A Comparative Study of the Performance of Finned Tube Air Cooled Condenser with Refrigerants R22 and R407C

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Abstract

Mathematical and numerical study of finned tube air cooled condenser for air conditioning unit with two ton refrigeration capacity using R22 as a base fluid and R407C an alternative fluid was investigated. Different parameters were considered in this work, such as condensing pressure, ambient temperature and refrigerant mass flow. A comparison of performance between two condensers when using R22 and R407C were performed. A redesign the condenser operates with the R407C to operate with the same system that operates with R22. The result showed the same behavior for the two refrigerants, the condensers are possible to work with R407C for the same geometry and some modifications in the structure of heat exchange with the same air velocity. The proposed model was validated with the outputs from the test data given in literature papers, derived from air cooled condensers with different dimensions. The results exhibited an agreement with the experimental results with a percentage of compatibility ± 10%.

Keywords: air cooled condenser, R22, R407C, numerical, alternative refrigerant.

1. Introduction

Air-cooled finned –tube condenser are widely used in refrigeration and air conditioning application. When compared with other types of heat exchanger, such as water cooled condenser, this type is economic. In the last years, the investigation on the heat exchangers servant in air conditioning systems has concentrated because of incorporation of the Protocol of Montreal prohibition the utilization of the halogenated refrigerants (HFCF's). Several alternatives for R22, one of the most expecting refrigerants is the tri- mixture R407C, "composed of HFC-32/125/134a, 23/25/52 wt. %". Air cooled condenser is defined as a bank of finned tubes arranged horizontally in rows and passes of definite numbers and sizes depending on the quantity of heat rejected to air. Condensing takes place in about 85% of the condenser area at a substantially constant temperature. **Traviss et al. [1]** experimental and theoretical studied the internal heat transfer coefficient for refrigerant R22 for a two-phase flow in condensation state. **Raymond et al. [2]** constructed a mathematical model for an air-cooled refrigerant condenser; the theoretical account is intended for design analyses or simulation of heat exchangers that have complex refrigerant circuiting or unusual air-side geometries. The model relies on a tube-by-tube computational approach, calculating the thermal and fluid flow performance of each tube in the heat exchanger individually, using local temperatures and heat transfer coefficients. **Rich [3]** conducted an experimental study on fin-and-tube heat exchangers. Nine coils tested in the study were found geometrically identical with the exception of fin pitch, which was varied in steps from 0 to 20 fins/in. **Shah [4]** performed an experimental study for predicting heat transfer coefficient during film condensation inside tubes. It has been verified by comparison with a wide variety of experimental data. **Anand and Tree [5]** modeled a condenser using finite difference method, fully implicit scheme thermodynamic and flow properties of the control volume are equal to the thermodynamic and flow properties at the end of the space step. **Schlager et al. [6]** developed equations for evaporation and condensation of refrigeration oil mixtures. **Lee et al. [7]** performed an empirical study of air cooled finned-tube condenser utilize several arrangement of the condenser path (Z and U style) employ R22 and R407C as working refrigerant fluids. Variant condenser capabilities were received from the numerical results and the empirical test, depending on the paths and the used of refrigerants. R22 preferable than R407C for the Z-style path arrangement, but there was no large difference between the results utilization the refrigerant in the U-style path arrangement. **Yunting and cropper [8]** developed a steady-state simulation model of a finned-tube air –cooled condenser by using the distributed model method. The model can be used to predict 3-D variations of parameters for both air and refrigerant side. **Ciro and Angelo [9]** conducted a numerical and experimental analysis for an air condenser working with the non azeotopic mixture R407C in steady state conditions. A homogenous model for the condensing refrigerant is considered to forecast the performances of the condenser. The results showed that the simplified model provides a reasonable estimation of the steady-state response, and this study presents a numerical model that is applicable to fin-and-tube condenses for air conditioning unit. The simulation scheme developed in this study considers the variations of fluid properties, friction factor, and heat transfer coefficient due to change of phase. This simulation predicts temperature, pressure, quality of the refrigerant, and the temperature of the air leaving the heat exchanger as a function of distance. Calculations were performed a tube by tube basis, and the results were compared with the experimental available in the literature.

2. Mathematical Model

A schematic diagram of an air-cooled condenser is shown in the Fig.1. A condenser consists of a bank of finned tubes arranged horizontally in rows and passes of definite numbers and size depending on the amount of heat rejected to air. A fan moves the air through the condenser, so the heat transfer coefficient is forced convection. The overall heat transfer coefficient for the condenser is represented by [10] as:

$$U = \frac{1}{(R_i + R_t + R_o)} \quad (1)$$

Where,

$$R_i = \frac{1}{h_i A_i} \quad (2)$$

$$R_t = \frac{ln\left(\frac{r_o}{r_i}\right)}{2\pi L k_t} \quad (3)$$

$$R_o = \frac{1}{h_{ao} A_o \eta_o} \quad (4)$$

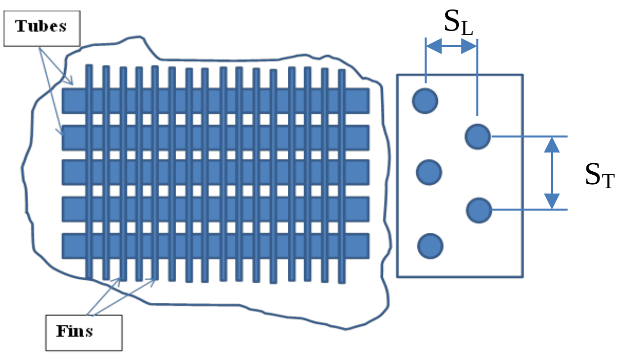

Figure 1: Finned tube heat exchanger configuration

The heat transfer in refrigeration condenser has three characterized zones on the refrigerant side; which are de-superheating, condensation, and sub-cooling zones. The heat is exchanged with a coolant air due to temperature difference a cross the condenser pipe, the temperature of both refrigerant and coolant varies across the flow path [11]. The total rate of heat transfer using the following equation is:

$$Q_{rej} = UA\Delta T \quad (5)$$

Along the tube section the flow orientation of the refrigerant was hired as a position the transfer of heat area into the calculation for the air division in the side of the air and the effects of temperature gradient by the refrigerant zeotropic blend, R407C into the side of refrigerant. This is shown in Fig.2 within a two-row condenser, where 36 tubes production two paths flow of refrigerant. Every tube was split to four parts, for a total of 144 parts along the path of the flow of refrigerant. A site planner of the heat exchanger is split to the segment of control volumes. In this scheme, the exit of one unit analysis becomes input to the other unit analysis. This provides more exact results, and is vastly utilize. The following assumptions are proposed to simplify the problem:

1. Steady-state condition in the refrigerant side and in the air side.
2. There is no conduction in the heat in the axis of tube and nearby the fins.
3. The heat transfer in the return bend was neglected, no heat flow through the return bend.
4. A homogeneous distribution of the air velocity facing to every section.
5. Thermal resistance between the fin and the tube was neglected.
6. After the first row, the velocity of air assumed a same as the entering velocity of air to the final row of the condenser.
7. Refrigerant fluid at any point is in case of thermal equilibrium.

The energy equation for the refrigerant side can be simply expressed as:

$$Q = \mathring{m}_r (h_{in} - h_{out}) \quad (6)$$

This equation of conservation is also appropriate by the air side. The velocity of facing air across the tubes are constant, an energy balance between the refrigerant fluid side and the air side for every segment, the method is utilized (NTU-ε) to implement part of the calculation of the transfer of heat.

$$Q = \mathring{m}_a cp_a [T_{a,i} - T_{a,o}] = \varepsilon C_{min} (T_{h,i} - T_{c,i}) \quad (7)$$

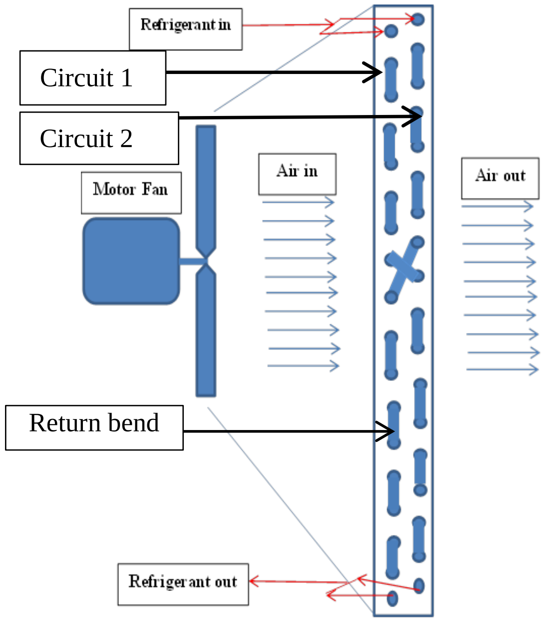

Figure 2: Circuit of the condenser

where the effectiveness ε can be calculated from [13] for cross flow heat exchanger.
For a single phase region:

$$\varepsilon = 1 - exp\left[\frac{NTU^{0.22}}{\frac{C_{min}}{C_{max}}}\left(exp\left(\frac{-C_{min}}{C_{max}} \times NTU^{0.78}\right) - 1\right)\right] \quad (8)$$

For a two-phase region:

$$\varepsilon = 1 - exp(-NTU) \quad (9)$$

$$NTU = \frac{U_o A_o}{C_{min}} \quad (10)$$

$$U_o A_o = \left[\frac{A_o}{h_i A_i} + \frac{A_o ln\left(\frac{D_o}{D_i}\right)}{2\pi L K_t} + \frac{1}{h_o \eta_o}\right]^{-1} \quad (11)$$

2.1. Calculation of Inside Heat Transfer Coefficient

i- For pure refrigerant R22, the correlation developed by Traviss et al. [1] for condensing heat transfer coefficients is based on analytical derivation assuming annular flow in a tube, the computations are as follows:

$$x_{tt} = \left(\frac{1-x_m}{x_m}\right)^{0.9}\left(\frac{\rho_v}{\rho_\ell}\right)^{0.5}\left(\frac{\mu_\ell}{\mu_v}\right)^{0.1} \quad (12)$$

A Reynolds number of the liquid phase is computed as:

$$Re_\ell = \frac{G_r(1-x_m)D_e}{\mu_\ell} \quad 50 < Re_\ell < 26000 \quad (13)$$

The condensing heat transfer coefficient is computed from the appropriate one of the following expressions:

$$h_{tp} = \frac{K_\ell[F(x_{tt})]\, Pr_\ell Re_\ell^{0.9}}{D_e F_2} \; for\; F(x_{tt}) < 1 \quad (14)$$

$$h_{tp} = \frac{K_\ell[F(x_{tt})]^{1.15} Pr_\ell Re_\ell^{0.9}}{D_e F_2} \; for\; 1 < F(x_{tt}) < 15 \quad (15)$$

The above expressions are used for the vapor quality range 0.1< x < 0.95. A factor, F_2, is calculated by the appropriate one of the following expressions:

$$F_2 = 0.707\; Pr_\ell\; Re_\ell^{0.5} \quad for\; Re_\ell < 50 \quad (16)$$

$$F_2 = 5Pr_\ell + 5\, ln\left(1 + Pr_\ell\left(0.09636\, Re_\ell^{0.585} - 1\right)\right)$$

$$for\; 50 < Re_\ell < 1125 \quad (17)$$

$$F_2 = 5Pr_l + 5\, ln(1 + 5Pr_\ell) + 2.5\ln\left(0.0031\, Re_\ell^{0.812}\right)$$

$$for\; Re_\ell > 1125 \quad (18)$$

Another factor, $F(x_{tt})$ is computed by:

$$F(x_{tt}) = \frac{0.15}{x_{tt}}\left(1 + 2.85 x_{tt}^{0.524}\right) \quad (19)$$

ii- For mixture refrigerant R407C, using the correlation of Dobso [12], during the annular flow the local heat transfer coefficient is calculated as:

$$Nu = 0.023\, Re_\ell^{0.8} Pr_\ell^{0.4}\left[1 + \frac{2.22}{x_{tt}^{0.89}}\right] \quad (20)$$

where,

$$Pr_\ell = \frac{cp_\ell \mu_\ell}{K_\ell} \quad (21)$$

$$h_{tp} = F_1[(1-x)^{0.8} + F_2 x^{0.8}] \quad (22)$$

where

$$F_1 = 0.023\left(\frac{G D_i}{\mu_\ell}\right) Pr_\ell^{0.4}\frac{K_\ell}{D_i} \quad (23)$$

$$F_2 = \frac{2.22\, x^{0.8}}{\left(\frac{\rho_\ell}{\rho_v}\right)^{0.445}\left(\frac{\mu_\ell}{\mu_v}\right)^{0.089}} \quad (24)$$

For single phase's liquid and vapor, the heat transfer coefficients calculation of the refrigerant flow inside tubes is based on the correlation of Dittus-Boelter [14].

$$Nu = \frac{h_i D_i}{K} = 0.023\, Re^{0.8}\, Pr^{0.3} \quad (25)$$

$$Re_\ell = \frac{G(1-x)D_i}{\mu_\ell} \quad 900 < Re_\ell < 27000 \quad (26)$$

$$Re_v = \frac{GxD_i}{\mu_v} \quad 12000 < Re_v < 370000 \quad (27)$$

$$Pr_\ell = \frac{\mu_\ell\, cp_\ell}{K_\ell} \quad 2.1 < Pr_\ell < 3.8 \quad (28)$$

$$Pr_v = \frac{\mu_v\, cp_v}{K_v} \quad 0.8 < Pr_v < 1.5 \quad (29)$$

2.2. Calculation of Outside Heat Transfer Coefficients

The heat transfer coefficient by convection at the outside of tube h_o is calculated as:

$$h_o = j\, G_{max}\, cp_a\, pr^{\frac{-2}{3}} \quad (30)$$

where, "G_{max} is the maximum air mass flux based on the minimum flow area, and j is the j-factor". To obtain the heat transfer coefficient of the air side, the j-factor is evaluated depending on the geometry of the fins. When the fin geometry is a plate, the correlation suggested by [16] is used.

$$j = j_4 \times 0.991 \left[2.24\, Re_b^{-0.092} \left(\frac{N}{4}\right)^{-0.031}\right]^{0.607(4-N)} \quad (31)$$

$$j_4 = 0.14 \times Re_D^{-0.32} \left(\frac{S_T}{S_L}\right)^{-0.502} \left(\frac{F_p}{D_o}\right)^{0.0312} \quad (32)$$

where,

$$Re_b = \frac{G_{max} S_L}{\mu_a} \quad (33)$$

$$Re_D = \frac{G_{max} D_o}{\mu_a} \quad (34)$$

$$G_{max} = \frac{\mathring{m}_{air}}{A_{min}} \quad (35)$$

In order to calculate the outside heat transfer resistance, which will be used to calculate the correction for the tube side heat transfer coefficient, and the local overall heat transfer coefficients, the extended surface efficiency, η_o, can be calculated as [17].

$$\eta_o = 1 - \frac{A_f}{A}\left(1 - \eta_f\right) \quad (36)$$

where η_f, the fin efficiency for a radial fin, can be found from the charts published in [18].

The pressure drop of the refrigerant side is calculating for the smooth tube, the equation of fanning with Pierre's correlation [19] were utilize for two-phase flow and single phase flow, respectively. When the flow of refrigerant is single phase, the equation of fanning is used as follow:

$$\Delta p = \left(\frac{2\, f\, G_r^2\, L}{D_i\, \rho}\right) \quad (37)$$

The friction factor for turbulent flow use the correlation proposed by [15].

$$f = 0.046\, Re^{-0.2} \qquad Re > 2300 \quad (38)$$

When the flow of refrigerant is two-phase, Pierre's is showed correlation as follows:

$$\Delta p = \left(f\, \frac{L}{D_i} + \frac{\Delta x}{x}\right) G_r^2 \times v_m \quad (39)$$

$$f = 0.0185 \left(\frac{K_f}{Re}\right)^{0.25} \quad (40)$$

$$K_f = \frac{\Delta h}{L\, g} \quad (41)$$

For bends pressure drop [20]:

$$\Delta P_{bends} = f\, \frac{G^2}{2\rho_i}\, \frac{L_{bends}}{D_i}\, N_{bends} \quad (42)$$

where,

L_{bends}: The length of the bends.

$$L_{bends} = \frac{\pi\, S_T}{2} \quad (43)$$

N_{bends}: Number of bends.

The properties of R22 and R407C are calculated from equation of state [21, 22], and using the subroutine program listed by [23, 24, 25] to calculate all thermodynamic and thermo physical properties for two refrigerants.

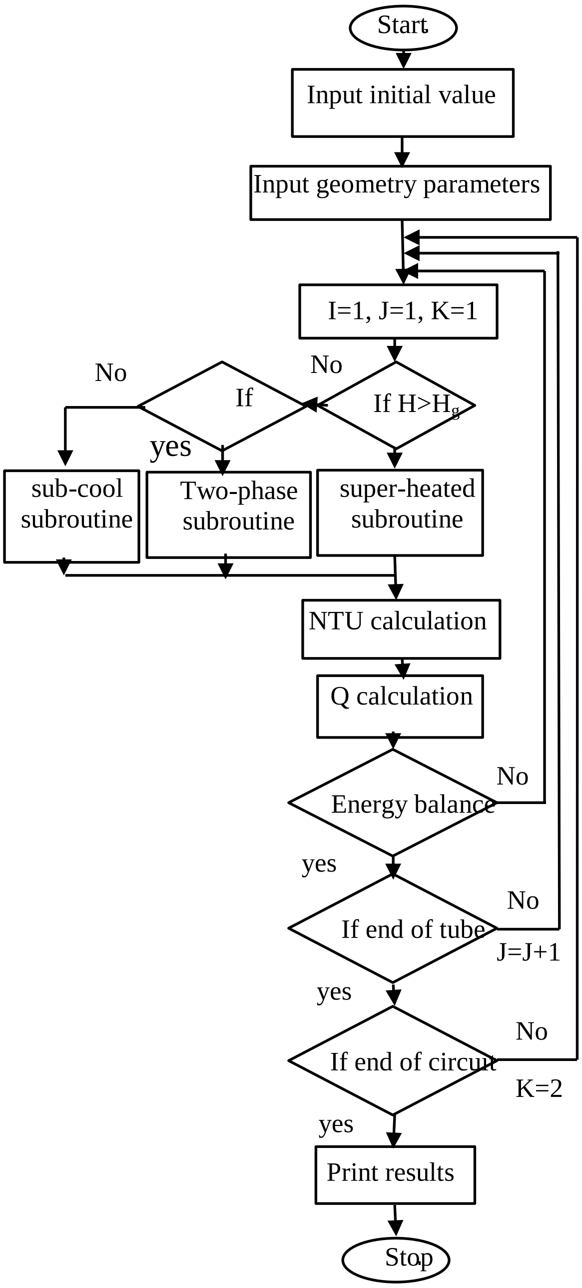

Figure 3: Flow chart of the simulation

3. Method of Solution

The method in this model is calculated tube-by-tube through the condenser. Two circuits in the present study were simulated starting with the refrigerant pipe and carrying on until the refrigerant outlet is reached. For each tube, the calculation is section-by-section until the end of tube. Energy balance for each element is reached between the refrigerant side and air side to correct the assumed temperature. The solution repeated until the difference between the temperatures is less than specified value. The logical flow chart for the iteration procedure was shown in the Fig. 3. The solution method was established by a program in a FORTRAN 90 language.

4. Model Validation

Experimental analysis was done by Khalid and Qusay [26] for condensing unit with finned-tube air cooled condenses using R407C, R290, R22 and R410A as refrigerants. The facility test was included benefit from the supply of the refrigerant system, which is equipped with an open channel with a temperature control system, and section test of heat exchanger. Table 2 lists the inlet parameters, such as pressure and temperature for the refrigerant and air temperature and facing velocity for air. These variables were used as inputs, and then from the simulation, the heat rejects from the condensers were obtained. A good agreement ±10% was found when comparing the results for heat rejects of the condensers, as exhibited in the Fig. 4.

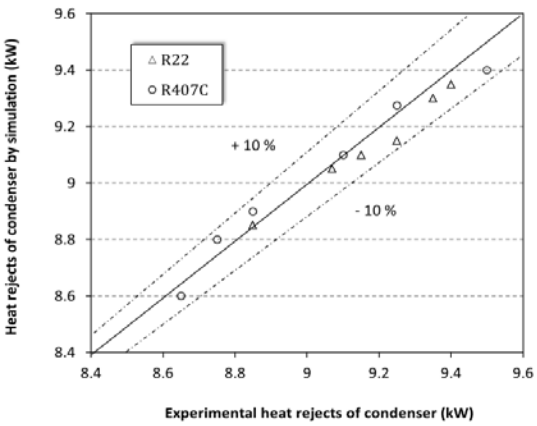

Figure 4: Comparison of experimental heat load of condenser with the predicted by simulation.

5. Results

The simulation results are shown in Figs. 5, 6 and 7 representing the variation of quality with the circuit length of condenser working with R22 and R407C, Fig.5 shows the distribution of quality for R22 and R407C for two circuits of condenser at the same operating condition, same behavior for two refrigerants through two circuits with a small difference in the two-phase flow region. Figures 6 and 7 exhibit the variation of quality along the circuit length for R407C and R22 at different ambient temperatures, respectively any increasing in the ambient temperature causes a change in the distribution of quality and changes the region of two-phase flow and liquid region. Figures 8 and 9 reveal the refrigerant quality with the number of tubes for circuit at different ambient temperatures and two circuits with mass flux for R407C and R22, respectively. For the two refrigerants, when any increasing in the ambient temperature and mass flux of refrigerant offsets increase in the two-phase region based on the sub-cooled region. The

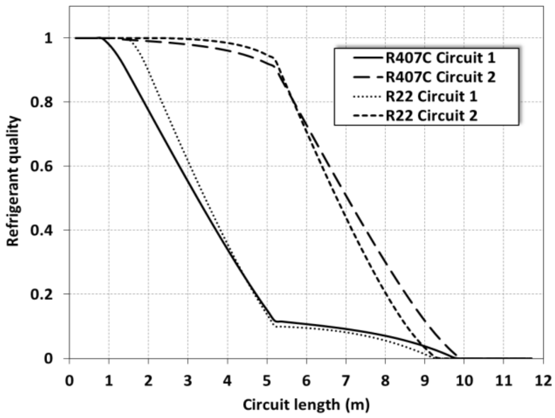

Figure 5: Variation of refrigerant quality with circuit length for R407C and R22.

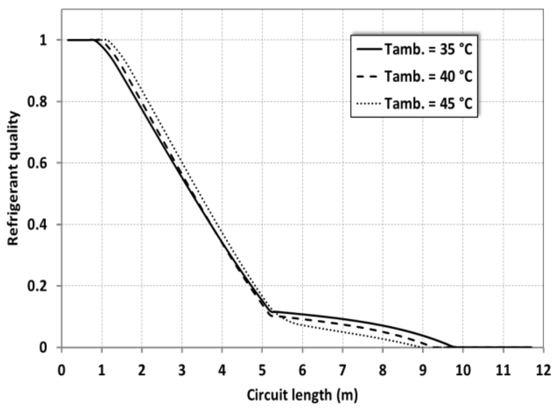

Figure 6: Variation of refrigerant quality with circuit length for R407C at different ambient temperatures.

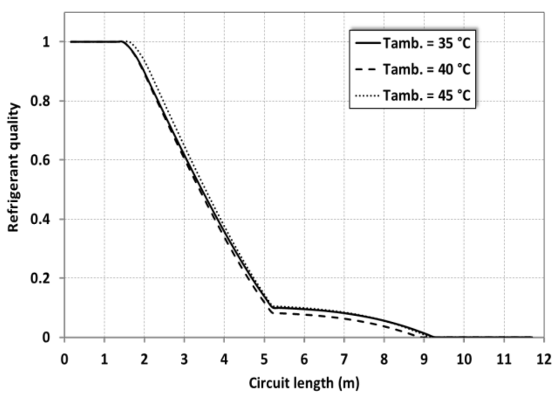

Figure 7: Variation of refrigerant quality with circuit length for R22 at different ambient temperatures.

sub-cooled region for R22 is longer than the sub-cooled region for R407C for the same unit, therefor re-designing in the condenser area for a new unit manufacture or increasing air flow rate for older unit when replacing the refrigerant and keeping the same area of condenser are to have same capacity for the same unit. Figures 10 and

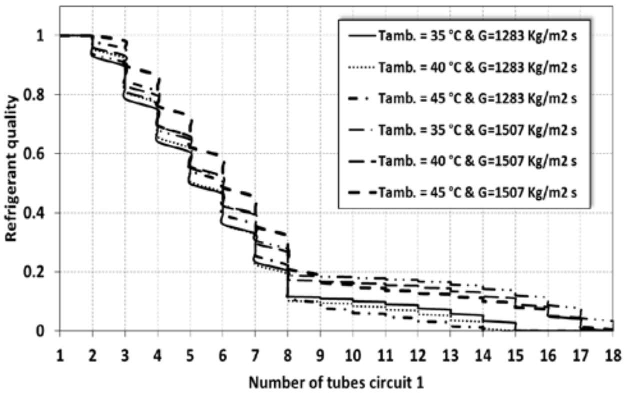

Figure 8: Variation of refrigerant quality no. of tubes for circuit1 for R407C at different ambient temperatures and mass fluxes.

11 represent the heat rejected through the condenser tubes at ambient temperature 35 ^oC and refrigerant mass flux 1283 kg/m^2 s for R407C and R22, respectively. Figures 12 and 13 represent the variation of refrigerant temperature, tubes wall temperature, outlet air temperature from each tube and inlet air temperature with the circuit length for R407C and R22, respectively. The refrigerant temperature for R407C in the two-phase region does not remain constant due to the temperatures glide for R407C zeotropic mixture. Figure 14 shows the enthalpy distribution for both refrigerant R407C and R22 through the circuit of condenser, for the two refrigerants, the enthalpy was found decreases along the circuit length. Figure 15 depicts the relationship of heat transfer rate with the ambient temperature. It is clear that the rate of heat reject usually decreases with the increase of ambient temperature. This is mostly due to the decrease in the properties of the liquid density and liquid thermal conductivity as the condensing temperature increases, consequently the heat transfer coefficients decrease, and performance of heat exchanger with R22 is usually best than R407C for the same conditions. Figure 16 exhibits the comparison of heat reject

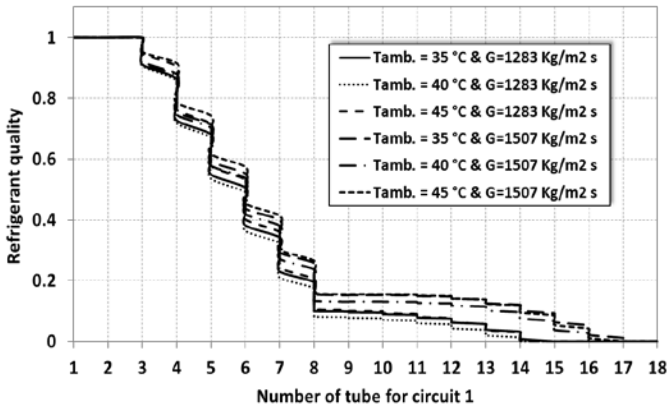

Figure 9: Variation of refrigerant quality with no. of tube for circuit1 for R22 at different ambient temperatures and mass fluxes.

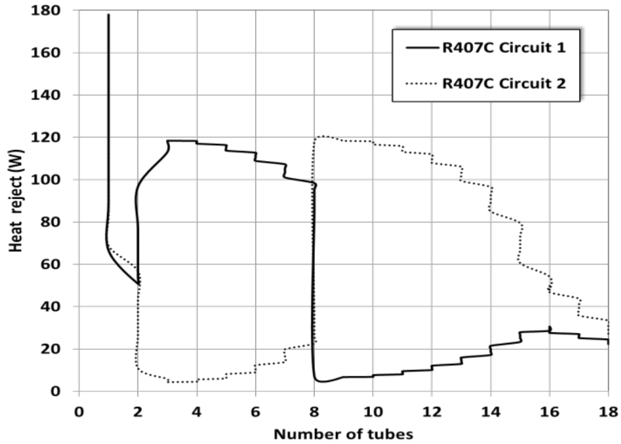

Figure 10: Variation of heat reject with the number of tubes for R407C at 35^oC ambient and 1283 kg/m^2s mass flux.

for R22 and R407C at different ambient temperatures, under the same geometry, the heat reject for a condenser operated with R22 is always greater than that operated with R407C, with the difference of 5%.

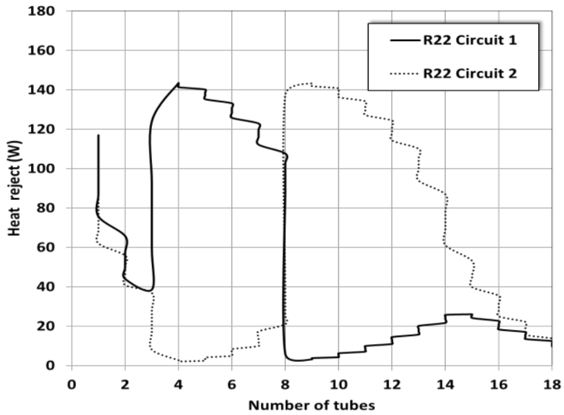

Figure 11: Variation of heat reject with the number of tubes for R22 at 35^oC ambient and 1283 kg/m^2s mass flux.

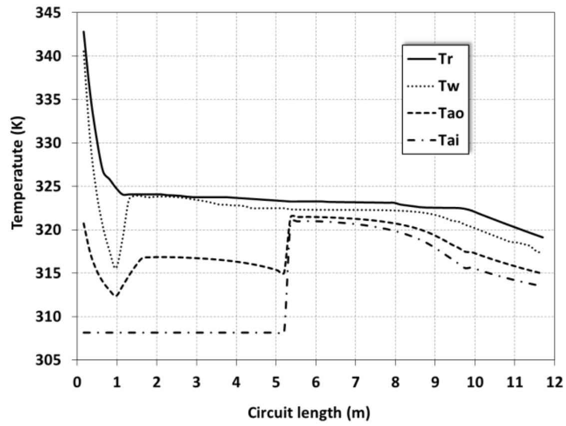

Figure 12: Variation of temperature with the circuit length for R407C at 35^oC ambient and 1283 kg/m^2s mass flux.

6. Conclusion

In this work numerical tested the effects of changing refrigerant type on the condenser performance of a finned- tube heat exchanger without any change in the geometry for a two ton window type air conditioning units. The numerical solution method is tube by tube method established by a program in utilizing a FORTRAN 90 language, the model is validated by comparing its outputs with the previous literature. Simulation is performed for cross-parallel flow type, for two types of refrigerant, R22 and R407C. The parameters used to predict the performance of condenser are the refrigerant mass flow rate and inlet air temperature with constant air velocity. According to the results the conclusions are summarized as follows:

(1) Same behavior for two refrigerants through two circuits with a small difference in the two-phase flow region.
(2) The heat reject for the condenser operated with R22 is always greater than that operated with R407C, with the difference of 5%, at given inlet air temperature.
(3) The condenser work with R22 is having better performance with 6.5% than those of R407C for the same unit.
(4) Increasing air flow rate for older unit, when replacing the refrigerant fluid and keeping the same area of condenser to have a same capacity for the same unit.

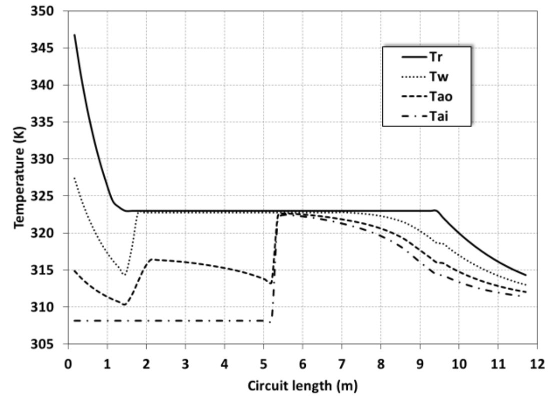

Figure 13: Variation of temperature with the circuit length for R22 at 35°C ambient and 1283 kg/m^{2}s mass flux.

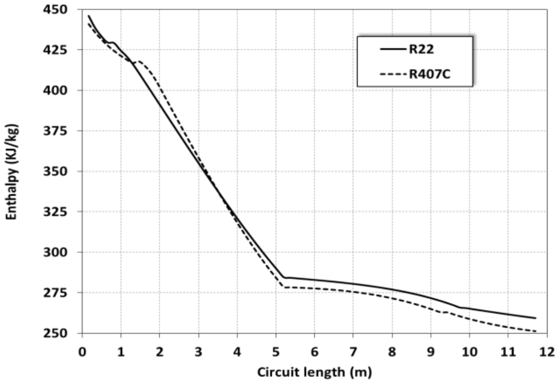

Figure 14: Variation of enthalpy with the circuit length for R407C and R22 at 35°C ambient and 1283 kg/m^{2}s mass flux.

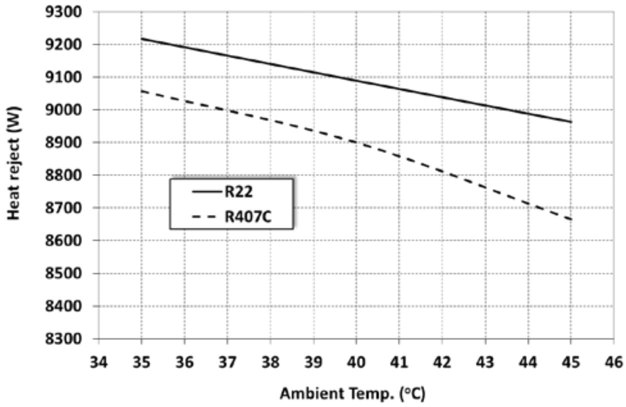

Figure 15: Variation of heat reject with the ambient temperature for R407C and R22 at 35°C ambient and 1283 kg/m^{2}s mass flux.

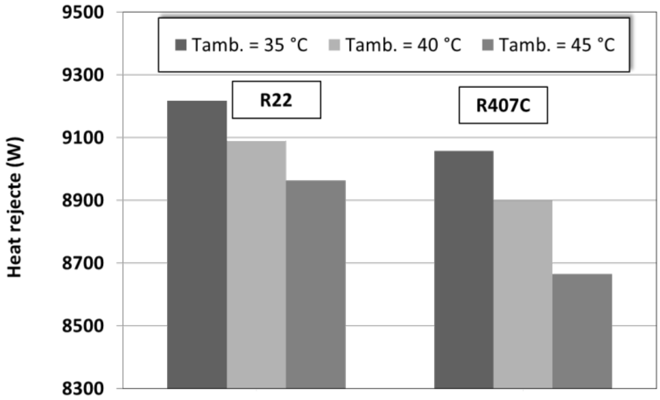

Figure 16: Comparison of heat reject for R407C and R22 at different ambient temperatures.

Table 1: Geometric dimensions of finned-tube heat exchanger.

Parameters	Specifications
Tube row arrangement	Staggered type
Tube position	Horizontal
Fin material	Aluminum
Tube material	copper
Fin thickness (mm)	0.12
Fin pitch (fin/in)	21
Surface configuration	smooth
Number of tubes per row	18
Number of tube row	2
Tube outside diameter (mm)	8
Tube inside diameter (mm)	6.3
Row pitch (mm)	18
Tube pitch (mm)	21
Total tube length (mm)	650
Face coil height (mm)	400
Face coil width (mm)	650
Circuits	See figure (2)

Table 2: Input conditions for the condenser

Refrigerant	T_{rin} (°C)	P_{rin} (bar)	T_{ain} (°C)	V (m/s)
R22	**80**	**19.43**	**35**	**2**
	87	**21.75**	**40**	**2**
	95	**24.27**	**45**	**2**
R407C	**75**	**19.8**	**35**	**2**
	80	**22.45**	**40**	**2**
	85	**25.29**	**45**	**2**

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Nomenclature

A	area[m^2]
cp	specific heat [KJ/kg K]
C	Heat capacity [W/K]
D	diameter [m]
D_e	hydraulic diameter [m]
D_i	inside diameter [m]
D_o	outside diameter [m]
f	friction coefficient
Fp	fin pitch [mm]
G	mass flax [kg/m^2 s]
g	gravitational acceleration [m/s^2]
h	enthalpy [KJ/kg]
h_i	inside refrigerant heat transfer coefficient [W/m^2 K)
h_{oa}	outside air heat transfer coefficient [W/m^2 K)
h_{tp}	refrigerant heat transfer coefficient in two phase region [W/m^2 K]
j	Colburn factor
K_f	Pierre's boiling number
k	thermal conductivity [W/m K]
L	tube length [m]
m°	mass flow rate [kg/s]
N	number of tubes
NTU	Number of transfer unit
p	pressure [pa]
Pr	Prandtl numper
q	heat flux [W/m^2]
Q	heat transfer rat [W]
R	thermal resistance [m^2 K/W]
Re	Reynolds number
r	radius [m]
S_T	traversing tube pitch [m]
S_L	longitudinal tube pitch [m]
T	temperature [K]
U	overall heat transfer coefficient [W/m^2 K]
v	specific volume [m^3/kg]
x	quality
Xtt	Lockhart-Martinelli parameter

Greek Symbols

ρ	density [kg/m^3]
η	efficiency
υ	viscosity [m/s]
Δ	difference
μ	dynamic viscosity [kg/m s]
ε	effectiveness

Subscripts

a	air
b	band
c	cooled, critical
f	fin
h	hot
i, in	in, inlet
ℓ	liquid
m, min, max	mean, minimum, maximum
o, out	out, outlet
r	refrigerant
rej	reject
t	tube

مقارنة اداء مكثف انبوب - زعنفة مبرد بالهواء يعمل بمائعي التثليج R22 و R407C

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الخلاصة

تم في هذا البحث دراسة عددية ورياضية لاداء مكثف ذو انابيب مزعنفة في وحدة تكييف هواء سعة اثنين طن تبريد تعمل بمائع التثليج R22 كمائع اساسي وR407C كمائع بديل. تمت دراست عدة متغيرات كضغط المكثف, ودرجة حرارة المحيط, ومعدل تدفق الكتلة لمائعي التثليج. عند مقارنة اداء المكثف عندما يعمل بمائعي التثليج R407C و R22 وجد انه يجب اعادة تصميم المكثف ليلائم العمل مع المائع R407C مع الابقاء على نفس اجزاء المنظومة الاخرى. اظهرت النتائج تشابه في سلوك مائعي التثليج لذا يمكن ان يعمل المكثف بمائع التثليج R407C بنفس الوحدة مع بعض التحسينات في تركيب المبادل الحراري ولنفس سرعة الهواء. تم مطابقة نتائج الانموذج المقترح مع النتائج العملية المنشورة في البحوث ولابعاد مختلفة , ووجد ان نسبة التوافق مع النتائج العملية لكمية الحرارة المطروحة في المكثف هي بحدود 10 % ± .