# Performance Evaluation of Small Scale Air-Conditioning System Using R22 and Alternative Refrigerants

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**ABSTRACT:**

    A mathematical model has been formulated to predict the influence of high outdoor air temperature on the performance of small scale air - conditioning system using R22 and alternative refrigerants R290, R407C, R410A.  All refrigerants were investigated in the cooling mode operation. The mathematical model results have been validated with experimental data extracted from split type air conditioner of 2 TR capacity.  This entailed the construction of an experimental test rig which consists of four main parts. They are, the refrigeration system, psychrometric test facility, measuring instrumentation, and auxiliary systems. The conditioned air was maintained at 25 $^0$C dry  bulb and 19 $^0$C wet bulb for all tests. The outdoor ambient air temperature was varied from 35 $^0$C to 55 $^0$C in 5 $^0$C increments. The study showed that R290 is the better replacement for R22 when the air conditioning system works under high ambient temperature. It has better performance as a drop in refrigerant.  R407C has the closest performance to R22, followed by R410A.

**KEYWORDS: Numerical and experimental refrigeration cycle performance; Mathematical model; Small air conditioner; Alternative refrigerants at high ambient temperature.**

## تقييم  أداء منظومة  تكييف هواء صغيرة الحجم باستخدام R22 وموائع بديلة

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**ألخلاصة :**

    تم بناء نموذج رياضي  يحاكي  تَأثيرَ ارتفاع درجة حرارة الهواء الخارجي على الأداء التشغيلي لمكيف هواء صغير السعة يعمل بمائع التثليج 22 R و بدائله R410A ,R407C, R290 و بطورِ التَبريد.  قورنت نتائج النموذج رياضي لهذه الدراسةِ بنتائج عملية من مكيِّف هواء مُنْفصل سعةِ 2 طن تبريد.  استلزم ذالك بناءَ جهاز اختبار تجريبي يتكون من أربعة أجزاء وهي نظام التثليج وحجرات الاختبار البيئية ومعدات القياس و أجهزة السيطرة على درجات الحرارة . تم تثبيت درجة حرارة الهواء الداخلية عند 25 °م بصلةً جافة  و19 °م بصلةً رطبة. بينما تغيرت درجات حرارة الهواء الخارجية مِنْ 35 °م إلى 55 °م بزياداتٍ مقدارها 5 °م كل مرة.  أثبتت الدراسةُ أنّ مائع التثليج  R290 هو البديل الأفضل إلى مائع التثليج 22 R عندما يَعْملُ نظامَ التكييف تحت درجات حرارة بيئية عالية.  كان مائع التثليج R407C اقربُ اداءاً إلى المائع 22 R  و يأتي بعده R410A.

**الكلمات الرئيسية: دراسة عددية و تجريبية لدورة التثليج، نموذج رياضي، مكيِّف هواء صغير السعة ، موائع بديلة عند ظروف بيئية قاسية.**

## 1.  INTRODUCTION

Many mathematical models have been proposed in the past for modeling refrigeration systems. These models can be classified as steady state and transient simulations. Steady state simulations are commonly used for  performance prediction and product design.  In other words, vapor compression systems are rated and designed using steady state performance evaluations.  Rice et al. (1981) used steady state simulation to explore optimization of conventional air-to-air heat pumps. The model was used to calculate the maximum coefficient of performance that can be attained, both with components that are presently available and with improved ones, for a range of heat exchanger sizes. Another model to predict the steady state performance of an air to air heat pump model was developed by Fisher and Rice (1983).  This model is known as the ORNL heat pump model.  The model was written using Fortran-IV computer program. In medium temperature refrigeration systems,  Spatz et al. (2004) evaluated the performance of R22 with its potential alternatives of R410A, R404A, and R290. The results show that R410A is an efficient and environmentally acceptable option to replace R22 in medium temperature applications. According to Cabello et al.(2005), a simplified steady-state model of a single stage vapor compression plant was presented based on empirical  and parametrical correlations. This model has been validated by experimental data obtained from a test bench using refrigerants R134a, R407C and R22. In 2007, Techarungpaisan et al. proposed a steady  state simulation model to predict the performance of a small split type air conditioner with an integrated water heater.  This model was also used to predict system parameters of interest such as hot water temperature, condenser exit air temperature, evaporator exit air temperature, mass flow rate of refrigerant, heat rejection in condenser and cooling capacity  of the system.

In addition to the above, other models have been proposed as simulation tools for vapor compression refrigeration cycles such as VapCyc, and CYCLE-D.  Richardson et al. (2004) presented VapCyc, which is a steady state simulation tool for the modeling and optimization of vapor compression refrigeration system and its components. The  simulation limits the user to the basic four component system. The simulation consists of a set of independent variables,  such as system charge, component model numbers and component independent variables, and results in a set of dependent system variables, such as COP, capacity, weight and volume. Another simulation tool called CYCLE-D offered by the National Institute of Standards and Technology (NIST) for the theoretical analysis of a vapor compression system was used by Domanski et al. (2005).  The goal of their results was to assess the impact of the evaporator performance on the COP for different refrigerants.

In this model,  a comparative performance of R22 and its alternatives R290, R407C and R410A were determined theoretically in an attempt to examine the possibility of substituting R22 in residential air conditioners used in summer hot arid climate.

## 2-MATHEMATICAL MODELING

The refrigeration cycle consists of  four major components; a rotary compressor running at constant speed, a finned-tube condenser coil, an evaporator coil, and capillary tubes.  These components are connected by pipes which may be surrounded by insulation.  Other components such as a filter-dries and an accumulator are neglected in this analysis due to their relatively small influence on the performance of the system (Jin and Spitler 2002).  These components form a complete refrigeration cycle.  In order to simulate the cycle, all models are interconnected with each other to form the complete model.  The thermodynamic properties of  R22 and the alternative refrigerants are calculated using the Engineering Equation Solver (EES) software (Klein  2006).  These properties are calculated from the equation of state (EOS).  The cycle presented in **Fig.1** is representative of a single-stage, vapor compression refrigeration cycle.
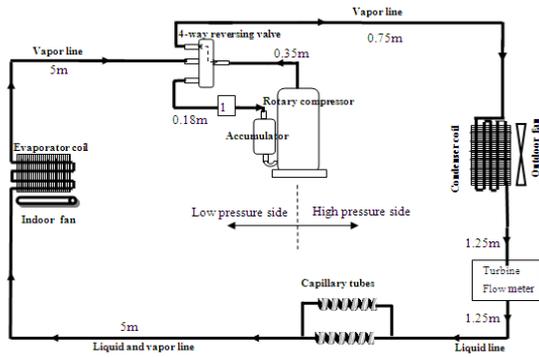
**Fig.1** Schematic diagram of a split type A/C system

The pressure-enthalpy diagram for an actual system with state points is shown in **Fig.2**.  It is noted that due to pressure drop across both the suction line and discharge line, the suction pressure ($P_1$) is slightly lower than the evaporator pressure ($P_e$), while the discharge pressure ($P_2$) is higher than the condenser pressure ($P_c$).
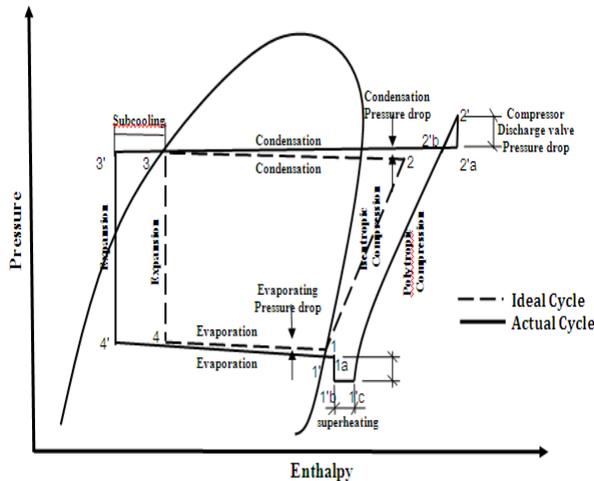


**Fig.2** Ideal and real refrigeration cycles.

## 2.1 Modeling of Rotary Compressors

The following assumptions for the rotary compressor are made:

1. Compressor operates at steady state conditions.

2. Compression process is assumed polytropic.

3. Changes in kinetic and potential energies are negligible.

4. Compressor motor and mechanical efficiency are taken to be 78% and 92.5%, respectively (Wakabayashi et al. 1982).

5. The temperature of the gas along with the shell is constant.

6. Pressure drop across the discharge valve is isenthalpic.

7. The flow through the compressor is homogeneous and one dimensional.

8. The oil effects on the refrigerant properties are neglected.

In an actual compression process, the entropy increases as the irreversibilities of the process increase.  Therefore, the compression process in the compressor is normally polytropic .  The  power of compressor is calculated according to the following equation (Winandy  2002):

$$W_{co} = \dot{m}_r . P_1 . v_1 . \frac{n}{n-1} \left[ \left( \frac{P_2}{P_1} \right)^{\frac{n-1}{n}} - 1 \right] \tag{1}$$

The theoretical power ($\dot{w}_{co}$) can  also  be  calculated in terms of the enthalpy change of refrigerant across the compressor, as follows:

$$\dot{W}_{co} = \frac{w_{is}}{\eta_{is}} = \dot{m}_r . (H_2 - H_1) \tag{2}$$

The power input to the compressor is determined as;

$$\dot{W}_i = \frac{\dot{W}_{co}}{\eta_{ov}} \tag{3}$$

The overall compressor efficiency determines the actual amount of electrical energy required to drive the compressor.  This efficiency is calculated by Cho and Jung ( 2010), and Sakaino et al. (1984) as;

$$\eta_{ov} = \eta_{in} \, \eta_{me} \, \eta_{mo} \tag{4}$$

The values of efficiencies $\eta_{mo}$ and $\eta_{me}$ are taken as 0.78  and 0.925 respectively (Duprez et. al. 2007),

whereas the indicated efficiency $\eta_i$ is calculated as (Ding and Zhang 2001):

$$\eta_{in} = \frac{\eta_v}{1 + 1.5.\Delta P_d \left(\frac{P_d}{P_s}\right)^{\frac{1}{n}} . v_s \Big/ (H_d - H_s)} \qquad (5)$$

The volumetric efficiency $\eta_v$ of the compressor is calculated from the well known equation:

$$\eta_v = 1 + C - C\left(\frac{P_d}{P_s}\right)^{\frac{1}{n}} \qquad (6)$$

It is well known that the rotary compressors have high clearance volumetric efficiencies due to the small clearance volume and corresponding low re-expansion losses of residual gas (Winandy et. al. 2002). In Eq.6, the clearance volume is taken at C=5%. The polytropic index n is given as;

$$n = \frac{\ln\left(\frac{P_d}{P_s}\right)}{\ln\left(\frac{v_s}{v_d}\right)} \qquad (7)$$

$\Delta P_d$ in Eq.5, is given by Ding and Zhang ( 2001) as;

$$\Delta P_d = 25(T_d - 273.15)^{-1.01}.10^{-0.15\frac{P_d}{P_s}} \qquad (8)$$

At steady state, the mass flow rate of refrigerant is assumed to be the same throughout the system and equal to the mass flow rate through the compressor ( Blanco et al. 2012 ). Most compressors operate at a fixed displacement rate. Therefore, the refrigerant mass flow rate will be a function of the suction specific volume as ;

$$\dot{m}_r = \frac{\eta_v.V_d.N}{v_1} \qquad (9)$$

The theoretical displacement volume of the compressor ($V_d$) is calculated from the difference in volume between the roller diameter and cylinder diameter as follows ( ASHRAE Handbook 2010);

$$V_d = \frac{\pi.L_{cy}.(D_c{}^2 - D_r{}^2)}{4} \qquad (10)$$

Where: $D_c$ and $D_r$ are the diameters of the cylinder and rolling piston respectively, and $L_{cy}$ the length or height of the cylinder.

In the compressor, the heat is transferred during the compression cycle due to the temperature difference between the cylinder walls and the refrigerant gas $Q_{su}$, on one hand, and between the shell and the ambient $Q_S$ on the other hand. It is difficult to study the heat transfer phenomena inside the rotary compressor since it requires knowledge of the internal dimensions and many factors concerning re-expansion of residual gas, mass of return flow, oil and gas leakage and so on. For these reasons, the heat transfer from the shell to the ambient is only considered in this study as in other works(Kim and Bullard (2001), Winkler (2009)). When the compressor shell temperature has reached a steady value, the rate of heat transfer from the refrigerant to the shell is equivalent to the rate of the heat transfer from the shell to the ambient. The heat transfer from the compressor shell can be obtained by applying the first law of thermodynamics across the compressor shell using the following equation for the steady state flow, neglecting the potential and kinetic energies (Duprez et. al., 2007).

$$Q_S = W_i - \dot{m}_r(H_d - H_s) \qquad (11)$$

## 2.2 Modeling of Finned Tube Condensers

On the refrigerant side, the condenser is divided into three heat transfer regions, a vapor phase de-superheating region, a two-phase condensing region and a liquid phase subcooling region. Among these three regions,de-superheating and subcooling occupy relatively a small portion. The de-superheating uses about 5 percent of the condensing surface area and subcooling uses 5 to 10 percent (Wang, 2000). The rest of the condenser area undergoes condensation at constant pressure and temperature. **Fig.3** shows a schematic diagram of the heat transfer processes through the condenser.
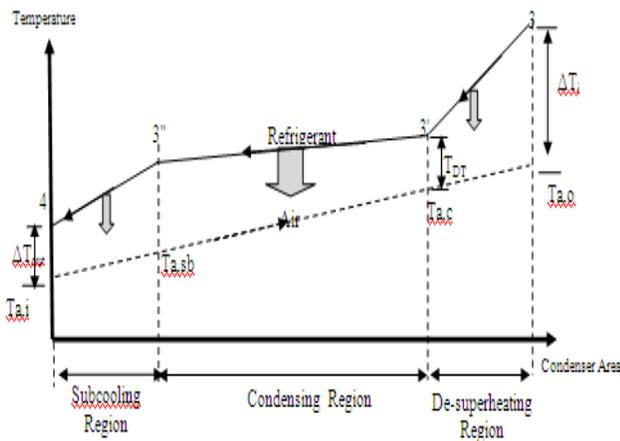
**Fig.3** Temperature profile in the condenser

The following assumptions are made in this study;

1-One dimensional steady flow,

2-Thermodynamic equilibrium between the liquid and vapor (separated  flow),

3-The condenser coils are assumed to be one dry path.

4-In the two phase region, the flow is assumed to be annular flow.

5-The air side heat transfer coefficient is constant over the whole condenser.

6-The air and refrigerant flow directions are in counter flow configuration.

7-The heat transfer and pressure drop effects are considered in the condenser tubes and they are neglected in bends.

The heat rejected from each region can be determined by evaluating the enthalpies at the inlets and outlets.

$$Q_{c,d} = \dot{m}_r \left( H_3 - H_{3'} \right) \tag{12}$$

$$Q_{c,t} = \dot{m}_r \left( H_{3'} - H_{3''} \right) \tag{13}$$

$$Q_{c,sb} = \dot{m}_r \left( H_{3''} - H_4 \right) \tag{14}$$

The effectiveness – number of transfer units ( $\varepsilon$ - NTU ) method is utilized to find the heat transfer for each region in the heat exchanger, using the temperature difference between the air and refrigerant sides as:

$$Q_{c,d} = \varepsilon_d C_{\min} \left( T_3 - T_{a,c} \right) \tag{15}$$

$$Q_{c,t} = \varepsilon_t C_{\min} \left( T_{3'} - T_{a,sc} \right) \tag{16}$$

$$Q_{c,sb} = \varepsilon_s C_{\min} \left( T_{3''} - T_{a,i} \right) \tag{17}$$

The condenser effectiveness ( $\varepsilon$ ) is defined as the ratio between actual heat transfer rate to maximum heat transfer rate:

$$\varepsilon = \frac{Q_a}{Q_m} \tag{18}$$

The effectiveness of cross flow heat exchanger with unmixed fluid is a function of the specific heat ratio ( $C_{\min} / C_{\max}$ ) and the number of transfer units ( Incropera and DeWitt, 1990 ).

For single phase region

$$\varepsilon = 1 - \exp\left\{ \frac{NTU^{0.22}}{C_{\min} / C_{\max}} \left[ EXP((-C_{\min} / C_{\max}) * NTU^{0.78}) - 1 \right] \right\}$$

$$\dots\dots\dots(19)$$

For two phase region

$$\varepsilon = 1 - \exp\left( -NTU \right) \tag{20}$$

Where the number of transfer units is calculated for each region to determine the heat transfer rate. This parameter is defined as follows:

$$NTU = \frac{U\,A}{C_{\min}}, \tag{21}$$

Where $C_{\min} = \min(C_m, C_{um})$, the refrigerant is modeled as mixed fluid and the air is modeled as unmixed fluid

$$C_m = \dot{m}_r \, Cp_r , \tag{22}$$

$$C_{um} = \dot{m}_a \, Cp_a$$

The  UA  value  denotes  the  overall  thermal conductance  of  any  region  in  the  condenser  in

W/$^0$C. The overall thermal conductance is computed as:

$$\frac{1}{UA} = \frac{1}{h_r A_{t,i}} + \frac{\delta_t}{k_t \, A_{t,m}} + \frac{1}{h_{tf} \, A_{t,o}} + \frac{1}{\eta_o \, h_a \, A_0} \tag{23}$$

By neglecting the very small resistances of the tube wall and the thermal contact resistance, the overall thermal conductance reduce to;

$$\frac{1}{UA} = \frac{1}{h_r A_{t,i}} + \frac{1}{\eta_o \, h_a \, A_0} \tag{24}$$

In single-phase region (liquid or vapor), the forced convection heat transfer coefficient of the refrigerant side for a smooth tube was determined using the Dittus-Boelter equation (Incropera and DeWitt, 1990)

$$h_{r,s} = 0.023 . Re^{0.8} . Pr^a \frac{k_r}{D_i} \tag{25}$$

For liquid phase: (a=0.3)

$$Re_L = \frac{G.d_i}{\mu_L}, \qquad Pr_L = \frac{Cp_L \, \mu_L}{k_L} \tag{26}$$

For vapor phase( a=0.4)

$$Re_v = \frac{G.d_i}{\mu_v} \qquad Pr_v = \frac{Cp_v \, \mu_v}{k_v} \tag{27}$$

In the two phase region, the condensation heat transfer coefficient correlations by Shah (1979) was employed. This correlation is calculated as a function of the vapor quality. It can be written as follows:

$$h_{r,t} = h_l \left[ (1-x)^{0.8} + \frac{3.8 \, x^{0.76} \, (1-x)^{0.04}}{P_{re}^{0.38}} \right] \tag{28}$$

Where the liquid heat transfer coefficient, $h_l$, is calculated using the Dittus-Boelter equation and $P_{re}$ is the reduced pressure of refrigerant ($P_{re} = P_{sa}/P_{cr}$). Furthermore, the average convection heat transfer coefficient in the condenser and evaporator is obtained by integrating local values over the length of the two phase region. The vapor quality of the

refrigerant is assumed to be linear throughout the length. Therefore, the integration is performed over the range of refrigerant vapor quality. The average condensing heat transfer coefficient, $h_{av,t}$, is thus given by:

$$h_{av,t} = \frac{1}{x_o - x_i} \int_{x_{in}}^{x_{out}} h_{r,t}(x) \, dx \tag{29}$$

Where $x_i$ and $x_o$ are vapor quality of refrigerant at the inlet and outlet of the condenser( $x_i$=1 and $x_o$=0).

The air side heat transfer coefficient for dry surfaces is based on the work of McQuiston (1981) and is calculated by the correlation :

$$h_{c,a} = \frac{J \, Cp_a \, G_m}{Pr^{2/3}} \qquad 3000 \prec Re \prec 15000 \tag{30}$$

The dry fin efficiency for a thin insulated tip with uniform rectangular section is based on the Schmidt method as described by Incropera and DeWitt (1990).

$$\eta_f = \frac{\tanh(ml)}{ml} \tag{31}$$

The parameters $m$ and $l$ are defined as:

$$m = \sqrt{\frac{2 \, h_a}{k_f \, \delta_f}} \tag{32}$$

$$l = r_e \quad \varphi = r_e \left( \frac{r_e}{r_0} - 1 \right) \left[ 1 + 0.35 \log\left( \frac{r_e}{r_0} \right) \right] \tag{33}$$

Where $r_e$ is equivalent radius, and $r_o$ is the outer radius of the fin. The equivalent radius parameter is given by the equation.

$$\frac{r_e}{r_0} = 1.28 \, \psi (\beta - 0.2)^{1/2} \tag{34}$$

$$\psi = \frac{S_t}{2r_0} \tag{35}$$

$$\beta = \frac{S_L}{S_t} \qquad \beta \geq 1 \tag{36}$$

Where: $S_L$ and $S_t$ are longitudinal and transverse pitch, respectively. $S_L$ is always selected to be greater than or equal to $S_t$.    Finally, the overall heat transfer rate for the entire condenser is the sum of the heat transfer rates for the three regions

$$Q_c = Q_{de} + \sum_{i=1}^{n} Q_{t,i} + Q_{sb} \qquad (37)$$

Where $n$ is number of condensing region segments.

## 2.3  Adiabatic Capillary Tube Modeling

The capillary tube is a constant area expansion device used widely in small scale air conditioner systems.  The refrigerant flow inside the capillary tube can be divided into the single phase and two-phase regions.  **Fig.4** shows a schematic diagram of the straight capillary tube connecting the outlet of the condenser and the inlet of  the evaporator.
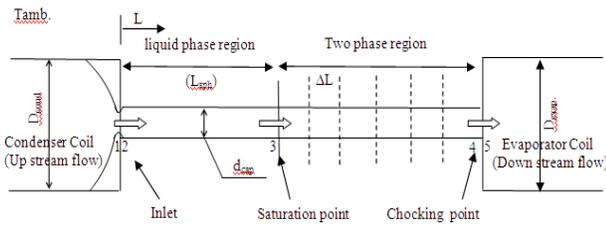


**Fig.4** Schematic diagram of a capillary tube

The following assumptions are made:

1. The capillary tube operates at steady state.

2. Single and two phase flow analysis are considered.

3. A subcooled liquid state enters the capillary tube .

4. The flow of refrigerant inside the capillary tube is one dimensional, homogenous and adiabatic.

5. Changes in kinetic and potential energies are negligible.

6. The capillary tube  is a straight and horizontal tube with a constant cross-sectional area.

The geometrical parameters and input conditions to the capillary tube model are inner diameter, temperature and pressure of the subcooled liquid refrigerant, and thermophysical properties of refrigerant.  Results  of the capillary tube model are the pressure and temperature at capillary tube exit and the capillary choking length.

By applying the Bernoulli equation between points 1 and 2, the minor pressure drop at the entrance is calculated by Sinpiboon and Wongwises (2002);

$$P_1 - P_2 = (\ 1 + k\ )\frac{\rho\,V^2}{2}. \qquad (38)$$

Where $k$ is the entrance loss factor. References are not in agreement about its value.  The value of $k=0.5$ is used here as given by Chen (2008).

In the single phase region (points 2→3), the pressure at point 2 decreases linearly until the saturation  pressure at point 3 according to the following equation (Munson and Young 2010).

$$P_2 - P_3 = f_s \frac{L_{c.s}}{d_{ca}}\frac{\rho\,V^2}{2}. \qquad (39)$$

By applying the steady flow energy equation for the single phase liquid region in the capillary tube with negligible work and heat exchange, we get:

$$H_2 = H_3 = H_{sc} \qquad (40)$$

Applying the steady flow energy equation for each element lying in the two phase region, between sections 3 and 4:

$$H_3 + \frac{V_3^2}{2} = H_f + x\,H_{fg} + \frac{G^2}{2}\left(v_f + x.v_{fg}\right)^2 \qquad (41)$$

Eq.41 is quadratic in the quality x.  It is given with other details and the choking length by Sukkarin and Somchai, 2011.

## 2.4  Modeling  of  a  Direct  Expansion Evaporator

The analysis for the evaporator model is similar to that of the condenser, but is assumed to contain two heat transfer regions on the refrigerant

side, namely a two phase evaporating region and a single phase superheated region. In addition, the air side cooling region is divided into two sections which are dry and wet. **Fig.5** shows the refrigerant flow through the evaporator.

The following assumptions are made for the evaporator model:

1. Fin edges are adiabatic

2. Fin efficiency is assumed to be 90%

3. The degree of superheating at the evaporator is assumed to be 6 $^0$C.

4. Dehumidification occurs on the air side at the entrance of two phase region.

5. Heat transfer and pressure drop effects due to bends are neglected.
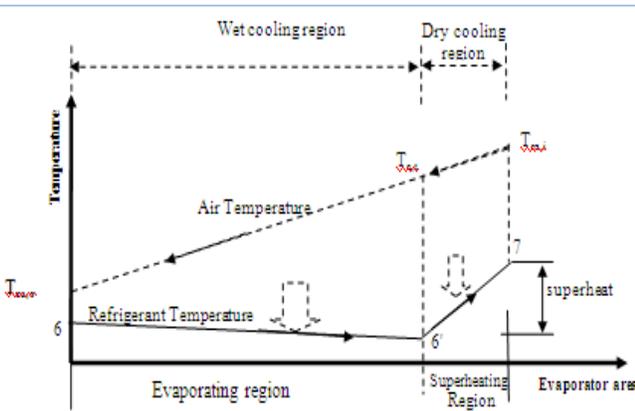


**Fig.5** Temperature diagram in an evaporator coil

The heat gain by the refrigerant side in the two phase and superheated regions are given by the formulae ;

$$Q_{e,t} = \overset{\bullet}{m}_r \left( H_{6'} - H_6 \right) = \varepsilon_t^* C_{min,t} \left( T_{a,t} - T_6 \right) \tag{42}$$

$$Q_{e,sp} = \overset{\bullet}{m}_r \left( H_7 - H_{6'} \right) = \varepsilon_{e,sp} C_{min} \left( T_{ro,i} - T_{6'} \right) \tag{43}$$

Where $H_{6'}$, $H_7$ and $H_6$ are refrigerant enthalpies at the locations in **Fig.5**. The modified heat

exchanger effectiveness $\varepsilon_t^*$ is given by Harms et al (2002) as;

$$\varepsilon_t^* = 1 - \exp \left( -\frac{UA^*}{m_a Cp_a} \right) \tag{44}$$

Where $UA^*$ is the overall heat transfer coefficient for the wet cooling region.

The overall thermal conductance of the wet region of the evaporator coil is computed as

$$\frac{1}{UA^*} = \frac{1}{h_r A_{t,in}} + \frac{1}{h_l \ A_0} + \frac{1}{h_{a,w} \ \eta_{o,w} \ A_0} \tag{45}$$

McQuiston and Tree (1973) assumed a linear relationship between the humidity ratio of the saturated air at the wet surface, and the surface temperature. These parameters are determined as follows:

$$h_l = \frac{k_{wa}}{\delta_{wa}} \tag{46}$$

$$h_{a,w} = h_{a,dry} \left\{ 1 + \frac{H_{fg,w} \left( w_a - w_w \right)}{L_e \ Cp_a \left( T_a - T_w \right)} \right\} \tag{47}$$

$$H_{fg,w} = \left( -2*10^{-5} \ T_w^3 + 0.0012 \ T_w^2 - 2.3804 \ T_w + 2501.6 \right) *10^3$$

$$\tag{48}$$

The moist air properties are obtained using relations given by Joudi,1991.

As in the condenser model, the Dittus-Boelter equation is used to compute the refrigerant side heat transfer coefficient in the single phase region:

$$h_{r,s} = 0.023.Re^{0.8}.Pr^{0.4} \frac{k_r}{D_i} \tag{25}$$

Whereas, the evaporating heat transfer coefficient in the two-phase region ($h_{r,t}$) is calculated as (Domanski and David, 1985) ;

$$h_{r,t} = 3.22.X_{tt}^{-0.3} \ h_{r,s} \tag{49}$$

The above correlation is applicable for annular flow at qualities from 10 percent to 90 percent.

The forced convection heat transfer coefficient on the air side of flat finned tube heat exchangers ($h_{a,dry}$) is calculated by the correlation proposed by Briggs and Yong, 1962.

$$h_{a,\,dry} = 0.134 . \frac{k_a}{D_0} \mathrm{Re}_a^{0.681} . \mathrm{Pr}_a^{0.333} \left(\frac{Z}{Y}\right)^{0.2} \left(\frac{Z}{t}\right)^{0.1134} \qquad (50)$$

The evaporator cooling capacity is the sum of the combination of  heat gains by the refrigerant side in the two phase and superheated regions and can be expressed as:

$$Q_e = \sum_{i=1}^{n} Q_{e,i} + Q_{e,sp} \qquad (51)$$

## 2.5 System Pressure Drop

Pressure drop in the system consist of pressure drops in the main components of the system and connecting pipes. Conventional equations are used with full details given by Al-Amir 2013.

## 3. SOLUTION PROCEDURE

The Engineering Equation Solver (EES) software is used in this work for computer programming.  The computer program of the actual cycle is more complex when compared to the ideal cycle program  and consists of a main program and a number of  procedures. These procedures are coupled with the main program by call commands. The main program contains system equations and call statements for all the preceding procedures. Basic inputs are required to the main program to simulate the actual vapor refrigeration cycle performance. These inputs are the physical dimensions of the components and operating conditions for cycle.  The output parameters of the system simulation model are the cycle state points, cooling capacity, heat rejection from the condenser, refrigerant pressure drops, consumed power, and the coefficient of performance. To plot the steady state characteristic performance of the actual refrigeration cycle on a p–h diagram, pressure and enthalpy data are taken from the simulation results.

For each refrigerant, five different settings of outdoor air temperature into the condenser were considered. The EES software is used to plot data on a p–h diagram.

## 4.EXPERIMENTAL WORK

The experimental test rig is shown in **Fig. 6**.



(a)top view
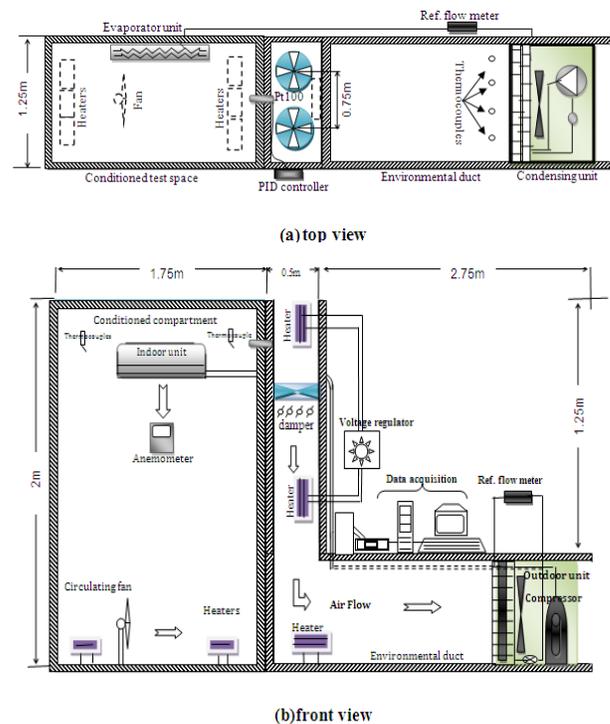


(b)front view

**Fig.6** A schematic diagram of the experimental test rig with dimension of conditioned compartment and environmental duct a)top view  b) front view

It consists of four main parts which are the refrigeration system, the psychrometric apparatus, measuring instrumentation, and auxiliary systems. Split type air conditioners with a 2 TR nominal cooling capacity model EL-26ITERH manufactured by Denka company were tested experimentally. This system required to replace the compressor for R410A refrigerant due to its higher operating pressures. The psychrometric apparatus consists of two sections, which are the indoor conditioned compartment and the outdoor environmental duct. These sections were locally constructed and their conditions were maintained

according to the ASHRAE standard 37-2009 for testing unitary air conditioners and heat pumps. The dry and wet bulb air temperatures for the interior are 27 $^0$C/19 $^0$C and outdoor ambient are  35 $^0$C/24 $^0$C.  The evaporator unit was mounted at a height  of 1.75 m.  Six electric heaters, 1200 W each, were placed inside the condition compartment for the heat load.  They are controlled by a Solid State Relay (SSR) to maintain the air temperature inside the compartment at 25 $^0$C throughout  the test period.  The environmental duct has an L shape and includes two circulation fans with dampers, which drive the air through the thermally insulated environmental duct. Three electric resistance heaters, 1200 W each, are located inside the duct for controlling the temperature of the air across the condensing unit. All the components such as system units, fans, electric heaters and voltage regulator are connected to a power supply. A sample of systematic test runs is summarized in Table 1.

**Table 1**Experimental runs at optimum charge under various outdoor air temperature

The instrumentation required for testing are divided into refrigerant and  air sides. Refrigerant side measurements include the thermocouples type-K with a range of -50  to 250  $^0$C and an accuracy of ± (0.4 % + 0.5 $^0$C), pressure transducers with a range of 0-10 bar or 0-40 bar and an accuracy of ±1% of full scale, turbine flow meter with a range of 0.02-1.2  m$^3$/h and an accuracy of ±0.58 of the reading. Whereas, air side measurements include thermocouples type–T with a range of -270 to 400 $^0$C and an accuracy of ± 0.3 $^0$C, and collection vane type anemometer with a range of 0.4 - 30 m/s and an accuracy of ± 3 %  full scale. All measurements were calibrated before their use.

# 5. RESULTS AND DISCUSION

**Fig.7** shows influence of outdoor air temperature on the compressor discharge tempera-ture. Clearly, increasing the outdoor air temperatures increases the compressor discharge temperature. It is noted from the same figure that discharge  temperature of the R22 is higher by about

21-23$^0$C than that of  R290 at outdoor temperature ranging from 35 $^0$C to 55 $^0$C.

**Fig.8** shows the effects of outdoor air temperature on power consumption of compressor when the system operates in the cooling  mode. The compressor power consumption increases with increasing outdoor air temperature from 35 °C to 55 °C with a constant 25 °C inside the conditioned compartment. This is because the compressor power is affected by two factors, the refrigerant mass flow rate and the compression ratio. The power consumption increased from 2.365 kW to 2.84 kW for R22, 2.04 kW to 2.65 kW for R290, 2.47 kW to 2.93 kW for  R407C, 2.74 kW to 3.32 kW for R410A for the same outdoor temperature range. Also, the power consumption increases with increased outdoor air temperature due to the rise in the temperature difference (lift) between the environmental duct and conditioned compartment.

The system has high cooling capacities at the standard rating conditions (35 $^0$C outdoor air temperature), but lose capacity as the outdoor air

| Ref. Type | Charge Level(gram) | No. of Run | Optimum Charge(g) | Outdoor Temp(C) | No. of Run |
|---|---|---|---|---|---|
| **R22** | 1500 | 2 | **R22 (2000g)** | 35 | 2 |
| | 1600 | 2 | | | |
| | 1700 | 2 | | 40 | 2 |
| | 1800 | 2 | | | |
| | 1900 | 2 | | 45 | 2 |
| | 2000 | 2 | | | |
| | 2100 | 2 | | 50 | 2 |
| | 2200 | 2 | | | |
| | 2300 | 2 | | 55 | |
| | | | | | 2 |
| **R290** | 200 | 2 | **R290 (900 g)** | 35 | 2 |
| | 300 | 2 | | | |
| | 400 | 2 | | | |
| | 500 | 2 | | 40 | 2 |
| | 600 | 2 | | | |
| | 700 | 2 | | 45 | 2 |
| | 800 | 2 | | | |
| | 900 | 2 | | 50 | 2 |
| | 1000 | 2 | | | |
| | 1100 | 2 | | 55 | |
| | | | | | 2 |
| **R407C** | 1500 | 2 | **R407C (1900 g)** | 35 | 2 |
| | 1600 | 2 | | | |
| | 1700 | 2 | | 40 | 2 |
| | 1800 | 2 | | | |
| | 1900 | 2 | | 45 | 2 |
| | 2000 | 2 | | | |
| | 2100 | 2 | | 50 | 2 |
| | 2200 | 2 | | | |
| | 2300 | 2 | | 55 | 2 |
| **R410A** | 1800 | 2 | **R410A (2800 g)** | 35 | 2 |
| | 2000 | 2 | | | |
| | 2200 | 2 | | 40 | 2 |
| | 2400 | 2 | | | |
| | 2600 | 2 | | 45 | 2 |
| | 2800 | 2 | | | |
| | 3000 | 2 | | 50 | |
| | | | | | 2 |
| | 3200 | 2 | | 55 | 2 |

temperature increases as shown in **Fig.9** . The

 cooling capacity of R410A has a higher value, at the standard test condition, than the other refrigerants.  This is because of the high volumetric refrigerating capacity, which is defined as the cooling capacity per unit vapor volume at  the exit of the evaporator in kJ/m$^3$.  Moreover, the mass flow rate of  R410A is higher than the other refrigerants at this condition. When the outdoor air temperature increases to 55 $^0$C, the cooling capacity of R410A drops faster than the other refrigerants because of its low critical temperature (72.5 $^0$C). Theoretical results for 2 TR system using R410A is higher by about 3.82 %  than that of R22 while it is lower for the alternative refrigerants R407C and R290 by about 1.66 % and  4.63 % than that of R22, respectively.

**Fig.10** indicates that the lower the outdoor air temperature, the higher is the COP. As the outdoor temperature increases, the COP  values decrease for all  refrigerants.  With  increasing  outdoor temperature, the COP of system charged with R410A drops much more than that of the  same system charged with other refrigerants. Which means, the COP of R410A system is more sensitive to changes in outdoor temperature.  On the other hand, the system charged with R290 has the highest COP among the selected refrigerants at standard conditions. The COP of  R407C seems to be similar to R22, as excepted. At maximum outdoor air temperature  (55 $^0$C), the coefficient of performance is maximum for R290 and minimum for R410A.

## 6. VALIDATION OF THE THEORETICAL MODEL WITH EXPERIMENTAL DATA

The normal way of validation of a theoretical model is by comparison with experimental results at the same operating conditions.   The deviations between the two results are calculated by the following formulas;

$$\text{Re} \textit{lative}  \textit{ deviation} \left(x \ \% \right) = \frac{\left(x_{\text{theoretical}} - x_{\exp \textit{erimental}}\right)}{x_{\exp \textit{erimental}}} \times 100 \quad (52)$$

$$\textit{Average  deviation.} \left(x_{av} \% \right) = \frac{1}{n} \sum_{i=1}^{n} \left[ \frac{\left(x_{i,\text{theoretical}} - x_{i,\exp \textit{erimental}}\right)}{x_{i,\exp \textit{erimental}}} \times 100 \right]$$
$$\ldots \ldots (53)$$

**Fig.11**    shows    the    comparison    between calculated  and  measured  compressor  discharge temperatures for outdoor air temperatures of 35ºC, 40 ºC, 45 ºC 50 ºC and 55 ºC. Fig.(11) include 40 experimental data points, 10 data points for each refrigerant. The compressor discharge temperature calculations agree well with experimental measure-ements. All the points are within -3.198% and +4.4% .

A  comparison  of  calculated  and  measured compressor  work  with  different  outdoor  air temperatures is shown in **Fig.12**.   The measured results, lie within a maximum deviation of 6.92% and a minimum deviation of -4.07 % .

Good  agreement  is  observed  between  the calculated  and  measured  cooling  capacity  at different  outdoor  air  temperatures  as  shown  in **Fig.13**. The maximum deviation in the system is +11.7%.  All the theoretical values are higher than the experimental values for all four refrigerants.

The    calculations    of    the    coefficient    of performance  for  the  actual  cycle  lead  to  good results,  as  illustrated  in  **Fig. 14**.   For R22, the average deviation in the coefficient of performance was 3.68% and minimum deviation of -3.38%.  The theoretical predictions for COP values are higher than the experimental values for the actual cycle.

The  actual  cycle  diagrams  (**Figs. 15** to **18**) clearly show the effect of  outdoor temperature at a relatively constant evaporator temperature. Pressure drops across suction line and discharge valve of compressor are very clear and the minor pressure drops  inside  the  condenser  and  evaporator  are obvious. These characteristics are, of course, not present in the assumptions for the ideal cycles.

## 7. CONCLUSIONS

1.At standard rating conditions, the cooling capacity for R410A was highest among the investigated refrigerants.

2. The compressor discharge temperature increase with increasing outdoor air temperatures.  R22 refrigerant has higher discharge temperature than the other refrigerants.

3. The system charged with R290 has the best COP than the other refrigerants.

4. The comparison shows that the power consumption of R410A and R407C are higher by about 15.8-16.9% and 4.4-3.17%, respectively and R290 lower by about 13.7-6.7% as compared to R22 under same operating conditions. Test results demonstrate that R290 is a good alternative to replace R22 in air conditioners from the standpoint of energy efficiency.

5. The results also show that the COP values decrease for all investigated refrigerants, as the outdoor temperature increases. The COP for 2 TR system using R410A is found to be the lowest among the four refrigerants at high outdoor air temperature. Therefore, R410A is not a good choice for air conditioner in high outdoor air temperature.

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## NOMENCLATURE

| | |
|---|---|
| $A$ | Area (m$^2$) |
| $C_p$ | Specific heat at constant pressure (J/kg.$^o$C) |
| $C_v$ | Specific heat at constant volume (J/kg.$^o$C) |
| $d_{ca}$ | Capillary tube diameter(m) |
| $D_h$ | Hydraulic diameter (m) |
| $f_s$ | Friction factor |
| $G$ | Mass flux (kg/s.m$^2$) |
| $h$ | heat transfer coefficient (W/m$^2$.$^0$C) |
| $H$ | Enthalpy(kJ/kg) |
| $J$ | Colburn J factor |
| $k$ | Thermal conductivity  (W/m.K) |
| $K$ | Entrance loss factor |
| $k_t$ | thermal conductivity of  wall tube (W/m.K) |
| $Le$ | Lewis number |
| $m$ | mass flow rate(kg/s) |
| $n$ | Polytropic index |

| | |
|---|---|
| $n_f$ | number of fins |
| $n_{ro}$ | number of rows. |
| $n_{tu}$ | number of tube |
| $N$ | Number of cycles per minute(N=3600) |
| $Nu$ | Nusselt number |
| $P$ | Pressure (kPa) |
| $Pr$ | Prandtl number |
| $P_{re}$ | Reduce pressure |
| $Q$ | Heat (kW) |
| Re | Reynolds number |
| $S_L$ | Longitudinal pitch (m) |
| $S_t$ | Tangential pitch (m) |
| $T$ | Temperature (K) |
| $U$ | Overall heat transfer coefficient (W/$^0$C) |
| $v_1$ | Specific volume at the comp. suc. (m$^3$/kg) |
| $V_d$ | Displacement Volume of the compressor (m$^3$) |
| $W$ | Compressor power (kW) |
| $x$ | Quality |
| $X_{tt}$ | Lockhart Martinelli parameter |

**Greek Symbols**

| | |
|---|---|
| $\gamma$ | Adiabatic index |
| $\varepsilon$ | Effectiveness |
| $\varepsilon_v$ | Void fraction |
| $\Delta P_d$ | Pressure drop across the discharge valve |
| $\eta_0$ | Efficiency |
| $\mu$ | Dynamic viscosity (kg/m˒s) |
| $\nu$ | Kinematic viscosity(m$^2$/s) |
| $\rho$ | Density (kg/m$^3$) |
| $\Phi_{Lt}$ | Two phase liquid multipliers |
| $\Phi_{vt}$ | Two phase vapor multipliers |
| $\delta_f$ | Fin thickness(mm) |
| $\delta_t$ | Wall thickness of the tube ( $\delta_t$=0.5mm). |
| $\sigma$ | Surface tension (N/ m$^2$) |
| $\Delta h$ | Enthalpy change (kJ/kg) |
| $X_{tt}$ | Lockhart Martinelli parameter |

**SUBSCRIPTS**

| | |
|---|---|
| c | condenser |
| co | compressor |
| d | discharge |
| e | evaporator |
| i | input |
| l | liquid |

| | |
|---|---|
| r | refrigerant |
| s | suction |
| S | Shell |
| t | Tangential |
| v | vapor |

**ABBREVIATIONS**

| | |
|---|---|
| COP | Coefficient Of Performance |
| EES | Engineering Equation Solver |
| GWP | Global Worming Potential |
| HC | Hydrocarbon |
| HCFC | Hydro chlorofluorecarbon |
| HFC | Hydro fluorocarbon |
| PID | Proportional Integral Derivative |
| TR | Tone of Refrigeration |
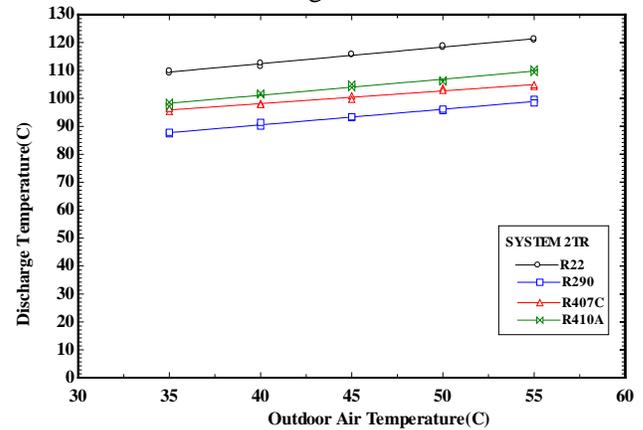


**Fig. 7** The effect of the outdoor air temperatures on discharge temperature.
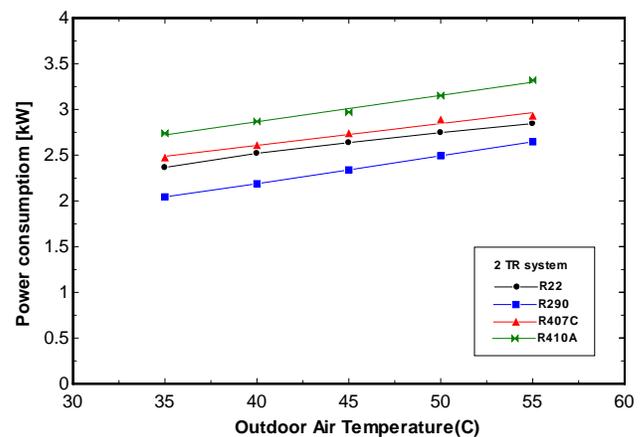


**Fig. 8** The effect of the outdoor air temperatures on compressor power consumption
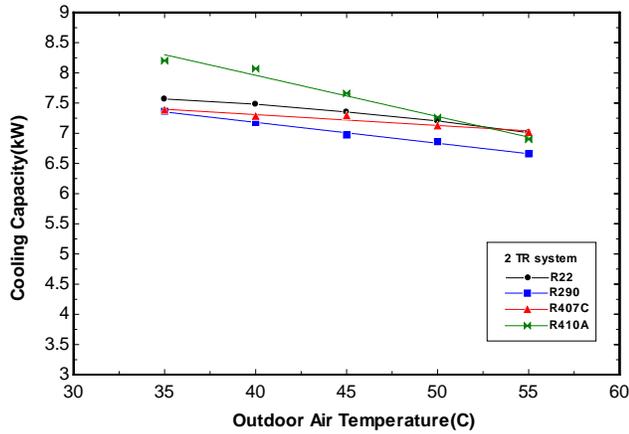
**Fig. 9** The effect of the outdoor air temperatures on evaporator cooling capacity .



**Fig.12** Comparison between calculated and. measured compressor power consumption for system working with R22 and its alternative.
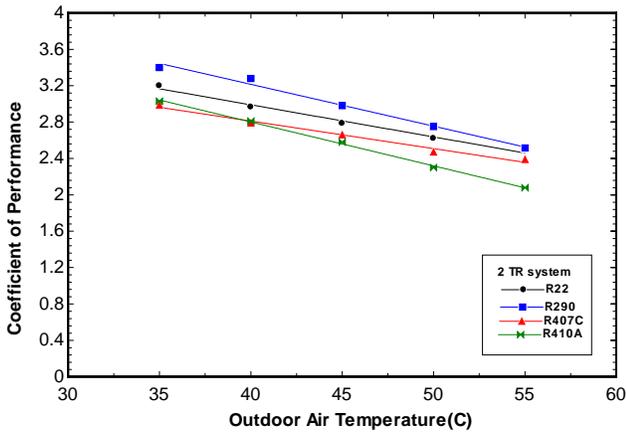


**Fig. 10** The effect of the outdoor air temperatures on COP of system.
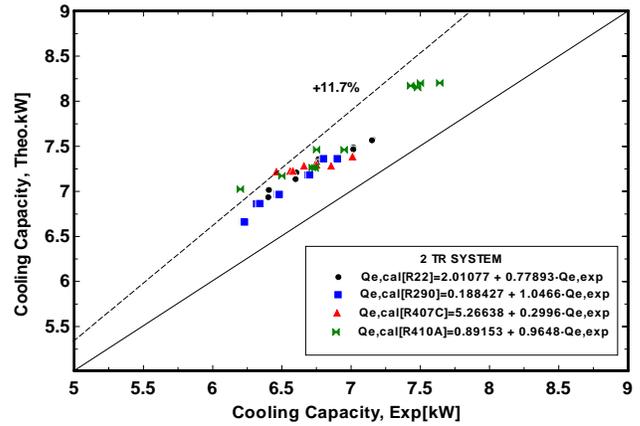


**Fig.13** Comparison between theoretical  and experimental cooling capacity  for system  working with R22 and its alternative.



**Fig.11** Comparison between theoretical and experimental compressor discharge temperatures system working with R22 and its alternative.
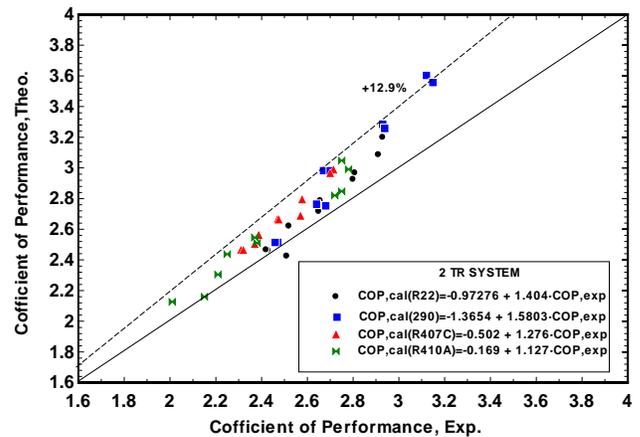


**Fig.14** Comparison between theoretical  and experimental coefficient of  performance for system working with R22 and its alternative

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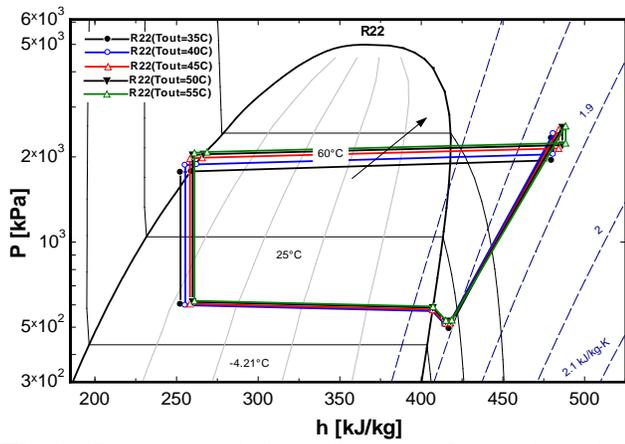
**Fig.15** Pressure-enthalpy diagram of actual cycle for R22 under different outdoor air temperatures
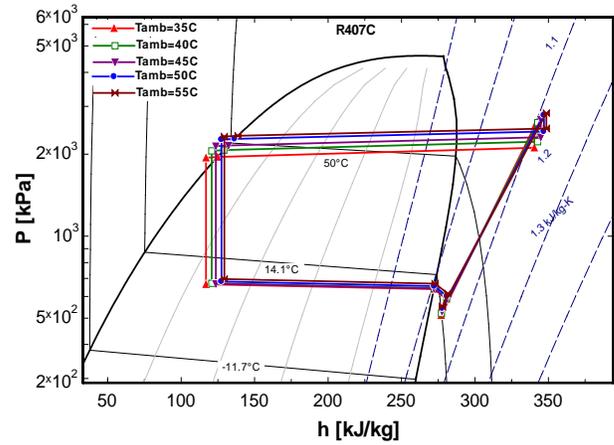


**Fig. 17** Pressure-enthalpy diagram of actual cycle for R407C under different outdoor air temperatures
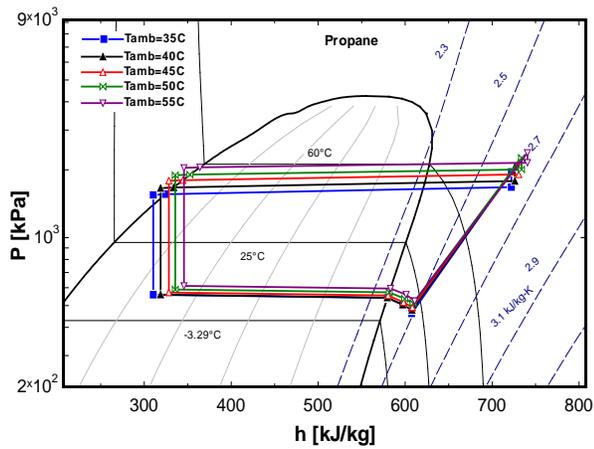


**Fig. 16** Pressure-enthalpy diagram of actual cycle for R290 under different outdoor air temperatures



**Fig.18** Pressure-enthalpy diagram of actual cycle for R22 under different outdoor air temperatures