Thermal analysis of air-cooled condensers of water chillers

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The objective of this study is to present a simulation model of air-cooled condensers of reciprocating vapor-compression refrigeration chillers using R-134a as a refrigerant at steady-state conditions. The study is primarily concerned with optimum design of air-cooled condensers. The study presents a generalized thermal design at different cooling loads for a finned-tube air-cooled condenser, which is based on mathematical equations for heat transfer and pressure drop. The condenser design was accomplished numerically and gives satisfactory results compared with commercial catalog data. The numerical results are presented in graphical charts giving the optimum size corresponding to cooling load at minimum cost. The present study shows the effect of ambient air temperature, condensing temperature, and number of rows in the condenser on the condensing unit annual cost. Also, the effects of condenser aspect ratio and fans speed on the pressure of refrigerant and annual cost are included.

الغرض من الدراسة هو تقديم نموذج يمثل مكثف مبرد بالهواء يعمل فى مبردات دورة التبريد بانضغاط البخار التى تستخدم مائع التبريد ١٣٤- أ أثناء حالة العمل المستقر وذلك بهدف تقديم التصميم الأمثل فى هذه الحالة. وتقدم الدراسة تصميم حرارى عام لمكثف ذو تبريد هوائى ومواسير ذات زعانف عند أحمال تبريد مختلفة باستخدام المعادلات الرياضية لانتقال الحرارة و انخفاض الضغط . وقد أعطى تصميم المكثف باستخدام الحلول العددية نتائج مرضية بالمقارنة مع البيانات المسجلة فى الكتالوجات التجارية. وقد تم عرض النتائج العددية على هيئة خرائط تعطى التصميم الأفضل المناظر لحمل التبريد مع أقل التكاليف. والدراسة تعطى تأثير كل من درجة حرارة الهواء الخارجى ، درجة التكثيف وعدد صفوف المواسير فى المكثف على التكاليف السنوية لوحدة التكثيف. أيضا تم دراسة تأثير كل من أبعاد المكثف وسرعة المراوح على ضغط مائع التبريد والتكاليف السنوية.

Keywords: Air – cooled condensers, Water chillers, Condenser design

1. Introduction

Due to the continued rise in atmospheric air temperature all around earth, because of ozone depletion, air-conditioning systems are extensively used to overcome this problem and to achieve adequate comfort for human life and industrial processes. Therefore, the consumption of energy and its cost will sharply increase. Recently, there are orientations towards an efficient design of units of refrigeration and air-conditioning systems to satisfy best performance during a change of load. Liquid chillers, as shown in fig. 1, are widely used to produce chilled water for air-conditioning purposes. The chillers are classified according to the type of cooling fluid that passes through the condenser. They may be classified as air-cooled or water-cooled chillers.

Air-cooled condenser coils are usually made from copper tubes, and aluminum fins.

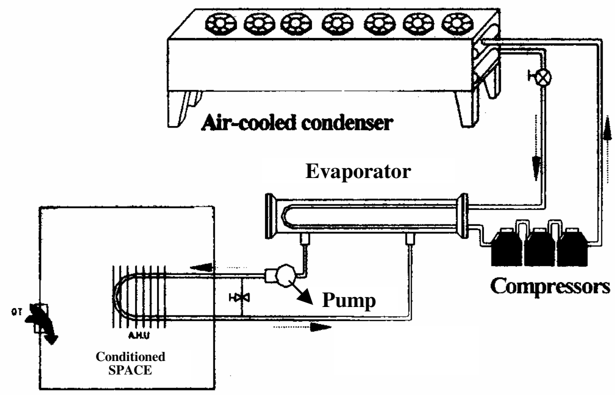

Fig. 1. Air-cooled chiller.

They are of the cross and counter flow heat exchanger type where the refrigerant flows inside tubes and an induced ambient air flows outside the tubes.

Fischer and Rice [1] proposed a model that predicts the steady-state and seasonal performance of heat pumps to optimize the design and performance. They modeled the heat exchanger using effectiveness versus number of transfer units, NTU method for cross-flow heat transfer. The model predicted heating or cooling capacity and electrical input power to the heat pump under different values of indoor and outdoor temperatures.

Armand et al. [2] developed a model of vapor compression cycle, with each component considered independently. The simulation model was based on heat transfer correlations assuming constant pressures inside both of the condenser and evaporator and energy balances for the cycle.

Damasceno, Rooke and Goldschmidt [3] compared three steady-state (air-to-air) heat pump computer models. The condenser and evaporator of the heat pump were analyzed by using the relation between effectiveness and NTU for the cross and counter-flow heat exchangers. They concluded that the prediction of capacity and coefficient of performance, COP are acceptable for all programs.

Cecchini and Marchal [4] developed a model that simulated the performance of refrigeration and air-conditioning equipment of all types: air-to-air, air-to-water, water-to-water and water-to-air. Pressure drops were neglected and constant subcooling and superheat degrees were assumed. The main aim of simulation was to calculate heating and cooling capacities and electrical power input at any operating conditions and compare it with experimental data.

Miyara et al. [5] presented simulation model analyses for a vapor-compression heat pump cycle (water-to-water) using nonazeotropic refrigerant mixtures of R-22 and R-114. Pressure loss in the condenser was neglected. The condenser and evaporator were of the double tube counter flow-type heat exchangers, in which refrigerant flows in the inner tube and water flows in the annular channel. The COP of refrigerant mixture was found to be higher than that of pure refrigerant when the heat transfer lengths of the condenser and evaporator were sufficiently long. On the other hand, the pressure drop increased when the tube length was long and the percentage composition of R-114 was high.

Bourdouxhe et al. [6] presented a simulation model for reciprocating chiller (water-to-water) to predict its performance. The model calculated the compressor power and refrigerant volume flow rates due to change of load in conditioned buildings during part-load that can be realized by cylinder unloading of the reciprocating compressor. Both condenser and evaporator were assumed to be isothermal on the refrigerant side and pressure drop was neglected in the heat exchanger.

Stefanuk et al. [7] presented a simulation model of a water-to-water vapor-compression cycle using refrigerant-22. They assumed no pressure drop inside the condenser and evaporator, and it occurred only through the valves of the reciprocating compressor. The main objective of this model was to predict system performance over the full operating range of a heat pump. They found that the control of superheat and refrigerant mass flow rate can improve the coefficient of performance for the heat pump.

Green and Roberts [8] studied the effect of heat exchanger design on the optimum total COP (cooling capacity over compressor and fan power) for the case of the outdoor air-coil of an air-to-water heat pump using R-22. The optimization study showed that optimum performance under frost-free conditions is achieved with an evaporator of large face area, as high an evaporating temperature as possible and no superheat across the coil.

Ouazia and Snelson [9] developed a simulation program to predict the performance of an alternative refrigerant (R-134a) operating in water-to-water heat pump. They assumed constant pressures inside condenser and evaporator and adiabatic compression for open-driven reciprocating compressors. The main objective was to calculate the performance characteristics including evaporator capacity, compressor shaft power, refrigerant mass flow rate, and cooling coefficient of performance. They also investigated the effect of subcooling degree on the coefficient of performance.

Linton and Snelson [10] described a series of tests using refrigerants (R-12, R-134a & R-152a). They studied the effect of condenser liquid subcooling on system performance for each of these refrigerants. The tests were carried out under controlled conditions in a well-instrumented (water-to-water) heat pump test facility. The tests covered an evaporating temperature range from -9 °C to 8 °C with a constant condensing temperature of 51.5 °C. The degree of subcooling was varied between 6 °C and 18 °C. The refrigerant superheat was maintained at 7.4 °C throughout the tests. It was found that subcooling increased COP.

2. Theoretical analysis

The condenser in a refrigeration system is a heat exchanger that usually rejects all the heat from the system. It is usually divided into three regions, namely, desuperheating, condensing and subcooling regions. Condensation takes place in approximately 85% of the condenser area at a substantially constant condensing temperature. Therefore, the governing equations, which are used for condenser design, are divided into two categories: a) the two-phase zone for condensing region, and b) the single-phase zone for desuperheating and subcooling regions.

The Mathematical expressions and correlations for a simulation design of air-cooled condensers are presented for heat transfer between the refrigerant and ambient air, as well as for pressure drop in refrigerant and air sides. In addition, different condenser coil areas are expressed in terms of coil geometry and layout.

2.1. Governing equations for the design

The main purpose of the design is to predict the size of the air-cooled condenser for given process data including the amount of heat rejection from the condenser. It is suggested to divide the surface area of the coil into equal control volumes, each of an elementary area δA_o and an elementary length δx as shown in fig. 2. The heat transfer in each control volume is given by $\delta Q_{i,j}$.

a) The continuity equations:

$$m_a = U_{fan}.\rho_{ai}.A_g \,, \tag{1}$$

$$m_r = G_r.A_i = (\pi/4)d_i^2.\ G_r.(W/\ S_t).(N_r/\ N_p) = Qe\ /\ (h_{sh} - h_{sub})\,. \tag{2}$$

b) The energy equations;

i- The energy equations for single-phase flow regions:

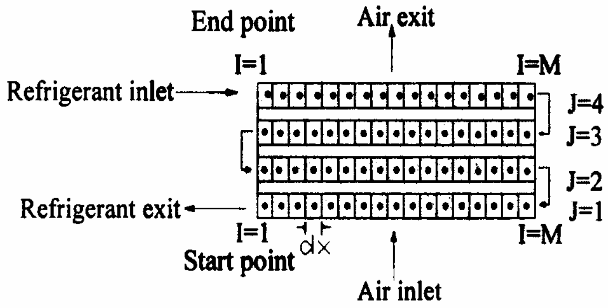

Fig. 2. The mesh grids of counter-cross flow condenser (4-rows & 4-passes).

$$\delta Q_{i,j} = \delta m_a\, Cp_a\, [Ta\ (j+1,i) - Ta\ (j,i)]\,, \tag{3}$$

$$\delta Q_{i,j} = \delta m_r\, Cp_r\, [Tr\ (j,i+1) - Tr\ (j,i)]\,, \tag{4}$$

$$\delta Q_{i,j} = U_{sp}\, \delta A_o\, (Tr_m - Ta_m)\,. \tag{5}$$

Where,

$Tr_m = [Tr\ (j,\ i+1) + Tr\ (j,\ i)]\ /\ 2$,

$Ta_m = [Ta\ (j+1,\ i) + Ta\ (j,\ i)]\ /\ 2$,

and,

$\delta A_o = \pi.d_o.\delta x$, $\delta x = L\ /\ M$, $\delta m_r = m_r/(N_c.\ Nr_p)$,

$\delta m_a = m_a\ /\ (N_c\ .\ M)$,

The above system of eqs. (3-5) contains three unknowns $(Tr\ (j,i+1),\ Ta\ (j+1,i)$ and $\delta Q_{i,j})$. Solving for these unknowns, we get:

$$\delta Q_{i,j} = S\ [Tr\ (j,i) - Ta\ (j,i)]\ /\ Xo\,, \tag{6}$$

$$Tr\ (j,i+1) = Tr\ (j,\ i) + Ni\ [Tr\ (j,\ i) - Ta\ (j,\ i)]\ /\ Xo, \tag{7}$$

$$Ta\ (j+1,i) = Ta\ (j,\ i) + No\ [Tr\ (j,\ i) - Ta\ (j,\ i)]\ /Xo\,. \tag{8}$$

Where,

$S = U_{sp}.\delta A_o$, $Ni = S\ /\ (\delta m_r.\ Cp_r)$, $No = S\ /\ (\delta m_a.\ Cp_a)$, $Xo = 1 + No\ /\ 2 - Ni\ /\ 2$

ii- The energy equations for two-phase flow region:

$$\delta Q_{i,j} = \delta m_a\, Cp_a\, [Ta\ (j+1,i) - Ta\ (j,i)]\,, \tag{9}$$

$$\delta Q_{i,j} = \delta m_r\, [hr\ (j,i+1) - hr\ (j,i)]\,, \tag{10}$$

$$\delta Q_{i,j} = U_{tp}\, \delta A_o\, (Tc - Ta_m)\,. \tag{11}$$

The pervious equations can be expressed as follows:

$$\delta Q_{i,j} = S_t\ [Tc - Ta\ (j,\ i)]\ /\ Xo_t\,, \tag{12}$$

$$hr\,(j,i+1) = hr\,(j,\, i) + Ni_t\,[Tc - Ta\,(j,\, i)]\,/\,Xo_t\,, \quad (13)$$

$$Ta\,(j+1,\, i) = Ta\,(j,\, i) + No_t\,[Tc - Ta\,(j,i)]\,/\,Xo_t\,. \quad (14)$$

Where,
$S_t = U_{tp}.\delta A_o$, $Ni_t = S_t / \delta m_r$, $No_t = S_t/(\delta m_a.\ Cp_a)$,
$Xo_t = 1 + No_t / 2$

2.2. Calculation of pressure drop for refrigerant-side and air-side

The refrigerant pressure drop in the condenser, ΔP_{tp} is evaluated using the following form of the Pierre correlation [11], as modified by Sami et al. [12]:

$$\Delta P_{tp} = \left[f + \frac{(x_o - x_i)}{x_{av}.L} \right] . \left[\frac{G_r^{\,2} \times L}{d_i \times \rho_{av}} \right] . \quad (15)$$

Where,

$f = 0.0185\ k_f^{0.35}.Re_i^{-0.25}$,

k_f is the Pierre number defined as : $k_f = (x_o - x_i).\ h_{fg}\ / L$, ρ_{av} and x_{av} are defined as :

$$\rho_{av} = \frac{\rho_v \cdot \rho_l}{x_{av}\rho_l + (1 - x_{av})\, \rho_v}\,, \quad x_{av} = \frac{x_o + x_i}{2} \quad .$$

The pressure drop of air-side consists of plenum losses, *Dpp* and condenser bundle losses, *Dpb.* Saunders [13] developed a correlation for plenum pressure drop, and another correlation for condenser bundle losses was presented by Kays and London [14]. These are given as:

$$Dpb = 0.5 \rho_{ai} U_{ms}^{\,2} \left[(1 + Far^2)(\frac{\rho_{ai}}{\rho_{ao}} - 1) \right] + F \left[\frac{A_t}{Far A_g} . \frac{\rho_{ai}}{\rho_{ao}} \right],$$

$$Dpp = 0.06\ \rho_{am} U_{fan}^2\ .$$

Where,

$F = -2.0E\text{-}22\ Re_o^4 + 6.0E\text{-}17\ Re_o^3$
$- 5.0E\text{-}12\ Re_o^2 + 1.0E\text{-}7\ Re_o + 0.0169$,

$$\rho_{am} = \left(\frac{\rho_{ai} + \rho_{ao}}{2} \right)\,, \quad U_{ms} = \frac{m_a}{\rho_{ai}\ A_x}\,.$$

2.3. Numerical solution method

The above analysis represents a system of equations to be solved numerically. It consists of the continuity eqs. (1,2) and the energy eqs. (3-5) for single-phase flow or the energy eqs. (9-11) for two-phase flow. This system can be solved numerically for selected values of the system input parameters L/W, N_r, N_p, U_{fan}, Qe, Tc, Te, and inlet air temperature, T_{ai}. All physical and thermodynamic properties are correlated by curve-fitting using tabulated data for *R-134a.* The set of equations for each phase, consists of three finite difference equations in three unknowns ($\delta Qi,\ j$, $hr\ (j,i+1)$ or $Tr\ (j,i+1)$, and $Ta\ (j+1,\ i)$) for a cross-counter, multi-rows refrigerant condenser. A computer program is developed to solve the above system of equations.

The simulation design of the model is also compared with manufacturers' data ("Carrier" catalogues [19]) for the following specifications

Qe = 350 kW,	d_o = 0.01341 m,	η_{is} = 0.82
Tai = 40°C,	di = 0.01259 m,	η_c = 0.7
Te = 1°C,	S_l = 0.0275 m,	U_{fan}= 2.5 m/s
Tc = 52.7°C,	S_t = 0.0318 m ,	L/W= 2.25
ΔT_{sup} = 6°C,	N_r = 3,	L = 5.9 m
ΔT_{sub} = 9°C,	N_p = 3,	W = 2.67 m
H = 2.33 m		

The model gives its converged solution at a face length, L = 5.7 m, which shows an acceptable deviation of 3.4 % for the same specifications.

2.4. Cost of condensing unit

The cost of a condensing unit is obtained from manufacturers' data "Carrier Company [19]". The condensing unit annual cost consists of two items, fixed and operating costs. Firstly, the fixed cost is the expense of condenser bundle and the semi-open compressor as well as taxes, insurance and depreciation. The capital cost of the condenser is correlated by curve-fitting from a set of data (bundle bare area and its cost).

Condenser cost = $-0.6287\ A_b^2 + 427.67\ A_b$
$+ 10052$ (L.E).

The capital cost of the compressor is obtained from "Copland" catalogues [20] according to the required compressor power. The annual charges of interest rate, taxes, insurance and depreciation are all estimated to be a bout 17% of the total capital cost of the condensing unit. Secondly, the operating cost is based on the energy consumption of both the compressor and fans during the operating hours of the condensing unit. The unit energy cost was assumed as 0.12 LE / kWh, and the number of operating hours per year is estimated as 2000 h.

3. Results

3.1. Effects of number of coil-rows and condensing temperature

The condensing temperature has two opposite effects. Increasing the condensing temperature will increase the temperature difference between the refrigerant and cooling air. This will reduce the required heat transfer area and face area, and therefore, the capital cost of the condenser is decreased. On the other hand, the running cost is increased due to the increase in the compressor power and also its cost. The increase in the number of rows, N_r leads to an increase in condenser depth, and therefore the face area is decreased and bundle cost is also decreased. Conversely, the fans power will increase due to an increase in bundle pressure drop, thus running cost will increase. Therefore, as N_r increases the annual total cost (capital and operating cost) will first decrease then increase showing an optimum minimum value as shown in figs. 3 to 5.

Fig. 3 shows the variations of annual cost of the condensing unit of air-cooled chillers as a function of the deep coil rows N_r for a range of condenser temperatures. As shown, an economical design is obtained when N_r equals 3 or 4 and T_c is about 48°C for an air-cooled condenser in an environment which has a dry-bulb temperature of 35 °C and with fan speed of 2.5 m/s. It is noticed that coil manufacturers [19] recommend 3-rows or 4-rows air-cooled condensers. Higher condensing temperatures over 50 °C for an ambient temperature of 35 °C, will increase the cost due to the increase in energy consumed by the compressor over the cost gained as a result of the decrease in the condenser size. The net outcome is the increase in the total cost of the condensing unit when it is designed to operate at higher condensing temperatures.

To show the effect of the weather conditions on the annual total cost of the condensing unit, figs. 4 and 5 are presented with the same system parameters as in fig. 3 but for T_{ai} = 40°C and 45°C, respectively. The results confirm that the optimum number of rows is 3 or 4-rows and the optimum rejection temperature is 52°C for T_{ai} = 40°C and 57°C for T_{ai} = 45 °C. This shows that optimum

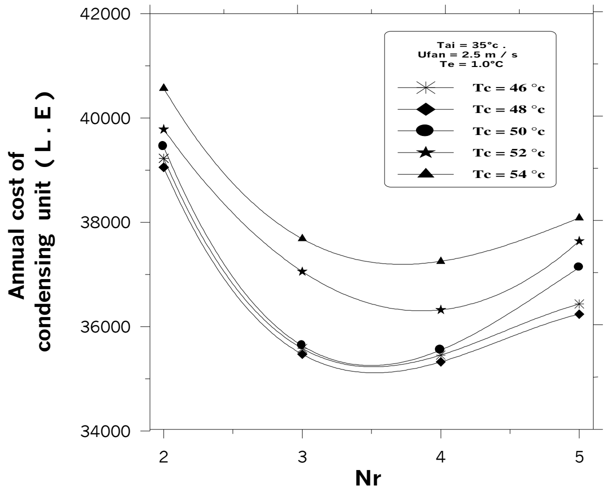

Fig. 3. Effects of *Nr* and *Tc* on annual cost of condensing unit for T_{ai} = 35°C.

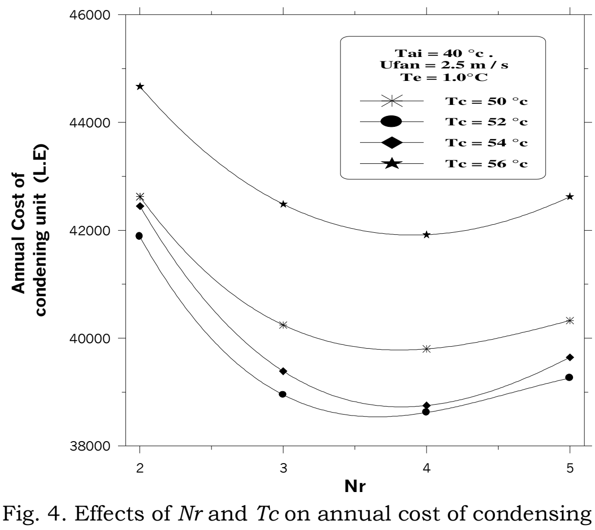

Fig. 4. Effects of *Nr* and *Tc* on annual cost of condensing unit for T_{ai} = 40°C.

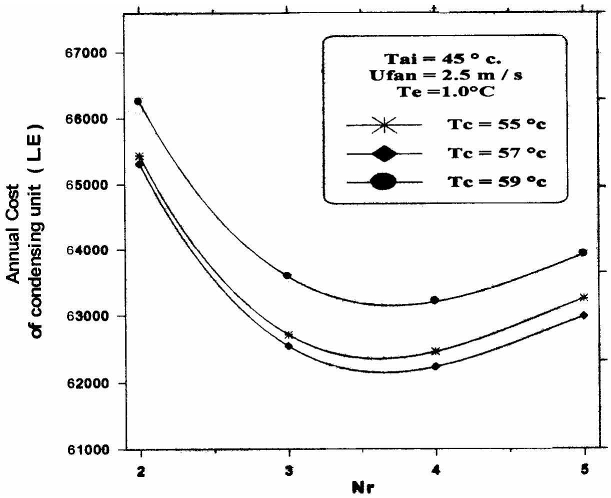

Fig. 5. Effects of *Nr* and *Tc* on annual cost of condensing unit for T_{ai} = 45°C.

temperature difference (Tc - T_{ai}) is between 12 and 13°C which is an acceptable range recommended by coils manufacturers [19].

3.2. Effects of aspect ratio and fan speed

The heat exchanger for single-phase flow differs from that for two-phase flow, where the aspect ratio (L / W) for the latter must be limited to ensure proper pressure drop inside the heat exchanger. As the aspect ratio increases, which means the use of small number of longer tubes, the mass velocity, G_r will increase. This leads to an improvement in the overall heat transfer coefficient on the expense of an increase in the refrigerant pressure-drop as shown in fig. 6. The increase in overall heat transfer coefficient will decrease the required condenser area as well as its cost. On the other hand, the increase in the pressure drop will increase the condensing temperature drop and therefore the bundle area will be increased and as well as its cost, without affecting the running cost significantly. Therefore, the aspect ratio must be chosen carefully to satisfy minimum annual cost and suitable pressure drop as shown in fig. 7. This is the distinct difference between the heat exchanger for single and two-phase flows. The increase in aspect ratio for single-phase heat exchanger is unlimited where the flow pressure drop does not affect the flow temperature and is limited only by pumping power and construction limit. The increase of fan speed will decrease

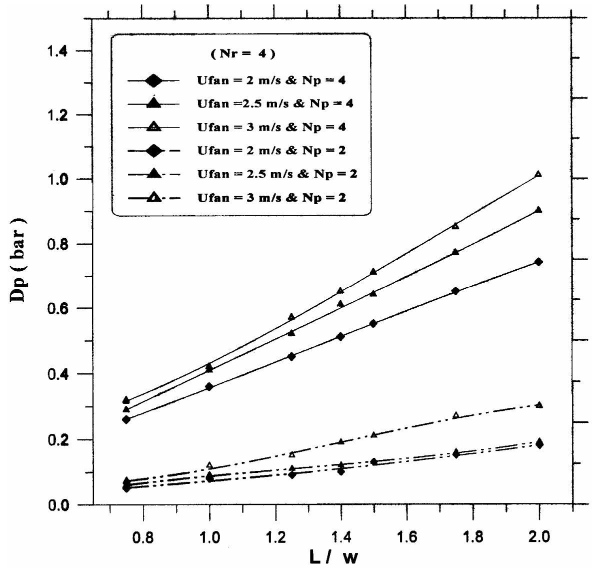

Fig. 6. Effects of aspect ratio and fan speed on refrigerant-side pressure drop.

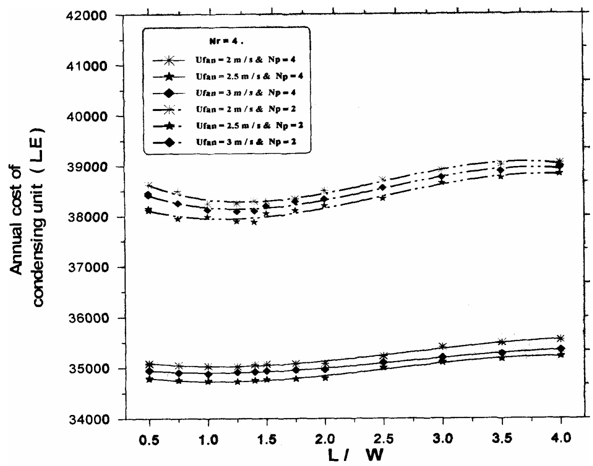

Fig. 7. Effects of aspect ratio and fan-speed on annual cost of condensing unit.

the initial cost of the condenser but the running cost definitely increases as a result of the increase of the air pressure drop. It is found that fan speed has a significant effect on the refrigerant pressure drop only at higher aspect ratios, L / W > 1.5. It is interesting to notice that the optimum fan speed is 2.5 m/s as shown in fig. 7.

4. Calculation procedure of the generalized design

The steps for evaluating the air-cooled condenser size with the known system parameters *(Qe,* η_{is} *,* η_c *,* N_r *,* N_p *,* d_o*,* d_i*,* S_l*,* S_t*, Te,* ΔT_{sup}*, and* ΔT_{sub}*)* are as follows:

1. Given the cooling capacity *Qe* (T.R), the number of coil-rows N_r, and for a selected fan speed in the range from 2 to 3 m /s, we can find the aspect ratio *L / W* from figs. 8 or 9. Fig. 8 is for coils with 3 or 4-rows with $N_p = N_r$ and fig. 9 is for 4-rows only with 2 or 4 tube-passes.
2. Knowing the ambient temperature, T_{ai}, and from fig. 10 (for $\eta_{is} = 0.82$ and $\eta_c = 0.7$ as given by compressor manufacturers [20]) we can find the optimum COP of the unit for minimum condensing unit cost.
3. Estimate the input electric power for the compressor, W_e and the actual adiabatic compressor power, W_{act} from the following:

$$W_e = \frac{Qe}{COP} \qquad , W_{act} = W_e \times \frac{\eta_c}{\eta_{is}} .$$

Where,

$$\eta_{is} = \frac{W_{isen}}{W_{act}} \qquad , \ \eta_c = \frac{W_{isen}}{W_e} .$$

W_{is} is the isentropic thermal power absorbed by the refrigerant leaving the compressor, η_{is} is the isentropic efficiency of the compressor, and η_c is the overall compressor efficiency.

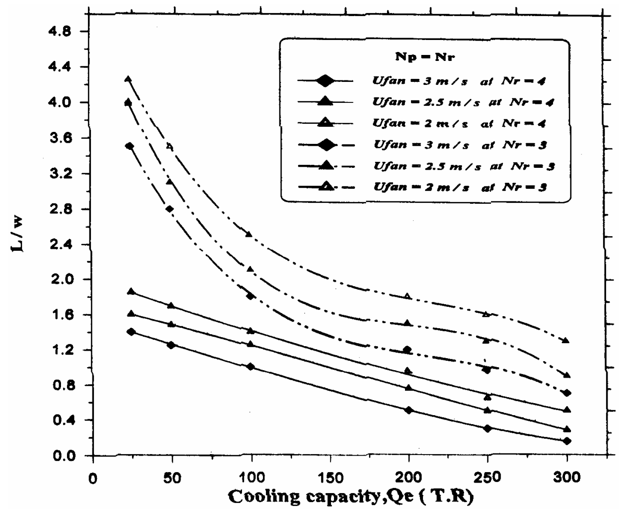

Fig. 8. The generalized design chart (cooling capacity vs. aspect ratio).

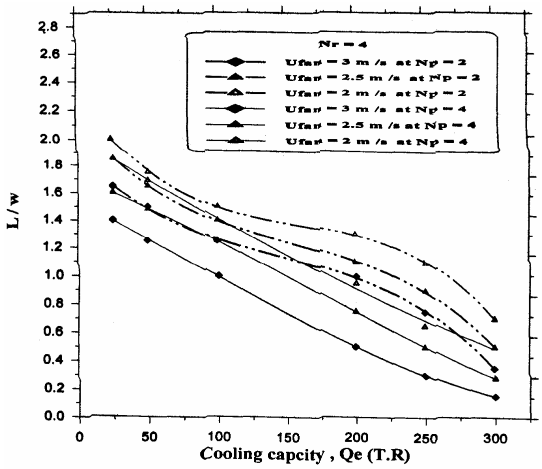

Fig. 9. The generalized design chart (cooling capacity vs. aspect ratio).

4. The actual condenser heat rejection, *Qc* which is the sum of the cooling capacity, *Qe* and the actual compressor work rate, W_{act} can be evaluated as:

$$Qc = Qe + W_{act} ,$$

$$\therefore Qc = Qe \left[1 + \frac{\eta_c}{\eta_{is} \cdot COP} \right] .$$

5. From fig. 11 or 12 (based on the given N_r and N_p), the condenser capacity *Qc* per unit face area A_g can be obtained in terms of the input data (U_{fan} , N_r, N_p, T_{ai}), and for $Tc = T_{ai} + 13$ °C.
6. Knowing *Qc* from step-4, the condenser face area can then be calculated.
7. Knowing the aspect ratio (*L / W*) and, Ag which equals *L.W*, then both *L* and *W* can be evaluated. The height of the coil, *H* can also be evaluated from $H = N_r . S_l$.
8. The number of columns N_c can be obtained in terms of the width *W* and the transverse tube spacing S_t as $N_c = W / S_t$.

The condenser is thus thermally designed with the minimum annual cost. The dimensions of the condenser given by (*L, W, H,* N_r *,* N_c, and indicated tube spacing and diameters) are completely specified .

5. Conclusions

A computer simulation program has been developed to predict the optimum size of air-

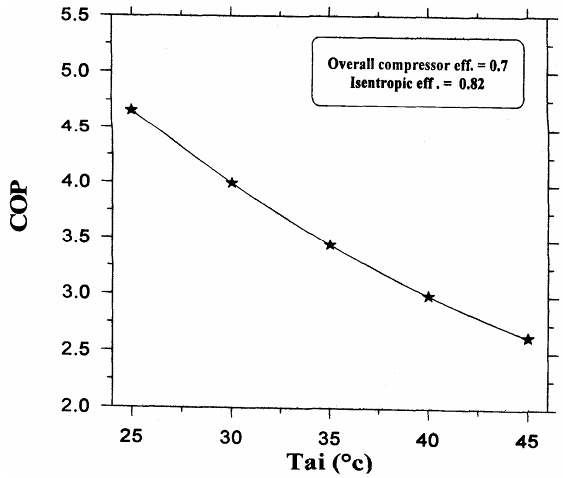

Fig. 10. Generalized design chart (optimum COP vs. ambient temperature).

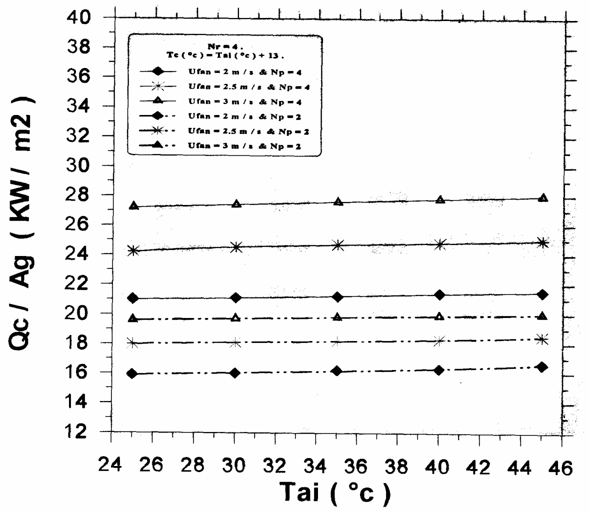

Fig.11. Generalized design chart (heat flux vs. ambient temperature).

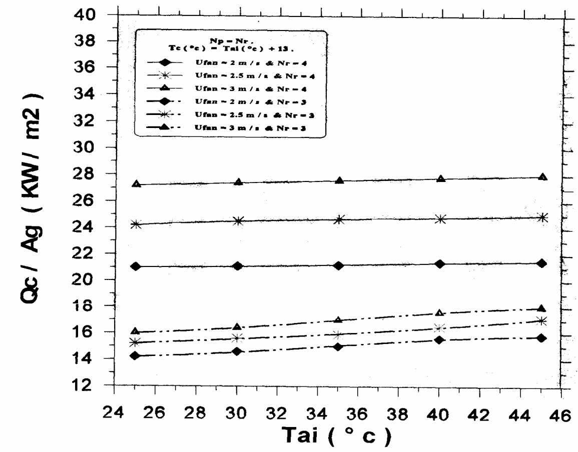

Fig. 12. Generalized design chart (heat flux vs. ambient temperature).

cooled condensers for cooling loads in the range from 25 to 300 tons of refrigeration. The optimum design is based on the main parameters, namely, condensing temperature relative to local ambient temperature, number of rows, condenser coil aspect ratio, number of passes, fan speed as well as cooling load. The effects of these parameters can be summarized as follows:

1. The condensing temperature mainly depends on the ambient air temperature. The optimum difference between the condensing and ambient temperatures is almost 13 °C, which gives satisfactory results for all different applications.
2. The effect of number of deep-rows, N_r has two sides, where an increase in N_r will be associated with a decrease in face area leading to an increase in fan power consumption. So, the recommended deep-rows is found to be about 3 or 4 rows.
3. The number of passes and the value of coil aspect ratio must be chosen carefully because they have a significant effect on condenser design. The increase in one of them will lead to a decrease in the condenser area that causes a significant increase in refrigerant pressure drop and in turn to a distinct drop in condensing temperature.
4. The increase of fan speed will decrease the face area and increase the fan power consumption. Therefore, the design value of fan speed should be within 2 to 3 m/s.

The simulation program has been validated by comparison with commercial data obtained from manufactures catalogs and gave satisfactory results.

Nomenclature

A_b Bare tubes surface area, $A_b = \pi . d_o . N_t . L$, m^2,

A_f Area of fins, $A_f = 2\,(H.W - \pi /4.N_t.d_o^2).\,N_f.\,L$, m^2,

A_g Face area, $A_g = W.\,L$, m^2,

A_i Inner surface area, $A_i = \pi .\, d_i.N_t.L$, m^2,

A_t Total outer surface area, $A_t = A_f + A_u$, m^2,

A_u Area of unfinned part of the tubes, $A_u = \pi .\, d_o.L.N_t.(1-N_f.t_f)$, m^2,

A_x Minimum free flow area, $A_x = (S_t - d_o).\, N_c.\, L.\,(1 - t_f * N_f)$, m^2,

COP Coefficient of performance of refrigeration cycle,
Cp_a Specific heat of air, J / kg. K,
Cp_r Specific heat of refrigerant, J / kg. K,
d_i Tube inside-diameter, m,
d_o Tube outside-diameter of condenser, m,
Far Minimum free-flow area / face area,
G_a Mass velocity of air, $G = m_a / A_x$, kg/m^2. s,
G_r Mass velocity of refrigerant, kg/m^2. s,
h_{fg} Latent heat of evaporation, kJ / kg,
h_r Specific enthalpy of refrigerant, kJ / kg,
h_{sh} Specific enthalpy at exit from evaporator, kJ / kg,
h_{sub} Specific enthalpy at exit from condenser, kJ / kg,
H Characteristic face height, m,
L Characteristic face length or condenser tube length, m,
M Number of control volumes in each row,
m_a Mass flow rate of air, kg / s,
m_r Mass flow rate of refrigerant, kg / s,
N_c Number of columns,
N_f Number of fins per unit length of coil, 1/m,
N_p Number of passes,
N_r Number of rows,
Nr_p Number of rows per pass,
N_t Number of tubes,
NTU Number of transfer units,
Qc Condenser heat rejection, kW,
Qe Cooling capacity, kW,
Re_i Reynolds number for refrigerant inside tubes, $Re_i = G_r.d_i / \mu_r$,
Re_o Reynolds number for air outside tubes, $Re_o = G_a.d_o / \mu_a$,
S_l Tube longitudinal pitch, m,
S_t Tube transverse pitch, m,
Ta Air temperature, °C,
Tc Mean condensing temperature, °C,
Te Mean evaporating temperature, °C,
Tr Refrigerant temperature, °C,
tf Fin thickness, m,
$Ufan$ Face velocity, m/s,
Usp Overall heat transfer coefficient (single-phase) flow, W /m^2 K,
Utp Overall heat transfer coefficient (two-phase) flow, W /m^2 K,
W Characteristic face width, m,
xi Refrigerant inlet quality, and
xo Refrigerant exit quality .

Greek symbols

ηis Compressor isentropic efficiency,
η_c Compressor overall efficiency ,
μ Dynamic viscosity, kg / m.s,
ρ_{ai} Density of inlet air to fans, kg / m^3,
ρ_{ao} Density of exit air from fans, kg / m^3,
ρ_l Density of saturated-liquid of refrigerant, kg / m^3, and
ρ_v Density of saturated-vapor of refrigerant, kg / m^3.

Subscripts

a Air,
i Inlet refrigerant or inside tubes,
o Exit or outside tube,
r Rows or refrigerant,
sh Superheated refrigerant,
sp Single-phase,
sub Subcooled refrigerant, and
tp Two-phase.

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