THE APPLICATION OF A STEP BY STEP TECHNIQUE FOR THE PERFORMANCE PREDICTION OF THERMAL POWER PLANT SURFACE CONDENSERS

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ABSTRACT:

In the present work, the step by step technique is used to predict the performance of the power plant condenser. The procedure includes the calculation of pressure distribution, condensation temperature, water inlet and outlet temperature, condensation load distribution and single phase heat transfer sub-cooling. A quasi two dimensions model is applied, one in the tube water direction and the other in the vapor direction. It is applied with different operating conditions of the condenser such as operating pressure, air percentage mixed with steam, cooling water temperature and fouling factor on tube side for summer and winter seasons.

The present model revealed that the fouling resistance has a great effect and plays the major part of the decline in the condenser performance. This is because it decreases the overall heat transfer coefficient and condensation rate. The performance of the condenser when the cooling water enters at the lower pass is better than the upper pass, due to the balance in the distribution of the heat transfer and the condensation rate between tube passes. The model is verified with field operating conditions of Southern Baghdad thermal power station. It has revealed that there is a good agreement between the field data and the present technique. The accuracy fell within **(98) %** and **(89) %** for the cooling water temperature prediction for summer and winter respectively, while it showed accuracy of **(98) %** and **(99) %** for the condensate exit temperature prediction for summer and winter respectively.

استخدام تقنية الخطوة - خطوة للتنبؤ بأداء المكثفات المستخدمة في محطات الطاقة الحرارية

الخلاصة

يتضمن البحث الحالي تطبيق طريقة الخطوة-خطوة للتنبؤ بأداء المكثفات المستخدمة في المحطات الحرارية لإنتاج الطاقة الكهربائية. الطريقة المقترحة تتضمن حساب توزيع الضغط ، درجة حرارة التكثيف ، درجة حرارة الدخول والخروج لماء التبريد، توزيع معدل التكثيف بالإضافة إلى انتقال الحرارة للطور الواحد لدرجة أقل من درجة حرارة التشبع. تم تطبيق نموذج ذو بعدين ، أحدهما باتجاه جريان ماء التبريد والأخر باتجاه

انسياب البخار. ظروف عمل مختلفة للمكثف تم اعتمادها مثل ضغط التشغيل ، النسبة المئوية للهواء المتسرب مع البخار ، درجة حرارة ماء التبريد عند الدخول للمكثف بالإضافة لمعامل الاتساخ لجانب الأنابيب لفصلي الصيف والشتاء.

لقد بين النموذج الحالي بان مقاومة الاتساخ الحرارية لها تأثيراً كبيراً وتلعب دوراً أساسياً في تدهور أداء هذه المكثفات. إن هذه المقاومة الحرارية تؤدي إلى تقليل معامل انتقال الحرارة الكلي و معدل تكثيف البخار في المكثف. أداء المكثف يكون أفضل عندما يدخل ماء التبريد إلى المكثف في الممر السفلي لكونه يؤدي إلى توزيع متجانس لمعدل التكثيف والحمل الحراري على ممرات ماء التبريد للمكثف. لقد تم التحقق من صحة النموذج الحالي باستخدام الظروف التشغيلية لمحطة كهرباء جنوب بغداد الحرارية. لقد بينت الدراسة توافق جيد بين القراءات التشغيلية وتلك المحسوبة بالعمل الحالي ، وبدرجة دقة تتراوح بين **(98)%** و **(89)%** لدرجة ماء التبريد لكل من فصل الصيف والشتاء على التعاقب. في حين كانت النتائج لدرجة حرارة بخار الماء المتكثف المحسوبة بالنموذج الحالي تقع ضمن المدى **(98)%** و **(99)%** لكل من فصل الصيف والشتاء على التعاقب.

KEY WORDS: Iterative, Power Plants, Surface Condenser, Performance, Prediction

INTRODUCTION:

The condenser is a device in which the vapor is converted to liquid and latent heat is transferred to coolant. A common type of condensers is indirect contact condensers in which vapor does not mix with the cooling fluid, the shell and tube type is the common one and the most widely used.

(**Jacob, 1959**) introduced the mean heat transfer coefficient for a vertical column of (N) horizontal tubes with the same temperature difference. The influence of the drag extended by the vapor on the condensate film was considered by (**Chen, 1961**), and (**Koh et al., 1961**), from numerical solution of the governing equations. (**Chisholm et al., 1965**), developed a numerical method of evaluation heat and mass transfer coefficient and local heat fluxes in surface condensers. (**Patankar and Spalding, 1974**), introduced a more practical approach by considering the tube nest as porous medium allowing coarser computational grid to be used and hence economize on computer requirements. (**Fujii, 1983**), treated condensation phenomenon in small simple tube banks as a basic problem for research and development on turbine condensers.

(**Zhang, 1994**), proposed quasi-three dimension numerical model to predict performance of large power plant condensers. The prediction was achieved by solving the governing mass, momentum and air concentration by using semi implicit consistent control-volume for simulation with different conditions in work of condenser. A test facility was constructed by (**McNeil, 1999**), and used to generate data for filmwise and dropwise condensation from steam and steam-air mixtures flowing downward across a bundle of tubes. Pressure drops in a dropwise bundle are not noticeably different from a filmwise bundle.

(**Karl and Hein, 1999**) found that the non-condensable gas accumulates in the vapor phase boundary layer and causes a high heat transfer resistance, especially with high pressures and low water temperatures. (**Seungmin, 2003**), examined numerically the annular filmwise condensation of vapors in a vertical tube with non-condensable gases. (**Liang et al., 2004**), presents accurate numerical solutions of the full two-dimensional governing equations for steady and unsteady laminar/laminar internal condensing flow. (**Tarrad and Kamal, 2004**), studied the performance prediction of Al-Daura steam power plant condensers, **Fig. 1**, in a quasi-two dimensional model. It revealed that cooling water resistance represents the greatest one among the whole resistances.

In the present work a step by step technique was used for the performance prediction of an existing power plant surface condenser. It provides a powerful tool for the performance assessment of the condenser working under various operation conditions at which the different measures that control the proper work of such equipments were investigated.

THEORETICAL TREATMENT MODEL:

The overall heat transfer coefficient that refers to the outside tube area may be expressed as, (**Davidson,1987**).:

$$U = \frac{1}{\dfrac{d_o}{d_i\ h_i} + \dfrac{d_o\ \ln\left(\dfrac{d_o}{d_i}\right)}{2\ k_w} + R_f + \dfrac{1}{h_c} + \dfrac{1}{h_v}} \qquad (1)$$

The heat flux at the condenser tube surface is calculated from:

$$Q = U\ LMTD \qquad (2)$$

FORCED CONVECTION IN TUBES:

Numerous relations have been proposed for predicting turbulent flow in tubes. The most popular correlation available for the prediction of the heat transfer coefficient is that of Petukhov cited in (**Cengel, 1998**) in the form:

$$Nu_i = \frac{Re_i Pr_i \left(\frac{\Gamma}{8}\right)\zeta}{1.07 + 12.7\left(Pr_i^{2/3} - 1\right)\sqrt{\frac{\Gamma}{8}}} \qquad (3.a)$$

$$Nu_i = \frac{h_i\ d_i}{k_i}$$

$$\zeta = \left(\frac{\mu_{ib}}{\mu_{iw}}\right)^{0.11}$$

$$\Gamma = (1.82\ log(Re_i) - 1.642)^{-2} \qquad (3.b)$$

For the Reynolds and Prandtl number ranges:

$10^4 \le Re_i \le 5\times10^6$ and $0.5 \le Pr_i \le 2000$

CONDENSATION HEAT TRANSFER COEFFICIENT:

The latent heat of condensation released as the vapor condenses must pass through a liquid film resistance before it reaches the solid surface and must be transferred to the medium on the other side.

NUSSELT EQUATION FOR A LAMINAR FILM:

The following relation for the heat transfer coefficient can be obtained from, (**Kreith and Boehm, 1999**) by the following expression:

$$h_{CN} = 0.925 \left(\frac{\rho_c \, (\rho_c - \rho_v) \, h_{fg} \, g \, k_c{}^3}{L_o \, \mu_c \, (T_{cs} - T_w)} \right)^{1/4} \tag{4}$$

With slight modification, the Nusselt analysis of laminar falling-film condensation over a flat plate can be adapted to film condensation isothermal horizontal cylinder. Doing so yields the following relation for the mean heat transfer coefficient as, (**Thome, 2004**):

$$h_{CN} = 0.728 \left(\frac{\rho_c \left(\rho_c - \rho_v\right) h_{fg} \, g \, k_c^3}{d_o \, \mu_c \left(T_{cs} - T_w\right)} \right)^{1/4} \tag{5}$$

Many investigators have made significant improvements to the original Nusselt theory to include the following effects to the condensation process:

Effect of Condensate Film Sub-cooling:

It has been shown that the cooling of the liquid below the saturation temperature during the condensation process can be accounted by using modified latent heat of condensation defined by (**Petukhov, 1998**) and (**Collier, 1972**) as:

$$h_{fg}^* = h_{fg} + 0.68 \, Cp_c \left(T_{cs} - T_w\right) \tag{6}$$

EFFECT OF VAPOR VELOCITY:

When the vapor surrounding a horizontal tube is moving at a high velocity, the analysis of filmwise condensation is affected by the surface shear and the vapor separation and its influence upon the condensate flow, (**Thome, 2004**). In practice, shear stress will be increased and the film is thinned, due to momentum transferred to the condensing vapor.

(**Shekriladze and Gomelauri, 1966**) analyzed the case of downward vapor flow over an isolated tube. They obtained an expression for which the vapors shear dominated conditions:

$$h_c = 0.9 \, \mathrm{Re}_{TP}^{1/2} \left(\frac{k_c}{d_o} \right) \tag{7.a}$$

$$\mathrm{Re}_{TP} = \frac{\rho_c\, u_\infty\, d_o}{\mu_c} \tag{7.b}$$

When gravity and vapor shear dominated conditions, the expression will be as:

$$h_c = 0.64\left[1 + \left(1 + 1.69\, F\right)^{1/2}\right]^{1/2} \mathrm{Re}_{TP}^{1/2} \left(\frac{k_c}{d_o}\right) \tag{8.a}$$

$$F = \frac{\mu_c\, h_{fg}^{*}\, d_o\, g}{u_\infty^2\, k_c\left(T_{cs} - T_w\right)} \tag{8.b}$$

(**Rose, 1984**) resolved the (**Shekrladze and Gomelauri, 1966**) problem and obtained an improved equation (without taking the effect of separation and circumferential pressure gradient for condensate) as:

$$h_c = \frac{0.9 + 0.728\, F^{1/2}}{\left(1 + 3.44\, F^{1/2} + F\right)^{1/4}} \mathrm{Re}_{TP}^{1/2} \left(\frac{k_c}{d_o}\right) \tag{9}$$

(**Fujii and Kurata, 1972**) modified eq. (9) to include a (generally small) correction for the fact that asymptotic shear stress expression has been used, thus:

$$h_c = \frac{0.9\left(1 + G\right)^{1/3} + 0.728\, F^{1/2}}{\left(1 + 3.44\, F^{1/2} + F\right)^{1/4}} \mathrm{Re}_{TP}^{1/2} \left(\frac{k_c}{d_o}\right) \tag{10.a}$$

$$G = \frac{\mu_c\, h_{fg}^{*}}{\left(T_{cs} - T_w\right) k_c} \left(\frac{\rho_v\, \mu_v}{\rho_c\, \mu_c}\right)^{1/2} \tag{10.b}$$

(**Fujii, 1981**) proposed the following experimental equations for the mean heat transfer coefficient for downward flow vapor as:

$$h_c = a\, F^{b}\, \mathrm{Re}_{TP}^{1/2} \left(\frac{k_c}{d_o}\right) \tag{11}$$

$$a = 0.96 \quad \mathrm{b} = 0.2 \qquad 0.03 \le \mathrm{F} \le 600$$

$$\mathrm{a} = 0.7 \quad \mathrm{b} = 0.25 \qquad \mathrm{F} \ge 600$$

(**Berman and Tumanov, 1962**) proposed from vertical down flow experimental data, corrective to Nusselt equation to take into account the effect of vapor shear as:

$$h_c = h_{CN}\left(1 + 0.0095\left(\mathrm{Re}_{mix}\right)^{\frac{11.8}{\sqrt{Nu_{CN}}}}\right) \tag{12.a}$$

This equation is restricted for $\dfrac{h_c}{h_{CN}} \le 1.5$

and

$$Nu_{CN} = \frac{h_{CN}\ d_o}{k_c} \tag{12.b}$$

$$\mathrm{Re}_{mix} = \frac{\rho_{mix}\ u_{\infty}\ d_o}{\mu_{mix}} \tag{12.c}$$

EFFECT OF INUNDATION IN TUBE BANKS:

The average heat transfer coefficient at the lower tube rows is smaller than that experienced from the above tube rows, (**Marto, 1988**).

(**Fuks, 1957**) and (**Kutateladze et al. ,1979**) derived a non-dimensional equation which accounted for the predominant physical mechanism as:

$$\frac{h_{c,n}}{h_{c,1}} = \left(\frac{\sum_{\mathrm{i=1}}^{\mathrm{n}} m_{\mathrm{cr,i}}}{m_{cr,n}} \right)^{-s} \tag{13}$$

(**Fuks, 1957**) showed that the index **S=0.07**, other authors as (**Wilson, 1972**) used **S=0.16**, (**Grant and Osment, 1968**) used **S=0.223** and (**Short and Brown, 1951**) used **S=0.25**.

Effect of Non-condensable Gasses:

Since only the vapor is condensed, the concentration of the non-condensable gas at the interface is higher than its value in the far ambient. This, in turn, decreases the partial pressure of the vapor at the interface below its ambient value. The resulting depression of the interface temperature generally reduces the condensation heat transfer rate below that which would result for pure vapor alone under the same conditions, (**Owen and Lee, 1983**).

(**Rose, 1980**) used procedure, which relies on a heat mass transfer analogy to obtain solutions for the corresponding mass transfer problem. The coefficient of mass transfer is evaluated from (**Rose, 1980**) in the form:

$$Sh = a_2\ \mathrm{Re}_{\mathrm{mix}}^{0.5}\ \prod^{-1/3}\ E_{\mathrm{v}}^{-\mathrm{b}_2} \tag{14.a}$$

$$\mathrm{a}_2 = 0.82 \qquad \mathrm{b}_2 = 0.6 \qquad \mathrm{Re}_{\mathrm{mix}} \geq 350$$
$$\mathrm{a}_2 = 0.52 \qquad \mathrm{b}_2 = 0.7 \qquad \mathrm{Re}_{\mathrm{mix}} \leq 350$$

$$\prod = \frac{P_v - P_{cs}}{P_{mix}} \tag{14.b}$$

$$E_v = \frac{P_a}{P_{mix}} \qquad (14.c)$$

The partial pressure of vapor is evaluated by:

$$\frac{P_{mix}}{P_v} = 1 + 0.622\,\frac{E_w}{1 - E_w} \qquad (15.a)$$

$$E_w = \frac{m_a}{m_v + m_a} \qquad (15.b)$$

By taking

$$Q = m_c\, h^*_{fg} \qquad (16)$$

(**Rose, 1980**) showed that:

$$Q = h^*_{fg}\, \rho_{mix}\, \frac{D}{d_o} \left(\frac{w_{cs} - w_\infty}{w_{cs}} \right) Sh \qquad (17)$$

And the heat transfer rate in eq. (16) may also be expressed as:

$$Q = {h^*_{fg}}^2\, \rho_v\, \frac{D_p}{d_o} \left(\frac{T_v - T_{cs}}{T_v} \right) Sh \qquad (18)$$

And the final expression for the heat transfer coefficient of the non-condensable gas, air, in the condenser has the form:

$$h_v = a_2 \frac{D_p}{d_o}\, Re_{mix}^{\;0.5} E_v^{\;-b_2}\, P_{mix}^{\;1/3} \left[\frac{\rho_v}{T_v}\right]^{2/3} {h^*}_{fg}^{\;5/3} \frac{1}{(T_v - T_{cs})^{1/3}} \qquad (19)$$

COMPUTATIONAL PROCEDURE:

The present work includes the prediction of the performance of the surface condenser used in the Southern Baghdad thermal power station. The object is to study the rating of an existing condenser and predict the total condensation load, condensation rate, pressure and temperature distribution on the steam side. The condenser is divided into a number of horizontal tube slices, rows, receive the steam and condensate from the above tube row. The first tube row is assumed to receive the steam directly from the exhaust turbine duct at its condition including mass flow rate, pressure and temperature. The condensation rate, vapor pressure, vapor temperature and overall heat transfer coefficient were calculated for each row.

The solution for the condenser rating was conducted a row by row, the exit condition from a tube row is assumed to be the entering condition to the next row and so on until the exit from the last tube slice. These calculations were based on an assumed arbitrary mass flow rate distribution for the bays along the heat exchanger. An iterative procedure was followed to establish the rate of condensation for each bay depending on its ability for condensation which is a function of the tube side condition and position. A computer program is built to establish the performance of power plant condenser incorporating all different variables involved in the performance prediction. Detailed computer program structure and solution marching procedure for the suggested model is presented by (**Majeed, 2007**).

CASE STUDY:

The case study considered for verification of the present model is the condenser design and operating conditions of the unit number (1) of the Southern Baghdad thermal station. The geometrical design characteristics of this condenser are shown in **Table 1** and the condenser tubes layout is shown in **Fig. 2**. For the present work the following cases will be considered:

- Case (1):- The data of this case is shown in **Table 2** and is taken in summer.
- Case (2):- The data of this case is shown in **Table 2** and is taken in winter.
- Case (3):- all data of this case are the same as case (1) except mass flow rate of air which is as (m_a=0.0092) kg/s as obtained from the field data of the manufacturer company.
- Case (4):- all data of this case are the same as case (1) except that the temperature of vapor inlet to condenser which is as (316.2) K corresponding to saturation pressure of (8644) Pa in summer season.
- Case (5):- all data of this case are the same as case (1) except the temperature of water inlet to condenser which is as (290.2) K.
- Case (6):- all data of this case are the same as case (1) except the fouling resistance and roughness as (R_f =0.00032) m^2.K/W and (ε=0.000036) m respectively as deduced from literatures.
- Case (7):- all data of this case are the same as case (1) except the water inlet to upper pass first.

RESULTS AND DISCUSSION:

Figure 3 shows the scheme of the longitudinal tube bundle arrangement for bay by bay condenser design. In all cases of performance prediction, Petukhov eq. (3) is used to calculate heat transfer coefficient inside tube, Fujii and Kurata eq. (10) to calculate condensate heat transfer coefficient and Rose eq. (19) to calculate air heat transfer coefficient.

Heat Load (*Q*):

As shown in **Figures 4** and **5**, the trend of the curves of heat transfer is the same for upper pass in case (1) and case (2), but the magnitude of reduction in heat transfer between rows is different according to pressure drop. For example, in case (1), the difference between rows in upper pass is (1163-1745) W and (2663-5003) W. For the lower pass, the trend of these curves is the same, but the magnitude of reduction between the first row in lower pass and the vent is different according to pressure drop. The extreme drop in curves for lower passes in the rows number (8) and (9). This is because that the number of tubes in these rows which are (19) and (20) are less than that of the other rows. This makes the quantity of cooling water less than in other rows, which explains the heat transfer decrease. As the pressure of vapor decreases towards the vent, the heat transfer rate of the tube rows and bays decreases. This can be explained by the decrease of temperature difference between vapor and cooling water which causes a decrease in heat transfer transmitted.

The variation of the condenser load with bay number for case (2) is shown in **Fig. 6**. The greatest heat transfer occurs in bay (11) for upper pass and bay (1) in lower pass because it has the lowest cooling water temperature, as compared with other bays, which leads to high temperature difference between vapor and cooling water and, hence, an increase in the heat transfer. The heat transfer rate in the lower pass is greater than that in the upper pass because of the coolest water temperature inlets to lower pass, except in three last bays in case (1) and four bays in case (2) since the temperature of cooling water is approximately equal, while the temperature of vapor in upper pass is greater than that in the lower pass.

As shown in **Fig. 7**, the trend of the curves of case (7) is opposite to those of case (1), because the cooling water enters the condenser at the upper pass which means high temperature difference between vapor and cooling water. This explains why the heat transfer rate in upper pass is higher than that in lower pass. On the other hand, the total heat transfer rate of case (7) is less than that in case (1) in all bays because the heat transfer rate in lower pass is lower than that in case (1). This can be explained by the small temperature difference between vapor and cooling water.

OVERALL HEAT TRANSFER COEFFICIENT (*U*):

As shown in **Figures 8, 9,** and **10**, the trend of curves is the same for all cases. Although the heat transfer rate in the upper pass is less than in lower pass, but the overall heat transfer coefficient in upper pass is higher than that in the lower pass. That is for a specified surface area, the heat flux, $U\Delta T_m$, for the upper pass is lower than that of the lower pass due to the decrease in temperature difference between vapor and cooling water. For case (7), the overall heat transfer coefficient, as shown in **Fig. 11**, increases from first bay to the last in upper and lower passes which is different from other cases. This is because the increase in heat transfer is more pronounced than the increase in temperature difference between vapor and cooling water. For example, the difference in overall heat transfer coefficient between bays for upper and lower passes of case (1) is (11.1-17.3) $W/m^2.K$ and for case (7) is (7.2-12.8) $W/m^2.K$.

The Rose correlation showed the highest overall heat transfer coefficient, whereas the **(Shekriladze and Gomelauri, 1966)** one expressed the lowest value, with a maximum corresponding deviation is about **(5) %**. The rest of correlations revealed a discrepancy in the range **(2-5) %** when compared to the minimum value given by the former correlation.

The results of the present work revealed that the cooling water resistance comprises the major part of the total resistance, it is within **(50-65) %**. The condensation resistance is ranged between **(15-20) %** which is followed by the fouling resistance with a percentage of **(9-15) %**. The mixture resistance showed the lower value among the various resistance types with a value fell within **(0.01) %**.

CONDENSATION RATE AND OTHER VARIABLES:

Condensation Rate (**m_{cr}**)**:** For case (1) and for upper pass, the greatest condensation rate occurs in bay (11), as shown in **Fig. 12**, because it has the lowest cooling water inlet to the bay compared with other bays. This leads to high temperature difference between vapor and cooling water which increases heat transfer rate. The trend of curves shows the reduction from inlet to tube bundle towards the last row in upper pass. This reduction is because of pressure drop through the rows. For example, the difference in upper pass is in the range of (0.0005-0.0016) kg/s.

Vapor Pressure (**P_v**)**:** The pressure drop in upper pass for all cases is greater than that of the lower pass. This is because the steam sustains the greatest velocity in the upper pass, whereas a sharp decrease in the steam velocity occurs in the lower pass. **Fig**. **13** shows extreme slope for pressure drop between bays. This can be explained by the large difference in steam velocity between bays. For example, in case (1), the predicted pressure drop for the upper pass was between (8.7-6.8) Pa and for the lower pass was between (1.5-.8) Pa, while in case (2) the pressure drop for the upper pass was between (12.8-8.8) Pa and for the lower pass was between (1.5-0.4) Pa.

Inlet Temperature of Cooling Water (T_i)

As shown in **Fig. 14**, the rise in cooling water is different from case to another due to the operating conditions considered. These are temperature difference between vapor and cooling water or mass flow rate of vapor or fouling resistance. The rise in case (3) is approximately equal to that in case (1), due to the little effect of the presence of air mass flow rate. For example, the rises in inlet cooling water in cases (1,2,3,4,5 and 6) are (11.4,16.6,11.4,10.1,14.2 and 12.7) °C. The results have also revealed that the rise of cooling water temperature in case (7) was (11.7) °C. This value was less than that of case (1) owing to its high decrease in temperature difference between vapor and cooling water.

COMPARISON WITH FIELD DATA:

The present model performance predictions was compared with the field data obtained from Southern Baghdad thermal power plant. It is suggested to consider the outlet cooling water temperature and condensate exit temperature as a measure for the accuracy of prediction. **Table 3** shows the predicted and measured temperature of the above two variables. It is obvious that there is a good agreement between the field data and the present work, with accuracy fell within **(98) %** and **(89) %** for the cooling water temperature prediction for summer and winter, while the accuracy fell within **(98) %** and **(99) %** for the condensate temperature prediction for summer and winter.

CONCLUSIONS:

The performance of power plant condensers is difficult and complicated art to be checked by single parameter, therefore a quasi two dimensional computer program has been built to depict the performance of this type of condensers. The principle findings of this investigation are listed below:-

- The procedure based on the step bay step method across tube bank of the condensers provides a powerful technique for the performance prediction of the condenser.

- The program is used to study different operating conditions, to analyze them and other parameters that affect the performance of Southern Baghdad power plant. From this study the followings are noticed:-

i. Best performance in winter owing to the large temperature difference between vapor and cooling water and the rise in the level of the river water which decreases the deposit inlet to the condenser tubes.
ii. Heat transfer, condensation rate, and overall heat transfer coefficient decrease towards the vent, because of vapor pressure drop. Heat transfer and condensation rate for the pass of condenser where river water enters first are higher than the other pass, while the overall heat transfer coefficient is lower. The increase of the heat transfer rate is mainly due to the increase in temperature difference between vapor and cooling water.
iii. The fouling resistance has an important effect on the condenser and plays a major part of decline of the condenser performance. This is due to the increase in the fouling resistance, leads to a decrease in the overall heat transfer coefficient and condensation rate.
iv. The performance of the condenser when the cooling water enters the condenser at the lower pass is better than that when it enters the condenser at the upper pass. This is because of the balance in the distribution of the heat transfer and condensation rate between passes.

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NOMENCLATURE:

a_n : Constant defined in eqs. (11) & (14.a)

b_n : Constant defined in eqs. (11) & (14.a)

cp: Specific heat (kJ/kg. K)

d : Tube diameter (m)

D : Molecular diffusion Coefficient (m^2/s)

D_p: Coefficient of vapor diffusion in gas (kg/(s.m.Pa))

E_v : Pressure ratio (air to mixture)

E_w : Mass ratio (air to mixture)

F : Parameter defined by eq.(8.b)

g : Gravitational acceleration (m/s^2)

h : Heat transfer coefficient (W/m^2 K)

h_{fg} : Latent heat of vaporization (J/kg)

h^*_{fg}: Modified latent heat of vaporization (J/kg)

k : Thermal conductivity (W/m.K)

L : Length of tube (m)

LMTD : Logarithmic mean temperature difference (K)

m : Mass flow rate (kg/s)

n : Number of certain row

Nu: Nusselt number

P : Pressure (Pa)

Pr : Prandtl number

Q : Heat flux (W/m^2)

R : Gas constant (J/kg.K)

R_f : Fouling resistance (m^2 K/W)

Re : Reynolds number

S : Index in eq. (13)

Sc : Schmidt number

Sh : Sherwood number

T : Temperature (˚C)

u : Fluid velocity (m/s)

U : Overall heat transfer coefficient (W/m^2 K)

w : Mass fraction

Greek Symbols:

ρ : Density (kg/m^3)

μ : Viscosity (Pa.s)

β : Mass transfer coefficient ($kg/s.m^2$)

ν : Kinetic viscosity (m^2/s)

Π : Parameter defined by eq. (14.b)

Γ : parameter defined by eq. (3.b)

Subscript Symbols:

a : Air

c : Condensate

cs : Vapor/Condensate interface

i : Coolant inside tube

ib : Inside tube bulk

iw : Inside tube wall

mix: Mixture

o : Outside or outlet

TP : Two phase

v : Vapor

$v\infty$: Vapor at free stream

w : Tube wall

∞ : Free stream

Table (1.a): Geometrical Design Characteristics Data for the Test Condenser.

Input data	Value
Internal diameter of tube (m)	0.0197358
external diameter of tube (m)	0.022225
Tube pitch (m)	0.02778125
Number of bays	11
Length of bay (m)	0.72736
Width of tube sheet (m)	1.228
Number of tube passes	2
Number of tubes for all condenser	2430
Number of rows for all condenser	57
Number of rows before the vent	34
Tube numbers for upper pass of condenser	1201
Tube numbers for lower pass of condenser	1229
Row numbers for upper pass of condenser	27
Row numbers for lower pass condenser	30

Table (1.b): Number of Tubes for Each Row in the Condenser.

Number of row	Number of tubes
From(1-27) odd rows	44
From(1-27) even rows	45
28-29-30	42
31-33	35
32-34	36
35	19
36	20
37-38-39	38
40-42	44
41-43-44…….-57	45

Table (2): Shell Side and Water Side Input Data for Case (1) and Case (2).

Input data	Case(1)	Case(2)
Total mass flow rate of vapor (kg/s)	15.0926	17.26852
Mass flow rate of air (kg/s) *	0.00472	0.00472
Air mass percentage %	0.0313	0.0273
Inlet vapor temperature to condenser (K)	319.69	314.861
Air vent capacity (kg/s)	0.0275	0.0275
Total mass flow rate of water (kg/s)	791.34376	791.34376
Inlet water temperature to condenser (K)	296.2	283.2
Thermal conductivity of tube (W/m.K)	300	300
Tube side fouling resistance (m^2.K/W) *	0.0004	0.00032
Roughness of tube (m) *	0.000061	0.000036
Pressure of water inlet to condenser (bar)	3.5	3.5

All data labeled by (*) are assumed values.

Table (3): Comparison Between Field Data and Present Work.

Data	Field data		Present work	
	Case(1)	Case(2)	Case(1)	Case(2)
Temperature of condensate at outlet from condenser (K)	319.5	313.8	318.6	313.6
Temperature of cooling water at outlet from condenser (K)	307.2	297.2	307.6	299.8

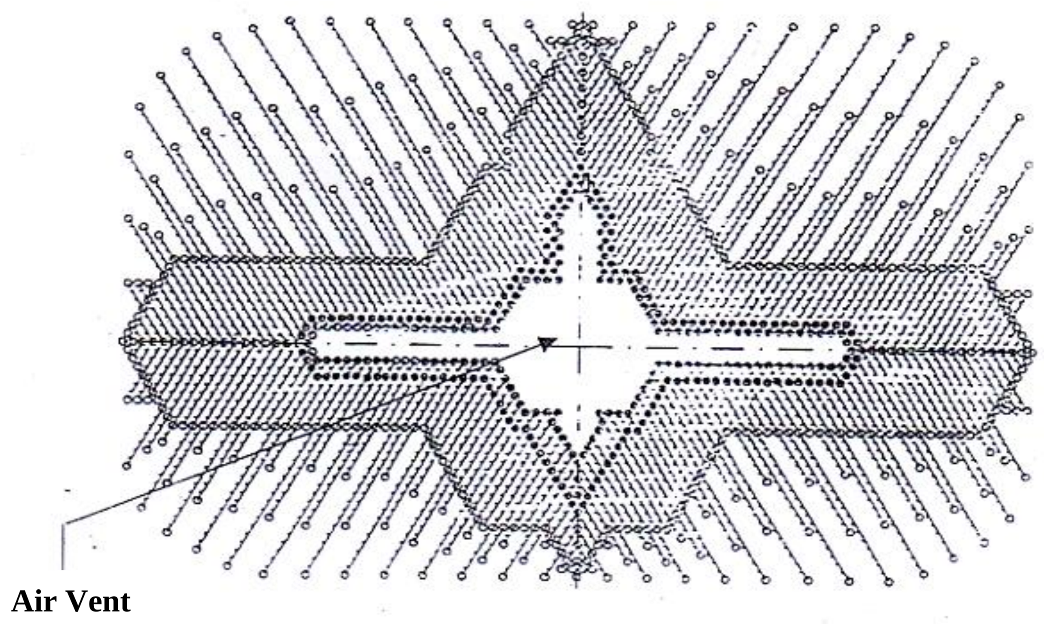

Figure (1): Arrangement of Tube Bundle in Al-Daura Thermal Power Station Condenser, (Tarrad and Kamal, 2004).

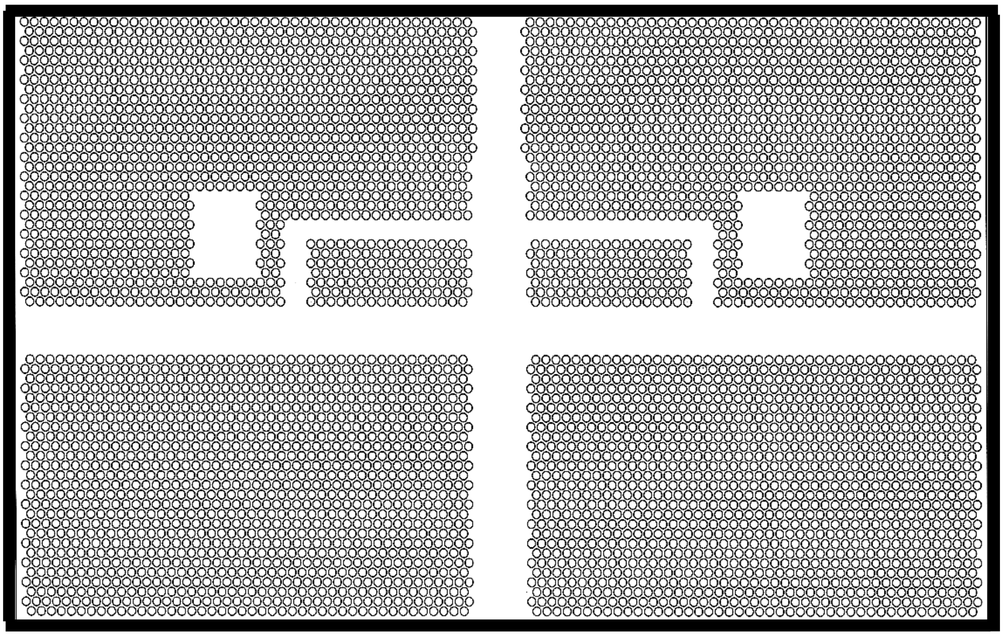

Figure (2): Geometrical Tube Arrangement of Southern Baghdad Power Plant Condenser.

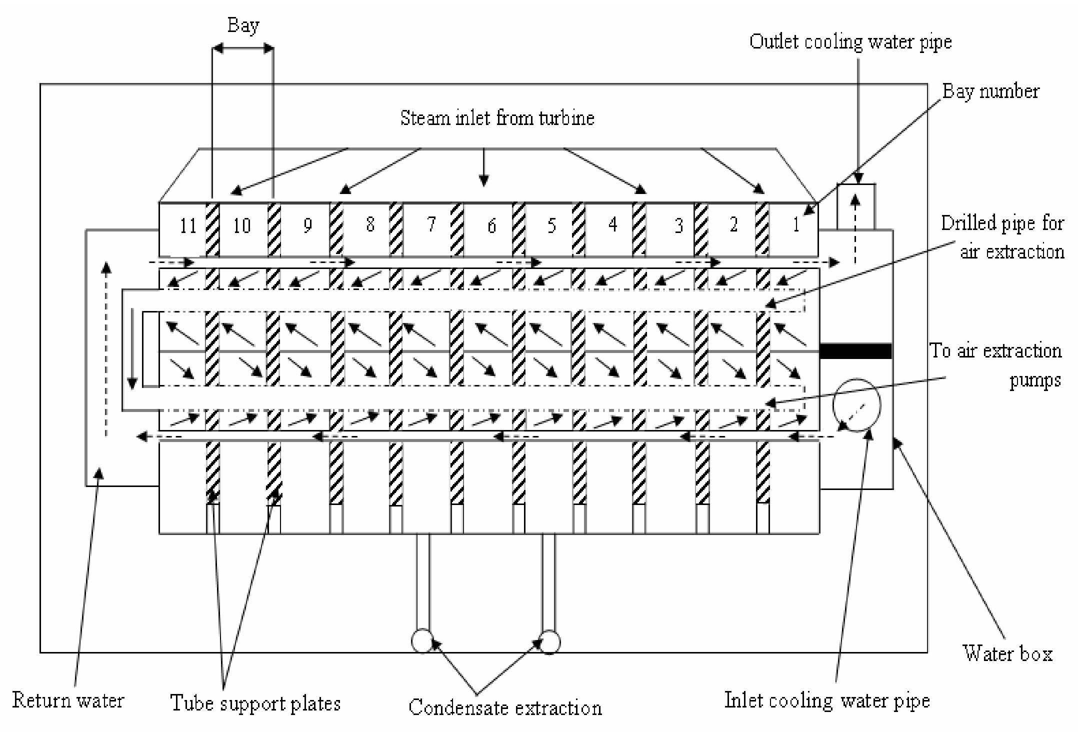

Figure (3): The Longitudinal Condenser Section for Bay and Tube Passes Arrangement.

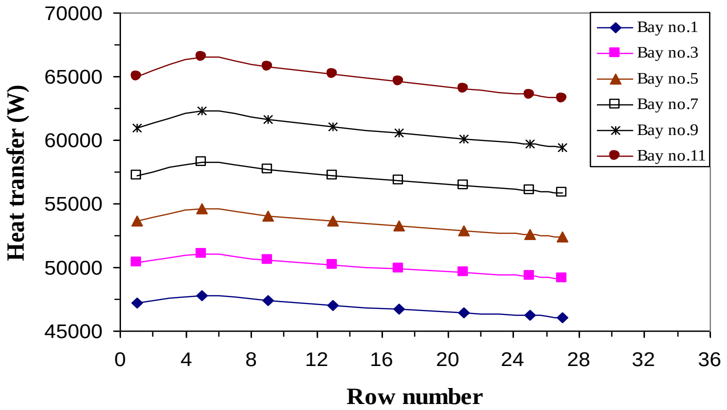

Figure (4.a):Heat Transfer Rate vs. Row Number for Upper Pass, Case (1).

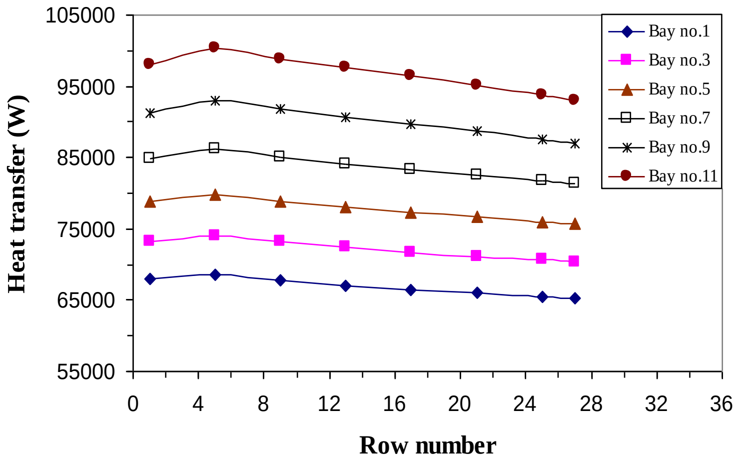

Figure (4.b):Heat Transfer Rate vs. Row Number for Upper Pass, Case (2).

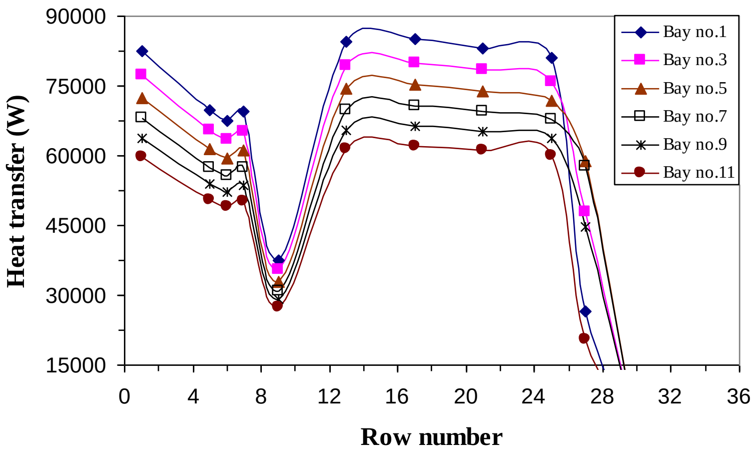

Figure (5.a):Heat Transfer Rate vs. Row Number for Lower Pass, Case (1).

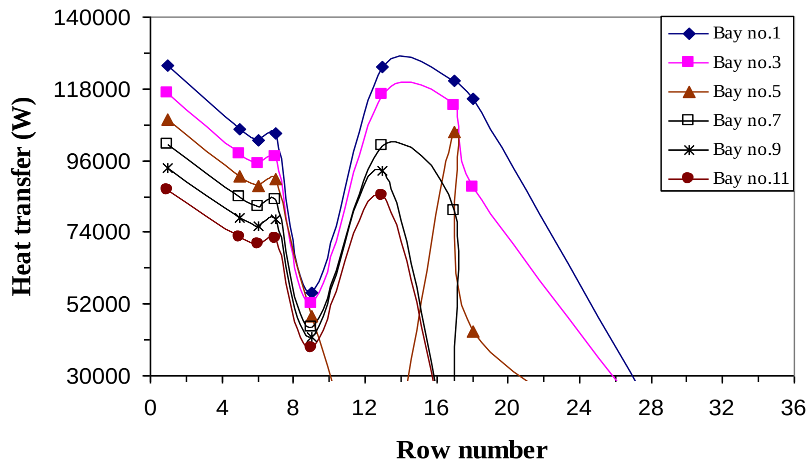

Figure (5.b):Heat Transfer Rate vs. Row Number for Lower Pass, Case (2).

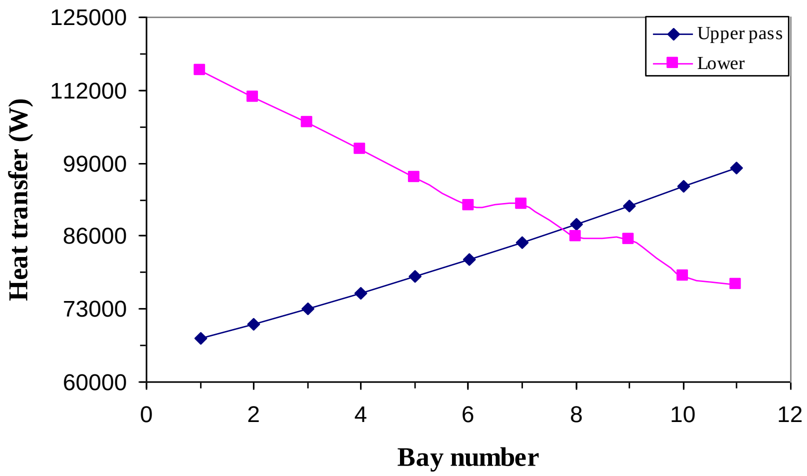

Figure (6): Heat Transfer Rate of Case (2) vs. Bay Number.

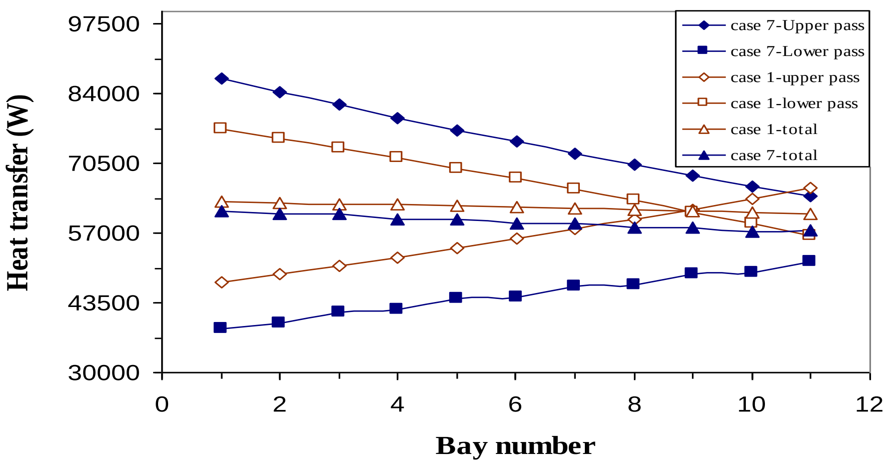

Figure (7): Heat Transfer Rate of Cases (1 & 7) vs. Bay Number.

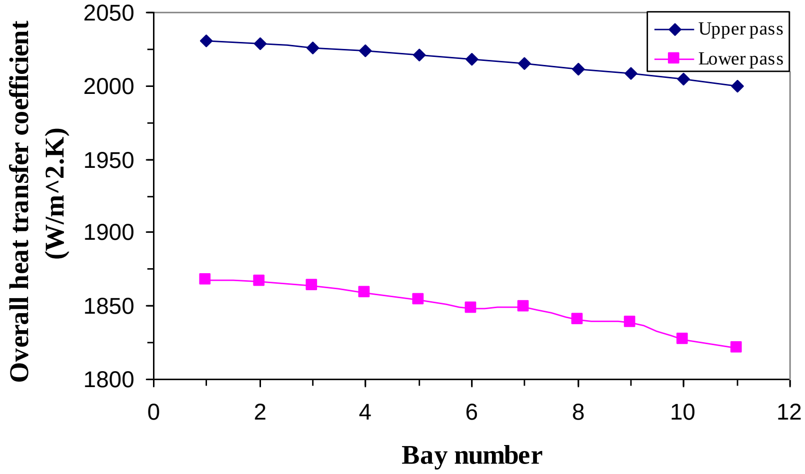

Figure (8): Overall Heat Transfer Coefficient of Case (2) vs. Bay Number.

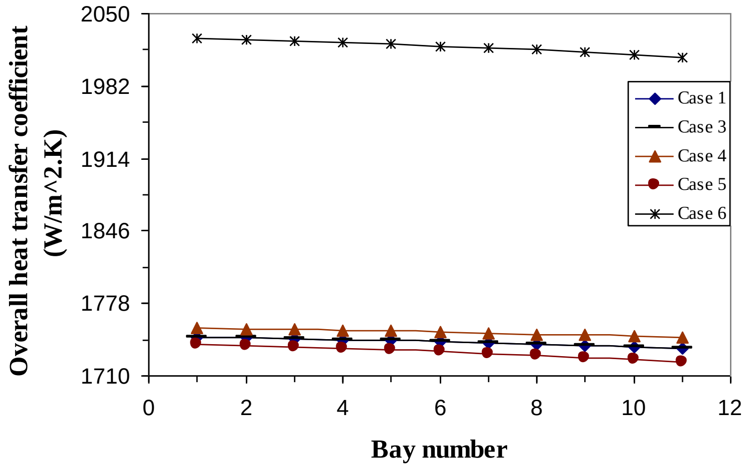

Figure (9): Overall Heat Transfer Coefficient of Various Cases vs. Bay Number for Upper Pass.

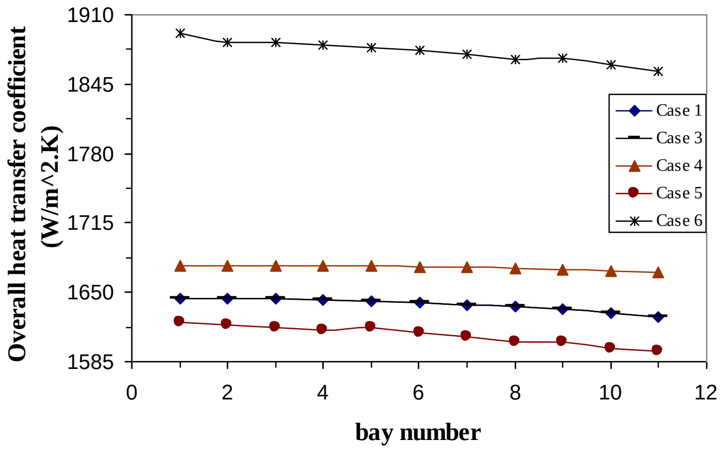

Figure (10): Overall Heat Transfer Coefficient of Various Cases vs. Bay Number for Lower Pass

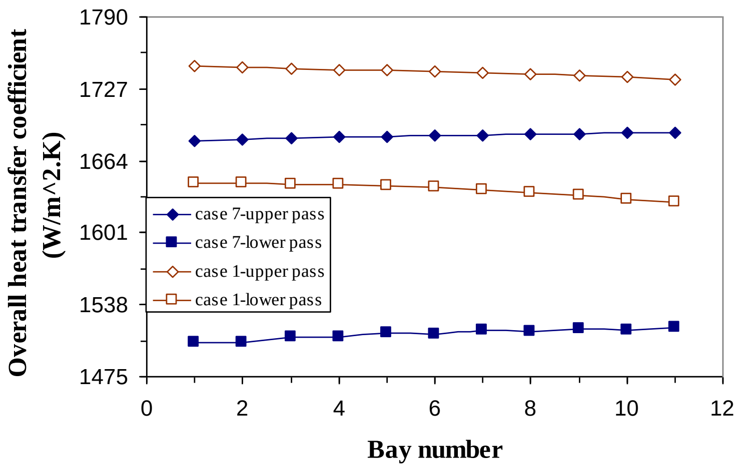

Figure (11): Overall Heat Transfer Coefficient of Cases (1 & 7) vs. Bay Number.

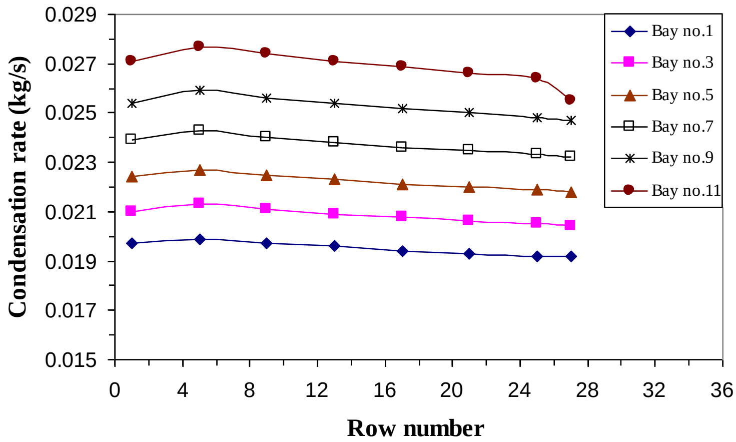

Figure (12): Condensation Rate Vs. Row Number of Case (1), Upper Pass.

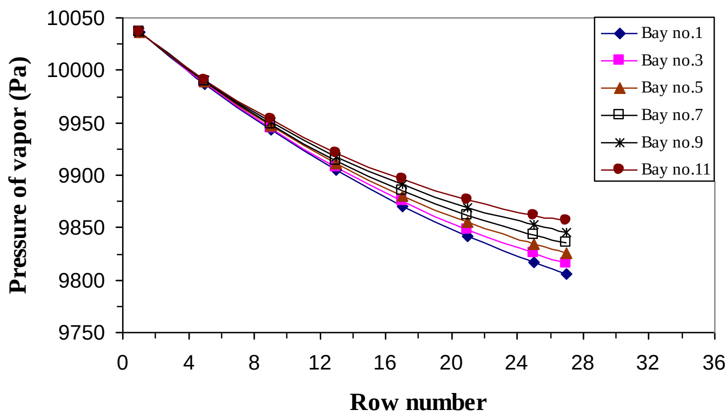

Figure (13.a): Pressure of Vapor vs. Row Number of Case (1) for Upper Pass.

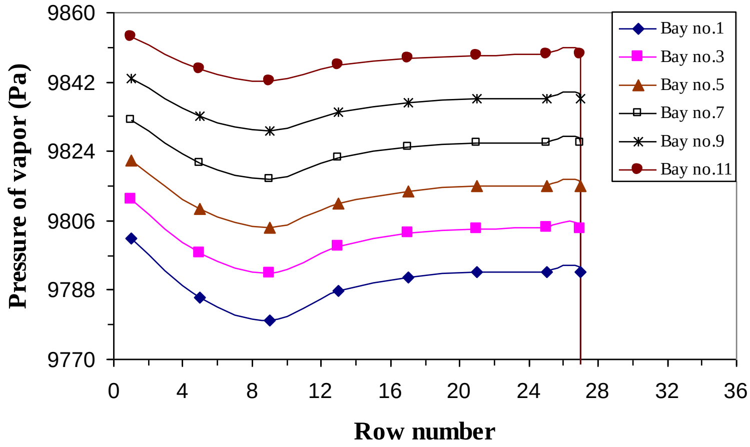

Figure (13.b): Pressure of Vapor vs. Row Number of Case (1) for Lower Pass.

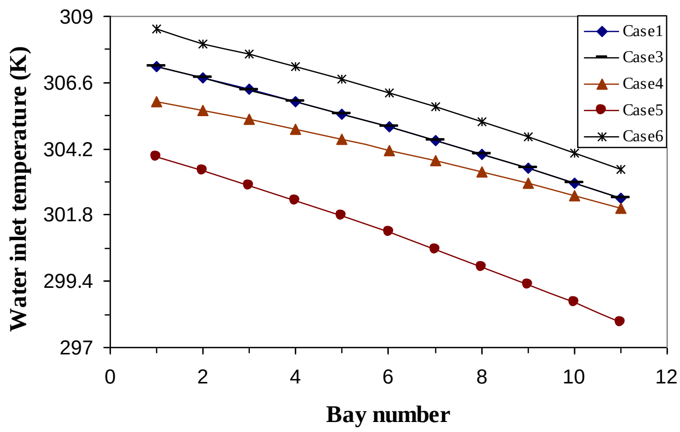

Figure (14.a): Water Inlet Temperature of Various Cases vs. Bay Number for Upper Pass.

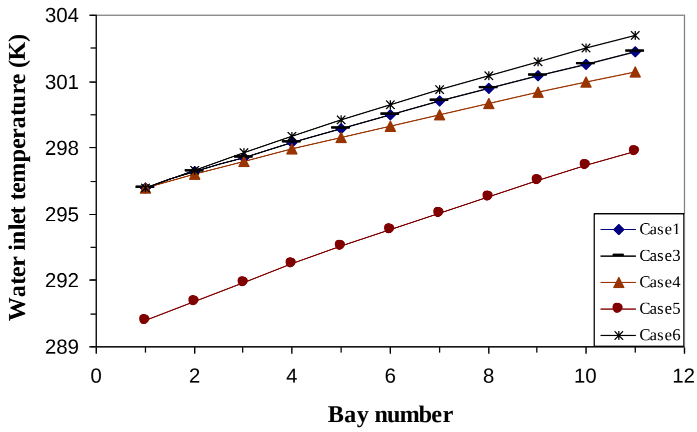

Figure (14.b): Water Inlet Temperature of Various Cases vs. Bay Number for Lower Pass.