EXPERIMENTAL PERSPECTIVE ASSESSMENTS FOR A PROPER REFRIGERANT ALTERNATIVE TO R-22 IN A WINDOW-TYPE AIR CONDITIONING UNIT

Ali Hussain Tarrad
Al-Mustansiriya University
College of Engineering
Mech. Eng. Department
Baghdad-Iraq

Ameera Yaseen Salim
Al-Mustansiriya University
College of Engineering
Mech. Eng. Department
Baghdad-Iraq

ABSTRACT

The present research is concerned with the future phase-out of halocarbon types of refrigerants used in the vapor compression refrigeration systems. A window-type air conditioner is selected for the tests to be conducted with two different refrigerants. It is well known that these cooling units are wide spread in their applications and are circulating ***R-22*** as a refrigerant. It is intended to replace this type of refrigerant by another type which is considered to be as environmental friendly refrigerant for smooth operation. The refrigerant selected for this object was ***R-407C*** for its favorable thermal properties and acceptable pressure and temperature ranges. The effect of replacing this refrigerant as a substitute on the performance of the cooling unit was studied. The performance characteristics comprise of cooling capacity ($\dot{Q}_{evap}$), power consumed by the compressor ($\dot{W}_{cons}$) and the coefficient of performance (***COP***) of the air conditioning unit were obtained. The results showed that the pressure ratio of the ***R-407C*** for a proper operation was higher than that of the ***R-22*** by a mean value of (**13**) **%.** The power consumed per ton of refrigeration by the unit when circulating ***R-407C*** was about (**19**) **%** greater than the ***R-22*** values for the whole range of tests. Further, the experimental work has revealed that the actual ***COP*** of the unit was reduced by (**17**) **%** when circulating the alternative ***R-407C*** for the test conditions. The results also showed that this alternative for the present refrigerant with existing cooling unit requires the attention to the mass flow rate of refrigerant circulated and the pressure and temperature ranges throughout the system.

KEY WORDS

Refrigeration, Alternatives, R-22, Window Type, Air Conditioning, R-407C

وجهة نظر لتقييم تجريبي لاستخدام مائع تثليج مناسب بديلاً لفريون-22 في وحدة تبريد هواء من النوع الشباكي

الخلاصة

يهتم البحث الحالي بالإيقاف المستقبلي لاستخدام موائع التثليج الهالوكاربونية والتي يدخل في تركيبها الكيميائي الغازات الهالوجينية في منظومات التبريد ألإنضغاطية. لقد تم اختيار مكيف هواء من النوع الشباكي لإجراء التجارب العملية ولمائعي تثليج مختلفين. ومن المعروف إن هذه الوحدات واسعة الانتشار بالاستخدام وتعمل بواسطة تدوير (**Freon-22**) كوسيط للتبريد. لقد كانت الرغبة باستخدام مائع تثليج قليل التأثير السلبي على البيئة وله القابلية على تشغيل الوحدة بصورة طبيعية. لقد وقع الاختيار على (**Freon-407C**) كبديل مناسب خلال البحث الحالي نظراً لما يتميز به من مواصفات حرارية مفضلة وله القابلية للعمل بمديات مقبولة من حيث درجة الحرارة والضغط. تم دراسة التأثير الناتج من استخدام هذا المائع على أداء وحدة تكييف الهواء موضوع البحث. إن دراسة الأداء لوحدة التبريد تتضمن حساب كلاً من حمل التبريد ($\dot{Q}_{evap}$)، الطاقة المستهلكة من قبل الضاغط ($\dot{W}_{comp}$) ومعامل الأداء (**COP**).

لقد بينت النتائج لهذه الدراسة إن نسبة الضغط بين جانب الدفع والسحب للضاغط لمائع التثليج (**Freon-407C**) ولعمل مستقر ومناسب للوحدة كان أعلى مما عليه في حالة استخدام (**Freon-22**) كمائع للتثليج وبمعدل (**13 %**). الطاقة المستهلكة لكل طن تثليج باستخدام (**Freon-407C**) كانت أعلى بمقدار (**19 %**) من مثيلتها لحالة (**Freon-22**) ولكل مدى التجارب الحالية. بالإضافة لذلك فان النتائج العملية قد بينت بان معامل الأداء لوحدة التكييف قل بمقدار (**17 %**) عند تدوير (**Freon-407C**) عبر منظومة التبريد. كما وبينت الدراسة الحالية بان البديل المقترح عند استخدامه في منظومات التبريد يتطلب الاهتمام بمعدل التدفق ومديات الضغط التشغيلي ودرجة الحرارة عبر المنظومة.

INTRODUCTION

For nearly sixty years, chlorofluorocarbons (***CFCs***), have been widely used as solvent, foam blowing agents, aerosols and specially refrigerants due to their preeminent properties such as stability, non-toxicity, non-flammability, good thermodynamic properties and so on. However, they also have some disadvantages of harmful effect on the Earth's protective ozone layer known as ozone depletion potential (***ODP***). Subsequently, it was discovered that (***CFCs***) also contributed significantly to the global warming potential (***GWP***). The reputation, reliability and maintainability of equipment normally improves with age, but this has not been true when it comes to refrigeration field. The (1990) Clean Air Act amendments imposed new regulations with landmark dates that industry was unable to meet. So stop gap solutions and lack of time to do research and development have resulted in a lack of standards and poor information.

The direct replacement of ***R-12*** with the pure refrigerant ***R-134a*** has been experienced much earlier and an excellent knowledge has been performed in the retrofit for the old machines. The phase out of ***R-22*** as working medium in refrigerating equipments and heat pumps began in the late of 1990s with terminating of their new installation and manufacturing throughout Europe. The new working fluids for the alternative of ***R-22*** are zoetrope refrigerants introducing further difficulties for the prediction of the cooling unit characteristics behavior during smooth operation.

The most attractive thermal properties of refrigerant that have significant effect on the performance of the vapor compression refrigeration system are the critical temperature and the molar heat capacity as pointed out by **(McLinden, 1987)** and **(Domanski, 1999)**. **(Meurer et al., 1999)** compared the performance of ***R-22*** and ***R-410A*** for elevated condensing temperature up to (60) ˚C. They have found that the variation of the (***COP***) of the system depends on the level of the condensing operating temperature of the refrigerant circulated. **(Lee et al., 2000)** investigated the performance of mixture refrigerants alternatives for the ***R-22***. They compared the coefficient of performance, ***COP***, the volumetric capacity of refrigeration, ***VCR*** and capacity for the alternatives ***R-407C*** and ***R-410*A** with those of ***R-22***. Their results showed that ***R-410A*** exhibited very close ***COP*** to that of the ***R-22*** within (3%) for the same pressure ratio. ***R-407C*** gave lower ***COP*** than that of the ***R-22*** by (9%). Motta and **(Domanski, 2000)** simulated a number of binary and ternary mixtures as alternatives for the use of ***R-22*** when working in elevated outdoor temperatures. Their results showed that fluids with a low critical temperature experience a large degradation of cooling capacity, while rate of compressor power increase is similar for all fluids. Johansson and **(Lundqvist, 2003)** presented a qualitative and quantitative literature survey for the alternatives for ***R-12*** and ***R-22*** used in Sweden refrigeration industry.

Other trend of investigations was devoted to study the effect of replacing the alternatives on the working components of the existing system. For this category, the reader is referred to the work published by **(Kuelh, 1987)**, **(Kim, 1993)**, **(Wolf et al., 1995)**, **(Sami and coworkers, 1998** and **2000)**. **(Sami et al., 2005)** studied the use of ***R-407C***, ***R-410A*** and ***R-410B*** in addition to the ***R-22*** for a good range of pressure and mass flow rate for three different sizes of capillary tubes. Their results showed that the pressure drop across the capillary tube is significantly influenced by the diameter of the capillary tube and the entering refrigerant conditions. Further, the pressure drop decreases with the increase of the capillary diameter and that alternatives in general experience higher pressure drop than that of ***R-22***.

EXPERIMENTAL APPARATUS:

The test apparatus selected for this project was a two refrigeration tons window-type air conditioner cooling unit manufactured by (***LG***) commercial company. The overall physical external dimensions of the cooling unit are (62 × 72 × 41) cm, **(*LG* Company Manual, 1999)**. The overall dimensions and characteristic specification of the evaporator and condenser are shown in **Table 1**. The air conditioner is using a capillary tube as expansion component made of copper tubing having four equally length paths of refrigerant with a geometrical dimensions of diameter of (3) mm and length of (80) cm each. A typical tube circuiting for the refrigerant circulating path through the evaporator and condenser for a window type air conditioner is shown in **Fig. 1**. The tube layout arrangement for the test evaporator and condenser of the air conditioning unit is shown in **Fig. 2**.

The experimental rig is shown schematically in **Fig. 3**. Two (6) mm stainless steel sheath thermocouples of (***J***) type are installed at the entering and leaving sides of the capillary tube. These temperature sensors are used to measure the temperature at the exit and inlet lines of the evaporator and condenser refrigerant paths respectively. The temperatures at the suction and discharge sides of the compressor were measured by fixing a temperature sensor at the pipe surface delivering the refrigerant. The pressure throughout the air conditioner was measured by the application of two pressure gauges in the range of (-1 to 18) bar for the low side and (-1 to 38) bar for the high pressure side. These gauges were installed at the inlet and exit sides of the all of the components of the cooling unit. The manufacturer equipped the air conditioner unit with a single reciprocating compressor.

In the present work, the pressure drop of the refrigerant flowing through the evaporator and condenser heat exchangers were also considered. The air side measurements were only conducted for the dry and wet bulb temperatures at the inlet and exit sides of both heat exchangers. The air dry bulb temperatures were measured by using a digital thermometer of the (***K***) type and the wet bulb values measured using the wet cotton wick at the bulb of a mercury glass thermometer. The thermocouples for the temperature measurements were calibrated against a mercury glass thermometer and showed an accuracy of about (0.1%). Further, the refrigerant side pressure gauges were also calibrated and exhibited an accuracy of about (0.2%).

TEST PROCEDURE:

On commencing of the tests, the air conditioner was allowed to operate for at least (10) minutes, which is the time required to approach the steady state conditions. The steady state was considered to be established when the temperature and pressure all around the cooling unit to be stable with time. This was achieved by the reading indications of the temperature sensors and pressure gauges for the refrigerant circulated. After that establishments, the temperature and pressure measurements were recorded all around the selected ports in the cooling unit. The data was collected for different operating conditions to assure the reproducibility of the measurements and to detect any variation in the environment effects. These tests were carried out for different ambient conditions concerning the air dry bulb and wet bulb temperatures. This was done to show the effect of ambient condition variation on the performance of the air conditioning unit.

After, finishing the tests with ***R-22***, the unit was discharged, cleaned, evacuated, and kept under (2) bar vacuum followed by recharging with the suggested alternative ***R-407C***. The proper amount of refrigerant required for circulation was limited by the allowable pressure of the cooling unit. Since, the alternative is assumed to work with the same heat exchangers and compressor, then the allowed volumetric flow rate of refrigerant that passing through the unit will be limited by that of the ***R-22*** value. This refrigerant amount was limited by the suction pressure of the compressor corresponds to a value close to (70) psig. The expansion component of the cooling unit comprises of copper capillary tube was used with the same length and inner diameter.

DATA REDUCTION

The measured operating conditions were used for the prediction of the air conditioner performance including the power consumed, refrigerating effect and condenser load calculated per unit mass of refrigerant for both tested refrigerants. Further the measure for the performance characteristics, the ***COP*** was predicted from the above parameters. The refrigerant thermal properties for both fluids were obtained from the published tables presented by **(*ASHRAE* Handbook, 1997)**.

The refrigerating effect achieved by the cooling unit through the evaporator can be expressed as:

$$\dot{q}_{evap} = h_1 - h_4 \quad (1)$$

The amount of the *R-22* refrigerant circulated through the air conditioner was obtained from the known cooling capacity of the unit in the form:

$$\dot{m}_r = \frac{Q_{evap}}{\dot{q}_{evap}} \quad (2)$$

Where

$$\dot{Q}_{evap} = \dot{m}_{air}\,(h_{ai} - h_{ao}) \quad (3)$$

Equation (2) was also used to predict the ***R-407C*** refrigerant mass flow rate circulated through the unit. The volumetric flow rate of air circulated through the cooling coil, evaporator is the same as that of the ***R-22*** circulating case.

It is worthwhile to mention here that the air conditioner capacity is not necessary to have the same value as that of the manufacturer for ***R-22***. However, the air mass flow rate passing through the evaporator and condenser coils are still constants and having the same values as those provided for the ***R-22*** case. The manufacturer manual reveals that the maximum indoor and outdoor circulating air rates are (12) m^3/min and (26) m^3/min respectively.

The work rate done by the reciprocating compressor may be estimated per unit mass of refrigerant from:

$$\dot{w}_{comp} = (h_2 - h_1) \quad (4)$$

The air conditioner condenser load is estimated by:

$$\dot{q}_{cond} = (h_2 - h_3) \quad (5)$$

At the exit of the condenser, the liquid enthalpy was calculated from the knowledge of the pressure and temperature as measured during the experiments. This liquid may be at sub-cooled condition depending on the measured condition. The enthalpy of the refrigerant mixture leaving the capillary tube was considered to be equal to that of the entering liquid to capillary tube. Here, the process was assumed to be isenthalpic through the capillary tube as deduced from the first law of thermodynamics.

Finally, the coefficient of performance of the cooling unit was obtained from:

$$COP = \frac{\dot{q}_{evap}}{\dot{w}_{comp}} \quad (6)$$

For the purpose of comparison between the refrigerants performance used in the present work a new parameter is suggested to be used having the expression:

$$WRR = \frac{(\dot{w}_{comp})_{R-407C}}{(\dot{w}_{comp})_{R-22}} \quad (7)$$

This parameter represents the ratio of the work rate between the alternative ***R-407C*** and the original circulating refrigerant ***R-22***. The parameters predicted from eq. (7) are to be based on the physical values required per ton of refrigeration. Another parameter can be defined for the ***COP*** comparison, that is the coefficient of performance ratio, ***COPR***. This represents the ratio of ***COP*** of the ***R-407C*** to that of the ***R-22*** in the form:

$$COPR = \frac{COP_{R-407C}}{COP_{R-22}} \qquad (8)$$

This expression may be estimated for both of the isentropic compression and actual cycles.

EXPERIMENTAL RESULTS AND DISCUSSION:

It is of the beneficial object to highlight some important aspects of the present work in this category. The following points should be considered:

i- The same air conditioner was used which means that the heat exchangers and compressor capacities are limited to those of the ***R-22*** case. This will limit the heat transfer area, pressure ratio and power consumed for the alternativ0e to those available of the existing equipments.

ii- The tube circuiting for the evaporator and condenser were unaltered. Therefore, the thermal characteristic parameter, (***UA***), is limited to that of the ***R-22*** refrigerant condition. In other words, the ***R-22*** characteristic value will be the controller of the operation for the alternative performance.

The above will impose some restrictions for the allowable mass flow rate of the alternative refrigerant ***R-407C*** circulating through the air conditioning unit and operating conditions. The pressure ratio of the compressor and its volumetric displacement are the controller for the charging amount of the refrigerant ***R-407C***. Further, the physical geometry of the available heat exchangers control the heat transfer rate through the air conditioner.

A typical measured results of the tests conducted during the circulation of ***R-22*** and ***R-407C*** are shown in **Tables 2** and **3** respectively. The data presented in these tables show the temperature and pressure measurements at the selected points through the cooling unit. The final results for the tests when circulating the refrigerants ***R-22*** and ***R-407C*** are shown in **Tables 4** and **5** respectively for the test conditions considered in the present study. In these tables the followings were considered to exist:

a- The (***IMPC***) represents the cycle in which the suction and discharge pressures were considered to be the mean value at the entering and leaving sides of the evaporator and condenser respectively. The condenser load and the compression work rate were calculated for the case of the isentropic compression for the refrigerant.

b- The (***AMPC***) represents the cycle at which the calculation of the performance was based on the actual measured pressure for all of the selected ports throughout the cycle.

The actual and isentropic (***p-h***) diagrams for the ***R-22*** and ***R-407C*** circulating refrigerants are shown in **Fig. 4**. Here, the dotted lines represents the isentropic mean pressure cycle (***IMPC***), whereas the solid lines express the actual measured pressures cycle (***AMPC***). The actual measured air side conditions for both of the evaporator and condenser of the cooling unit obtained during the tests are shown schematically on the psychrometric chart in **Fig. 5**. All the performance assessment of the present work was based on a (12) m^3/min of indoor air circulation, **(LG Company Manual, 1999)**.

REFRIGERANT R-22 TESTS:

It is clear that the actual apparent compression of the compressor is lower than that of the isentropic value due to the friction losses and heat losses during the operation of the compression process. The final results for the actual measured data obtained for the cycle (***abcd***), **Fig. 4.a**, are shown in **Table 4.b**. In this table, the power consumption, $\dot{W}_{cons}$, during the operation of the unit was estimated from the measured values of the current and voltage of the power supply. During the steady state conditions, the current was ranged between (11.2) and (11.4) amperes while the voltage was almost constant at (220) volts.

$$\dot{W}_{cons} = I \times V \times \cos\varphi \qquad (9)$$

In which the power factor (***cos φ***) was assumed to be (0.94) according to the manufacturer manual.

The isentropic compression through the compressor is higher than that of the actual compression for all of the tests conducted in the present work. The ratio of the actual to that of the isentropic compression, $\dot{W}_{ratio}$, expressed mathematically as:

$$\dot{W}_{ratio} = \frac{W_{meas}}{W_{cs}} \qquad (10.a)$$

is ranged between (69) and (91) % for the whole range of the test conditions as shown in **Table 4.b**. The efficiency of the actual compression is defined as:

$$\eta_{ca} = \frac{W_{meas}}{W_{cons}} \qquad (10.b)$$

This parameter shows the ability of the unit to make advantage of the power input to the compressor. This parameter revealed that the actual useful work of the unit was ranged between (41) and (52) %. These values were lower than those of the isentropic compression where (η_{cs}) was ranged between (53) and (65) %. Clearly, this was due to the irreversibility of the compression process where it has a deviation from the ideal behavior.

The measured pressure ratios (p_r) of ***R-22*** tests were ranged between (3.73) and (3.84) as shown in **Table 2.a** and the ***COP*** of the cooling unit was ranged between (3.8) and (4.3) when the calculation based on the (***IMPC***) mode, **Table 4.a**. The corresponding actual values based on the consumed power (***AMPC***) was ranged between (2.2) and (2.5), **Table 4.b**, as estimated from:

$$COP_{cons} = \frac{Q_{evap}}{W_{cons}} \qquad (10.c)$$

It is obvious that the (***IMPC***) has higher coefficient of performance than that of the (***AMPC***) for the whole range of the test conditions to be within the range of (36-47) %. This reveals that the actual compression operating process follows a path different from that of the constant entropy process on the (***p-h***) diagram. Further, the useful work for compression in the latter mode of calculation was lower than that of the former mode which represents the expected behavior for the irreversible process mode through the compressor.

The measured parameters for the air side of test number (1) are shown schematically on the psychrometric chart in **Fig. 5.a**. Here, the process throughout the evaporator exhibited almost a straight horizontal line revealing a major part of the cooling load to be a sensible heat only, process line no. (2). As the usual operation process through the condenser in which it exhibits a sensible load only, line no. (2) represents the air side process on the psychrometric of test no. (1).

Fig. 6 shows the (***IMPC***) and (***AMPC***) performance variables variation with the cooling unit capacity. These variables include, the power consumption per ton of refrigeration, ***COP***, condenser load, power consumed and the calculated work rate of the compressor. The last column of **Table 4** represents the estimated condenser air flow rate for both of the modes of calculations. It is obvious that the actual required air flow rate for (***AMPC***), **Table 4.b**, was higher than that of the ideal (***IMPC***) cycle, **Table 4.a**, by a range of (10-15) %. This was due to the increasing load required for the condenser as a result of the higher power required to run the compression unit.

Refrigerant R-407C Tests:

All of the above parameters were calculated for the alternative refrigerant ***R-407C*** applying the same relations and modes of the considered category of the estimation procedure. The (***p-h***) diagram of both of the (***IMPC***) and the (***AMPC***) results are shown in **Fig. 4.b** for test number (1) presented in **Table 3.a** for the refrigerant side of the cycle. The air side processes are presented schematically on the psychrometric chart in **Fig. 5.b** for both of the evaporator and condenser heat exchanger sides of test no. (1).

For all of the tested pressure ratio conducted using this refrigerant, the isentropic useful work was higher than that of the actual compression due to the irreversibility of the compression process. The (η_{ca}) values were ranged between (44) % and (55) %, **Table 5.a**, whereas, the corresponding values of the (η_{cs}) were within the range of (47 - 56) %, **Table 5.b**. The isentropic coefficient of the performance exhibited a range of (3.8 – 3.9) with a mean value of (3.86). The corresponding values for the actual cycle (***AMPC***) were in the range of (1.60) to (2.16) having a mean value of (1.79) which lower than that of the (***IMPC***) by (53) %. The trend of the data showed almost constant power consumption for the tested cooling load capacity in this work, **Fig. 7.a**. The actual ***COP*** showed an increase when the cooling unit load was increased, **Fig. 7.b**. It was observed that the power consumption per ton increases with load reduction. This was also noticed during the tests when circulating ***R-22*** as shown in **Table 4.b**.

RESULTS ASSESSMENT:

For the purpose of comparison of the results, **Table 6** was prepared for the ratios presented in eqs. (7 & 8) for the (***AMPC)*** and the (***IMPC)*** cycles. These results based on a mean values of the predicted variables of **Tables 4 & 5**. It is useful to construct the following argument about the experimental assessments of the present work:

i- The data showed that the coefficient of performance of the ***R-407C*** refrigerant was lower than that of the ***R-22*** results for both modes of calculations. Moreover, the ***COPR*** was ranged between (**0.94**) and (**0.83**) for the (***IMPC***) and (***AMPC***) modes respectively. This means that the ***COP*** of the unit was lower for the ***R-407C*** than that of the ***R-22*** circulating refrigerant.

ii- Accordingly, the power consumed per ton of refrigeration, WRR_{cons}, for the ***R-407C*** was higher than that of the ***R-22*** by (**19**) **%** although the useful work rate of the latter was lower than that of the former by (**5**) **%.** This is partly due to the higher working pressure range of the ***R-407C*** than that of the ***R-22*** during the steady state conditions, **Table 6**.

iii- For all tests of both refrigerants, the air mass flow rate for the (***IMPC***) and (***AMPC***) modes to pass through the condenser showed an increase with cooling capacity of the unit due to the increase of the condenser load.

iv- The circulated refrigerant per ton of refrigeration for the ***R-407C*** refrigerant was in the range between (**81-88**) kg/hr. This was greater than that of the ***R-22*** case which has the range of (**79 – 82**) kg/hr. This is related to the physical properties of both refrigerants and essentially is not dependent on the operating pressure ratios.

v- The experimental data shown in **Tables 4** and **5** reveals that the work rate ratios ($\dot{W}_{ratio}$) for the ***R-407C*** tests were higher than those of the ***R-22*** results. In other words, the calculated useful work rate ($\dot{W}_{meas}$) was closer to the isentropic ($\dot{W}_{cs}$) condition for ***R-407C*** than that of the ***R-22*** circulating case.

CONCLUSIONS:

The experimental work conducted in the present work revealed the following findings:

i- The suggested alternative refrigerant ***R-407C*** is one of the promising substitute for the ***R-22*** in the air conditioning cooling systems. This object can be targeted without a major alteration in the geometrical and physical dimensions of the existing unit.

ii- Higher pressure ratio has been experienced during this work for the ***R-407C*** than that of the present circulating refrigerant ***R-22*** for the object of system operation with the existing geometrical cooling unit design.

iii- The power consumed per ton of refrigeration was higher for the present alternative than that of the ***R-22*** by about (**19**) **%** calculated as a ratio of power consumed with respect to the base refrigerant ***R-22***..

iv- The above was accompanied by a reduction of the actual ***COP*** of the air conditioning unit by (**17**) **%** and an increase in the circulated mass flow rate of refrigerant per ton.

v- The assessment formulated in the present work recommends to use ***R-407C*** as a direct alternative for the refrigerant ***R-22*** in this type of air conditioning equipments.

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NOMENCLATURE:

COP : Coefficient of Performance defined by eq.(7)
COPR : Coefficient of Performance Ratio
h : Enthalpy, (kJ/kg)
$\dot{m}$: Mass Flow Rate, (kg/s)
p : Pressure, (bar)
p_r : Pressure Ratio
P : Work rate per ton of refrigeration, (kW/TR)
$\dot{Q}$: Heat Transfer Rate, (kW)
$\dot{q}$: Heat Transfer per Unit Mass, (kJ/kg)
T : Temperature, (ºC or K)
WRR: Work Rate Ratio
$\dot{W}$: Work Rate, (kW)

Greek Letters:

Δp : Pressure Difference, (bar)
η : Efficiency

Subscript:

a : Air
ca : Actual Compression Value
cond : Condenser
comp : Compressor
cons : Consumed Value
cs : Isentropic Compression Value
d : Dry Bulb
evap : Evaporator
i : Inlet
o : Outlet Value
r : Refrigerant
w : Wet Bulb

Table (1): The Characteristic Physical Dimensions of the Evaporator and Condenser Geometry.

Dimension specification	Evaporator	Condenser
Tube Length (mm)	390	570
Inner Tube Diameter (mm)	7.93	6.34
Outer Tube Diameter (mm)	9.52	7.94
Transverse Tube Pitch (mm)	19.25	17.32
Longitudinal Tube Pitch (mm)	28	20
Number of Tube Circuits	4	3
Number of Tubes per Circuit	11	23
Total Number of Tubes	44	76
Number of Tube Rows	3	4
Tube Metal	Copper	Copper
Tube Metal Thermal Conductivity (W/m.K)	386	386
Inner Tube Surface	Smooth	Smooth
Fin Thickness (mm)	0.2	0.2
Fin Pitch (mm)	1.5	1.5
Number of Fins per Inch (FPI)	17	17
Fin Type	Flat Plate	Flat Plate
Fin Metal	Aluminum	Aluminum
Fin Thermal Conductivity (W/m.K)	202	202

Table (2): Experimental Data of Air Conditioning Unit Circulating *R-22*.

Table (2.a): Refrigerant Side Measured Operating Conditions.

Test No.	Δp_{cond} (bar)	Δp_{evap} (bar)	P_1 bara	P_2 Bara	P_3 bara	P_4 bara	T_1 (ºC)	T_2 (ºC)	T_3 (ºC)	T_4 (ºC)	p_r (---)
1	0.3	0.1	5.71	22.51	21.81	5.81	10.0	81.0	43.6	9.0	3.846
2	0.7	0.3	5.71	22.71	22.01	6.01	11.0	82.0	42.0	8.5	3.814
3	0.7	0.1	5.61	21.51	20.81	5.71	9.0	82.0	42.0	9.0	3.737
4	0.8	0.1	5.51	21.31	20.51	5.61	9.0	81.0	42	9.0	3.76

Table (2.b): Air Side Measured Operating Conditions.

Test No.	Heat Exchanger Type	T_{di} (˚C)	T_{wi} (˚C)	T_{do} (˚C)	T_{wo} (˚C)
1	Evaporator	30.8	16	9.3	5. 8
	Condenser	35.2	22	58.5	28
2	Evaporator	31.2	18	10.2	9.0
	Condenser	34.5	22	60.0	28.5
3	Evaporator	29.7	15	8.2	5.5
	Condenser	31.8	25	52.2	30
4	Evaporator	28.2	14	8	5.0
	Condenser	32.4	18	57.4	25
5	Evaporator				
	Condenser				

Table (3): Experimental Data of Air Conditioning Unit Circulating *R-407C*.

Table (3.a): Refrigerant Side Measured Operating Conditions.

Test No.	Δp_{cond} (bar)	Δp_{evap} (bar)	P_1 bara	P_2 bara	P_3 Bara	P_4 bara	T_1 (ºC)	T_2 (ºC)	T_3 (ºC)	T_4 (ºC)	p_r (---)
1	0.3	0.1	5.51	24.31	24.01	5.61	18.0	89	45	9.7	4.34
2	0.5	0.1	5.21	23.51	23.01	5.31	16	89	49	12	4.42
3	0.5	0.2	5.31	24.01	23.51	5.51	15	90	50	12	4.31
4	0.6	0.15	5.66	25.11	24.51	5.81	15.6	89	53	11	4.32

Table (3.b): Air Side Measured Operating Conditions.

Test No.	Heat Exchanger Type	T_{di} (˚C)	T_{wi} (˚C)	T_{do} (˚C)	T_{wo} (˚C)
1	Evaporator	25.0	13	6.6	5.0
	Condenser	30.6	16	54.6	23.5
2	Evaporator	25	12	6	3.0
	Condenser	28	16	54	24
3	Evaporator	26	13	6	3.0
	Condenser	29	16	55	24
4	Evaporator	26	13	6.1	3.0
	Condenser	32	17	57	24.5

Table (4.a): The Performance Criteria of (*IMPC*) Mode Circulating *R-22*.

Test No.	$\dot{Q}_{evap}$ (kW)	$\dot{W}_{cs}$ (kW)	$\dot{Q}_{cond}$ (kW)	COP_s (----)	$\dot{m}_r$ (kg/h)	$\dot{W}_{cons}$ (kW)	P_s (Kw/TR)	η_{cs} (%)	$\dot{m}_{acs}$ (kg/h)
1	5.796	1.533	7.330	3.84	134.6	2.358	0.930	65.0	1082
2	5.520	1.314	6.834	4.20	124.5	2.316	0.837	56.7	846.1
3	5.324	1.226	6.55	4.34	119.32	2.296	0.810	53.4	1126
4	4.913	1.231	6.144	3.99	110.8	2.275	0.881	54.1	864.6

Table (4.b): The Performance Criteria of (*AMPC*) Mode Circulating *R-22*.

Test No.	$\dot{Q}_{evap}$ (kW)	$\dot{W}_{cons}$ (kW)	$\dot{Q}_{cond}$ (kW)	COP_{cons} (----)	$\dot{W}_{meas}$ (kW)	$\dot{W}_{ratio}$ (%)	P_{cons} (kW/TR)	$\dot{m}_r$ (kg/h)	η_{ca} (%)	$\dot{m}_{ac}$ (kg/h)
1	5.796	2.358	8.154	2.460	1.060	69.1	1.408	131.64	45.0	1203.2
2	5.520	2.316	7.840	2.383	1.195	91.0	1.475	123.6	51.6	971.6
3	5.324	2.296	7.620	2.319	1.041	85.0	1.516	119.0	45.4	1310.4
4	4.913	2.275	7.188	2.160	0.936	76.0	1.628	110.45	41.1	1012.0

Table (5.a): The Performance Criteria of (*IMPC*) Mode Circulating *R-407C*.

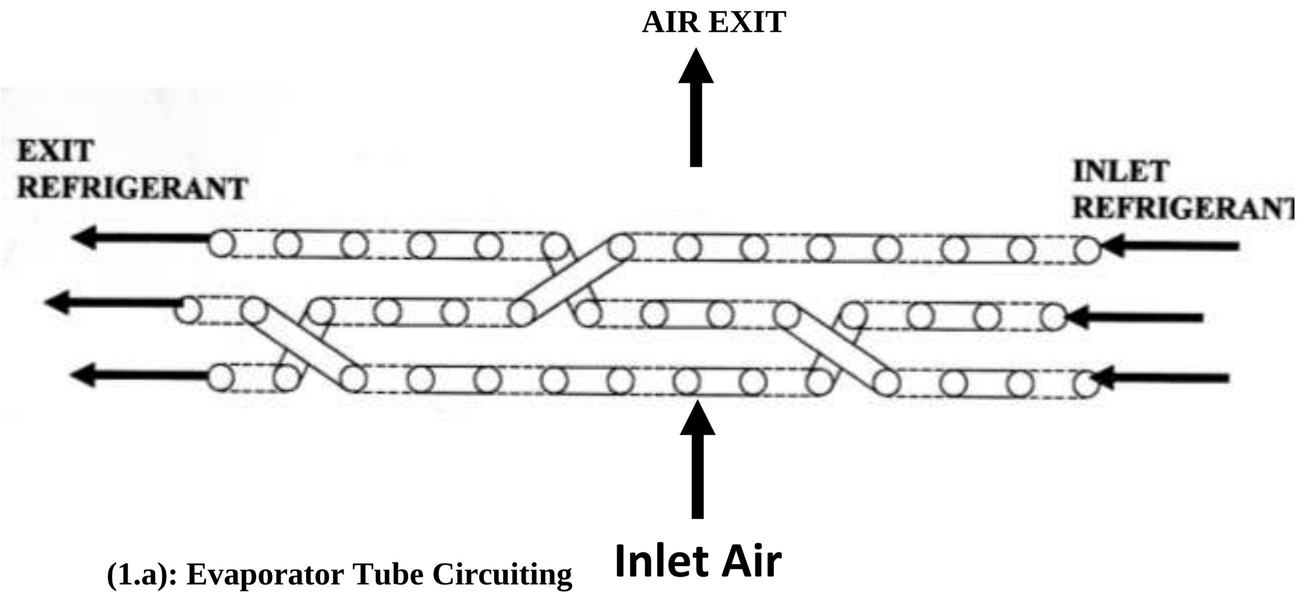
Test No.	$\dot{Q}_{evap}$ (kW)	$\dot{W}_{cs}$ (kW)	$\dot{Q}_{cond}$ (kW)	COP_s (----)	$\dot{m}_r$ (kg/h)	$\dot{W}_{cons}$ (kW)	P_s (kW/TR)	η_{cs} (%)	$\dot{m}_{acs}$ (kg/h)
1	4.235	1.111	5.346	3.811	97.57	2.378	0.922	46.7	740.2
2	4.471	1.141	5.61	3.92	105.3	2.399	0.898	47.6	734.4
3	5.17	1.352	6.523	3.823	124.83	2.399	0.920	56.4	869.7
4	5.20	1.337	6.583	3.890	130.1	2.440	0.903	54.8	923

Table (5.b): The Performance Criteria of (*AMPC*) Mode Circulating *R-407C*.

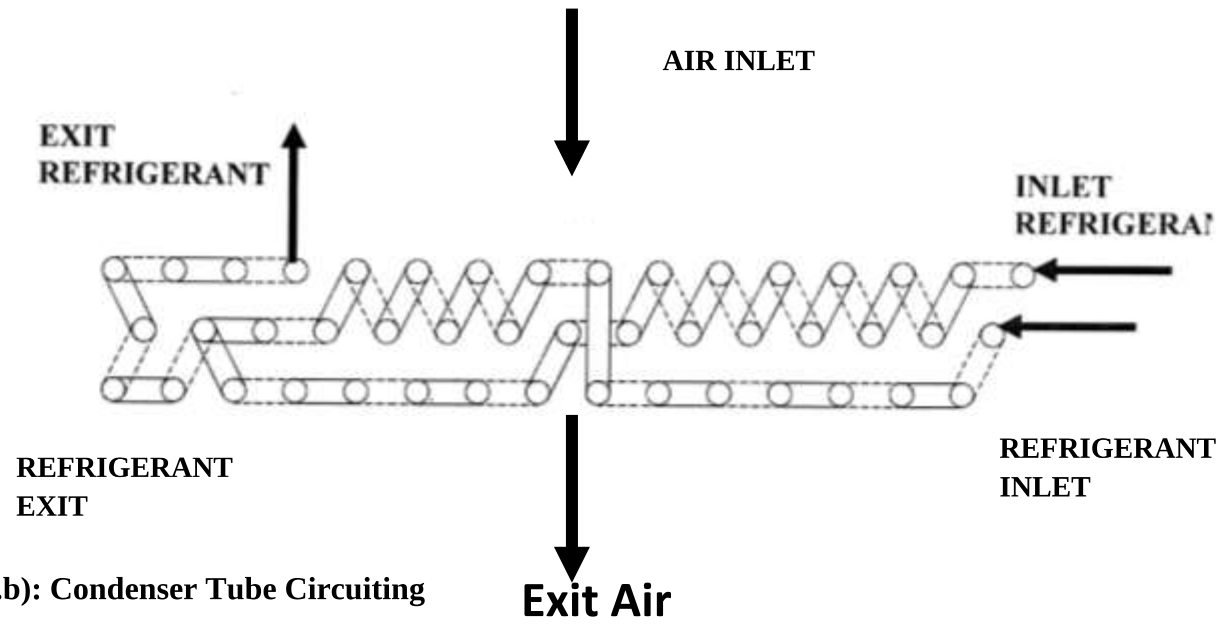
Test No.	$\dot{Q}_{evap}$ (kW)	$\dot{W}_{cons}$ (kW)	$\dot{Q}_{cond}$ (kW)	COP_{cons} (----)	$\dot{W}_{meas}$ (kW)	$\dot{W}_{ratio}$ (%)	P_{cons} (kW/TR)	$\dot{m}_r$ (kg/h)	η_{ca} (%)	$\dot{m}_{ac}$ (kg/h)
1	4.235	2.378	6.613	1.580	1.051	94.6	1.973	97.50	44.2	915.7
2	4.471	2.399	6.870	1.864	1.063	93.2	1.889	104.8	44.3	899.3
3	5.17	2.399	7.570	2.160	1.310	97.0	1.631	124.7	54.6	1009.3
4	5.20	2.44	7.640	2.131	1.147	85.8	1.649	129.9	47.0	1078.6

Table (6): A Comparison for the Work Rate and (*COP*) Ratios of *R-22* and *R-407C*.

Refrigerant Type	p_r (----)	$COPR_s$ (----)	$COPR_{cons}$ (----)	WRR_s (----)	WRR_{cons} (-----)
R-22	3.73 ---3.84	1.00	1.00	1.00	1.00
R-407C	4.31--- 4.42	0.94	0.830	1.05	1.19



(1.a): Evaporator Tube Circuiting



(1.b): Condenser Tube Circuiting

Figure (1): A Typical Heat Exchangers Tube Circuiting of an Air Conditioner.

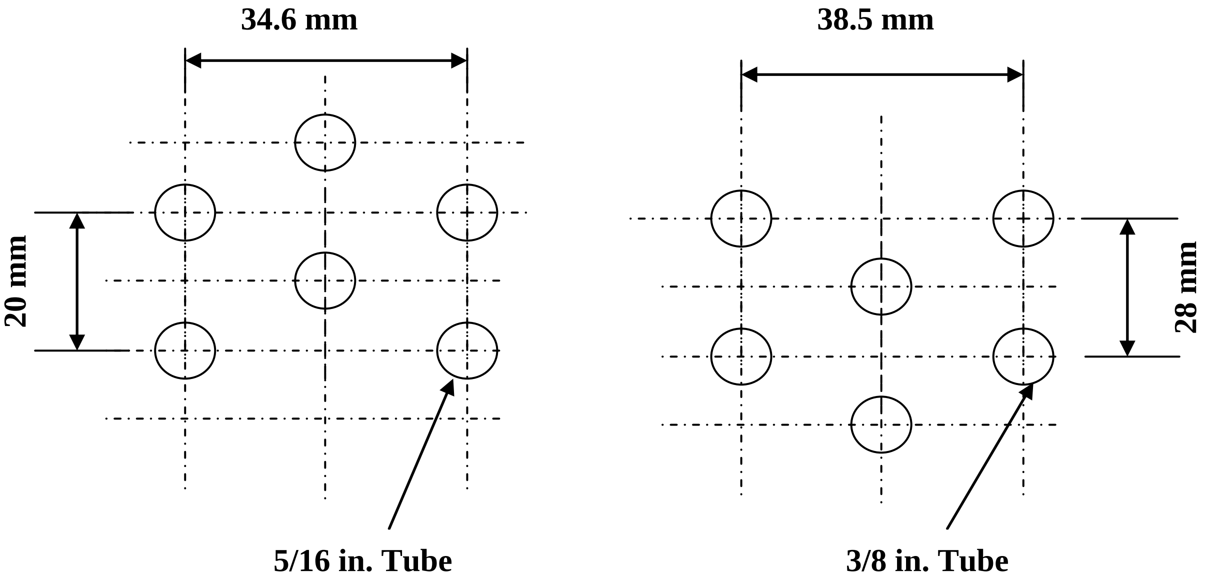


Figure (2.a): Condenser Tube Arrangement **Figure (2.b): Evaporator Tube Arrangement**

Figure (2): The Tube Layout Arrangement of the Test Air Conditioner.

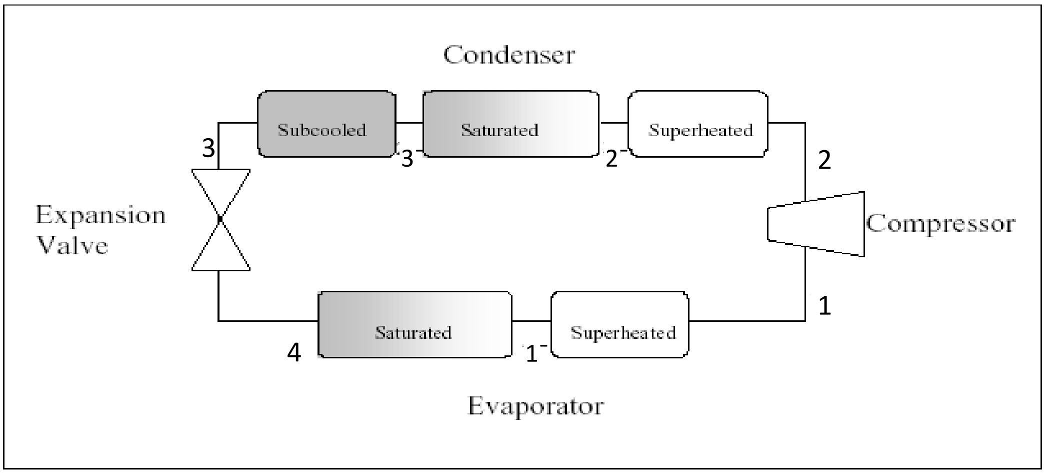


Figure (3.a): A Schematic Diagram for a Vapor Compression Refrigeration System.

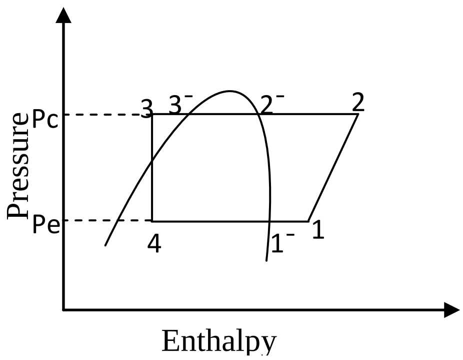


Figure (3.b): A Standard Vapor Compression Cycle Considering Superheating and Sub-cooling.

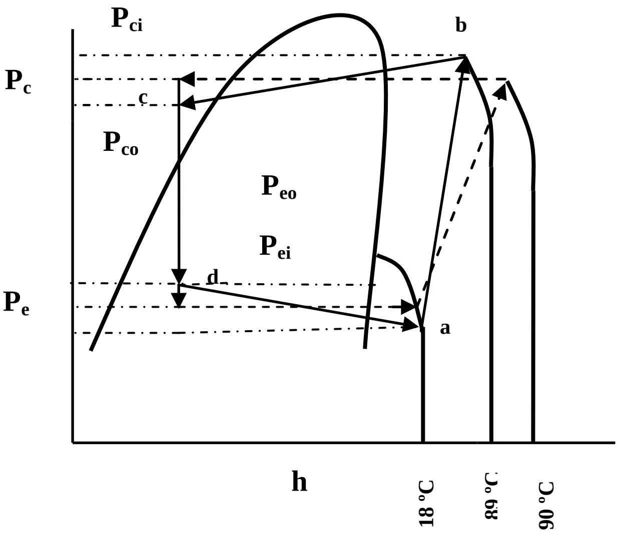


Figure (4.b): Test Number (1) for *R-407C*

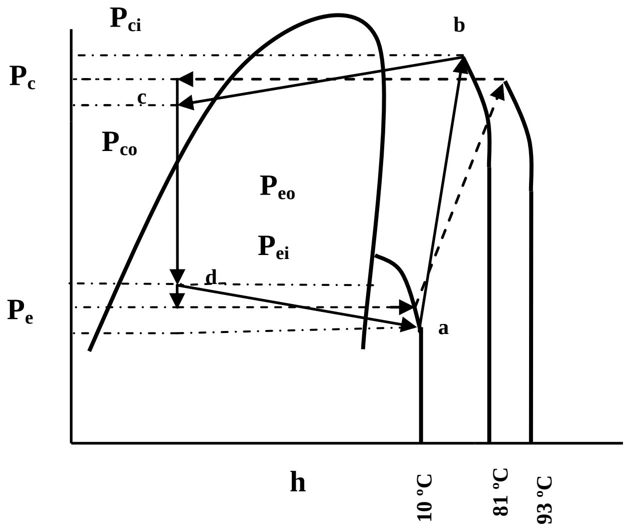
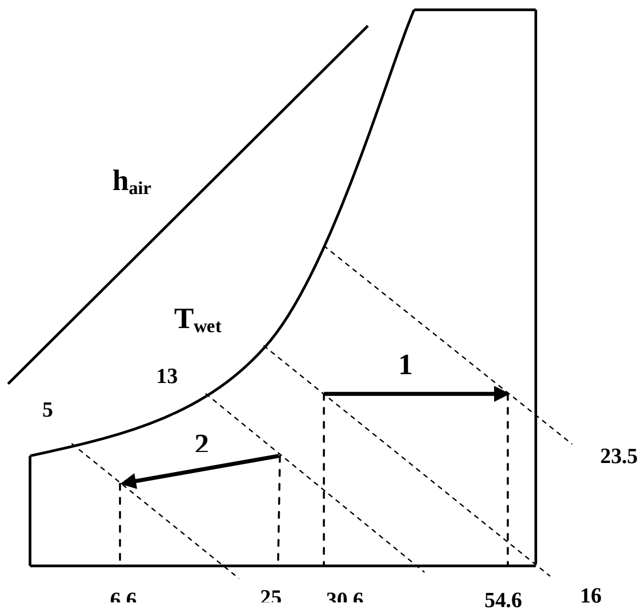


Figure (4.a): Test Number (1) for *R-22*

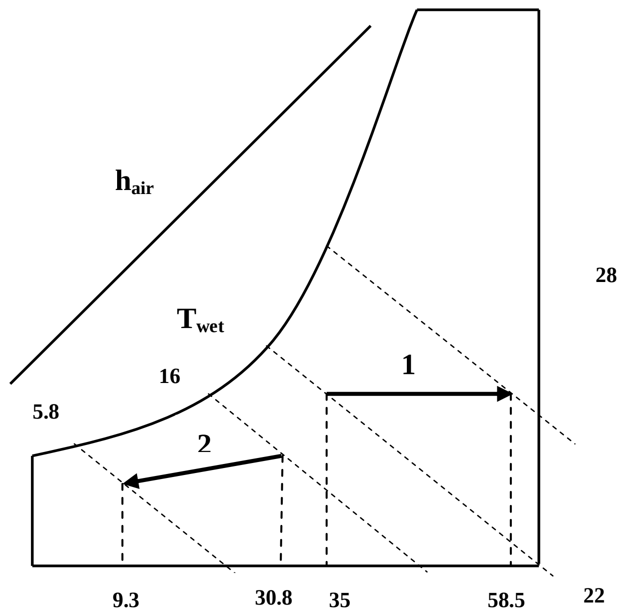
Figure (4): A Typical (p-h) Diagram of the *R-22* and *R-407C* Refrigerants Tests.



(1) Condenser Process
(2) Evaporator Process

Dry-Bulb Temperature ºC

Figure (5.b) Test No. (1), *R-407C*



Dry-Bulb Temperature ºC

(1) Condenser Process

Figure (5.a) Test No. (1), *R-22*

Figure (5): A Typical Psychrometric Process Chart of the *R-22* and *R-407C* Refrigerants Tests.

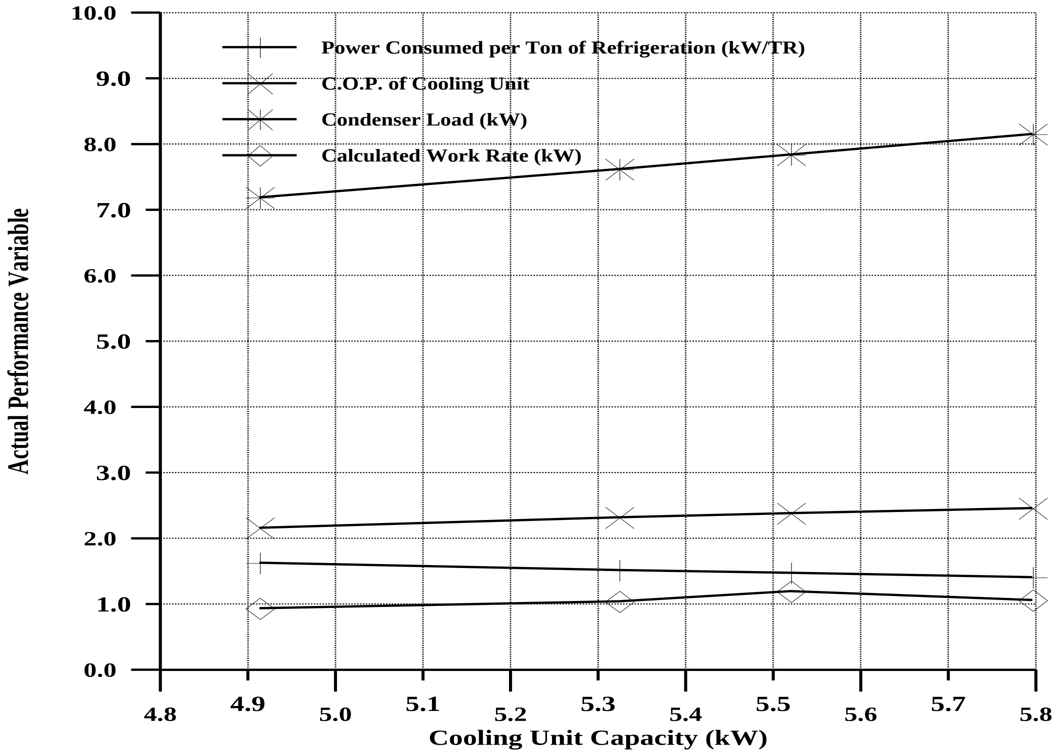


Figure (6.a): IMPC Cycle, *R-22*.

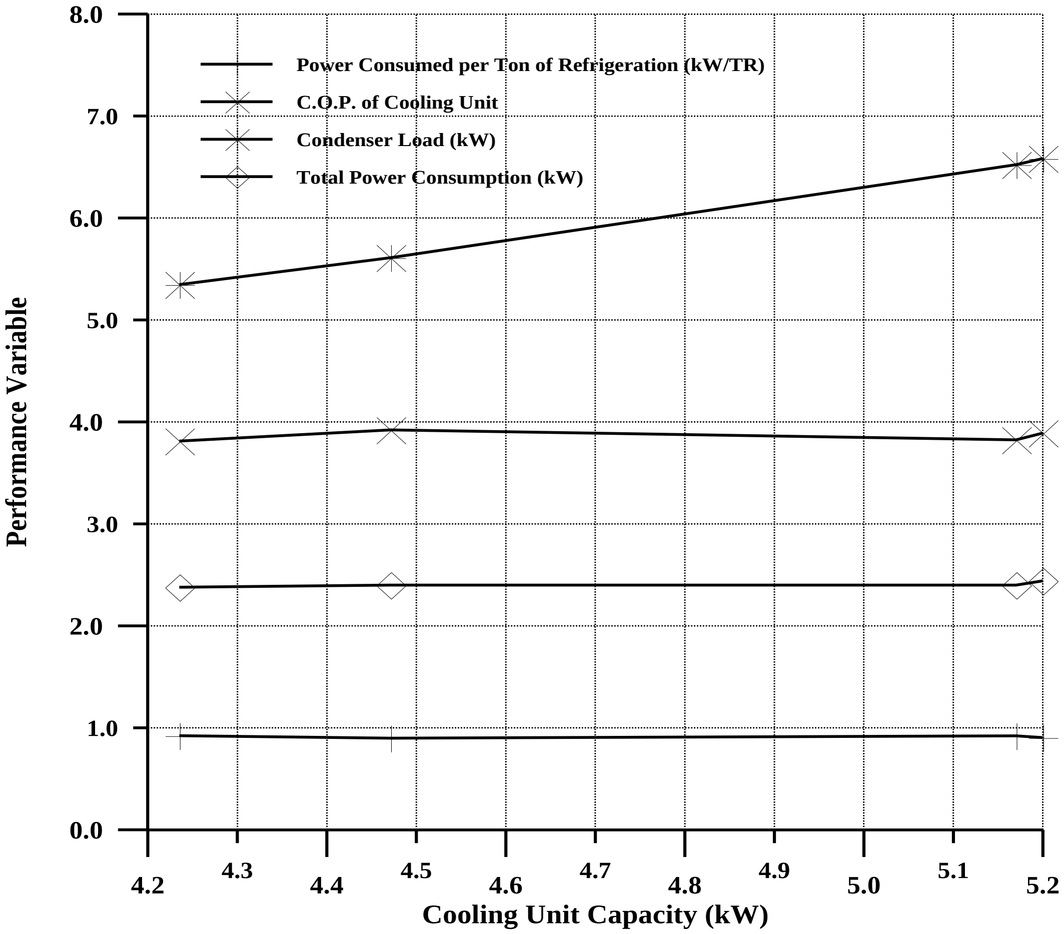


Figure (6.b): AMPC Cycle, *R-22*.

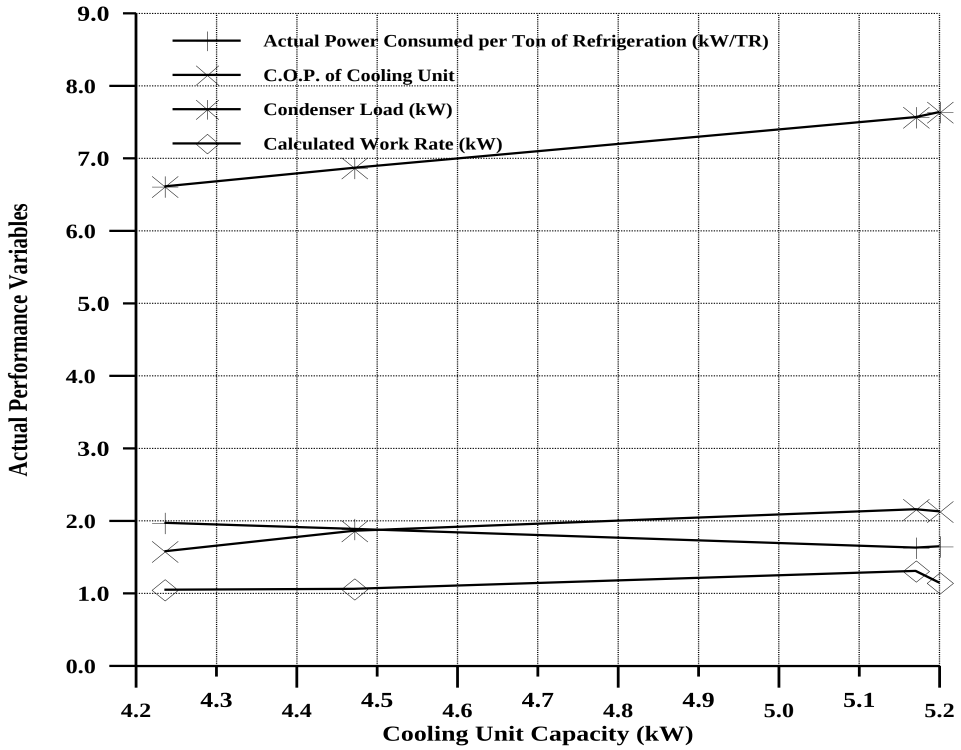


Figure (7.a): IMPC Cycle, *R-407C.*

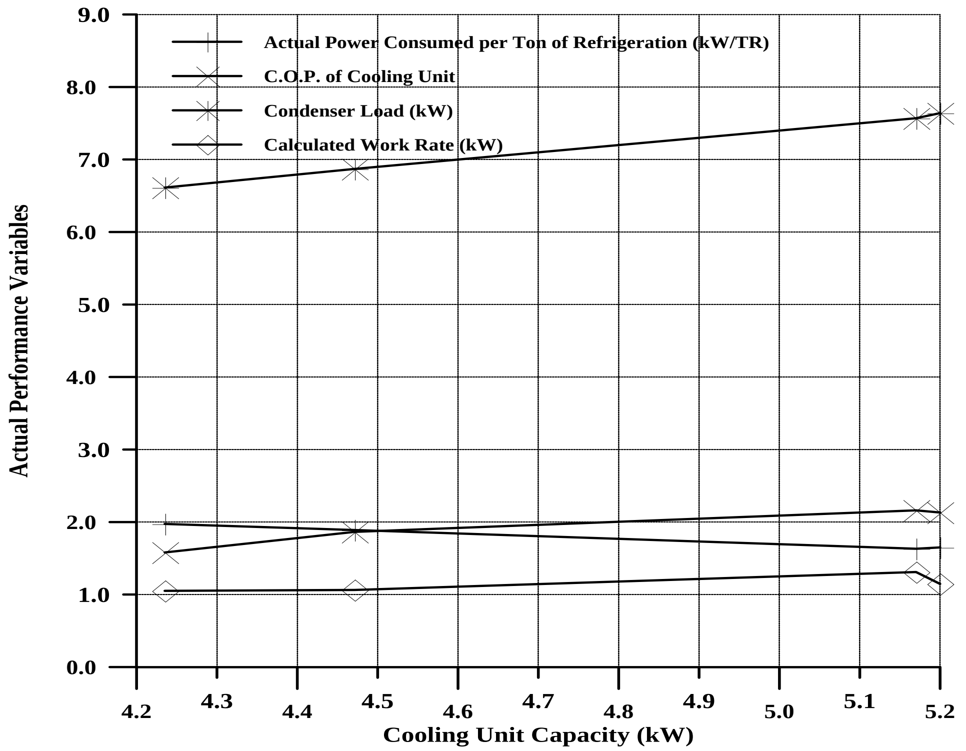


Figure (7.b): AMPC Cycle, *R-407C.*