads to possibility of replacing the refrigerant R-22 by R-407C and R-404.

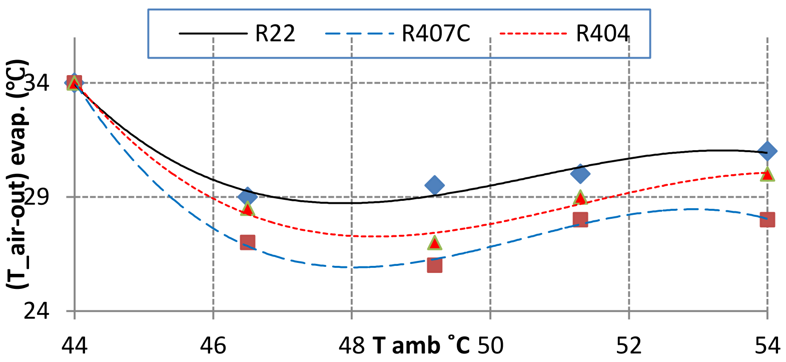

Figure 9. Evaporative Temperature for R-404 ,R-407C & R22

By using EES software (Engineering Equation Solver) for calculation and drawing charts, "Fig. 10, 11 & 12", show the (p-h) diagram of R-22 , R-407C and R-404 respectively, so due to data convergence that measured from the experimental test rig; The two shapes of (p-h) diagram are converged.

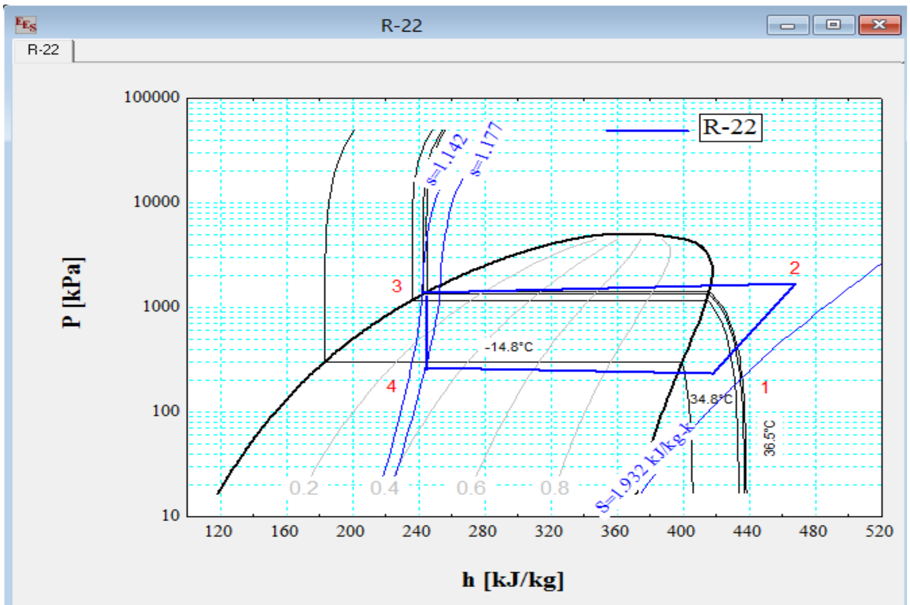

Figure 10. P-h Diagram of R-22

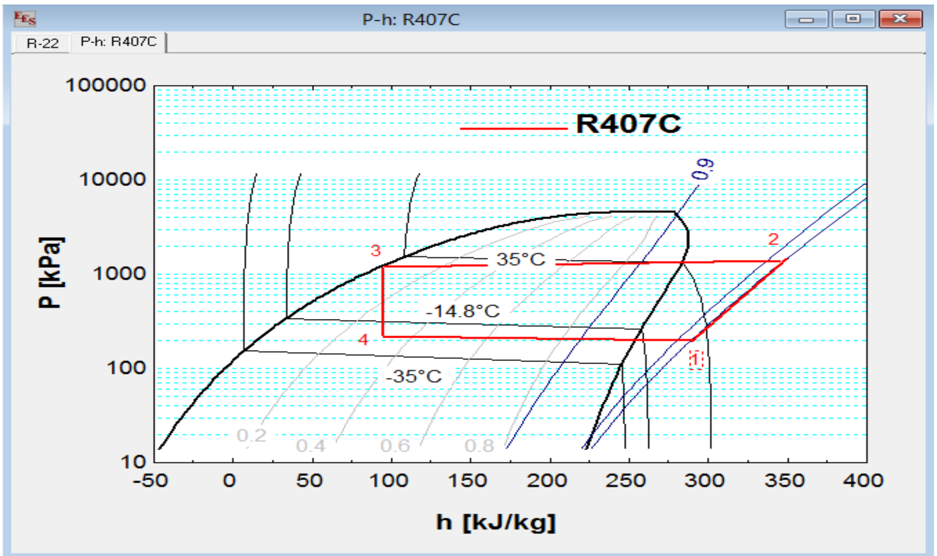

Figure 11. P-h Diagram of R-407C

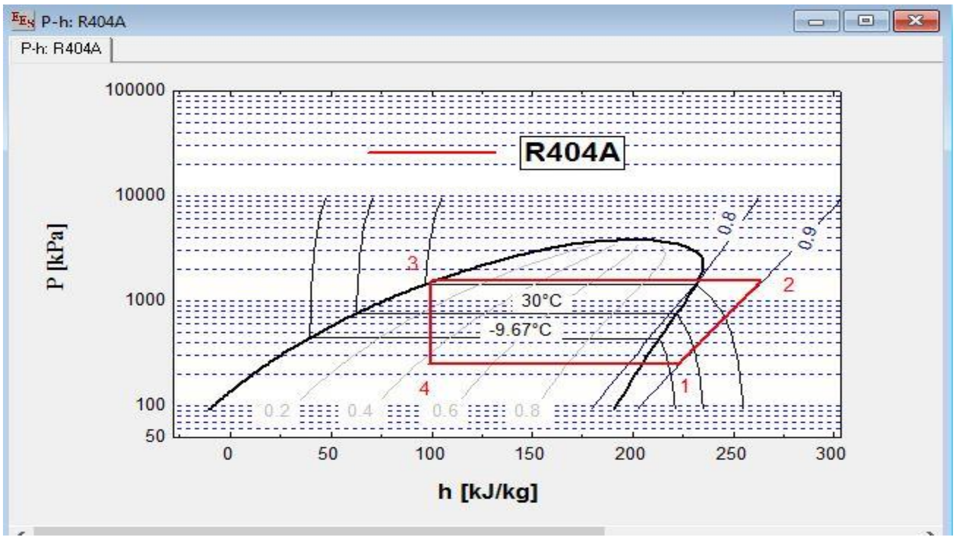

Figure 12. P-h Diagram of R-404

"Fig. 13", shows the COP behavior of the three types of the refrigerants R22, R407C and R404. However; The system using R407C has COP of 8% less than COP of system using R22, also; there is about COP of 12% for system using R404 compared with system using R22. So, it can be concluded that it is applicable to replace R-22 by R407C or R-404 in same air-conditioning unit.

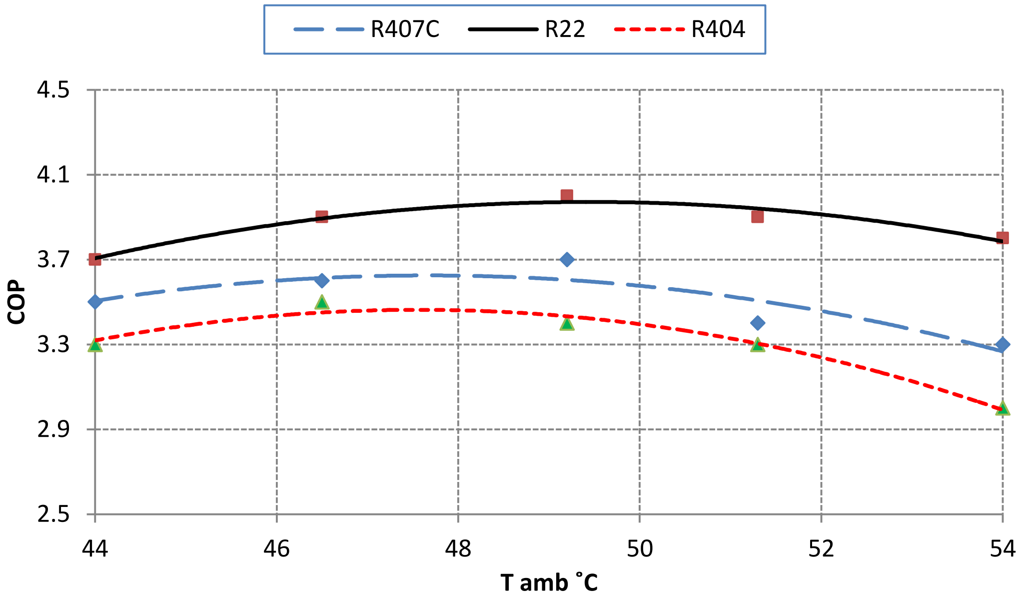

Figure 13. Coefficient of performance for R-22, R-407C& R-404

5. Conclusions

It is found that there are data convergence between the results of the compressor ratio of system using R-22 and system with R407C and R404. The compressor ratio of system using R407C is more than the compressor ratio of system using R22 by 4% and The compression ratio for R-404 is more than R-22 by 2%. The outlet air temperature of the evaporator for system using R-22 is more than the outlet air evaporative temperature with system using R-407C and R-404 by 8% and 5% respectively. The COP of the system using R-22 is more than the system using R-407C and R-404 by 8% and 12% respectively. So it can be concluded that it is possible to replace system refrigerant using R-22 by R-407C and R-404 as an alternative refrigerant that considered friendlier to environment.

Abbreviations

A	Area (m^2)
COP	Coefficient of Performance
C_p	Specific heat at constant pressure (J/kg. K)
D	Pipe Diameter (m)
h	Convection heat Transfer coefficient ($W/m^2.°C$)
h	Enthalpy (kJ/kg)

k	Thermal Conductivity (W/m.°C)
Nu	Nusselt Number
P	Pressure (kPa)
Pr	Prandtl Number
Q	Heat transfer (W)
Re	Reynolds Number
s	Entropy (kJ/kg)
S_n	Normal direction of tube bank centers (m)
S_p	Parallel direction of tube bank centers (m)
T_s	Surface temperature (°C)
V	Velocity (m/s)

Nomenclature

a	Air
Alt	Alternative
Btu/h	British Thermal unit per hour
c	Condenser
com	Compressor
e	Evaporator
HE	Heat Exchanger
m	mean
r	Refrigerant
sa	Saturated
sc	Sub Cooled
sh	Superheated

Greek Symbols

$\dot{m}$	Mass flow rate (kg/s)
μ	Kinematic Viscosity (Pa.s)
ρ	**Density** (kg/m3)
$\dot{V}$	Volume flow rate (m^3/s)

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