A STUDY THE EFFECTS OF SOME OPERATION PARAMETERS ON CONDENSER PERFORMNACE IN SMALL A.C. UNIT WITH (R134a)+

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Abstract

Finned-tube heat exchangers are widely used in space conditioning systems, as well as any other applications requiring heat exchange between liquids and gases. The only significant opportunity for electrical power use reduction residential in air conditioners is in technology improvement of the finned-tube heat exchangers

In this investigation, two operating parameters that a effect the air cooled condenser, air velocity and ambient temperature are studied. The velocity of air flow over the condenser ranges between 0.5 m/s and 3 m/s and the ambient temperatures are (20,28,and35)°C. It is found that condensers employing tubes of smaller diameters yield the best system performance. The effect of pressure drop for refrigerant side and air side on performance unit has been studied also.

It is concluded that when the air velocity riss the condensing temperature is reduced and the work of compressor is reduca in the cycle, then the refrigerating effect rises and the coefficient of performance increases also. And when the ambient temperature rises the condensing temperature rises too and the refrigerating effect is reduce as result of the coefficient of performance decrease. Also when the air velocity over condenser rises the number of heat transfer units rises too therefore the effectiveness of condenser rises.

المستخلص

المبادلات الحرارية من نوع الانبوب والزعنفة كثيرة الاستعمال في منظومات المكيف، بالإضـــافة إلـــى تطبيقات أخرى تَتطلّب استخدام مبادل حراري بين السوائل والغازات. إنّ الفرصةَ الهامّة الوحيدة لتخفـــيض استعمال الطاقة الكهربائية منْ مكيّفات الهواء المنزلية هو تحـــسينِ تقنيـــةِ المبـــادل الحـــراري للأنبـــوب المزعنف.

في هذا البحث تم دراسة حدين من الحدود المؤثرة على المكثف وهو سرعة تـــدفق الهـــواء ودرجـــة حرارة الجو. عند دراسة سرعة تدفق الهواء تم اختيار عدة سرع تبدأ من ٠,٥ م/ثا إلى ٣م/ثا ولـــدرجات حرارة جو مختلفة هي (٢٠، ٢٨، ٣٥) درجة مئوية. وقد وجد إن المكثفات التي تستخدم أنابيب بأقطـــار الصغيرة تعطي أداء جيد للمنظومة.و قد تمت دراسة تأثير انحدار الضغط لجهة مائع التثليج و الهواء.

نستنتج من هذا البحث عند زيادة سرعة الهواء تقل درجة حرارة التكثيف يقلل شغل الضاغط عندئـــذ يزداد التأثير ألتثليجي و زيادة معامل أداء المنظومة أيضا.وكذلك عند زيادة درجة حـــرارة المحـــيط تـــزداد

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درجة حرارة التكثيف أيضا مما يقلل التأثير ألتبريدي ونتيجة لذلك يقل معامل أداء المنظومة. وكـــذلك عنـــد زيادة سرعة الهواء المار على المكثف فان عدد مرات انتقال الحرارة (NTU) وكـــذلك الفعاليـــة للمكثـــف سوف تزداد.

NOMNECLATURE

A	Area	m^2
A_{fr}	Frontal area of condenser (Width x Height)	m^2
A_{fin}	Surface area of the fins	m^2
A_o	Total heat transfer area on the air side (A_{fin}+A_t)	m^2
A_s	Surface area	m^2
A_t	Surface area of the tubes	m^2
C	Heat capacity	kJ/s. °C
C_r	Ratio of heat capacities (C_{min}/C_{max})	-
C_p	Specific heat at constant pressure	kJ/kg. °C
COP	Coefficient of Performance	-
D	Diameter	m
D_h	Hydraulic Diameter ($4A_{min}D_{depc}/Ao$)	m
Di	Inside tube diameter	m
D_{depc}	Depth of the condenser in the direction of air flow (rows x Xl)	m
Eu	Euler number	-
F	Darcy friction factor	-
F	Correction factor	
G	Mass velocity	$kg.m^2/s$
G_{max}	Mass velocity through minimum air flow area	$kg.m^2/s$
h_a	air-side convective heat transfer coefficient	W/m^2. °C
J	Colburn j-factor	-
J_4	j-factor for 4 row coil	-
J_z	j-factor for fewer than 4 rows (z = number of rows)	-
J_p	McQuiston & Parker parameter	-
K	Thermal conductivity	w/m. °C
L	Length of tube	m
LMTD	Log mean temperature difference	°C
M	Mass flow rate	kg/s
NTU	Number of transfer units	-
Nu	Nusselt number	-
P	Pressure	Pa
ΔP	Pressure drop	Pa
ΔP_{fin}	Pressure drop in fin side	Pa
ΔP_{tot}	Total Pressure drop	Pa
ΔP_{tub}	Pressure drop in side tube	Pa
Pr	Prantl Number (ν/α)	-
Q	Total heat transfer	W
q_{cst}, r_{cst}, s_{cst}, t_{cst}, u_{cst}	Rich power series parameters	-
R	Radius	m
R	Radius of a circular fin	m
Re	Reynolds number	-
Rel	Reynolds number based on longitudinal tube spacing, Xl	-
St	Stanton Number (Nu/RePr)	-
T	Temperature	°C
T_{hi} ,T_{ho}	Inlet and outlet temp. for hot fluid	°C
T_{ci} , T_{co}	Inlet and outlet temp. for cold side	°C
UA	Overall heat transfer coefficient * heat transfer area	W/° C
ε	Heat exchanger effectiveness	-

μ	Dynamic viscosity	Kg/m.s
P	Density	Kg/m^3
Σ	Minimum free flow area / frontal area	-

Introuduction

The condenser in a refrigeration system is a heat exchanger that generally rejects all the heat from the system. This heat consists of heat absorbed by the evaporator plus the heat from the energy input to the compressor. The compressor discharges hot, and high-pressure refrigerant gas into the condenser, which rejects heat from the gas to some cooler medium. Thus, the cool refrigerant condenses back to the liquid state and pumped from the condenser to continue in the refrigeration cycle.

Different experimental arrangements were used [1,2], to study the combustibility of R-134a at high temperature and pressure. It was noted that the results depend on the nature of the equipment used in the tests. They showed that R-134a is combustible at pressures above 100 kPa (gage) at room temperature and air concentrations are greater than 80% by volume. At 177°C, combustibility was observed at pressures above 36 kPa (gage) and air concentrations above 60% by volume. Both researchers found R-134a to be nonflammable at ambient conditions and under the likely operating conditions of air-conditioning and refrigeration equipment. Blends of R-22/152a/114 showed combustion at temperatures at 82°C at atmospheric pressure and above, with air concentrations above 80% by volume [1].

The major benefit of R-134a introduction, of course, is elimination of the ozone depletion concern. In addition, it offers an unanticipated and adventitious benefit relating to global warming impact of the automotive air conditioning system. R-12 is an intrinsically powerful greenhouse gas whose emission from the automotive air conditioning system results in the total equivalent warming impact (TEWI) that is about 50% of the TEWI of the tailpipe emissions of the entire vehicle. Introduction of R-134a sharply reduces

the TEWI of the automotive air conditioning system. It turns out that the TEWI of the present R-134a automotive air conditioning systems is only a small fraction of the TEWI of the tailpipe greenhouse gases put out by the motor vehicles. Furthermore, the TEWI of the automotive air conditioning systems becomes negligible when viewed in the context of the overall TEWI of the greenhouse gases due to human activity including fossil fuel combustion, agriculture and deforestation. A recent investigation provides a comprehensive look at the entire issue of global warming with particular emphasis on the contribution of automotive air conditioning systems[2].

An experimental study[3] was conducted on the air-side performance for two specific louver fin patterns and their plain plate fin counterparts. This study investigated the effects of fin pitch, longitudinal tube spacing and tube diameter on the air-side heat transfer performance and friction characteristics.

An experimental investigation[4] was conducted to the heat transfer and friction characteristics of plate fin-and tube heat exchangers having (7mm) diameter tubes. In this study, 8 samples of commercially available plate-fin-and-tube heat exchangers were tested. It was found that the effect of varying fin pitch on the air-side heat transfer performance and friction characteristics was negligible for 4-row coils. However for 2- row coils, the heat transfer performance increased with a decrease in fin pitch.

An experimental data[5] was collected on a plate-fin-and tube heat exchanger configuration. They examined the effect of the number of tube rows, fin pitch, tube spacing, and tube diameter on heat transfer and friction characteristics. This study found that the effect of fin pitch on the air-side friction pressure drop was negligibly 7 times smallesr than that for air-side Reynolds numbers greater than 1000. It was also found that the heat transfer performance was independent of fin pitch for 4-row configurations.

One of the earliest and most complete investigations of heat exchanger heat transfer and pressure drop characteristics was performed[6] experimentally. An extensive amount of experimental heat transfer and friction pressure drop data was complied for several different plate-fin-and-tube heat exchanger configurations as part of this study. However, no optimization of the heat transfer surfaces and geometry was performed.

The effect of various geometric variations on 1-row plate fin-and-tube coils was tested experimentally[7] . The investigation concludes the effects of varying the fin spacing, fin depth, tube spacing, and tube location on the heat transfer performance of the coil. The results of Shepherd's study showed that as the fin pitch increased, the air-side heat transfer coefficient, for a given face velocity, increased only slightly. He also found that as the fin depth and tube spacing increased, with all other variables constant, the air-side heat transfer coefficient decreased.

Some others[8] studied the effect of varying the fin spacing on the heat transfer and friction performance of multi-row heat exchanger coils. Rich found that over the range from 3 to 14 fins per inch, the air-side heat transfer coefficient was independent of fin pitch. Neither Rich's nor Shepherd's investigations involved the optimization of the heat exchanger operating conditions and geometric parameters.

All of the above studies provide valuable insight into the effects of varying different geometric parameters on the heat transfer and friction performance of plate-fin-tube heat exchangers. However none of the above works has investigated the effect of varying these geometric parameters has on the optimization of a complete air-conditioning system.

Expermental Work

General refrigeration unit is operating with refrigerant 134a. The condenser is in the high side, where the heat is released from the refrigerant. The compressor is of a hermetic type 1/5 hp. The refrigerant liquid enters the evaporator from the expansion valve type Tn_2 (-40/ +10C) PB28bar (Danfass).

The forced circulation evaporator is a compact arrangement of refrigerant- cold tube and fins. A fan driven by electric motor, blows air over it.

The condenser is with a heat–exchanging surface of the sum of its fins surface minus the holes of the tubes, plus the surface of the same tubes. Fig.1 shows the condenser.

The specifications of the condenser under consecration are:

Number of tubes (n) = 40, Length of tubes (l) = 0.29m, Length of the external elbow = 0.045m, Number of elbows = 38, Diameter of tubes (D) = 0.01m, Number of fins (N) = 66, Dimensions of fins = 0.275 * 0.085 = 0.024m^2, The overall heat – transfer surface is equal to (3.378m^2)

The measuremens that have bean done as follows:

The temperature of refrigerant was measured at the inlet, outlet of the condenser, and the temperature of refrigerant at inlet of the evaporator and the temperature of air after the condenser. The air velocity after the condenser was measured by anemometer. Pressure at the suction and discharge of compressor was measured by pressure gages.

Fig.1. The condenser of the unit under

Theory

As known, the condenser is a heat exchanger. The heat rejected from the refrigerant to the cold fluid (the air) is dependent on the heat exchanger effectiveness and the heat capacity of each fluid [9 & 10]:

$$Q = \varepsilon C_{\min}(T_{hi} - T_{ci}) \quad (1)$$

where ε is the heat exchanger effectiveness; C_{min} is the smaller heat capacity for the hot or cold fluids; T_{hi} is the inlet temperature of the hot fluid; and T_{ci} is the inlet temperature of the cold fluid. The heat capacity C, is expressed as:$C = m\ C_p$: where m is the mass flow rate of fluid and C_p is the specific heat of the fluid.

The actual heat transfer may be computed by calculating either the energy lost by the hot fluid or the energy gained by the cold fluid. For the cross flow {9 & 10]:

$$Q_{act} = m_h C_h \left(T_{hi} - T_{ho}\right) = m_c C_c \left(T_{co} - T_{ci}\right) \tag{2}$$

$$Q_{\max} = \left(mC\right)_{\min} \left(T_{hi} - T_{ci}\right) \tag{3}$$

$$\varepsilon = \frac{Q_{act}}{Q_{\max}} \tag{4}$$

The minimum fluid may be either the hot or cold fluid, depending on the mass-flow rate and specific heats.

For the cross flow heat exchanger the effectiveness is as[9]:

$$\varepsilon = \frac{1 - \exp\left[\left(1 - \frac{UA}{C_{\min}}\right)\left(1 - \frac{C_{\min}}{C_{\max}}\right)\right]}{1 - \left(\frac{C_{\min}}{C_{\max}}\right)\exp\left[\left(-\frac{UA}{C_{\min}}\right)\left(1 - \frac{C_{\min}}{C_{\max}}\right)\right]} \tag{5}$$

It is now possible to develop expressions which relate the heat exchanger effectiveness to another parameter referred to as the number of transfer units (NTU). The value of NTU is defined as [9]:

$$NTU = \frac{UA}{C_{\min}}\ ,\quad Cr = \frac{C_{\min}}{C_{\max}} \tag{6}$$

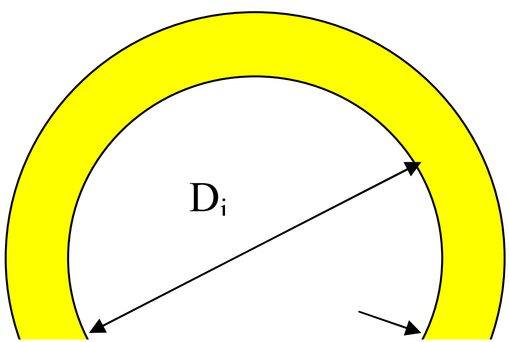
It is now a simple matter to solve a heat exchanger problem where

$$\varepsilon = f\left(NTU, Cr\right) \tag{7}$$

Numerous expressions have been obtained which relate the heat exchanger effectiveness to the number of transfer units. The handout summarizes a number of these solutions and the special cases which may be derived from them.

A heat exchanger analysis always begins with the determination of the overall heat transfer coefficient. The overall heat transfer coefficient may be defined in terms of individual thermal resistances of the system. Combining each of these resistances in series gives[10]:

$$\frac{1}{UA} = \frac{1}{(hA)_i} + \frac{X}{k} + \frac{1}{(hA)_o} \tag{8}$$



The log mean temperature difference (LMTD) is derived for a parallel flow or counter flow arrangement. The LMTD has the form[9]:

$$LMTD = \frac{\Delta T_2 - \Delta T_1}{\ln \frac{\Delta T_2}{\Delta T_1}} \qquad (9)$$

where ΔT_1 and ΔT_2 represent the temperature difference at each end of the heat exchanger. The LMTD expression assumes that the overall heat transfer coefficient is constant along the entire flow length of the heat exchanger. If it is not, then an incremental analysis of the heat exchanger is required. The LMTD method is also applicable to cross flow arrangements when used with the cross flow correction factor (F) whicl is obtained to correct the value of LMTD. The heat transfer rate for a cross flow heat exchanger may be written as:

$$Q = UAF\ LMTD \qquad (10)$$

For a constant surface heat flux for single phase laminar flow when buoyancy force has no effect , the Nusselt number can be approximated by the following expression[9].

$$Nu_d = 4.364 \qquad (11)$$

In the turbulent region, however, there are a number of expressions available for the Nusselt number. One of the more commonly used correlations for turbulent flow is the Dittus-Boelter equation. This correlation is valid for fully developed flow in circular tubes with moderate temperature variations (Incropera & DeWitt, 1996)[10]. For refrigerant cooling in a condenser, the Dittus-Boelter equation is expressed as:

$$Nu_d = 0.023\ Re^{0.8}\ Pr^{0.4} \qquad (12)$$

This mathematical relation has been confirmed by experimental data for the following conditions: $0.7 \leq Pr \lesssim 160$, $Re \gtrsim 10{,}000$, $L/D \gtrsim 10$

The most important aspect of the condenser model is the equations used to describe the heat transfer coefficients and pressure drops on both the air-side and refrigerant-side of the heat exchanger. On the refrigerant side, the heat transfer coefficient in the single phase regions (sub-cooled and superheated) region is calculated:

$$St = \frac{Nu}{\mathrm{Re}\,\mathrm{Pr}} = \frac{h}{GCp} \tag{13}$$

where St : Stanton number and G : the total mass velocity

The pressure drop on the refrigerant side in the single-phase regimes is calculated by the standard equation for pressure drop in circular pipe flow[9,10]:

$$\Delta P = \frac{fG^2 L}{2\rho D} \tag{14}$$

By using a Darcy friction factor for fully developed laminar flow:

$$f = \frac{64}{\mathrm{Re}} \tag{15}$$

where the Reynolds number is based on the inside tube diameter (D_i)

The method for calculating the pressure drop inside the tube bends is the same as that used by Wright, using the work of Chisholm (1983). For single-phase flow, the pressure drop in tube bends is calculated simply by assigning an equivalent length to each bend based on the flow diameter and the bend radius.

The first step in computing the pressure drop in a tube bend is to determine the equivalent length of the bend. The equivalent tube length, *y*, is a function of the relative radius, R_r: where $R_r = \frac{R_b}{D_i}$ and R_b is the tube centerline radius of the bend.

The single-phase pressure drop in a bend was approximated[11] by simply substituting the equivalent length of the bend, *y*, for the straight pipe length in the standard pressure drop equation:

$$\Delta P_{b,sp} = \frac{fG^2}{2\rho}\left[\frac{Y}{D_i}\right] \tag{16}$$

where .$\Delta P_{b,sp}$; the single phase pressure drop in the bend.

While the velocity distribution of the air over the coil is assumed to be uniform, the complex airflow pattern across the fin-and-tube surfaces makes the theoretical predictions of the heat-transfer coefficient and friction factor from first principles very difficult; therefore such relations are usually empirical. However, these empirical correlation are limited by the range of data used in their development, and these ranges are not always published along with the correlation. Sometimes is required to obtain the experimental data from other publications and examining the ranges used in the correlation development.

Many of the early studies that provide heat transfer and friction correlation for the air side of finned-tube heat exchangers were based on larger tube diameters and tube spacing than those more commonly used today. Heat exchanger manufacturers found that the use of smaller heat transfer tubes, smaller transverse tube pitch, and smaller longitudinal tube pitch can effectively reduce the air side resistance as well as saving resources and can lead to a much more compact fin-and-tube heat exchanger design. Benefits of using smaller diameter tubes include smaller form drag caused by the tube, higher refrigerant side heat transfer coefficients due to smaller hydraulic diameter, and less refrigerant inventory in the system.

The important test results [12] are used to evaluate the air side convective heat transfer coefficient for a plate finned-tube heat exchanger with multiple rows of staggered tubes. The model is developed for dry coils. The heat transfer coefficient is based on the Colburn j-factor, which is defined as:

$$J = St\ Pr^{2/3}$$

(17)

Substituting the appropriate values for the Stanton number, *St*, gives the following relationship for the air-side convective heat transfer coefficient, *ha*:

$$h_a = \frac{JCpG_{\max}}{\Pr^{3/2}}$$

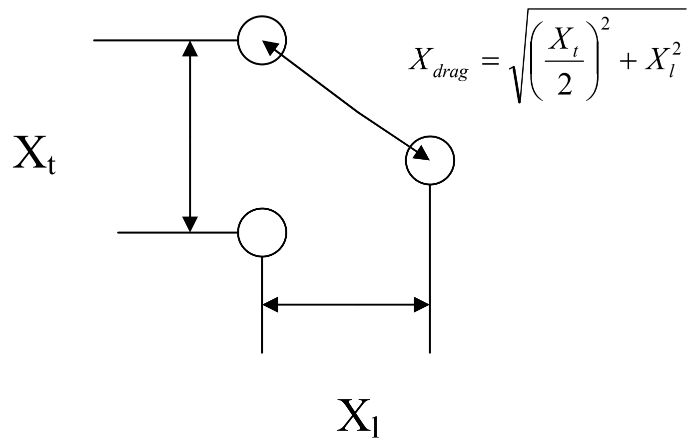
(18)

where C_p ; the specific heat, and G_{max} :-the mass velocity of air through the minimum flow area which is expressed as:

$$G_{\max} = \frac{m_{air}}{A_{\min}}$$

(19)

The minimum free flow area, A_{min}, is the passage height (fin spacing – fin thickness) multiplied by the minimum of the distances *Xt* or 2*X_{diag} as shown in the diagram below



In this test a 4-row finned tube heat exchanger is used as the baseline model, and the Colburn j-factor for a 4-row finned-tube heat exchanger is defined as:

$$J_4 = 0.2675 J_p + 1.325 \times 10^{-6} \qquad (20)$$

and the parameter j_p is defined as:

$$j_p = \mathrm{Re}^{-0.4} \left[\frac{A_O}{A_t} \right]^{-0.15} \qquad (21)$$

where At ; the tube outside surface area, and Ao ; the total air side heat transfer surface area (fin area plus tube area). The Reynolds number, Re in the above expression is based on the tube outside diameter, Do, and the maximum mass velocity, G_{max}. The area ratio can be expressed as:

$$\frac{A_o}{A_{tube}} = \frac{4}{\pi}\frac{X_L}{D_h}\frac{X_t}{D_{depc}}\sigma \qquad (22)$$

where X_l the tube spacing parallel to the air flow (longitudinal), X_t :- the tube spacing normal to the air flow (transverse), D_{depc} : the depth of the condenser in the direction of the air flow, D_h : the hydraulic diameter defined as:

$$D_h = \frac{4A_{\min}D_{depc}}{A_o} \qquad (23)$$

σ is the ratio of the minimum free-flow area to the frontal area:

$$\sigma = \frac{A_{\min}}{A_{fin}} \qquad (24)$$

The J-factor for heat exchangers with four or fewer rows can then be found using the following correlation:

$$\frac{J_z}{J_4} = \frac{1-1280Z\,\mathrm{Re}_L^{-1.2}}{1-(1280)(4)\mathrm{Re}_L^{-1.2}} \qquad (25)$$

where z : the number of rows of tubes, and Re_l : the Reynolds number based on the longitudinal tube spacing:

$$\mathrm{Re}_L = \frac{G_{\max}X_L}{\mu_{air}} \qquad (26)$$

According to Rich[8] , the air-side pressure drop can be divided into two components, the pressure drop due to the tubes, $.p_{tubes}$, and the pressure drop due to the fins, $.p_{fin}$. The work of Rich is used to evaluate the air-side pressure drop due to the fins, which is expressed as: $\Delta P_{fin} = f_{fin}V_{m,air}\frac{G_{\max}^2}{2}\frac{A_{fin}}{A_{\min}}$

(27)

where, f_{fin} is the fin friction factor, $V_{m,air}$ is the mean air specific volume, and A_{fin} is the fin surface area. In experimental tests, Rich found that the friction factor is dependent on the Reynolds number, but it is independent of the fin spacing for fin spacing between 3 and 14 fins per inch. In this range of fin spacing, Rich expresses the fin friction factor as:

$$f_{fin} = 1.7\,\mathrm{Re}^{-0.5} \tag{28}$$

To determine the pressure drop over the banks tubes, the relationships developed by [13] are used .

$$\Delta P_{tube} = Eu \frac{G_{\max}^2}{2\rho_{air}} Z \tag{29}$$

where z is again the number of rows, and Eu is the Euler number. Rich expresses the Euler number as a function of the Reynolds number and the tube geometry. For staggered, equilateral triangle tube banks with several rows, Rich expresses the Euler number by a fourth order inverse power series by the following:

$$Eu = q_{cst} + \frac{r_{cst}}{\mathrm{Re}} + \frac{S_{cst}}{\mathrm{Re}^2} + \frac{t_{cst}}{\mathrm{Re}^3} + \frac{u_{cst}}{\mathrm{Re}^4}$$

where Re is the Reynolds number based on the outer tube diameter. The coefficients q_{cst}, r_{cst}, s_{cst}, t_{cst}, and u_{cst} are dependent on the Reynolds number and the parameter "a"(Table(1)), which is defined as the ratio of the transverse tube spacing to the tube diameter (X_t/D_o). The coefficients for a range of Reynolds numbers and spacing to diameter ratios have been determined from experimental data [13] and are expressed in Table (1):

The total pressure drop over the heat exchanger is then simply the sum of the pressure drop over the tubes and the pressure drop over the fin

$$\Delta P_{tot} = \Delta P_{tube} + \Delta P_{fin} \tag{31}$$

Result and Disscution

The experimental results for two cases were studied here. The first case is the study of the effect of air velocity on the condenser performance. The second case is the study of the effect of ambient temperature on the condenser performance. The results for different speed of condenser fan were recorded at ambient temperature 20°C.

Table(1): Coefficients for the Euler Number Inverse Power Series

A	Reynolds Number	q_{cst}	r_{cst}	S_{cst}	t_{cst}	u_{cst}
1.25	$3<Re_{Do}<10^3$	0.795	0.247E3	0.335E3	-0.155E4	0.241E4
	$10^3<Re_{Do}<2x10^6$	0.245	0.339E4	-0.984E7	0.132E11	-0.559E13
	$3<Re_{Do}<10^3$	0.683	0.111E3	-0.973E2	0.426E3	-0.574E3

1.5	$10^3 < Re_{Do} < 2x10^6$	0.203	0.248E4	-0.758E7	0.104E11	-0.482E13
	$7 < Re_{Do} < 10^2$	0.713	0.448E2	-0.126E3	-0.585E7	0.000
2.0	$10^2 < Re_{Do} < 10^4$	0.343	0.303E3	-0.717E5	0.880E7	-0.380E9
	$10^4 < Re_{Do} < 2x10^6$	0.162	0.181E4	-0.792E8	-0.165E13	0.872E16
	$10^2 < Re_{Do} < 5x10^3$	0.330	0.989E2	-0.148E5	0.192E7	0.862E2
2.5	$5x10^3 < Re_{Do} < 2x10^6$	0.119	0.848E4	-0.507E8	0.251E12	-0.463E15

Fig. 2, 3, 4, and 5 show the effect of the air velocity on LMTD , effectiveness, overall heat transfer coefficient, and Rdynolds number of air respectively at 20°C ambient temperature and from these it note is that at air velocity (3m/s), gets the best effectiveness.

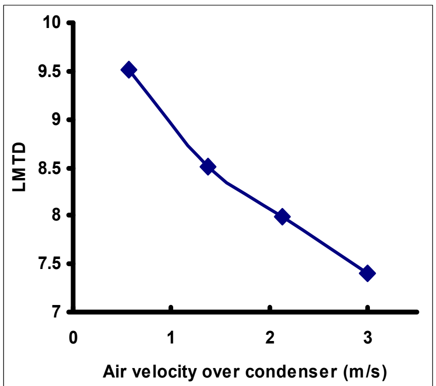


Fig.2: Effect of air velocity on LMTD at ambient Temperature = 20C

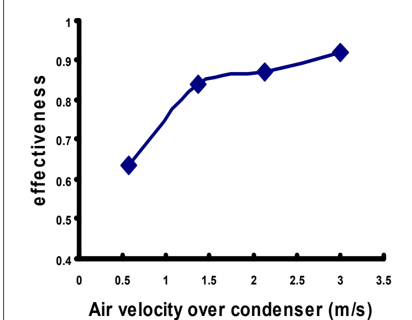


Figure 3: Effect of air velocity on effectiveness at ambient Temperature = 20C

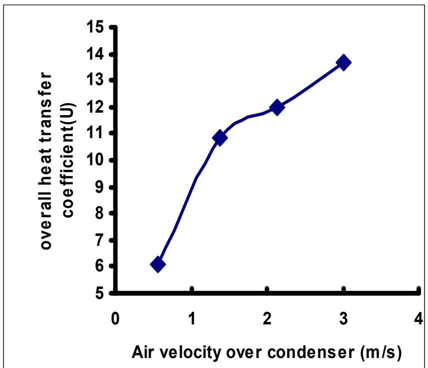


Fig. 4: Effect of air velocity on overall heat transfer coefficient at ambient Temperature = 20C

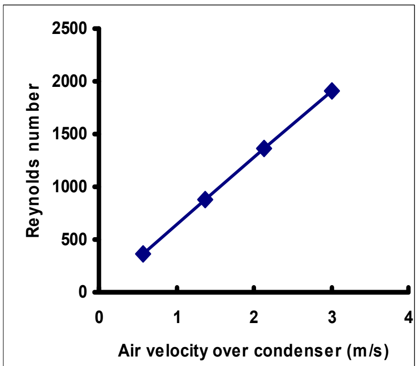


Fig.5 : Effect of air velocity on Reynolds number of air at ambient Temperature = 20C

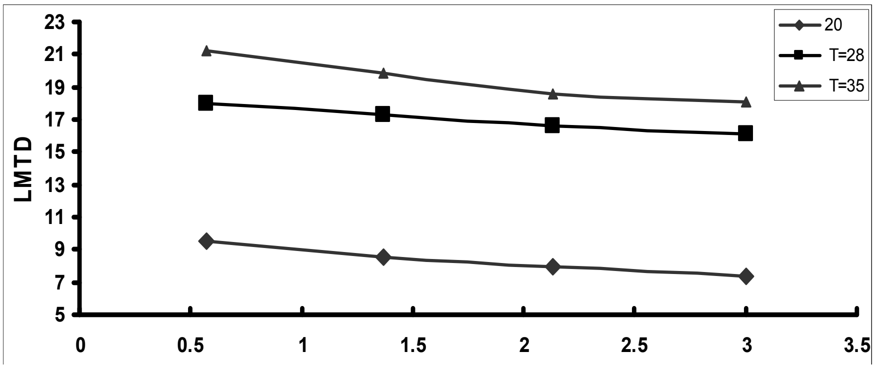


Fig. 6 shows the effect of air velocity on LMTD for various ambient temperatures. It is noted that LMTD decreases with air velocity. The ambient temperature affects the LMTD inversely. Fig. 7 shows the effect of air velocity on the effectiveness of the condenser. The effectiveness is increased with air velocity increasing.

Fig. 8 and Fig. 9 show the effect of air velocity on the compressor power and COP for various ambient temperature. The condenser fan power increases proportionally with the cubic power of the velocity. The COP is insensitive to changes in the velocity, however, in this range of velocities, as the condenser fan power requirement is increasing, the required compressor power is decreasing by approximately the same amount of power due to increasing in heat transfer energy from the condenser to the ambient air, so the condenser temperature and pressure decrease and the that effect of the compressor power to decreases also . As the air velocity over the condenser increases, the condensing temperature decreases, and the inlet enthalpy to the evaporator also decreases. This causes a reduction in the mass flow rate of refrigerant required to maintain the evaporator cooling capacity. Hence, the amount of compressor work is decreased, the condensing temperature of the refrigerant can never be lower than the inlet air temperature. Thus, there is a minimum power requirement for the compressor. As the air velocity increases the power required for the condenser fan begins to grow rapidly. At this point, the decrease in the compressor power requirement will not compensate for this increase in the condenser fan requirement.

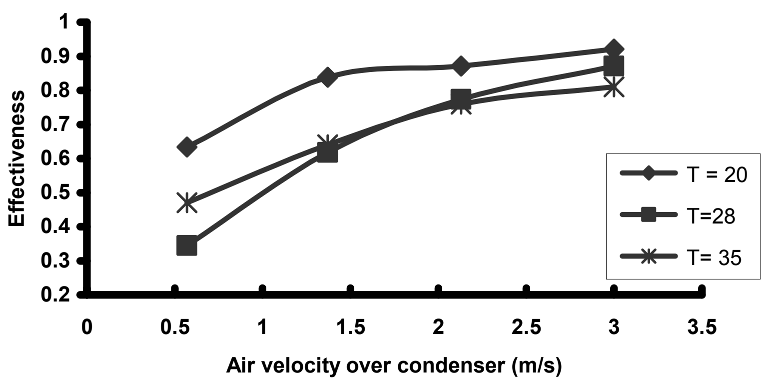


Fig.7 : Effect of air velocity on Effectiveness for various ambient Temperature

As the ambient temperature decreases, the saturation pressure (high pressure) in the condenser also decreases. Therefore, the compressor power consumption is also decreased. Thus, the enthalpy of the refrigerant entering the evaporator is reduced. The decrease in the enthalpy of the refrigerant entering the evaporator that is produced by the decrease in the ambient temperature causes the evaporator cooling capacity to increase and the enthalpy of the refrigerant entering the evaporator to decrease which also causes a reduction in the mass flow rate of refrigerant required to maintain the evaporator cooling capacity. Hence, the amount of compressor work is decreased. Therefore, the ultimate result of decreasing the ambient temperature is an increase in the COP of the system.

As a result, the inlet quality is also lower and more of the refrigerant in the evaporator exists in the liquid state. The total mass of refrigerant in the entire system is constant. Hence, as the ambient temperature decreases, the mass of refrigerant in the evaporator increases and the mass of refrigerant in the condenser decreases. When the mass of refrigerant in the condenser decreases, the volume of the condenser that contains vapor refrigerant increases and the volume of liquid refrigerant in the condenser decreases, causing an overall decrease in the mass of the refrigerant in the system (refrigerant charge). Thus it is possible for the mass of refrigerant in the condenser to drop to very low levels such that complete condensation does not occur. In these cases the refrigerant enters the expansion valve, while the valve goes to its wide open position, and a fixed superheat cannot be maintained. This reduces the COP of the system. As a result, more sub-cool at 35°C is needed to maintain some sub-cool at the lower ambient temperatures (i.e. as the ambient temperature decreases, the degrees of sub-cool also decrease).

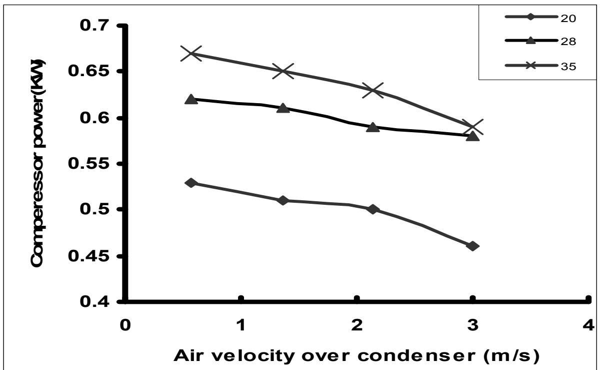


Fig .8: Effect of air velocity on Comperessor power for various ambient Temperature

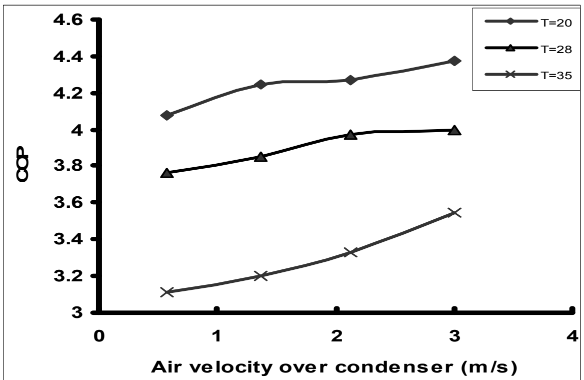


Fig.9: Effect of air velocity on COP for various ambient Teperature

Fig. 10 and Fig. 11 show the effect of air velocity on Reynolds number of refrigerant and pressure drop of refrigerant for various ambient temperature. It notes that when the ambient temperature is rise the Reynolds number and pressure drop rise too.

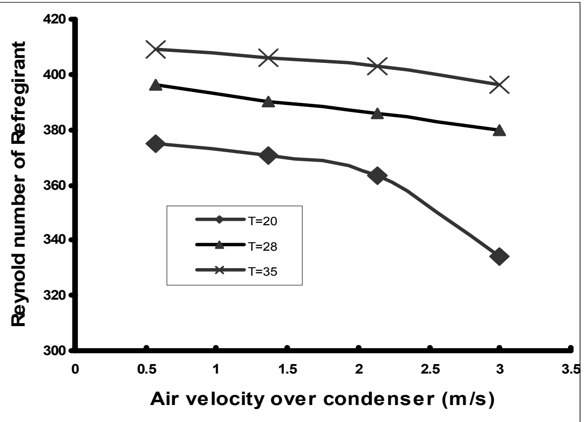


Fig.10: Effect of air velocity on Reynold number of Refregirant for various ambient Temperature

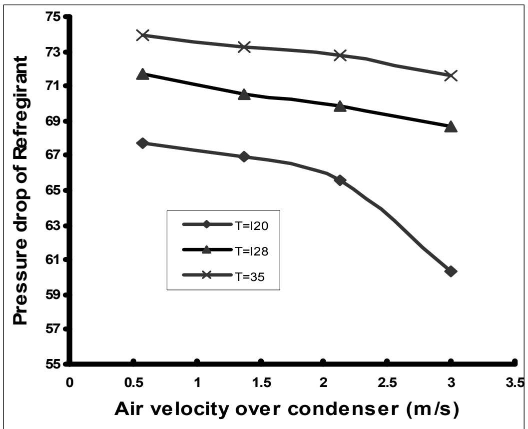


Fig.11: Effect of air velocity on Pressure drop of Refregirant for various Tambient Temperature

Fig. 12 shows the effect of air velocity on ratio of pressure cedrop at different ambient temperatures. It indicas that the air velocity causes the same different in the ratio of pressure drop at different ambient temperatures.

Fig. 13 and Fig. 14 show the relation between the mass flow rate ratio (m_r/m_a) and the condenser effectiveness and COP. They indicate that when the ratio increase the effectiveness and COP decrease. It means that when the mass of refrigerant flow in system increass the pressure and temperature of condenser also increase. These will increase the refrigerating effect so the effectiveness and COP decrease.

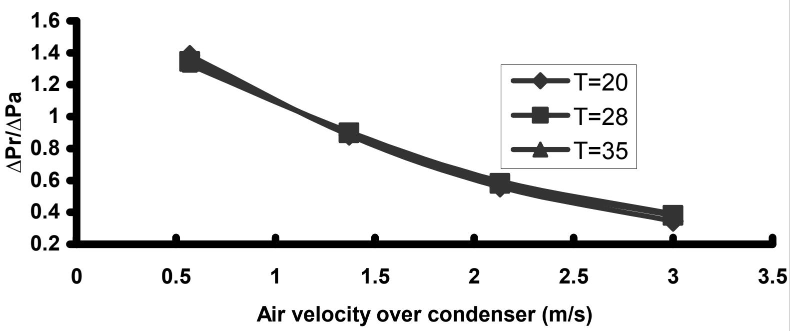


Fig.12: Effect of air velocity on DPr/DPa for various Tambient Temperature

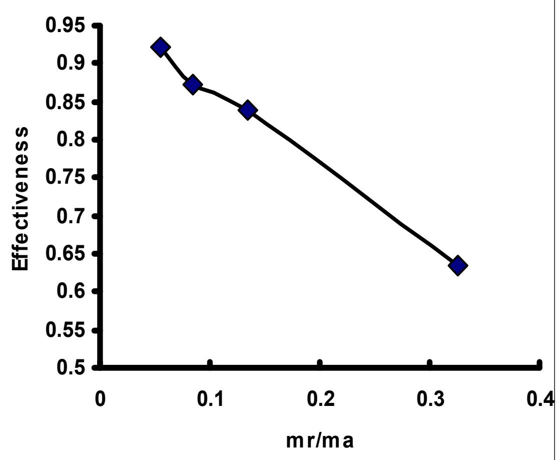


Fig.13: Effect of mr/ma on Effectiveness at ambient Temperature =

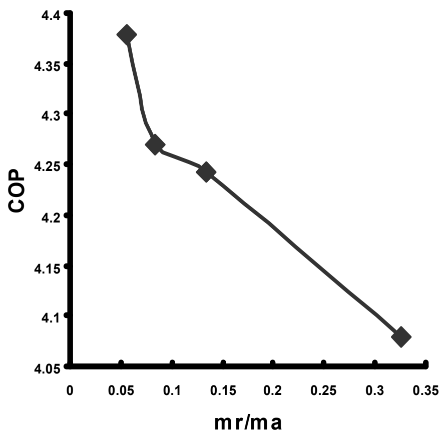


Fig.14: Effect of mr/ma on COP at ambient Temperature = 20C

Fig.15 shows the relation between the mass flow rate ratio (m_r/m_a) and the ratio of pressure drop. It indicates that when the ratio of mass flow rate increases the pressure drop ratio increases also. It means that when the mass of refrigerant flow is increased the pressure drop refrigerant increases too (shown in Fig. 16) then the ratio of pressures drop increase when the ratio of mass flow rate increases.

Fig. 17 shows the relation between the ratio of heat capacities and the number of heat transfer unit. It indicates that when the ratio of heat capacities increases the number of heat transfer unit decreases. It means that the effectiveness decreases (as showns in Fig. 18 and Fig. 19).

Fig.20 shows the effect of compressor power on condenser effectiveness and it is notedwhen compressor power is reduced the condenser effectiveness increases.

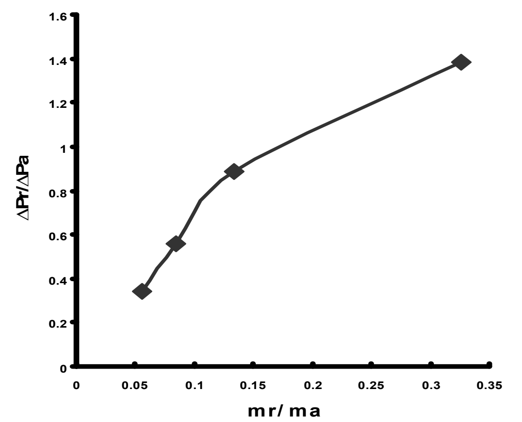


Fig. 15: Effect of mr/ ma on DPr/DPa at ambient Temperature = 20C

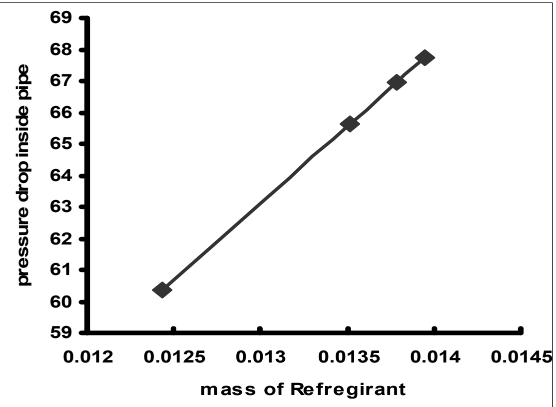


Fig16: Effect of mass Refregirant on pressure drop total inside pipe at ambient Temperature = 20C

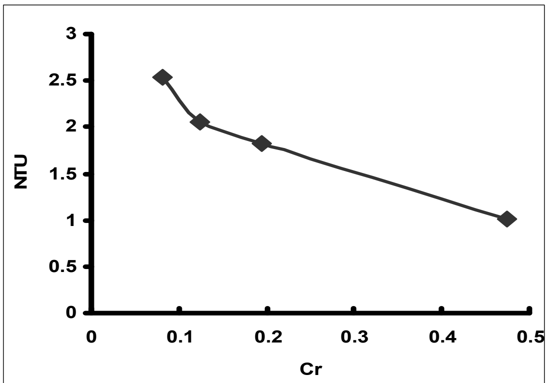


Fig.17: Effect Cr on NTU at ambient Temperature = 20C

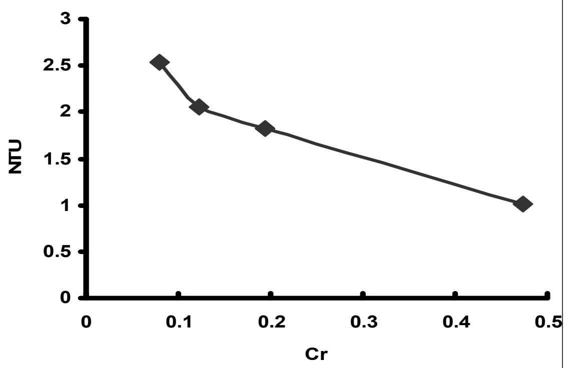


Fig.18: Effect Cr on NTU at ambient Temperature = 20C

Conclusions

From the discussion it is concluded that:

When the velocity air flow on the condenser is increased the condensing temperature, the work of compressor in the cycle and mass flow rate of refrigerant are decreased at constant pressure during evaporator operation. The coefficient of performance is increased due to the increase in the refrigerating effect.

When the ambient temperature is raised the condensing temperature and pressure are increased. Therefore the work of compressor is increased. The mass flow rate of refrigerant increaseds. The refrigerating effect and the coefficient of performance is decreased.

When the air velocity over condenser is raised the heat transfer coefficient is raised too therefore the effectiveness of condenser raise.

In general the air velocity on condenser, condenser temperature, and refrigerant mass flow rate affect the performance of the cycle.

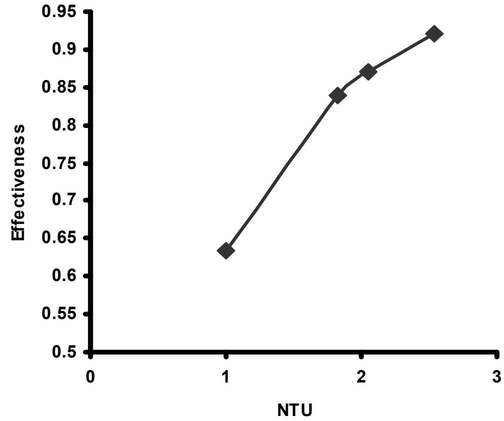


Fig.19: effect NTU on effectiveness at ambinet Temperature =20C

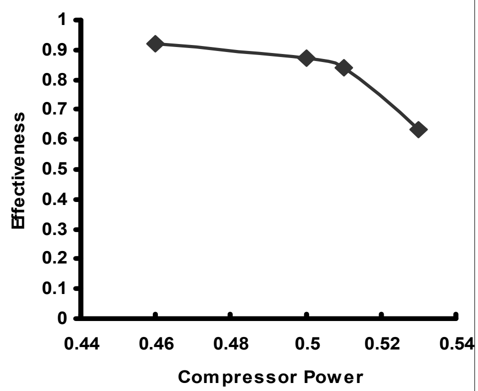


Fig.20: Effect of Compressor Power on Effectiveness at ambient Temperature = 20C

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