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Experimental Investigation of Condensation of Refrigerants R134a and R12 in Air Cooled Horizontal Condenser

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Abstract :

With the phasing out of ozone depleting refrigerants under the Montreal protocol 1987 researches on alternatives to new refrigerants (HFC's) and (HC 's) are made to replace refrigerants (CFC) and (HCFC) in air-conditioning and cooling systems that destroy the ozone layer.

Most of the refrigerating systems specially the domestic units replaces R12 by R134a, this study make a comparison of performance between the two cycles specially in the condenser where the main difference between R12 and R134a, while in the evaporating where the pressure of the refrigerant determines the evaporator temperature and therefore the cooling capacity of the system ,the two refrigerants are close in behavior.

In this study, the experimental results of heat rejected at condenser, coefficient of performance, the experimental condensing heat transfer coefficients of R12 and R134a, in a horizontal copper finned tube with an outer diameter of 10mm are presented at different mass fluxes ,and different ambient air temperatures during condensation under annular flow . A correlation has been developed from the data obtained, the refrigerant side heat transfer coefficients obtained from experimental study is different by 5%-12% from that computed from the correlations developed by Shah , cited in [1,2].

دراسة تجريبية لعملية التكثيف لمائعي التبريد R134a و R12 في مكثف أفقي مبرد بالهواء

الخلاصة :

بعد معاهدة مونتريال 1987 لأيقاف أنتاج الموائع الضارة بالبيئة, أجريت بحوث كثيرة على موائع صديقة للبيئة مثل (HFC's) و (HC 's) لاستبدال الموائع الضارة بالبيئة مثل (HCFC) و (CFC) في مجال التبريد والتكييف.
ويعتبر(R134a) البديل الأساسي لمائع التثليج (R12) الذي كان مستخدما في أجهزة التبريد بكثرة, بالأخص المنزلية منها . تم في هذا البحث إجراء مقارنة بين أداء الجهاز عند استخدام R12 كمائع تبريد و R134a خاصة في المكثف حيث الفرق الرئيسي في المواصفات الحرارية بين (R12) و(R134a) . أما في مجال المبخر فأن المواصفات الحرارية للمائع R134a (العلاقة بين درجة الحرارة والضغط الذي يحدد سعة التبريد للمنظومة) مقاربة جدا لمائع التثليج (R12). تم في

هذه الدراسة الحصول على نتائج عملية للحرارة المفقودة من المكثف ومعامل أداء المنظومة و معامل انتقال الحرارة لمائعي التبريد (R12) و (R134a) داخل أنابيب مكثف أفقي نحاسي , قطر الأنبوب الخارجي 10 ملم عند معدلات مختلفة لتدفق المائع ودرجات مختلفة لحرارة المحيط الخارجي خلال الجريان الحلقي ووضعت معادلات لبعض تلك النتائج . كما أظهرت النتائج العملية أن معامل انتقال الحرارة يختلف بنسبة 5%-12% عن النتائج باستخدام العلاقة التجريبية المقدمة من قبل Shah . الواردة في[1,2] .

Keywords: Condensation: R12; R134a ; Horizontal condenser

Nomenclature:

Re	Reynolds number $\rho vD/\mu$	
Nu	Nusselt number hD/k	
Pr	Prandtl number $C_p\mu/k$	
$h_{r,sp}$	single phase refrigerant-side heat transfer coefficient	$W/m^2.K$
h_{tp}	two-phase heat transfer coefficient	$W/m^2.K$
h_{tpm}	mean heat transfer coefficient from Shah correlation	$W/m^2.K$
hr	refrigerant-side heat transfer coefficient	$W/m^2.K$
hi	enthalpy of refrigerant inter condenser	kJ/kg
ho	enthalpy of refrigerant at outlet of condenser	kJ/kg
T_{sat}	saturation temperature of refrigerant	ºC
T_{si}	temperature of inner tube surface	ºC
T_{so}	average outside tube surface temperature	ºC
T_{ai}	temperature of air inlet across condenser	ºC
T_{ao}	temperature of air outlet	ºC
Q	rate of heat flow	Watt
U	overall heat transfer coefficient of condenser	$W/m^2.K$
x_t	tube thickness	m
A_o	outside area of tube	m^2
A_i	inside area of tube	m^2
A_m	mean circumferential area of tube	m^2
k_t	conductivity of tube metal	W/m .K

C_p	specific heat at constant pressure	kJ/kg .K
G	total mass flux	$kg/m^2.s$
h_{sub}	enthalpy of subcooled refrigerant	kJ/kg
h_{sup}	enthalpy of superheated refrigerant	kJ/kg
h_{satl}	enthalpy of saturated liquid	kJ/kg
h_{satv}	enthalpy of saturated vapor	kJ/kg
m	refrigerant mass flow rate	kg/s
μ_l	saturated liquid viscosity	kg/ms
μ_v	saturated vapor viscosity	kg/ms
ρ_l	saturated liquid density	kg/m^3
ρ_v	saturated vapor density	kg/m^3
x	vapor quality	

Introduction:

Several researches about condensation heat transfer in tube are carried ,some of these are, Shah [3], develop a simple dimensionless correlation for predicting heat-transfer coefficients during film condensation inside pipes. These include fluids such as water, R-11, R-12, R-22, R-113, methanol, ethanol, benzene, toluene, and trichloroethylene condensing in horizontal, vertical, and inclined pipes of diameters ranging from 7 to 40mm, saturation temperatures from 21 to 310 ºC, liquid Reynolds numbers from 100 to 63 000, and liquid Prandtl numbers from 1 to 13. Thome et al. [4],studied a new general flow pattern/flow structure based heat transfer model for condensation inside horizontal, plain tubes. The model predicts local condensation heat transfer coefficients for the following flow regimes: annular, intermittent, stratified-wavy, fully stratified and mist flow. Graham et al. [5], performed condensation experiments over a mass flux range of 75–450 kg m^{-2} s^{-1} in an 8.91 mm inside diameter, axially grooved, micro-fin tube with R134a . At 75 kg m^{-2} s^{-1}, the axially grooved tube performs marginally better than a smooth tube. Wang et al. [6], presented a comprehensive comparison of eight previously proposed correlations with available experimental data for the frictional pressure drop during condensation of refrigerants in helically grooved, horizontal microfin tubes. SHAO et al.[7], presented an experimental investigation on condensation heat transfer of R-134a in horizontal straight and helically coiled tube-in-tube heat exchangers. For refrigerant mass flux varying from 100 kg/m^2 s to 400 kg/m^2 s and the vapor quality ranging from 0.1 to 0.8 . Gayurrul et al. [8], made a general correlations to predict boiling and condensation heat transfer coefficients of refrigerants such as

R-22, R-12, R-113, and R-134a in both the low-fin tubes and the micro-fin tubes. Cavallini et al. [9], presented a new simple model for the prediction of the heat transfer coefficient applied to condensation in horizontal microfin tubes of halogenated and natural refrigerants . Giovanni [10], presented an experimental tests on HFC-134a condensation inside a small brazed plate heat exchanger, the effects of refrigerant mass flux, saturation temperature and vapor super-heating are investigated . Ozden et al. [11], presented an experimental results of the condensing heat transfer coefficients of R600a ,a hydrocarbon refrigerant, in a horizontal smooth copper tube with an inner diameter of 4mm and outer diameter of 6mm at different vapor quality and different mass fluxes during condensation under annular flow conditions. Azam [12], study the condensation of R407C in finned tube condenser for refrigeration system of one ton refrigeration capacity. This study performs the calculations and comparison of the refrigerant side heat transfer coefficient for R134a and R12 flowing inside air cooled finned tube condenser at different mass fluxes, and different ambient air temperatures during condensation under annular flow .

Theoretical Aspects:

The condenser is a heat exchanger that usually rejects all the heat from the refrigeration system .This includes not only the heat absorbed by the evaporator but also the energy input to the compressor. The condenser accept hot , high -pressure refrigerant ,usually a super heated gas, from the compressor and reject heat from the gas to some cooler substance ,usually air or water. As energy is removed from the gas it condenses and this condensate is drained so that it may continue its path back through the expansion valve or capillary to the evaporator [13].

The condenser is usually made of copper or steel tubing with fins attached which increase the effective area of heat dissipation surface ,for domestic use the condenser is usually air cooled by natural or forced convection using motor driven fan to force air over the condensing tubing and to increase the cooling effects on the condenser [14].

In the condenser , three zones , corresponding to refrigerant desuperheating ,condensation and Subcooling are considered .

In the superheating zone the surface temperature is above the saturation temperature so there is no condensation in this region .

Kays and London ,cited in[2,15], have developed a heat transfer correlation for single phase turbulent flow. This correlation was developed using empirical data taken from a variety of refrigerants in circular heat exchanger tubes under several thermodynamic conditions. The correlation is expressed as:

$$St\, Pr^{2/3} = a_{st}\, Re^{\,b_{st}} \quad \text{------------- (1)}$$

where the coefficients a_{st} and b_{st} are as follows:

Laminar flow	$Re < 3,500$	$a_{st} = 1.10647$	$b_{st} = -0.78992$
Transition	$3,500 \leq Re \leq 6,000$	$a_{st} = 3.5194 \times 10^{-7}$	$b_{st} = 1.03804$
Turbulent flow	$6,000 < Re$	$a_{st} = 0.2243$	$b_{st} = -0.385$

and the Stanton number, St is expressed as:

$$St = \frac{Nu_D}{Re\,Pr} = \frac{h_{r,sp}}{G\,C_p} \qquad \text{-------------- (2)}$$

The real condensation of refrigerant occurs in the condensation zone , where two phase flow (a combination of liquid and vapor refrigerant) exists .

A large number of techniques for predicting the heat-transfer coefficients during condensation inside pipes have been proposed .These range from very arbitrary correlations to highly sophisticated treatments of the mechanics of flow .

The two-phase flow heat transfer model developed by Shah is a simple correlation , cited in[1,2,15],that has been verified over a large range of experimental data. In fact, experimental data from over 20 different researchers has been used in its development. For this model, at any given quality, the two-phase heat transfer coefficient is defined as:

$$Nu=Nu_l\left[(1-x)^{0.8} + \frac{3.8x^{0.76}(1-x)^{0.04}}{pr^{0.38}}\right] \qquad \text{-------------- (3)}$$

pr is the reduced pressure = condenser pressure / critical pressure

$0.002 < pr < 0.44$

$0 < x < 1$

$Re_l > 350$

$Pr_l > 0.5$

$$Nu_l = 0.023\, Re_l^{0.8}\, Pr_l^{\,0.4} \qquad \text{-------------- (4)}$$

Nu_l is the liquid phase refrigerant side Nusselt number .

The two-phase heat transfer coefficient is defined as:

$$h_{tp}= h_l\left[(1-x)^{0.8} + \frac{3.8x^{0.76}(1-x)^{0.04}}{pr^{0.38}}\right] \qquad \text{------------ (5)}$$

The average heat transfer coefficient h_{tpm} is obtained by integrating equation (5) along the tube length. The vapor quality of refrigerant is assumed that it is directly proportional with the tube length therefore the average heat transfer coefficient [2,15]:

$$h_{tpm} = \frac{h_l}{(x_o - x_i)} \left[\frac{-(1-x)^{0.8}}{1.8} + \frac{3.8}{pr^{0.38}} \left[\frac{x^{1.76}}{1.76} - \frac{0.04x^{2.76}}{2.76} \right] \right]_{x_i}^{x_o} \quad \text{-- (6)}$$

h_l is the heat transfer coefficient of liquid calculated from equation (4).

For complete condensation, (x varying from 1 to 0),the mean heat transfer coefficient reduces to the following expression:

$$h_{tpm} = h_l \left[0.55 + \frac{2.09}{pr^{0.38}} \right] \quad \text{------------- (7)}$$

Experimental Work:

The refrigeration apparatus shown in Figure(1) is used in this work. Measurements of evaporator and condenser pressures ,temperature at inlet and outlet of evaporator, and at points as shown in Figure(2) along the condenser surface. Volume flow rate of refrigerant ,inlet and outlet temperature of air across the condenser ,air velocity ,and the readings of current ,voltage ,power consumed are taken using refrigerant R134a at different volume flow rate at ambient temperature of (32ºC,27.5 ºC,19.5ºC) .The same measurements are taken for R12 at ambient temperature of(32ºC,27.5ºC,22ºC) for comparison purposes. The condenser is supplied with glass tubes to show the phase of refrigerant along the condenser.

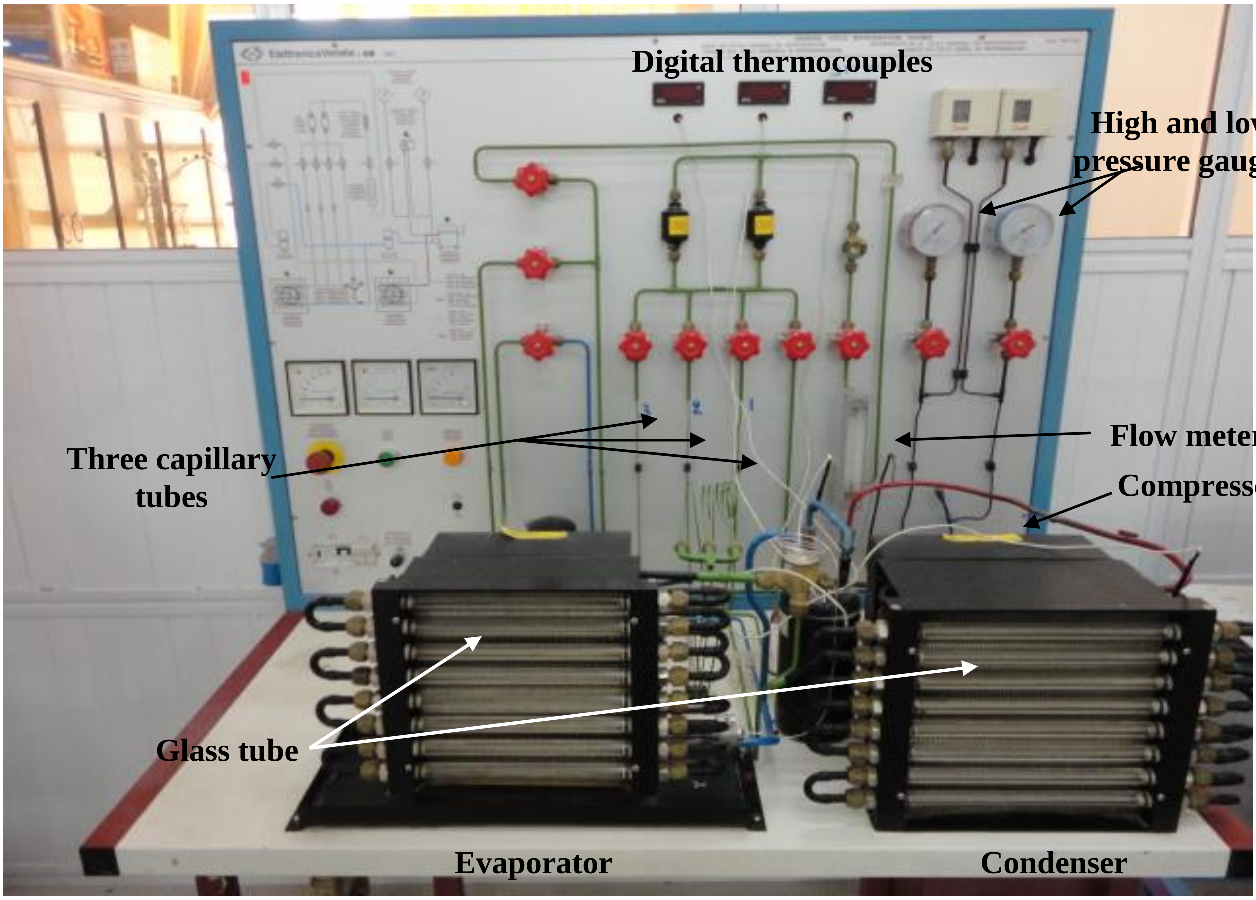
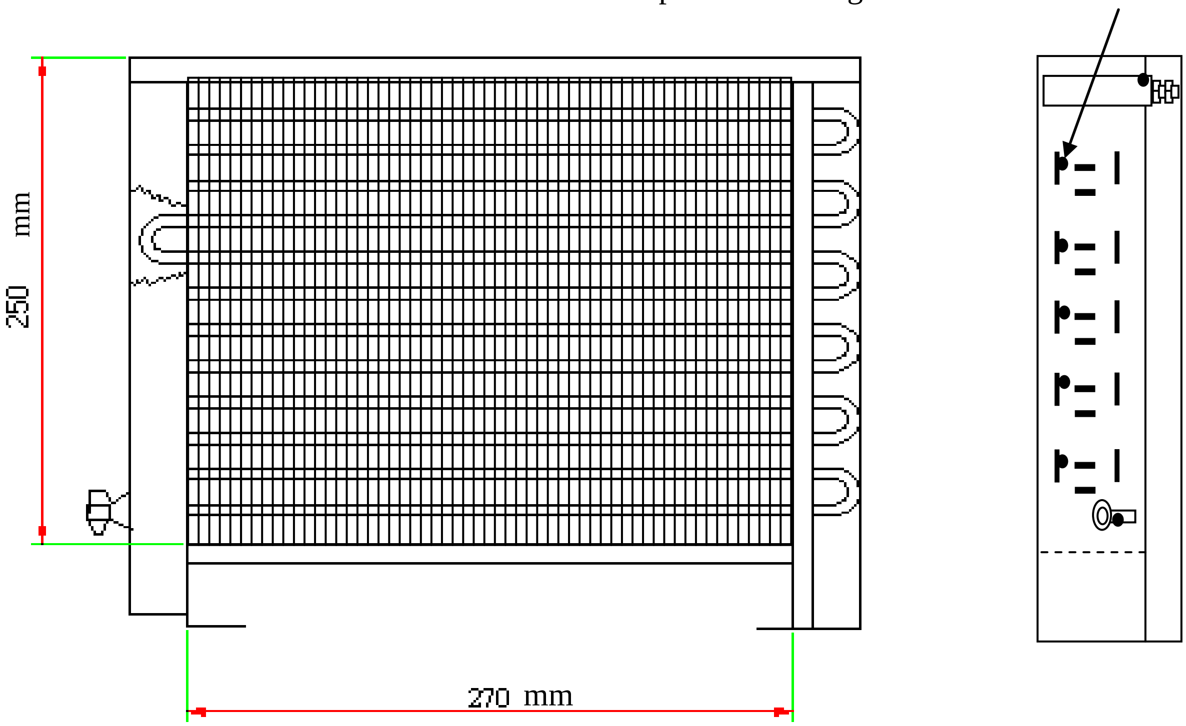


Figure (1): Components of the experimental apparatus .



Figure(2):The model of air cooled condenser

The condenser shown in Figure(2) has an outer diameter of .01m,while the length of condenser tube is 13.31m and the surface area of the condenser is calculated as:
Overall heat-exchange surface area= surface of the tube +fins=3.378 m^2 -- (8)

The condenser used in this study is air cooled fin tube heat exchanger ,and the refrigerants used are R134a , and R12.The fan forces air between the fins and over the tubes . When the refrigerant exits from the compressor, it enters the condenser as a superheated vapor and exits as a sub-cooled liquid.
The amount of heat per unit mass of refrigerant rejected from each section can be expressed as the difference between the refrigerant enthalpy at the inlet and at the outlet of each section.

The total heat rejected from the hot fluid, which in this case is the refrigerant R134a , or R12 to the cold fluid, which is the air, is dependent on the heat exchanger effectiveness and the heat capacity of each fluid , mass flow rate, and ambient temperature .

$$q=(h_{sup} - h_{satv}) +(h_{satv} -h_{satl}) + (h_{satl} - h_{sub}) \qquad \text{---------} (9)$$

Once the temperature of refrigerant enters and leaves the condenser, condenser pressure and volume flow rate are measured then the heat rejected from condenser Q is calculated as,

$$Q= m (hi- ho) *1000 \quad \text{------------- (10)}$$

Under steady state conditions, the rate of heat transfer Q is the same from the outside surface to the inside surface of the tube and from the inside surface of the tube to the refrigerant .
Although the difference between average outside tube surface temperature T_{so} and inner tube surface temperature T_{si} is small ,the inner tube surface temperature is calculated as :

$$Q=(k_t/x_t)\, A_m\, (T_{so} – T_{si}) \quad \text{-------------- (11)}$$

The average values of experimental heat transfer coefficient are calculated at average surface temperature of the condenser as[11]:

$$Q= h_r\, A_i\, (T_{sat} –T_{si}) \quad \text{--------------- (12)}$$

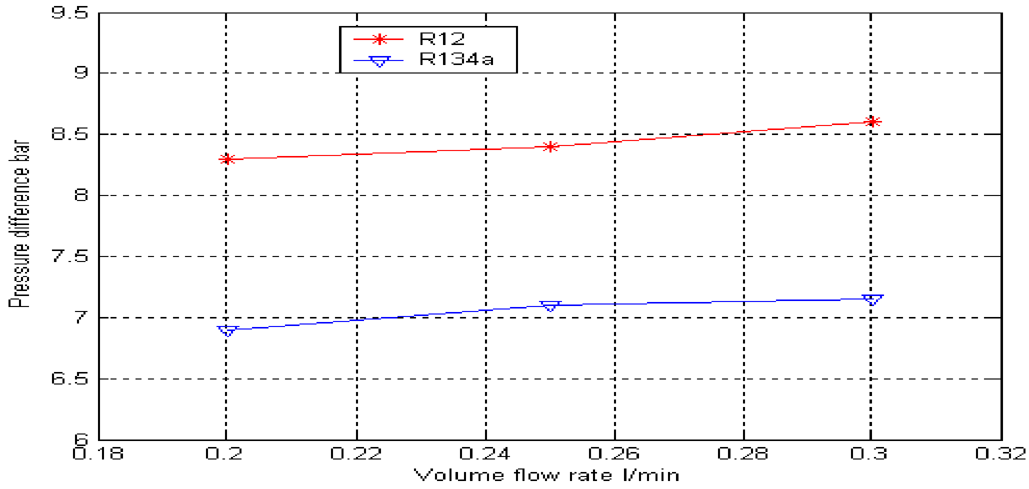
The overall heat transfer coefficient of the condenser U is equal to the ratio between heat flow rate Q, overall heat-exchange surface area A_o and logarithmic mean temperature difference ΔT_{ln}, computed as [10] :

$$U=Q/(A_o\, \Delta T_{ln}) \quad \text{--------------- (13)}$$

$$\Delta T_{ln}=((T_{sat} - T_{ai})- (T_{sat} - T_{ao}))/(\ln(T_{sat} - T_{ai})/ (T_{sat} - T_{ao})) \quad \text{-------- (14)}$$

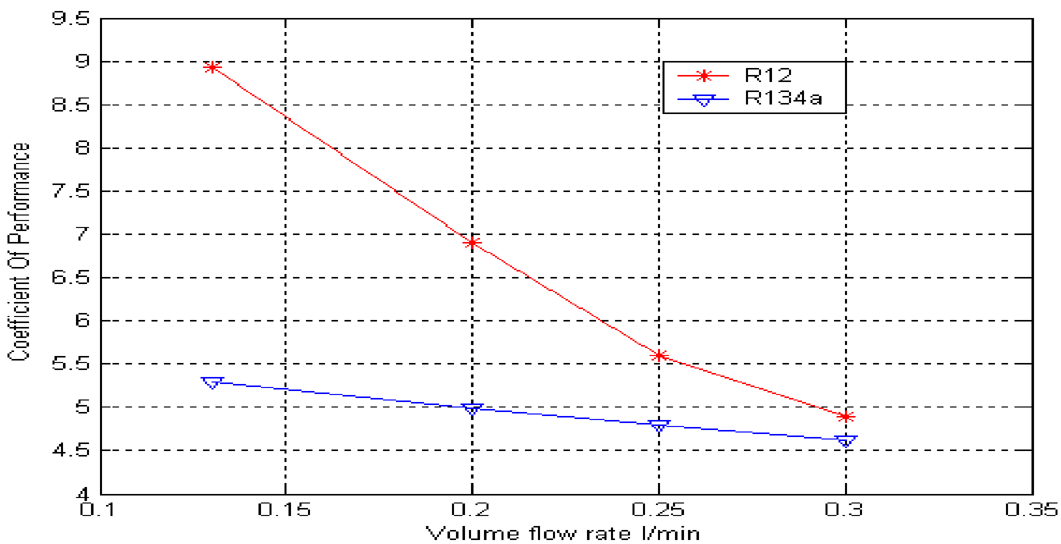
Results and discussion:

Figure(3) shows that the pressure difference when the apparatus is charged with R12 is more than pressure difference using R134a.This is because the specific volume of R12 is less than specific volume of R134a ,there for compressor of less capacity can be used for R12, to achieve the same pressure difference ,so the coefficient of performance is affected.



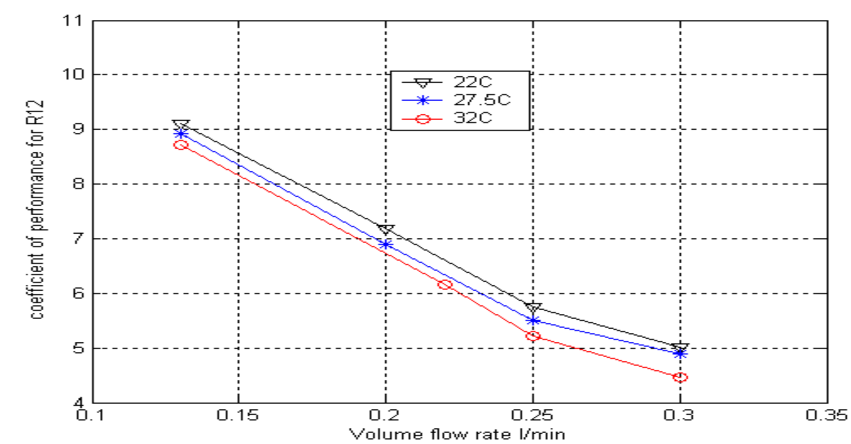
Figure(3): Pressure difference of R12 and R134a at the same ambient temperature

Figure (4) shows the variation of the coefficient of performance of both R12 and R134a at several volume flow rates at 27.5ºC ambient air temperature for the same experimental condenser and evaporator pressure ,outlet temperatures from evaporator and condenser for R12 .



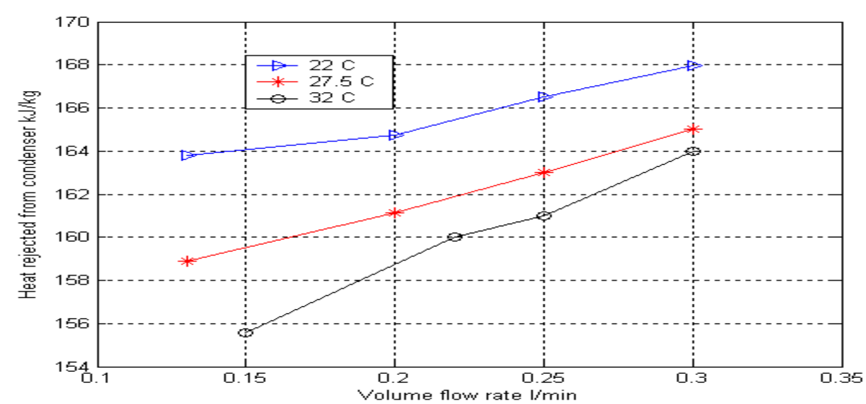
Figure(4):The coefficient of performance of both R12 and R134a at the same condenser and evaporator pressure at different values of refrigerant flow rate

Since in most of the domestic refrigeration systems the condenser is air cooled the effect of ambient air temperature on the condenser performance should be considered specially for places with large variation of outside air temperature even through the same day .
The amount of heat rejected from condenser is decreased with increasing outside air temperature , and condenser pressure increased hence the amount of cooling capacity will decreased, decreasing the coefficient of performance of the apparatus as shown in Figure(5) .

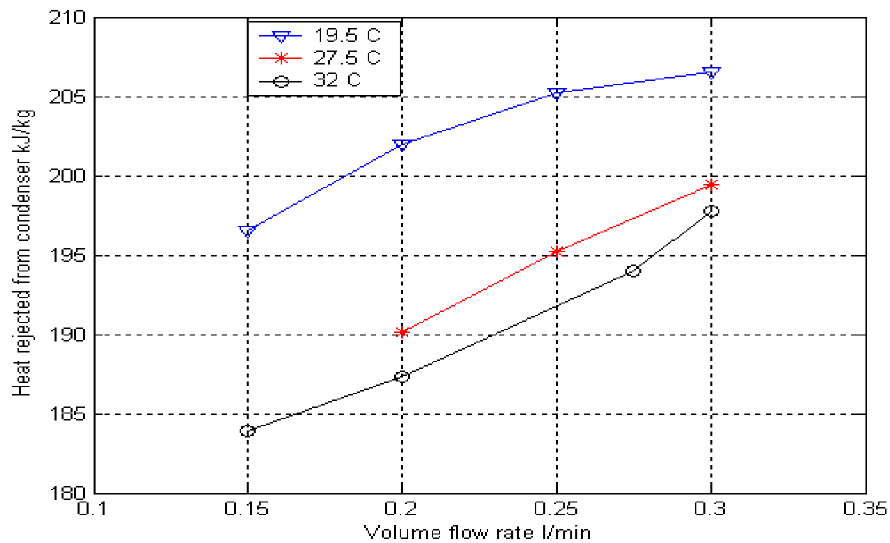


Figure(5): Effect of outside air temperature on coefficient of performance at different flow rate of R12

The variation of heat rejected from condenser at different ambient temperature is shown in Figure (6) for R12 and Figure (7) for R134a at different volume flow rate of refrigerant.

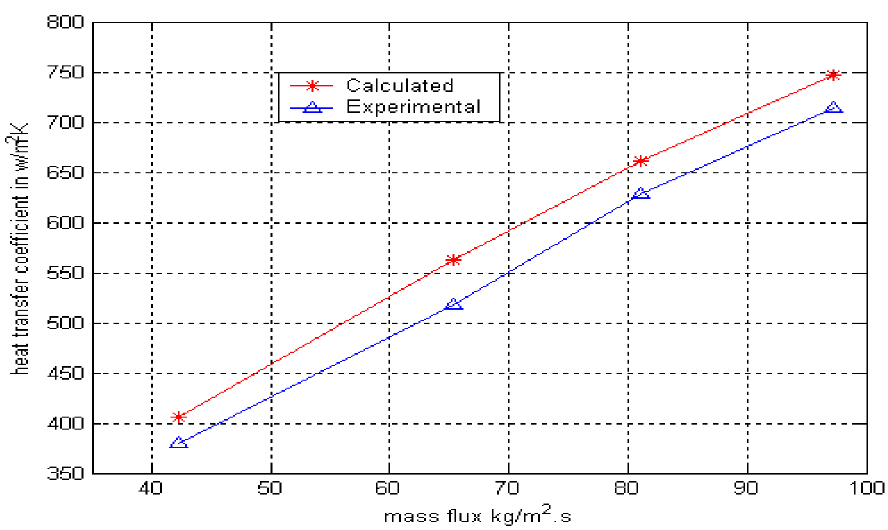


Figure(6):Heat rejected from condenser at ambient air temperatures (22ºC,27.5ºC,32ºC) for R12



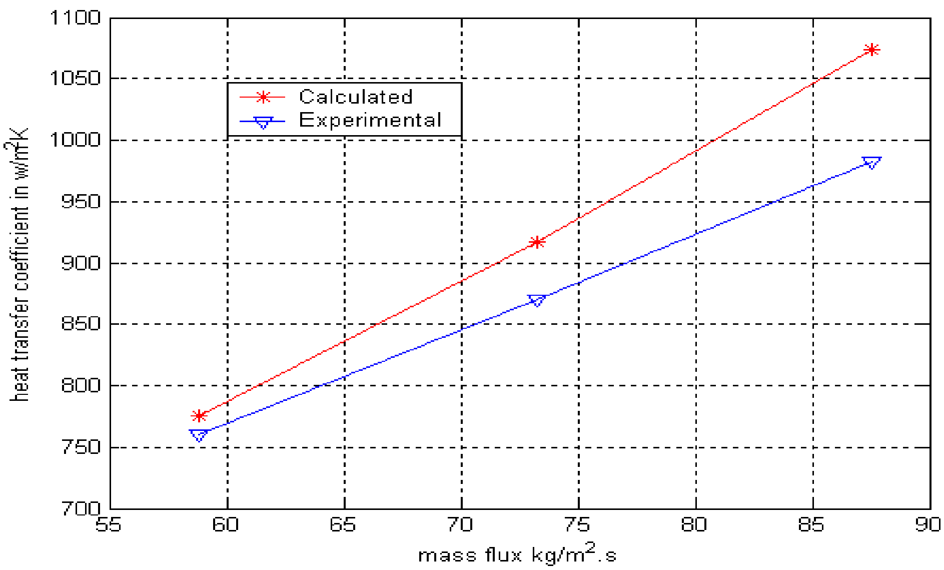
Figure(7):Heat rejected from condenser at ambient air temperatures (19.5ºC,27.5ºC,32ºC) for R134a

The calculated values of refrigerant side average heat transfer coefficient of R12 from Shah correlations compared with experimental values of (hr) computed from equation(12) is shown in Figure(8) .



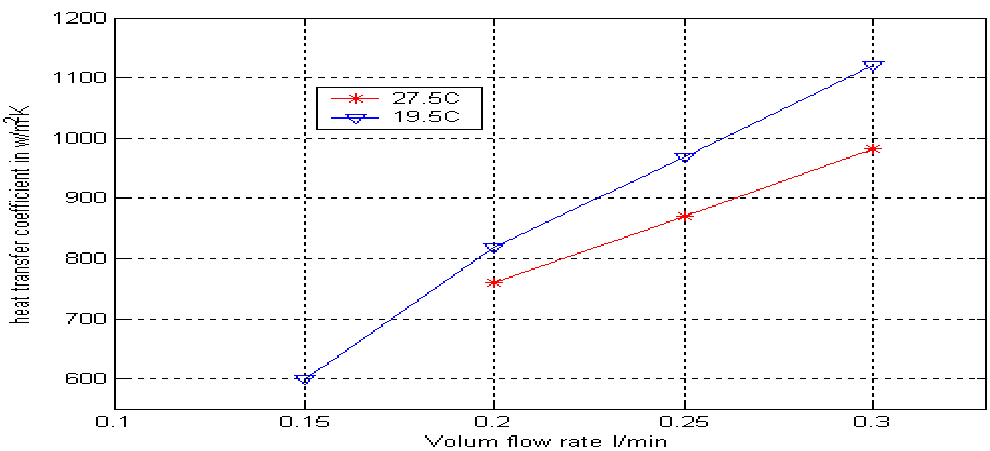
Figure(8):Difference between experimental and calculated values of average heat transfer coefficients of R12 at ambient air temperature of 27.5ºC

The theoretical values of refrigerant side average heat transfer coefficient of R134a computed from Shah correlation at experimental data compared with experimental values of (hr) computed from equation(12) shown in Figure(9), indicates less than 12% difference between them.



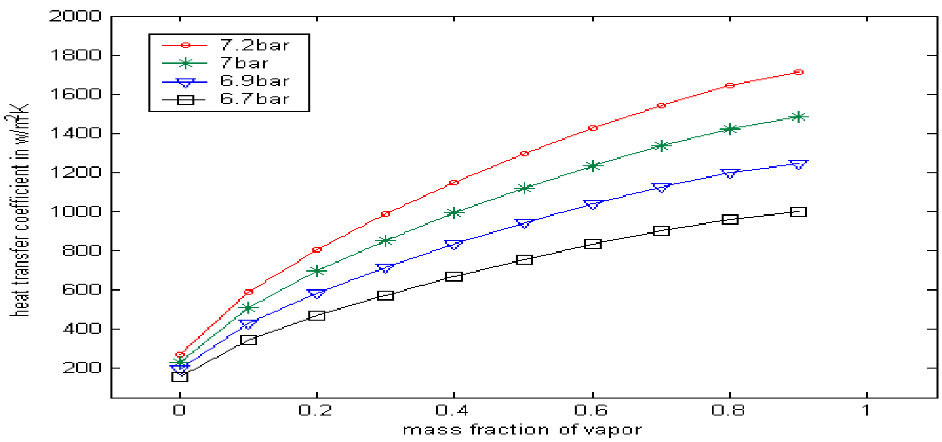
Figure(9):Difference between experimental and calculated values of average heat transfer coefficients of R134a at ambient air temperature of 27.5ºC

Figure(10) shows that the experimental values of heat transfer coefficient of R134a is decreased with increasing the ambient air temperature at different volume flow rate.



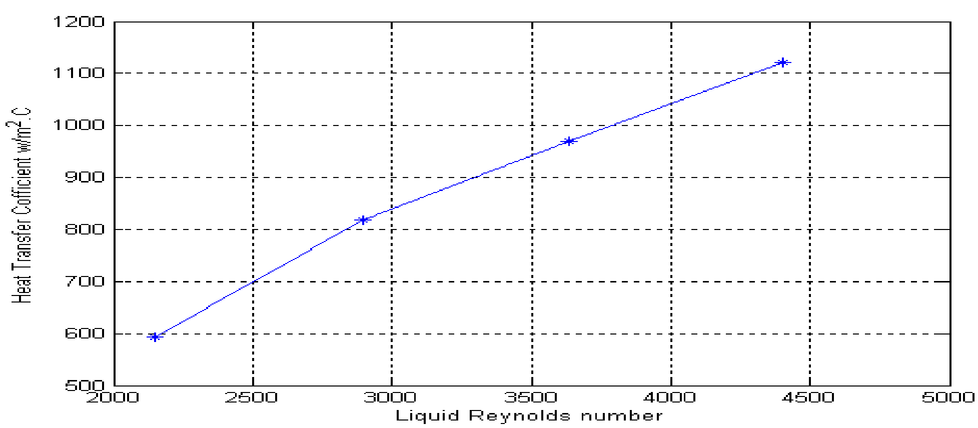
Figure(10):Variation of experimental values of heat transfer coefficient of R134a at different values of volume flow rate at ambient air temperature of (27.5 º C and 19.5 º C)

As a large density results in a thinner growth of condensate film , high mass flux and high density of thinner condensate film will increase the value of the mixture Reynolds number which increases the value of Nusselt number and hence increasing the heat transfer coefficient[16], so the heat transfer coefficient increases with increasing dryness fraction as shown in Figure (11), results calculated from shah correlation equation(3) at many experimental values of condenser pressure for different mass fluxes at ambient air temperature of 19.5ºC.



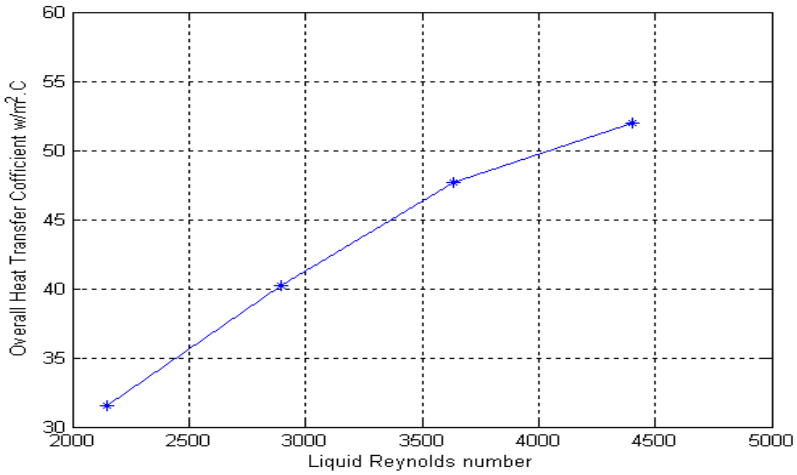
Figure(11):Heat transfer coefficient calculated from shah correlation at experimental values of condenser pressure for R134a at ambient air temperature of 19.5ºC

Figure (12) shows the relation between experimental average heat transfer coefficient and liquid Reynolds number for R134a.



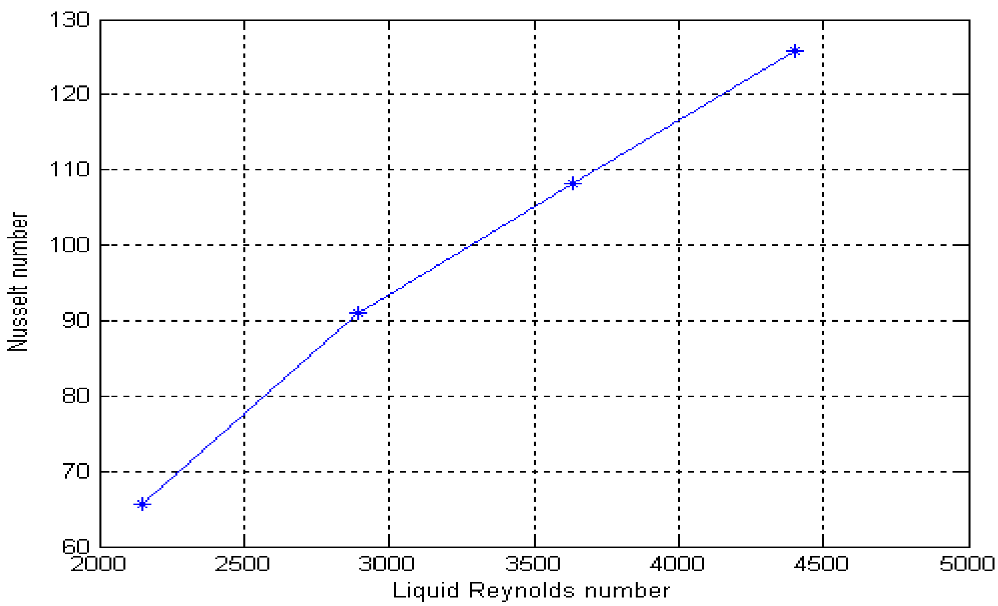
Figure(12):Variation of experimental values of heat transfer coefficient and Reynolds number for R134a

Figure(13) shows the variation of overall heat transfer coefficient computed from equation (13) and liquid Reynolds number for R134a.



Figure(13):Variation of overall heat transfer coefficient and liquid Reynolds number for R134a

Figure (14) shows the variation of experimental results of Nusselt number and Reynolds number .



Figure(14):Relation between experimental (liquid Reynolds number and Nusselt number) for R134a

All the thermodynamic properties of R12 and R134a at any state are taken from [17,18] , and Microsoft EES (Engineering Equation Solver) .While MATLAB program is performed to calculate and draw the results .

Conclusion:

The following observations are indicated from this study :

1. The coefficient of performance of R12 is better than the coefficient of performance of R134a at the same condensing and evaporating pressures ,but because R12 is one of the refrigerants that destroy the ozone layer, it was replaced by R134a.
2. The heat rejected from condenser and the coefficient of performance of the apparatus are decreased with increasing ambient air temperature.
3. The difference between the experimental results of refrigerant side average heat transfer coefficient and that computed from Shah correlation is 5%-12%.
4. The values of heat transfer coefficient is decreased with increasing the ambient air temperature, and increased with increasing refrigerant mass flux.
5. The overall heat transfer coefficient is increased with increasing Reynolds number.
6. From the experimental work of this study ,an empirical formula is predicted to relate the Nusselt number and Reynolds number at different volume flow rate:

$$Nu= -3.7e\text{-}006Re^2 + 0.05Re -26$$

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