Global Journal of Researches in Engineering: A
Mechanical and MechanicsEngineering
Volume 15 Issue 3 Version 1.0 Year 2015
Type: Double Blind Peer Reviewed International Research Journal
Publisher: Global Journals Inc. (USA)
Online ISSN:2249-4596 Print ISSN:0975-5861

Experimental and Numerical Model for Thermal Design of Air Cooled Condenser

By Ali Hussain Tarrad & Ali Farhan Al-Tameemi
Private Consultant Engineer, Denmark

Abstract- The present work outlines a simple procedure for the thermal design of air cooled heat exchanger. The step by step numerical technique is implemented a long the steam flow direction to ratea vertical orientation single pass two tube rows heat exchanger. A saturated steam at atmospheric pressure of flow rate ranged between (18-36) kg/hr was passed throughout the tubes to provide a steam velocity in the range between (3.5) and (7) m/s. The condenser entering air dry bulb temperature was ranged between (21) and (42) °Cand condensation load capacity fell in the range of (11) and (22.5) kW. The air flow rate was (1200) cfm and (2400) cfm corresponding air face velocities of (3 and 6) m/s. The simulated data showed excellent agreement with the measured rating parameters regarding the heat exchanger load duty and exit air cooling temperature. The respective discrepancy for the heat duty was within (12) % and (-5) % and the exit air temperature was underestimated by (5) %.

Keywords: condensation, heat exchangers, air cooled, steam condensers, modeling.

GJRE-A Classification : FOR Code: 091307

EXPERIMENTALANDNUMERICALMODELFORTHERMALDESIGNOFAIRCOOLEDCONDENSER

Strictly as per the compliance and regulations of:

Experimental and Numerical Model for Thermal Design of Air Cooled Condenser

Ali Hussain Tarrad [α] & Ali Farhan Al-Tameemi [σ]

Abstract- The present work outlines a simple procedure for the thermal design of air cooled heat exchanger. The step by step numerical technique is implemented a long the steam flow direction to ratea vertical orientation single pass two tube rows heat exchanger. A saturated steam at atmospheric pressure of flow rate ranged between (18-36) kg/hr was passed throughout the tubes to provide a steam velocity in the range between (3.5) and (7) m/s. The condenser entering air dry bulb temperature was ranged between (21) and (42) °Cand condensation load capacity fell in the range of (11) and (22.5) kW. The air flow rate was (1200) cfm and (2400) cfm corresponding air face velocities of (3 and 6) m/s. The simulated data showed excellent agreement with the measured rating parameters regarding the heat exchanger load duty and exit air cooling temperature. The respective discrepancy for the heat duty was within (12) % and (-5) % and the exit air temperature was underestimated by (5) %.

Keywords: condensation, heat exchangers, air cooled, steam condensers, modeling.

I. Introduction

In ACC, heat is transferred from the process fluid (steam) to the cooling medium (air) through the fin tube bundle. It depends on the temperature difference (driving force) between air and steam so that the dry bulb temperature of air is a key control of the ACC performance. Therefore, the dry cooling system with ACC is less efficient in hot ambient.

Kutscher and Costenaro (2002) [1]developed a model to assess the cost and performance of different methods for using supplemental evaporative cooling to boost the summer performance of air cooled condenser in geothermal power plants. A system in which water directly contacts the condensate tubes has the highest performance and is economically the most attractive. However, consideration of scaling and corrosion must be addressed.

Jabardo and Mamani (2003) [2] developed a simulation model based on dividing the condenser into three zones as superheating, condensing and sub-cooling.Each region was treated as an independent heat exchanger. The discrepancy between the experimental data and the simulation results has shown a good agreement.

Author α: Private Consultant Engineer, Mechanical Engineering (PhD) Copenhagen-Denmark. e-mail: dr.alitarrad@yahoo.com
Author σ: Mechanical Engineer (MSc.) AdhwaAlshamal Contracting and General Trading Baghdad-Iraq. e-mail: engineerali.85@gmail.com

Gadhamshettyet al. (2006) [3], proposed a new approach to alleviate the performance decline in air cooled condenser with increasing the air dry-bulb temperature. A chilled water thermal energy storage system is used to pre-cool the inflow air to the ACC whenever the ambient air temperature increases above (20 °C). The proposed procedure used the test 171 MW plant saves (2.5%) of the power (4.2MW) without using any water or incurring any water treatment cost.

The work of Tarrad and coworkers[4], [5] and [6] was concentrated on the heat transfer performance and modeling of air cooled heat exchangers. Their work showed that the thermal enhancement is a dependent measure of the fin geometric variables and row intensity of the air cooled heat exchanger. Tarrad (2010) [7] developed a numerical model for performance prediction of dry cooling of the air cooled condensers applied in power plants technology. A computer code was built that depends on the idea of using the row by row technique for estimating the heat transfer coefficient, air temperature and air physical properties distribution in the air flow direction from row to row. The model results showed an improvement in the condensation load up to (23%) when air pre-cooling mode applied to inflow air to the ACC to lower the dry-bulb temperature from (45) to (28) °C at air face velocity of (3.6) m/s.

Tarrad and Khudor(2015)[8] have presented quite a simple and adaptable correlation for the air side heat transfer coefficient in the form of dimensionless group criteria. It depends on the fin geometry, row and tube intensity and operating conditions. They concluded that their correlation predicts the heat duty and overall heat transfer coefficient of the case study heat exchangers with total mean absolute errors of (13%) and (10%) respectively. More recently, Tarrad and Al-Nadawi (2015) [9] presented a model for the air cooled condenser. Its strategy depends on the tube by tube technique implemented for a window type air conditioning unit circulating different refrigerant such as R-22, R-407C and R-407A. They postulated that the predicted heat duty of these refrigerants by their model has showed excellent agreement and was within the range of (-5%) and (+1.7%).

II. Experimental Work

a) Test Rig

An experimental facility was constructed to allow two types of condensing system worked as a test arrangement, Altameemi (2007) [10]. Each one represents a separate unit having all the specifications and instruments which allows condensation data to be collected over a range of operating conditions, figure (1). Air cooled condenser and shell and tube heat exchanger were used as a single or in a hybrid arrangement.The experimental apparatus was modified after a first set of data to ensure an accurate operation condition compatibility to weather criteria in Iraq in hottest period. This was accomplished by adding an electrical heating coil works as air heater for a set exit temperatures.

b) Heat Exchanger

Air cooled condenser is a finned tube heat exchanger, typically used for the process which consists of a finned-tube bundle with rectangular box headers on both ends of tubes. The "ACC" used in the present work was a vertical type, a single pass having two rows with flat side tubes occupied each rowas shown in figure (2).Flat side tube geometry used to enhance heat transfer inside tube with extra heat transfer area and reduce pressure drop outside tube. The tubes material are brass which has excellent physical properties [11], compared with other materials.

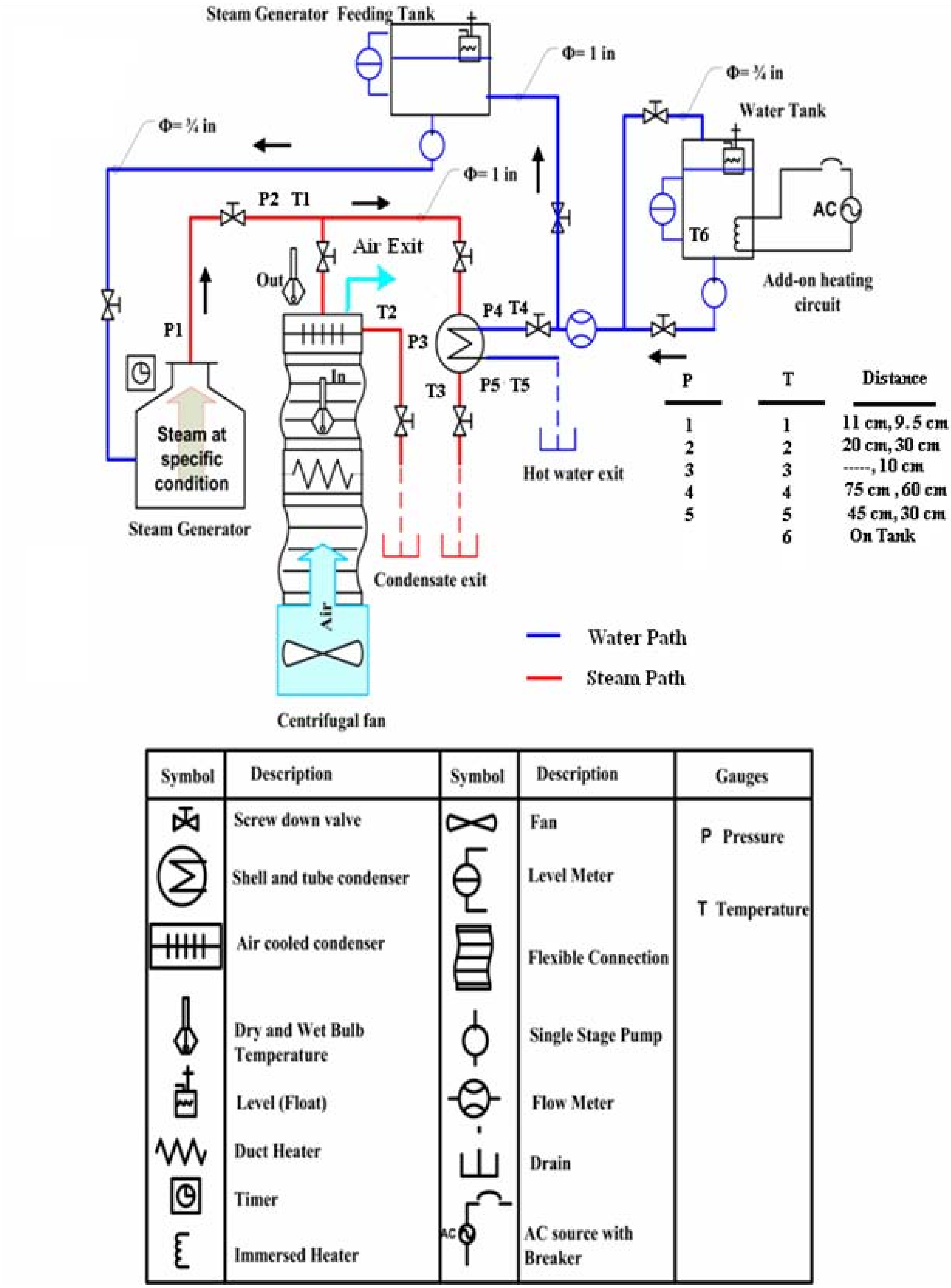

Figure (1) : A Schematic diagram for the test rig, Altameemi [10]

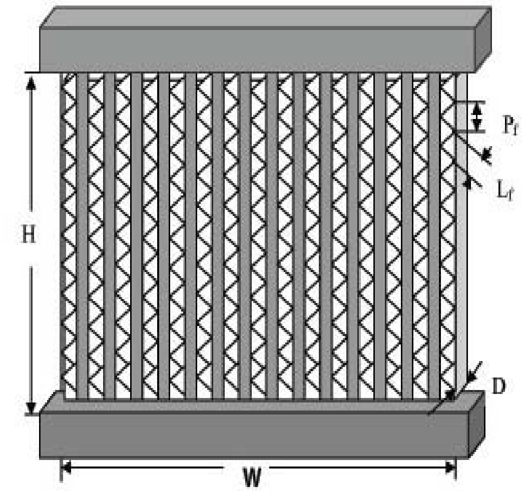

Figure (2.a) : Heat exchanger dimension designation

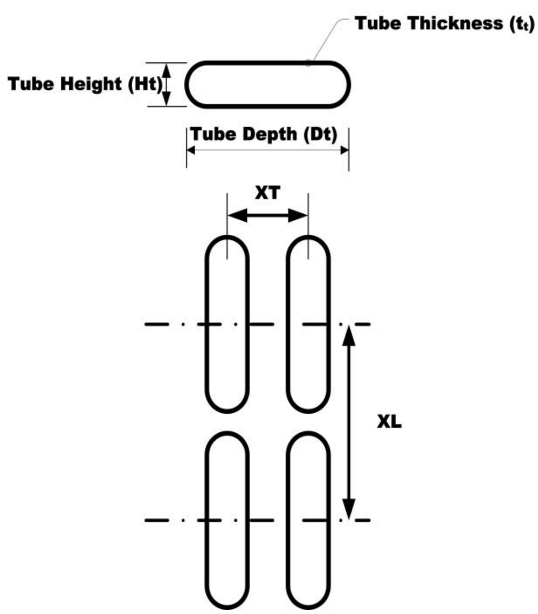

Figure (2.b) : tube layout

Figure (2.c) : Tube shape

Figure (2) : Physical geometry of heat exchanger, Altameemi [10]

The fins are constructed to the tubes so each fins row contains (70) fins attached to tube in equal pitch about (4.18) mm.The fins material is a copper, the physical characteristicand dimensions are listed in table (1).

Table (1) : Geometry and physical characteristic of heat exchanger

Parameter		Value
Core	Width (W), mm	590
	Depth (D), mm	30
	Height (H), mm	320
	No. of Tubes	110
	No. of Tubes/Row	55
	No. of Rows	2
	No. of fins/ Tube (on both sides)	140
	Transverse Distance (X_T) to flow, mm	11.25
	Longitudinal Distance (X_l) to flow, mm	15
	Frontal Area (A_{face}), m^2	0.1888
Fin	Pitch (f_P), mm	4.18
	Length (l_f), mm	8.05
	Thickness(t_f), mm	0.18
	Area of a Single Fin (A_f), mm^2	36.12
	Material	Copper
	Thermal Conductivity (W/m.C), [12]	388
Tube	Height (H_t), mm	2.35
	Depth (D_t), mm	12
	Thickness (t_t), mm	0.24
	Material	Brass
	Inner tube surface	Smooth
	Thermal Conductivity (W/m.°C), [12]	119
Area	Total Surface Area (A_{total}), m^2	3.935
	Total Fin Surface Area, m^2	3.029
	Total Bare Tube Area, m^2	0.906

The whole assembly is mounted on legs with a rubber and fasteners to well-set during operation.

The uncertainty percentage for the measurement was estimated to be within (± 2 %) for the whole tests range in this work, Altameemi [10].

III. Model Methodology

a) Model Technique and Assumptions

The air cooled steam condenser is shown in figure (3).The steam flows inside the flat tube. The air flow is perpendicular to the tubes across the fins so that both process and service fluids pass in a cross flow pattern on both sides of the exchanger wall. A row by row technique was implemented and each row in the single pass condenser is subdivided into segments in the height direction of the heat exchanger, figure (3). Each segment is treated as a small condenser with a specified geometry. It is associated with steam parameters such as mass flow rate, pressure and temperature. The air side has also specific air mass flow rate and inlet air temperature.

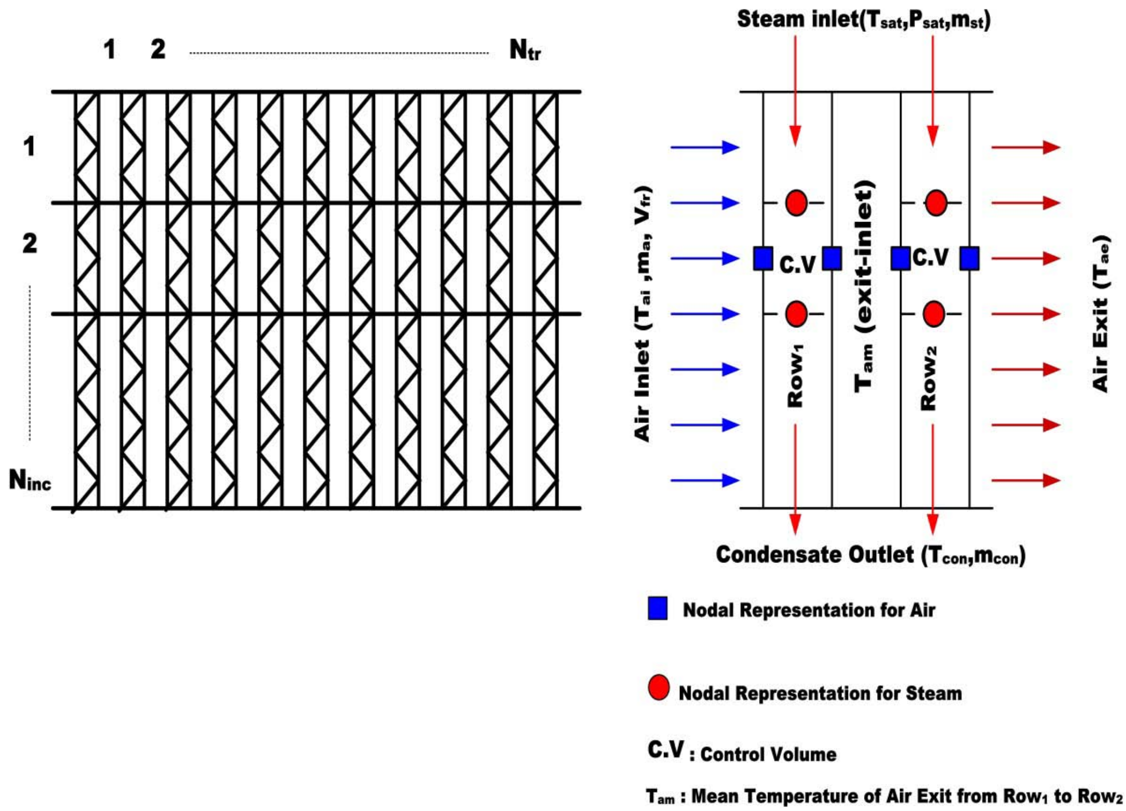

Figure (3) : A step by step modeling representation

The assumptions of the ACC modeling are:

1. The air mass flow rate is assumed to be distributed uniformly over the whole face of the air cooled steam condenser. To ensure this assumption, the inlet duct configuration and mal-distribution was checked.
2. Homogenous temperature distribution of air all over the frontal face area of the heat exchanger and hence for each segment.
3. The mean air exit temperature of each row is considered to be the inlet to the next row.
4. The steam temperature variations between the rows were assumed to be negligible.
5. The inlet air velocity for each row was assumed to be uniform and is represented by a specified value.
6. The steam mass flow rate from the main header is divided equally for each row.
7. Heat transferred away by convection and radiation is neglected.
8. Material properties of the finned tube bundle are constant with temperature variations.

The air side dry-bulb exit temperature that leaves the condenser is considered as the target parameter of the present model.

b) Steam Side Heat Transfer Coefficient

i. *Condensation Mode*

When the vapor velocity is high, the vapor will pull the liquid along the interface. This is because the vapor velocity at the interface must drop to the value of the liquid velocity. If the vapor flows downward this addition force will increase the average velocity of the liquid and thus the film thickness is decreased. This in turn, will decrease the thermal resistance of the liquid film and thus increasing heat transfer. For condensation inside tubes in air cooled condenser model the tube inside diameter is very small and the influence of tube wall curvature is important in condensation phenomenon. The correlation postulated by Boyko-Kruzhilin (1967) [13] was used to estimate the mean condensing coefficient for a steam between inlet quality x_i and outlet quality x_o inside tubes as below:

$$h_{c,BK} = h_{lo}\left[\frac{\sqrt{\left(\frac{\rho}{\rho_m}\right)_i}+\sqrt{\left(\frac{\rho}{\rho_m}\right)_o}}{2}\right] \qquad (1)$$

Where:

$$\left(\frac{\rho}{\rho_m}\right)_i = 1 + \frac{\rho_l-\rho_v}{\rho_v}x_i \qquad (2.a)$$

$$\left(\frac{\rho}{\rho_m}\right)_o = 1 + \frac{\rho_l-\rho_v}{\rho_v}x_o \qquad (2.b)$$

h_{lo} is the sensible heat transfer coefficient assuming that the total fluid is flowing with condensate properties (condensate filled the tube and was flowing alone). This can be evaluated with the implementation of any available correlation for forced convection in tubes.

Global Journal of Researches in Engineering (A) Volume XV Issue III Version I Year 2015

Boyko and kruzhilin [13]suggested the following correlation:

$$h_{lo} = 0.021\left(\frac{k_l}{D_i}\right) Re_{lo}{}^{0.8} Pr_l{}^{0.43} \quad (3)$$

In a condenser, where the inlet stream was assumed to be saturated vapor and the vapor will be totally condensed,for these conditions equation (1) becomes, Sinnott (2005) [14]:

$$h_{c,BK} = h_{lo}\left[\frac{1+\sqrt{\frac{\rho_l}{\rho_v}}}{2}\right] \quad (4)$$

In the present work it is suggested to divide the tube length into equal increments. The quality change across each increment of length (Δz) is calculated to be (Δx),(x_i –x_e). The heat transfer of steam in two phase region depends on the quality change as a demonstrative factor to describe the amount of heat released at constant temperature. After the determination of the quality limits at each increment, it was utilized to calculate the local condensing heat transfer coefficient at the mid-quality magnitude of each (Δx). Assuming that the local value is constant over particular quality range (Δx), then:

$$\left(\frac{\rho}{\rho_m}\right)_{eq} = \frac{1}{2}\left[\left(\frac{\rho}{\rho_m}\right)_i + \left(\frac{\rho}{\rho_m}\right)_o\right] \quad (5.a)$$

$$\left(\frac{\rho}{\rho_m}\right)_{eq} = \frac{1}{2}\left[1 + \left(\frac{\rho_l-\rho_v}{\rho_v}\right)x_i + 1 + \left(\frac{\rho_l-\rho_v}{\rho_v}\right)x_o\right] \quad (5.b)$$

$$\left(\frac{\rho}{\rho_m}\right)_{eq} = \left[1 + \left(\frac{\rho_l-\rho_v}{\rho_v}\right)x_{eq}\right] \quad (5.c)$$

Where x_{eq} represents the mid-quality which is equal:

$$x_{eq} = \frac{(x_i+x_o)}{2} \quad (5.d)$$

So that equation (1) becomes:

$$h_{c,BK} = h_{lo}\left[\sqrt{1 + \left(\frac{\rho_l-\rho_v}{\rho_v}\right)x_{eq}}\right] \quad (6)$$

At high condensing loads, with vapor shear dominating, tube orientation has no effect, and equation (1) may also be utilized for horizontaltubes, Perry (1999) [15].

ii. *Single Phase Forced Convection*

Turbulent flow:

Numerous relations have been proposed for predicting fully developed turbulent flow in uniform cross-section tubes. The Dittus and Boelter correlation is suitable for moderate temperature variation,Incropera and Dewitt (1996) [16]. TheDittus-Boelter is usually given in the form:

$$h = 0.023\left(\frac{k_f}{D_i}\right) Re^{0.8} Pr^n \quad (7)$$

Where Reynolds and Prandtl numbers are estimated from:

$$Re = \frac{\rho\, u_t D_i}{\mu} = \frac{G\, D_i}{\mu} \quad (8.a)$$

$$Pr = \frac{\mu\, cp}{k_f} \quad (8.b)$$

And $n = 0.4$ for heating ($T_s > T_m$) and $n = 0.3$ for cooling ($T_s < T_m$).

This mathematical relation has been confirmed experimentally for the following ranges of conditions:

0.7≤ Pr ≤ 160 Re_D ≥ 10,000 and L/D_i ≥ 10

For the case where the condensing load is small, vertical tube condenser may maintain sub-cooling in the bottom end of the tube. For this condition, if the temperature difference at the inlet and exit is greater than 10° C, then the moderate temperature variation assumption is invalid. However, for flows characterized by large property variations, Sieder and Tate correlation was used to calculate the heat transfer coefficient. Incropera and Dewitt (2006) [17] recommends:

$$h = 0.027\left(\frac{k_f}{D_i}\right) Re^{0.8} Pr^{1/3}\left(\frac{\mu}{\mu_s}\right)^{0.14} \quad (9)$$

0.7≤ Pr ≤ 16,700 Re_D ≥ 10,000 and L/D_i ≥ 10

Laminar flow:

Below a Reynolds number of about (2000), the flow in pipes will be considered to fall in the laminar region. Sieder and Tate (1930) [18] recommends the following simple correlation:

$$h = 1.86\,(Re\, Pr)^{0.33}\left(\frac{D_i}{L}\right)^{0.33}\left(\frac{\mu}{\mu_s}\right)^{0.14} \quad (10)$$

Here (L) is the length of the tube or conduits.

For steam condensation where the condensed fluid is water, a special correlation for water could be used for more accurate estimation for inside coefficient. Eagle and Ferguson (1930)[19] data bank was adapted by Sinnott (1999) [20] to develop specifically correlation for water as follows:

$$h_i = 4200\,(1.35 + 0.02\, t_m)\left(\frac{u_t{}^{0.8}}{D_i{}^{0.2}}\right) \quad (11)$$

Equation (11) is applicable for all range of Reynolds number. However, if the estimated inside heat transfer coefficient by this equation is large compared with another used correlation, as described above, the smallest value will be taken.

For noncircular tubes all the above equations may be applied by using an effective diameter as a characteristics length. It is termed as the hydraulic diameter and is defined as:

$$D_h = \frac{4\,A_c}{P} \tag{12}$$

Where A_c and P are the flow cross sectional area and wetted perimeter.

c) Air Side Heat Transfer

i. *Heat Transfer Coefficient*

Heat transfer performance of the tube bank is determined by flow pattern, which is strongly dependent on the arrangement of the tubes. The longitudinal tube spacing and transverse tube spacing could influence the thermal characteristics performance of heat exchanger. This has been studies by Tarrad et al. [6], Grame [21] and Jones [22].On the other hand fin spacing, fin thickness and fin height also affect the performance of finned tube heat exchangers as reported by Briggs (1963) [23] and Tarrad and Khodur(2015) [8]. In the present work, the forced convection heat transfer coefficient for the triangular fins could be predicted approximately using correlation of internal flow in ducts for laminar or turbulent flow as stated above. Reynolds number for air flow inside a triangle fin with using a hydraulic diameter approach is calculated as below:

$$Re = \frac{\rho\, u_{max}\, D_h}{\mu} \tag{13.a}$$

The velocity $u_{max}$is calculated at the minimum cross flow area S_m in tube bundle, hence:

$$u_{max} = \frac{\dot{m}_a}{S_m\,\rho_a} \tag{13.b}$$

In the present work minimum cross flow area was calculated according to the air side geometry as:

$$S_m = \frac{1}{2} F_p (X_T - H_t) \tag{13.c}$$

$$\dot{m}_{a,inc} = \frac{\dot{m}_a}{H_{inc}} \tag{13.d}$$

ii. *Steam Side Tube Characteristics*

The cross section of the ACC tube is as shown in figure (4.a).The sides of the tube are assumed to be semicircular with a diameter equal to the tube height.

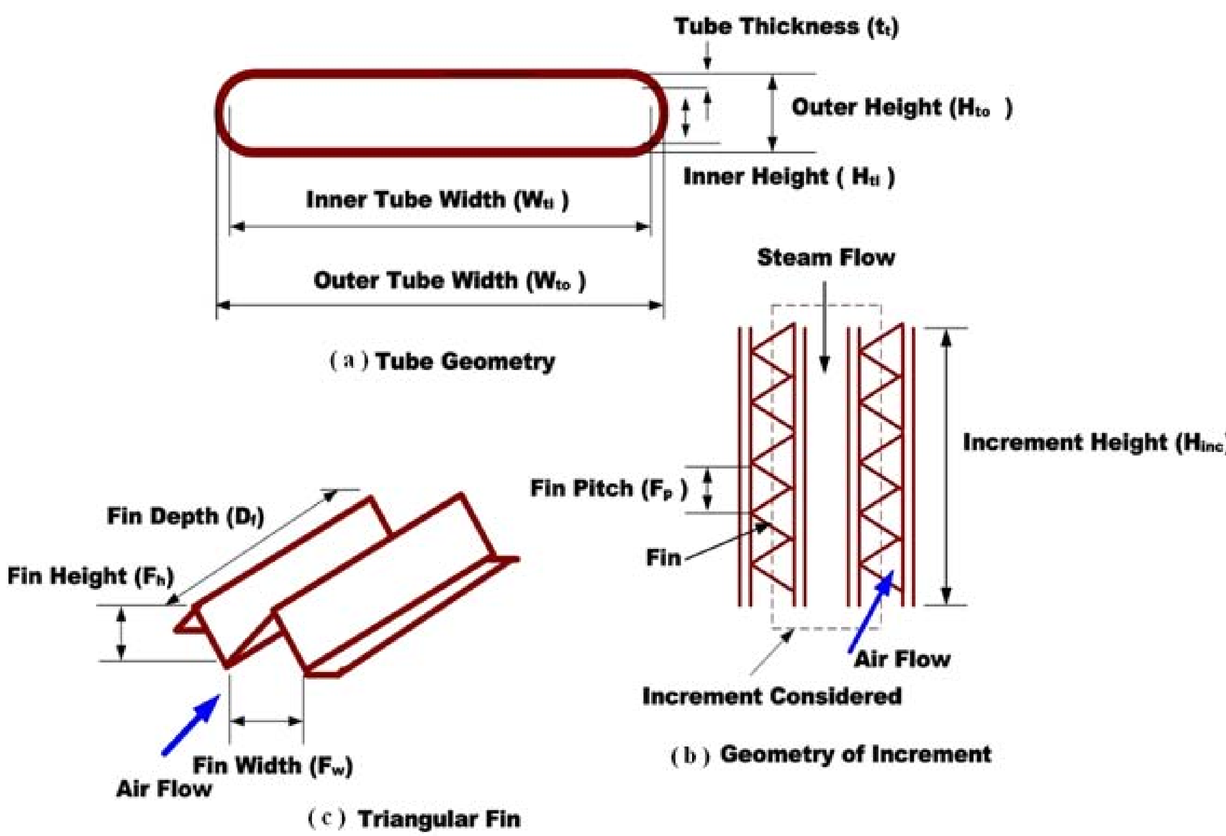

Figure (4) : Geometrical details of air cooled condenser

Therefore, the tube cross-sectional area is calculated as follows:

$$A_t = H_{t,i}(W_{t,i} - H_{t,i}) + \frac{\pi}{4} H_{t,i}^{\;2} \tag{14}$$

The tube wetted perimeter is the total tube perimeter in contact with the steam. It is given by:

$$P_t = \pi\, H_{t,i} + 2\,(W_{t,i} - H_{t,i}) \tag{15}$$

The hydraulic diameter is given by:

$$D_{h,t} = \frac{4\,A_t}{P_t} \tag{16}$$

The tube walls wetted by direct contact with steam transfer heat directly from the steam side to the outside air,hence the heat transfer area:

$$A_s = 2\left(W_{t,i} - H_{t,i}\right)L_t + \pi\, H_{t,i} L_t \tag{17}$$

Each fin attached to the tube with approximately (1 mm) thickness along the depth. The contact area is calculated as:

$$A_{att} = t_{att}\, D_{tube} \tag{18}$$

The total heat transfer area is given by:

$$A_{o,t} = A_s - A_{att} \tag{19}$$

iii. *Air Side Tube Characteristics*

The cross section of the tube outer surface is shown in figure (4.b). The face area of one tube and fin set in one increment, figure (4.c) is defined as:

$$A_{fa,inc} = \left(H_{tube,o} + F_h\right)H_{inc} \tag{20}$$

Therefore, the area blocked by the fins is given by:

$$A_{ba} = \frac{H_{inc}}{F_p}\left(l_f t_f + l_f t_f\right) \tag{21}$$

The area available for air flow is represented as the total area less than the area blocked by the fins together with the area occupied by the tube for steam flow as follows:

$$A_a = A_{fa,inc} - \left(A_{ba} + H_{tube,o} H_{inc}\right) \tag{22}$$

The perimeter of the tube which is directly in contact with air is given by:

$$P_a = \left[2\,\frac{H_{inc}}{F_p}\left(l_f - t_f\right) + \left(H_{inc} - H_{inc}\,\frac{t_f}{F_p}\right)\right] \tag{23}$$

The hydraulic diameter is:

$$D_{h,a} = \frac{4\,A_a}{P_a} \tag{24}$$

The tube wall which is in contact with steam on the inside surface and with air on the outside surface directly transfers heat from the steam to the outside air. This constitutes the heat transfer area:

$$A_{a,s} = \left[2\left(D_{t,o} - H_{t,o}\right)\left(1 - \frac{t_f}{F_p}\right) + \pi\, H_{t,o}\right] H_{inc} \tag{25}$$

The fins also confirm the heat transfer from the steam to air. Therefore fin surface area is defined as follows:

$$A_{f,s} = 2\,\frac{H_{inc}}{F_p}\left(l_f D_f\right) \tag{26}$$

The total heat transfer area become:

$$A_o = A_{a,s} + \eta_f A_{f,s} \tag{27}$$

Here η_f *is* the fin efficiency, when $\left(\frac{dT}{dx} = 0, x = l\right)$ then the efficiency is given by, Kreith (1999) [24]. In the present work, the fin is represented by a schematic diagram shown in figure (5), hence:

$$\eta_f = \frac{\tanh\, ml}{ml} \tag{28.a}$$

$$m = \sqrt{\frac{h_a P_a}{k_a A_a}} \tag{28.b}$$

Therefore, the total surface efficiency of the fin, η_o is therefore expressed as below:

$$\eta_o = 1 - \frac{A_f}{A_o}\left(1 - \eta_f\right) \tag{29}$$

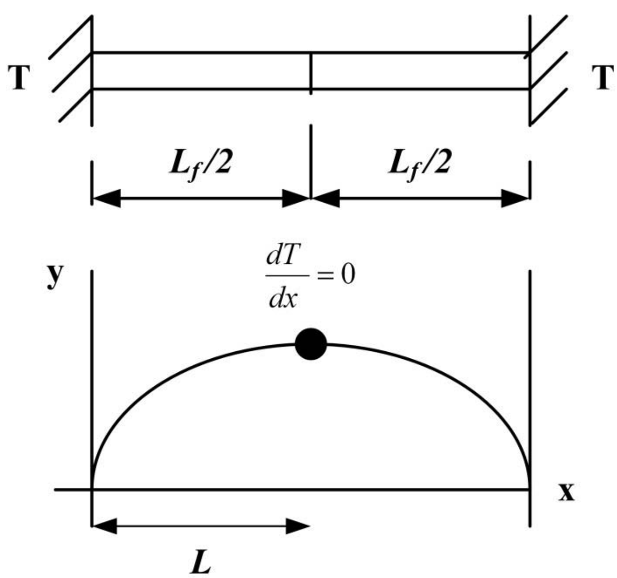

Figure (5) : Representation of fin midpoint temperature variation *d) NTU Effectiveness Relations:*

The NTU method offers more advantages for analyzing heat exchangers. It shows that iterative procedure is not required when inlet and outlet temperatures are unknown. For any heat exchanger, the total heat rejected from the hot fluid to the cold fluid is dependent on the heat exchanger effectiveness and also on the heat capacity of each fluid. This can be calculated as follows:

$$Q = \varepsilon \, c_{min} \left(T_{h,i} - T_{c,i}\right) \qquad (30.a)$$

The heat capacity, c, the extensive equivalent of the specific heat, determines the amount of heat a substance absorbs or rejects per unit temperature change, where:

$$c = \dot{m} \, cp \qquad (30.b)$$

The effectiveness is the ratio of the actual amount of heat transferred to the maximum possible amount of heat transferred and defined as:

$$\varepsilon = \frac{Q}{Q_{ma\,x}} \qquad (31.a)$$

In the present work, the heat duty of each increment is calculated by the effectiveness-NTU method. The effectiveness relation for single-phase fluid cross flow is given below, Holman (2002) [25]:

$$\varepsilon = 1 - exp\left[\frac{exp\,(-N\,C_r\,n) - 1}{C_r\,n}\right] \qquad (31.b)$$

And for Counter flow:

$$\varepsilon = \frac{1 - exp\,[-N\,(1 - C_r)]}{1 - C_r\,exp\,[-N\,(1 - C_r)]} \qquad (32.a)$$

Where:

$$n = N^{-0.22} \qquad (32.b)$$

$$C_r = \frac{c_{min}}{c_{max}} \qquad (32.c)$$

$$N = NTU = \frac{UA}{c_{min}} \qquad (32.d)$$

This is the effectiveness relationship for a cross flow, single-pass heat exchanger with both fluid unmixed. For tow-phase fluid flow, the effectiveness relation is:

$$\varepsilon = 1 - exp(-NTU) \qquad (33)$$

The overall heat transfer coefficient (U), takes into consideration the total thermal resistance to heat transfer between two fluids. Even though the convective heat transfer coefficients may be different on the two sides of the heat exchanger, the (UA) product is the same on either side. This is because all of heat taken from hot side must be transferred to the cold side. Overall heat transfer coefficient defined as,Sinnott (1999) [20]:

$$\frac{1}{U_o} = \frac{1}{h_o \eta_o} + \frac{1}{h_{fo,o} \eta_o} + \frac{D_o \ln\left(\frac{D_o}{D_i}\right)}{2\,k_{wall}} + \left(\frac{A_o}{A_i}\frac{1}{h_{fo\;i}\eta_i}\right) + \left(\frac{A_o}{A_i}\frac{1}{h_i \eta_i}\right) \qquad (34.a)$$

By neglecting the effect of fouling on both sides of the heat exchanger, and inside surface efficiency, the overall heat transfer coefficient is reduced to:

$$\frac{1}{U_o} = \frac{1}{h_o \eta_o} + \frac{D_o \ln\left(\frac{D_o}{D_i}\right)}{2\,k_{wall}} + \left(\frac{A_o}{A_i}\frac{1}{h_i}\right) \qquad (34.b)$$

e) Model Scheme

The suggested model in the present work was prepared in the form of a computer program designated as ACCRP. It was written with the aid of Liberty Basic Language, [26]. Multi iteration schemes were implemented to obtain the final rating data of the air cooled condenser. These were related to the air stream side and steam flow inside the tubes. A complete description and flow diagram of this code could be found in Altameemi [10].

IV. Model Verification

The experimental results are compared to the theoretical simulation results using the ACCRP computer program. The results showed a good agreement with the experimental data.The present model was verified by the implementation of the experimental data collected from [10] at atmospheric pressure. The model calculation scheme depends on the prediction of the air exit dry bulb temperature that on the lee side of the condenser under steady state conditions. Hence, this parameter was considered as an indication for the uncertainty percentage of the simulation process of the present model. Accordingly, the accuracy of the present row by row and step by step strategy is related to how far be the predicted air temperature and the condenser load from the experimental data. The uncertainty of each parameter and its discrepancy or scatter from the experimental data was estimated from:

$$\Phi^{\pm} = \frac{\Phi_{theoretical} - \Phi_{experimental}}{\Phi_{experimental}} \qquad (35)$$

Here Φ represents either the exit air temperature or the heat duty of the condenser.

a) Thermal Load

The experimental data collected in the present work showed its dependency on the air velocity and its temperature. Figure (6) shows the variation of the condenser load with entering temperature at two different velocities for a steam mass flow rate of (33.5) kg/hr. It is obvious that increasing air velocity or reducing the entering condenser air temperature will enhance the condenser load.Increasing the air velocity

Global Journal of Researches in Engineering (A) Volume XV Issue III Version I Year 2015

showed an increase in the condenser steam mass flow rate of about (17.3 %) and (14 %) at entering air dry-bulb temperature of (20.7°C) and (42°C) respectively. The corresponding condenser thermal load was improved by (1.21) and (1.18) times as shown in figure (6).

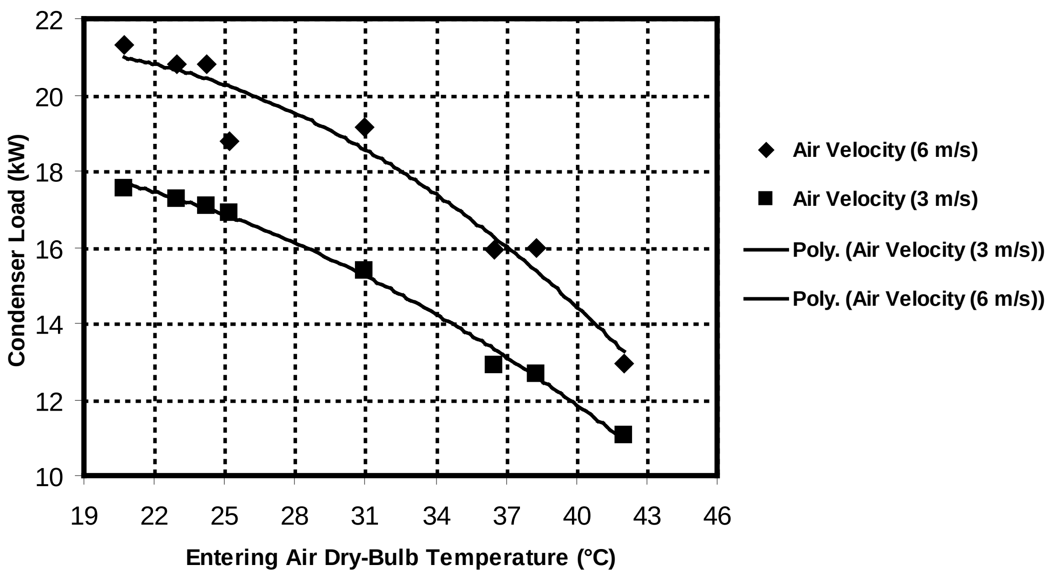

Figure (6) : Air cooled condenser load variation with entering air dry-bulb temperature

The ACC thermal loads for different experimental conditions exhibited a maximum deviation of (+8 %), and a minimum deviation of (-5 %) for air velocity of (3) m/s.The corresponding values for (6) m/s, the maximum discrepancy of (+ 11 %), as shown in figures (7) and (8). It is quite clear that the uncertainty in the model prediction was better at the low air velocity than that at the higher one.

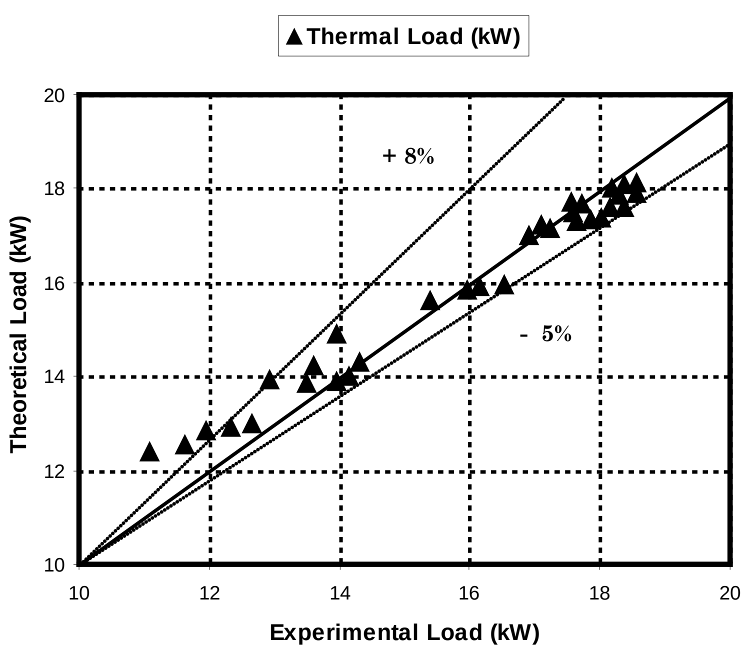

Figure (7) : Comparison of experimental and theoretical ACC thermal load at air velocity of (3) m/s

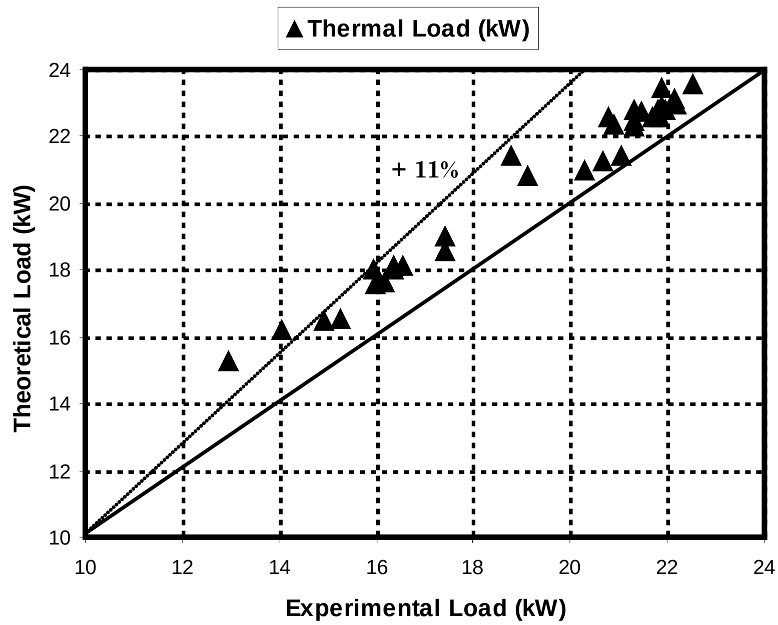

Figure (8) : Comparison of experimental and theoretical ACC thermal load at air velocity of (6) m/s

Figure (9) shows the comparison between the measured experimental heat duty and that predicted by the present modal for the entire operating conditions considered in the present work. The model predicted the condensation load with a high confidence and it lies with a scatter of (+12) % and (-5) % for more than (98) % of the data points.

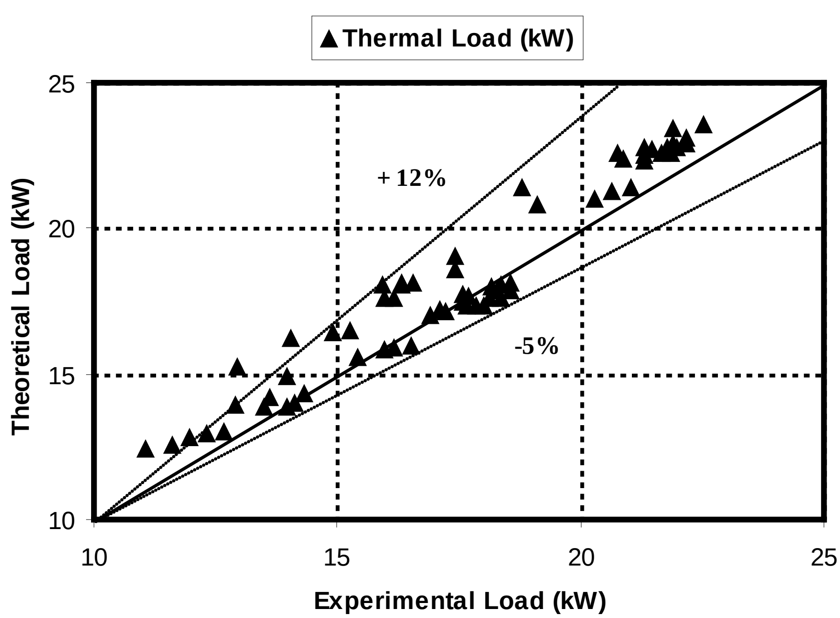

Figure (9) : Comparison of experimental and theoretical ACC thermal load at air velocity of (3) m/s and (6) m/s

b) *Exit Air Temperature*

Verification of the air exit temperature samples for (3 and 6) m/s are shown in figure (10). It is obvious that all of the results are under predicted by (5 %). The most attractive feature of the present model is its response to the variation of the operating conditions. These conditions are related to the condensation loading, air temperature and its flow rate.

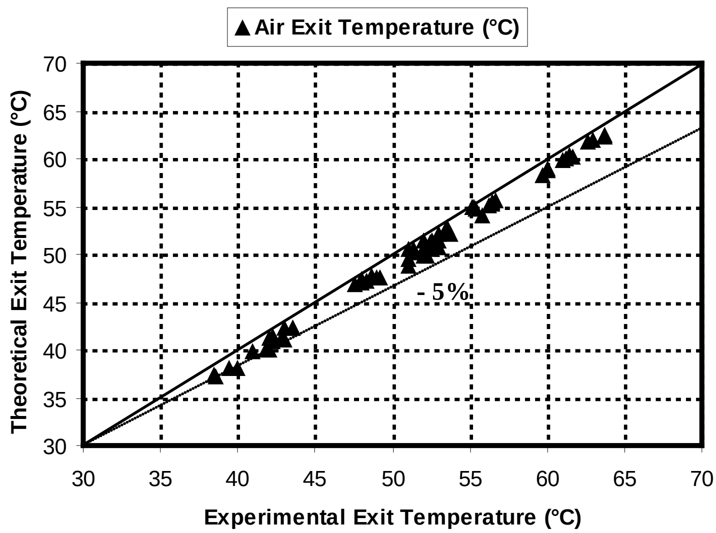

Figure (10) : Comparison of experimental and theoretical ACC exit dry-bulb temperature for both test air velocities at different steam flow rates

c) Overall Heat Transfer Coefficient Distribution

The distribution of the overall heat transfer coefficient with the tube length is illustrated in figures (11),(12) for the ACC under low and high air velocity respectively. The present model showed that the local overall heat transfer coefficient for air cooled condenser varies with increment and row position in the air direction towards the exit side of the condenser. The variation with increment position is due to the condensation heat transfer variation with vapor fraction at each section. The variation with row position is due to the variation in air side heattransfer coefficient in the air direction.

It is obvious that the choice of increasing the air flow rate and flow velocities with (100%) by fanning a higher air volumetric flow may be considered to achieve high heat transfer. This is due to the increaseof air side heat transfer coefficient, especially for hot ambient condition when performance in a credible manner was required.The air velocity is an important character in air cooled condenser overall design and determination of the face velocity. However it is usually falls within the range of (1.5) to (4) m/s, Tarrad (2010) [7]. It is recommended for the face velocity of the air a value which is not exceeding (3) m/s, for pressure drop consideration, Wilber and Zammit (2005) [27].

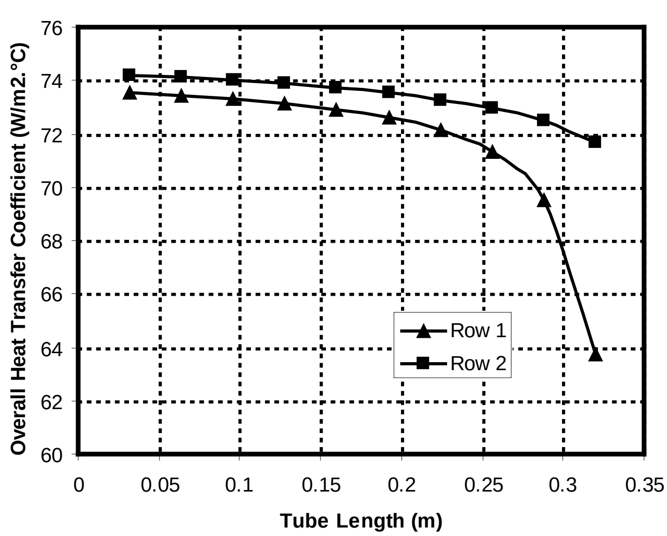

Figure (11) : Overall heat transfer coefficient distribution for ACC under air flow velocity of (3) m/s at inlet air temperature of (31°C) and steam flow rate of (33.4) kg/hr

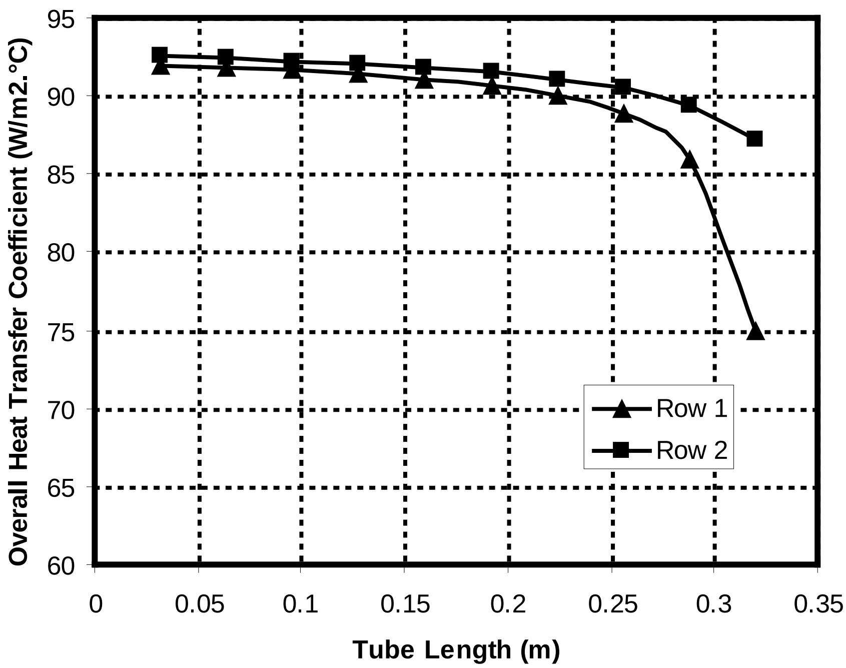

Figure (12) : Overall heat transfer coefficient distribution for ACC under air flow velocity of (6) m/s at inlet air temperature of (31 °C) and steam flow rate of (33.4) kg/hr

V. Conclusions

The present work revealed the following findings:

1. Thermal rating model for the ACC has been built successfully to predict the thermal load and temperature distributions across the heat exchanger. It showed excellent agreement between the experimental and predicted data of the exit air dry bulb temperature and condensation load.
2. The simulation model results showed that the local overall heat transfer coefficient (U_o) is changed slightly with the increment and row position. It is conservative to assume that (U_o) is of a constant value for each row.
3. When the air flow rate was doubled, the ACC average steam mass flow rate is increased by (17.5%) and the average condenser thermal load is increased by (17.6%) with air dry-bulb temperature reduction of (42 to 20.7) °C.

Nomenclature

A	Area	m ^{2}
C_r	Heat Capacity Ratio	-
cp	Specific Heat at Constant Pressure	J/kg.°C
D	Diameter or Depth	m
F_p	Fin Pitch	m
g	Gravitational Acceleration	m/s ^{2}
h	Specific Enthalpy	kJ/kg
h_c	Steam Condensing Heat Transfer Coefficient	W/m ^{2} °C
h_o	Outside Heat Transfer Coefficient	W/m ^{2} °C
H	Height	m
k	Thermal Conductivity	W/m °C
L	Tube Length	m
l_f	Fin Length	m
$\dot{m}$	Fluid Mass Flow Rate	kg/s
n	Parameter Defined by Equation (32.b), Correlation index	-
N	Number	-
Nu	Nusselt Number	-
p	Operating Absolute Pressure of Steam	bar
P	Perimeter of Tube	m
Pr	Prandtle Number	-
$\dot{Q}$	Condensation Load	kW
Re	Reynolds Number	-
S_m	Hollow Fin Cross Sectional Area	m ^{2}
t	Thickness	m
T	Fluid Temperature	°C or K
ΔT	Temperature Difference	°C
u	Fluid Velocity	m/s
U_o	Overall Heat Transfer Coefficient	W/m ^{2} °C
W	Air Cooled Condenser Width	m
x_{eq}	Steam Local Quality	-
X_T	Transverse Tube Pitch to the Flow Direction	m
X_L	Longitudinal Tube Pith in Flow Direction	m
z	Heat Exchanger Cooled Length	m

Greek Symbols

η_f Fin Efficiency

η_o Total Surface efficiency

ε Effectiveness of the Condenser

ρ Density (kg/m ^{3})

μ Viscosity (kg/m.s)

Φ Parameter defined by eq. (35)

Subscripts

a Air
ass. Assumed Value
att. Attached
c Condensate , cold
cBK Boyko-Kruzhilin condensation coefficient
cal. Calculated Value
cond Condenser, Condensation
e Exit or Equivalent
eff. Effective
eq. Equivalent
est. Estimated Value
f Fouling , Fin or fluid
g gas
h Hydraulic diameter, hot fluid
i Inside or Inlet
inc increment
l Liquid
lo Liquid Only
m mean
max maximum
min minimum
o Outside
r ratio,row
row Row Value
s Steam Value or Surface
t Tube
tr tube per row
Tube Tube Value
ν Vapor
w Wall Value

Abbreviations

ACC	Air Cooled Condenser
ACCRP	Air Cooled Condenser Rating Program
NTU	Number of Heat Transfer Unit
OHTC	Overall Heat Transfer Coefficient

References Références Referencias

1. Kutscher C. and Costenaro D., "Assessment of Evaporative Cooling Enhancement Methods for Air-Cooled Geothermal Power Plants", presented at the Geothermal Resources Council, (GRC) ,Annual Meeting Reno, Nevada, September, (2002).
2. SaizJabarrdo, J.M., and Mamani,W.G.,"Modeling and Experimental Evaluation of Parallel Flow Micro Channel Condensers", J.Braz., Soc., Mech., Sci and Eng., Vol.25, No.2, Reo de Janeiro, Apr./June, 2003.
3. Gadhamshetty, V., Nirmalakhandan, N., Myint, M. and Ricketts, C.," Improving Air-Cooled Condenser Performance in Combined Cycle Power Plants", Journal of Energy Engineering, Vol.132,No.2,August 1, (2006).
4. Tarrad A. H., Shehhab U. S.,"The Prediction of Environment Effect on the Performance of a Vapour Compression Refrigeration System in Air Conditioning Application", Journal of Engineering and Development. Vol.11, No.1, March 2007, pp.169-189.
5. Tarrad, A. H., Khudor, D. S., and Abdul Wahed, M.," A Simplified Model for the Prediction of the Thermal Performance for Cross Flow Air Cooled Heat Exchangers with a New Air Side Thermal Correlation ", Journal of Engineering and Development, Vol. 12, No.3, 2008.
6. Tarrad, A. H., Saleh, F. A., and Abdulrasool, A.A., "A Simplified Numerical Model for a Flat Continuous Triangle Fins Air Cooled Heat Exchanger Using a Step by Step Technique", Engineering and Development Journal, Vol. 13, No. 3, pp. 38-59, Al-Mustansiriya University, Baghdad, (2009).
7. Tarrad, A.H.,"A Numerical Model for Performance Prediction of Dry Cooling Conditions of Air Cooled Condensers in Thermal Power Plant Stations", Journal of Engineering and Technology, Vol.28, No.16, Baghdad, Iraq, (2010).
8. Tarrad, A. H. and Khudor, D. S., "A Correlation for the Air Side Heat Transfer Coefficient Assessment in Continuous Flat Plate Finned Heat Exchangers", Transactions of the ASME, Journal of Thermal Science and Application, Vol. (7), No. 2, Paper No. TSEA-14-1194, DOI: 10.11151/1.4029459, 2015.
9. Tarrad, A. H. and Al-Nadawi, A. K., "A Rating Model of Finned-Tube Condenser using Pure and Zeotropic Blend Refrigerants", accepted for publication in German Journal of Mechanical Engineering Research, 2015.
10. Altameemi, A. F., "Study and evaluation of the operation characteristics for the condensation load distribution in hybrid systems on the condenser side", MSc. Thesis, Al-Mustansiriya University, Baghdad, June 2011.
11. Jana, W.S.,"Engineering Heat Transfer", 2nd edition, CRC press, LLc, USA, 2000.
12. Jung D., D., and N. Assanis," Numerical Modeling of Cross Flow Compact Heat Exchanger with Louvered Fins using Thermal Resistance Concept", SAE. Paper, 2006-01-0726, University of Michigan, (2006).
13. Boyko-Kruzhilin, "Heat Mass Transfer", Inc. J., Vol.10, pp. 361, 1967.
14. Sinnott, R.K.,Chemical Engineering, Vol.6, Heaton, A.W. and Co. LTD, Exeter, (2005).
15. Robert, H.P. and Green, D.W.,"Perry's Chemical Engineering Hand Book ", 8th Edition, McGraw-Hill Company, New York, (1999).
16. Incropera, F. P. and DeWitt, D. P., "Fundamentals of Heat and Mass Transfer", 4th Edition, John Wiley & Sons, New York, (1996).
17. Incropera, F. P. and DeWitt, D. P., "Fundamentals of Heat and Mass Transfer", 6th Edition, John Wiley & Sons, New York, (2006).
18. Sieder, E.N., and Tate, C.E.," Heat Transfer and Pressure Drop of Liquids in Tubes", Ind. Eng. Chem., Vol.28, p.1429, 1936.
19. Eagle, A., and Freguson, R.M.," On the coefficient of Heat Transfer from the Internal Surface of Tube Walls", Proc.R. Soc. Lond. A June 2, 1930.
20. Sinnott, R.K.,Chemical Engineering, Vol.6, Heaton, A.W. and Co. LTD, Exeter, (1999).
21. Gram, A.J., Mackey, C.O., and Monroe, E.S. Jr., "Convective Heat Transfer Banks, II- Correlation of Data for Tem Row Deep Tube Bank", ASME Trans., Vol.80, p.25, 1958.
22. Jones, C.E., and Monroe, E.S. Jr.," Convective Heat Transfer and Pressure Drop of Air Flowing across In-Line Tube Banks, I- Apparatus, Procedure and Special Effects", ASME Trans., Vol.80, p.18, 1958.
23. Briggs, D.E., and Young, E.H.," Convective Heat Transfer and Pressure Drop of Air Flowing across Triangular Pitch Banks of Finned Tubes", Chem. Eng. Progress Symposium Series- Heat Transfer, Vol.59, No.41, pp.1-10, 1963.
24. Kreith, F., "Thermal Engineering", CRC press, LLc, USA, 1999.
25. Holman, J. P., "Heat Transfer", 9th Edition, McGraw-Hill Book Company, (2002).
26. Liberty Basic Program, V4.04, Modern Basic Language to Produce Custom Windows Programs, available from World Wide Web: www.libertybasic.com .
27. Wilber,K. R. and Zammit, K.,"Development of Procurement Guidelines for Air-Cooled Condensers", CEC/EPRI, Advanced Cooling Strategies/Technologies Conference, Sacramento, California, (2005).