Study of the Condenser Performance in Al-Nassiriyah Thermal Power Plant

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Abstract

This paper includes a study of the performance of Al-Nassiriyah thermal power plant condenser. Which includes a calculations of the main variables of condenser (heat transfer, condensation rate, overall heat transfer coefficient and vapour pressure) by applying two dimensional mathematical model and depending on the empirical equations of heat transfer.

Comparison was made between the theoretical results for Al-Nassiriyah thermal power plant condenser before operation it (standard power plant) with practical data which taken from the plant for the second unit in year 2007. The comparison showed that there was a large difference between theoretical results and the values taken from the plant as a result to the effect of operation conditions.

Keywords : Thermal power plant , comparison, condensation.

المستخلص

يتضمن هذا البحث دراسة أداء مكثف محطة الناصرية الحرارية. تضمنت الدراسة حساب المتغيرات الرئيسية للمكثف (معدل انتقال الحرارة، معدل التكثيف، معامل انتقال الحرارة الإجمالي، ضغط البخار) وذلك باستخدام نموذج رياضي ثنائي الأبعاد بالاعتماد على المعادلات التجريبية لانتقال الحرارة. أجريت مقارنة بين النتائج النظرية لمكثف محطة الناصرية قبل تشغيلها مع قراءات أخذت من المحطة للوحدة الثانية ولسنة 2007 وقد تبين من خلال المقارنة ان هناك انحراف كبير بين النتائج النظرية والقيم المأخوذة من المحطة نتيجة لتأثير الظروف التشغيلية.

1. Introduction

Thermal power plants are responsible for the production of most of the electric power in the world, and even small increases in thermal efficiency could cause large saving of the fuel consumption. Therefore, large efforts were made to improve the efficiency of the power plant and one mean is by proper operation of the condenser. Surface condenser has a shell whose ends are covered by the plates with condenser tubes. The ends of tubes are communicated with water boxes. The water boxes are divided by a partition into two sections of tubes, which form passes for water. Water is fed into the water boxes through an inlet pipe connection and first runs through the below partition. In the opposite water box, which has no partition, water flow is turned and moves in the opposite direction through the upper section condenser tubes, above the partition. When completing second pass, water enters the first water box and is drained through the outlet pipe connection [1].

Steam from the final stage of the steam turbine enters into the steam space of the condenser and moves, in cross flow, over the tubes. With cooling water, steam is condensed on the outer surface of the tubes causing a sharp drop of the specific volume of steam and, as a consequence, of low pressure (vacuum) is created in the condenser. The condensate rate runs down into the lower portion of the condenser and is collected in hotwell [2].

Chisholm et al., 1965**,** [3] developed a numerical method of evaluation heat and mass transfer coefficient and local heat fluxes in surface condensers. Al-ka'abiy, 2000**,** [4] Study The effect of Tigris water quality upon fouling rate in the condenser of Al-Daura power plant. AL-Chalaby and Rishack, 2001, [5] studied the reasons behind the loss in vacuum pressure of the condenser in Al-Hartha power plant. Tarrad and kamal, 2004, [6] studied the performance prediction of thermal power plant condensers in a quasi-two dimensional model and the proposed model showed a good agreement with field data. An experimental and analytical study was performed by Seungmin and Revankar, 2005, [7] to investigate the effect of non-condensable gas in a passive condenser system for three operation modes.

Al-Nassiriyah power plant was constructed by Technoprome (Russian) company which commissioned in 1979. It consists of four units. The electrical generating power (design power) for each unit is 210 MW. The unit condenser is of the surface tubular type and consists of two identical exchangers. Each carrying half load of the cooling water is forced through 8380 tubes, arranged in two passes and in a staggered configuration, the number of bays is 9 and the number of rows is 94 for all condenser. The design heat load is 210 MW and 27°C inlet cooling water temperature. The centrifugal pump with a maximum flow rate is 20000 m^3/h for each pump and 1000 kW power input was used for cooling water. The condensers of Al-Nassiriyah power plant are surface condenser [8].

Table (1). Geometrical input data for condenser of Al-Nassiriyah thermal power plant.

Geometrical input data	value
External diameter of tube (m)	0.028
Internal diameter of tube (m)	0.026
Tube pitch (m)	0.035
Width of tube sheet (m)	4.45
Number of bays	9
Length of bay (m)	0.984, 0.876, 1.042, 1.042, 1.042, 1.042, 1.042, 0.876, 0.984
Number of tubes for condenser	8380
Number of rows for upper pass of condenser	46
Number of rows for lower pass of condenser	48
Number of tubes for upper pass of condenser	3874
Number of tubes for lower pass of condenser	4506

The main objectives of the present work are to:

1. Build a computer program (Fortran 90 language) to study the performance of thermal power plant condensers by calculating the main variables under various operating conditions which have direct effect on condenser performance, such as overall heat transfer coefficient, heat transfer rate and vacuum pressure.
2. Compare the theoretical results with the obtained data from Al-Nassiriyah power plant to know the reasons of the difference between them.

2. Theory

In the heat transfer analysis of the surface condenser, various thermal resistance in the direction of heat flow from the vapour to the cooling water are combined into the overall heat transfer coefficient (U_o). This thermal resistance includes cooling water resistance, fouling (inside tube) resistance, tube wall resistance, condensing resistance and non-condensable gas resistance [9].

The heat flux can be calculated from:

$$Q = U_o \; LMTD \tag{1}$$

The overall heat transfer coefficient that refers to the outside tube area may be expressed as:

$$\frac{1}{U_o} = \frac{d_o}{d_i\, h_i} + R_{fou} + \frac{d_o \ln\left(\frac{d_o}{d_i}\right)}{2\, k_w} + \frac{1}{h_c} + \frac{1}{h_v} \tag{2}$$

Where:

R_{fou}: fouling resistance inside tube, in the present work, fouling factor is taken as (0.0-0.0005 $\frac{m^2\ °C}{W}$), [10].

2.1 Forced Convection inside Tube

Petukhov as cited by [11] presented the results of a theoretical analysis of heat transfer coefficient in pipes to variable property fluids as:

$$h_i = \frac{Re_i\, Pr_i \;\left(\frac{\eta}{8}\right)}{1.07 + 12.7\left(Pr_i^{\frac{2}{3}} - 1\right)\sqrt{\frac{\eta}{8}}}\left(\frac{k_i}{d_i}\right)\left(\frac{\mu_{i,b}}{\mu_{i,w}}\right)^{0.11} \tag{3}$$

Where:

$$Re_i = \frac{\rho_i\, u_i\, d_i}{\mu_i}$$

and

$$\eta = (1.82\log(Re_i) - 1.64)^{-2}$$

$\mu_{i,b}$: Dynamic viscosity evaluated at bulk temperature.

$\mu_{i,w}$: Dynamic viscosity evaluated at wall temperature.

2.2 Nusselt equation for a laminar film

The following relation of Nusselt theory for the heat transfer coefficient for circular tube [12]:

$$h_{cN} = 0.725\left(\frac{\rho_c(\rho_c-\rho_v)\,h_{fg}\,g\,k_c^3}{d_o\,\mu_c\,(T_{cs}-T_w)}\right)^{1/4} \qquad (4)$$

The cooling of the liquid below the saturation temperature can be accounted by replacing (h_{fg}) by the modified latent heat of condensation (h_{fg}^{*}) which is defined as [13]:

$$h_{fg}^{*} = h_{fg} + 0.68\,Cp_c\,(T_v - T_w) \qquad (5)$$

The condensate heat transfer coefficient can be calculated from Fujii equation as cited by [14]:

$$h_c = \frac{0.9\,(1+G)^{1/3} + 0.728\,F^{1/2}}{(1+3.44\,F^{\frac{1}{2}}+F)^{1/4}}\;Re_{TP}^{1/2}\left(\frac{k_c}{d_o}\right) \qquad (6)$$

Where:

$$Re_{TP} = \frac{\rho_c\,u_\infty\,d_o}{\mu_c},$$

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

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

2.3 Effect of inundation in tube banks

During shell side condensation in tube bundles, the conditions are much different than for a single tube. As condensate flows by gravity to lower tubes in a bundle, the thickness of the liquid film at lower tubes becomes much bigger, this effect known as inundation effect. Kern derived a non-dimensional equation which accounted for the predominant physical mechanism as [15]:

$$\frac{h_{c,n}}{h_{c,1}} = \left(1 + \frac{\sum_{i=1}^{n-1} m_{cr,i}}{m_{cr,n-1}}\right)^{-0.16} \qquad (7)$$

2.4 Mass transfer coefficient and Sherwood number

The dimensionless mass transfer coefficient or Sherwood number is defined as [16]:

$$Sh = \frac{\beta \ d_o}{\rho_{mix} \ D} \tag{8}$$

For condensation of vapour in the presence of a non-condensable gas, the condition that the surface is impermeable to the gas gives:

$$m_c = \beta \ (w_{v\infty} - w_{vcs})/(1 - w_{vcs}) = \beta \ (w_{cs} - w_{\infty})/w_{cs} \tag{9}$$

Where (m_c) is towards the surface, total vapour mass flux or condensation rate at the vapour condensate interface. Equations (8) and (9) give:

$$Sh = \frac{m_c \ d_o}{\rho_{mix} \ D} \left(\frac{w_{cs}}{w_{cs} - w_{\infty}} \right) \tag{10}$$

By taking

$$Q = m_c \ h_{fg}^{*} \tag{11}$$

Combining equations (10) and (11) give:

$$Q = h_{fg}^{*} \ \rho_{mix} \ \frac{D}{d_o} \left(\frac{w_{cs} - w_{\infty}}{w_{cs}} \right) Sh \tag{12}$$

Where (Sh) can be obtained from Rose equation [17]:

$$Sh = \left[\left(1 + 2.28 \ Sc^{1/2} \frac{(w_{cs} - w_{\infty})}{w_{\infty}} \right)^{1/2} - 1 \right] \frac{w_{cs}}{2 \ (w_{cs} - w_{\infty})} Re_{mix}^{1/2} \tag{13}$$

Where:

$$Re_{mix} = \frac{\rho_{mix} \ u_{\infty} \ d_o}{\mu_{mix}}$$

and

$$Sc = \frac{\mu_{mix}}{\rho_{mix} \ D}$$

Assuming vapour as an ideal-gas mixture, the interface equilibrium conditions give [18]:

$$w_{cs} = \frac{P_{mix} - P_{cs}}{P_{mix} - \left[1 - \left(\frac{M_v}{M_a} \right) \right] P_{cs}} \tag{14}$$

The diffusion coefficient is obtained by [17]:

$$D = \frac{0.926}{P_{mix}}\left(\frac{T^{2.5}}{T + 245}\right) \tag{15}$$

Where T is taken at $(T_v + T_{cs})/2$ in (K) and P_{mix} in (Pa).

2.5 Velocity distribution in shell side of surface condenser

As vapour passes across the tubes in the shell side of the surface condenser, its velocity would change due to two reasons. Firstly; the reduction in the mass flow rate of vapour where it condensates around the tubes. Secondly; the change of flow area between the rows. Accordingly, the vapour velocity for the first row differs from that for other rows below [19]:

a) For the tube in the first row:

$$u_{\infty} = \frac{m_v + m_a}{\rho_v \, A_{vd}} \tag{16}$$

Where:

$$A_{vd} = w_{ts} \, L$$

b) For the next tube in the other rows:

$$u_{\infty(n+1)} = \frac{(m_v + m_a) - \sum_{i=1}^{n} m_{cr,i}}{\rho_{mix} \, A_{mv}} \tag{17}$$

Where A_{mv} is the mean void area of flow in the vapour space for an equivalent triangular pitch-tube layout, which defined as [19]:

$$A_{mv} = L\,(NTR_n - 1)\left(Pt - \frac{\pi \, d_o^2}{2\sqrt{3}Pt}\right) \tag{18}$$

The pressure amount at each tube in the column can be calculated from this equation [20]:

$$P_{mix_{n+1}} = P_{mix_n} - \left[\left(0.2\, c_r^2 \, \frac{\ddot{m}_v^2}{\rho_v}\right)_n - \left\{\left(\frac{\ddot{m}_v^2}{\rho_v}\right)_n - \left(\frac{\ddot{m}_v^2}{\rho_v}\right)_{n+1}\right\}\right] \tag{19}$$

Where:

$$c_r = \frac{A_{vd}}{A_{mv}}$$

and

$$\ddot{m}_v = \rho_v \, u_{\infty}$$

$$P_{vac} = \left[\frac{(P_{atm} - P_{mix})}{P_{atm}}\right] * 760 \qquad (20)$$

2.6 Calculation method

The step by step method (this method divides the condenser into a number of bays and a number of rows, the calculation starts from upper pass where the vapour inlets to condenser and continuous from row to row towards the bottom in specified bay) is applied to calculate the performance of power plant condenser, the bays are numbered from cooling water inlet to condenser and the rows numbered from the top of condenser to the bottom.

Considering the heat flux which includes the convection inside tube, fouling resistance and wall resistance is:

$$Q = h_{iwf}\,(T_w - T_{in}) \qquad (21)$$

Where:

$$h_{iwf} = \left(\frac{d_o}{d_i\,h_i} + \frac{d_o\,ln\left(\frac{d_i}{d_o}\right)}{2\,k_w} + R_{fou}\right)^{-1} \qquad (22)$$

Where (h_i) calculate from equation (3).

Considering heat flux across condensate film as:

$$Q = h_c\,(T_{cs} - T_w) \qquad (23)$$

Combining equations (23), (4), (6) and (7) give:

$$Q = C\,(T_{cs} - T_w)^{3/4} \qquad (24)$$

Where (C) is parameter defined as:

$$C = Y * Z \qquad (25)$$

Where:

$$Y = 0.725\left(\frac{\rho_c\,(\rho_c - \rho_v)h_{fg}\,g\,k_c^3}{d_o\,\mu_c}\right)^{1/4}\left(1 + \frac{\sum_{i=1}^{n-1} m_{cr,i}}{m_{cr,n-1}}\right)^{-0.16} \qquad (26)$$

The parameter (Z) is defined as:

$$Z = \frac{h_c}{h_{CN}} \tag{27}$$

$$Z = \frac{0.9\,(1+G)^{1/2} + 0.728\,F^{1/2}}{h_{CN}\,(1 + 3.44\,F^{\frac{1}{2}} + F)^{1/4}}\; Re_{TP}^{1/2}\left(\frac{k_c}{d_o}\right) \tag{28}$$

Considering heat flux through bulk of mixture as:

$$Q = h_v\,(T_v - T_{cs}) \tag{29}$$

Combining equations (29), (12) and (13) give:

$$Q = S\,(T_v - T_{cs})^{2/3} \tag{30}$$

Where (S):

$$S = h_{fg}^{*}\,\rho_{mix}\,\frac{D}{2\,d_o}\left[\frac{1}{T_v - T_{cs}}\right]^{2/3}\left[\left(1 + 2.28Sc^{1/3}\left(\frac{W_{cs} - W_{\infty}}{W_{cs}}\right)\right)^{1/2} - 1\right] Re_{mix}^{1/2} \tag{31}$$

Combining equations (21), (24) and (30) gives:

$$\left[\frac{Q}{S}\right]^{3/2} + \left[\frac{Q}{C}\right]^{4/3} + \left[\frac{Q}{h_{iwf}}\right] = T_v - T_m \tag{32}$$

Or

$$Q_{new} = (T_v - T_m)\Big/\left[\frac{Q_{old}^{1/2}}{S^{3/2}} + \frac{Q_{old}^{1/3}}{C^{4/3}} + \frac{1}{h_{iwf}}\right] \tag{33}$$

The overall heat transfer coefficient between vapour and cooling water is evaluated from:

$$U_o = \frac{Q}{T_v - T_m} \tag{34}$$

And The outlet temperature of cooling water (T_o) is evaluated from heat balance over tube as:

$$m_i\,Cp_i\,(T_o - T_i) = \pi\,d_o\,L\,U_o\,LMTD \tag{35}$$

Where:

$$LMTD = \frac{(T_v - T_i) - (T_v - T_o)}{ln\left(\frac{T_v - T_i}{T_v - T_o}\right)} = \frac{T_o - T_i}{ln\left(\frac{T_v - T_i}{T_v - T_o}\right)} \tag{36}$$

Combining equation (35 and 36) gives:

$$T_o = T_v - \frac{T_v - T_i}{exp\left(\frac{U_o\,\pi\,d_o\,L}{m_i\,Cp_i}\right)} \tag{37}$$

Calculate effectiveness of condenser which defined by the ratio of cooling water temperature differential throughout the condenser to the maximum temperature differential [5]:

$$\varepsilon_{eff} = \frac{T_o - T_{in}}{T_{vt} - T_{in}} \tag{38}$$

3. Results and discussion

a. Heat transfer

Figure (1) shows that the general trend of the curves for heat transfer rate with row number in upper pass, undergoes increasing from the first row to the row number (11), because of the number of tubes in the first row which have (58 tubes) is less than that in the row number (11) which have (84 tubes). Then the trend shows after decreasing as going down across the upper pass. This is because of steam pressure drop between the two rows results in a reduction in the overall heat transfer coefficient and, hence, a decrease in heat transfer rate.

Figure (2) shows that the heat transfer increases from the first row to the row number 16 in lower pass, this is because the number of tubes in the first row (78 tubes) is less than that in row number 16 which have (96 tubes). The trend of the curves is the same. The extreme drop in curves for lower passes in the rows number (35) and (36), because the number of tubes in these rows (38 tubes) less than that in the other rows. This makes the quantity of cooling water less than that in other rows, and that explains decreasing of heat transfer rate. The heat transfer rate approaches zero at row number (46) because of the condensation of all vapour and causing subcooling from this row to the last row.

b. Overall Heat Transfer Coefficient

Figure (3) shows the variation of the overall heat transfer coefficient (U_o) across the rows in the upper pass. The general trends of curves show a reduction in (U_o) as going down across the rows in upper pass. This is attributed to the increase in condensation resistance. The overall heat transfer coefficient in bay number (1) is larger than that in the other bays because of the small temperature difference between the vapour and cooling water.

Figure (4) describes the variation of overall heat transfer coefficient across the rows in the lower pass. There is a decrease in (U_o) as going down across the rows. This is due to the increase in all resistances as going down across the rows in the lower pass. The extreme drop in curves for lower pass in the rows number (35) and (36), since the heat transfer in this rows are less than that in the other rows. The overall heat transfer coefficient in bay number (1) is larger than that in the other bays because it has the largest heat transfer rate.

c. Vacuum pressure

Figure (5) shows that the curves of vacuum pressure increase from the first row towards the end of the rows at the same trend for upper pass because of large pressure drop in this pass and this leads to a decrease in the vapour pressure and an increase in vacuum pressure and a decrease in the temperature of vapour.

Figure (6) illustrates the distribution of vapour vacuum pressure through the rows for lower pass. The vacuum pressure curves increases from first row to the vent and then decreases to the end of the rows because the pressure drop at the vent increases so that the pressure of vapour decrease and the vacuum pressure increase. The reason of the decrease in the vacuum pressure after the vent is due to the decrease in mass flow rate of vapour.

d. Comparison of results

Figure (7) shows that the calculated results which obtain from computer program (assume new power plant with fouling resistance equal to zero) and the operational results which obtained from the log sheet for Al-Nassiriyah power plant (five days per a month for several months in year 2007). The average was taken for these days in the month. The figure shows that the vacuum pressure for performance curve (new power plant) is greater than that for operational data and the difference between two curves about (20 mm Hg). The reason of this difference is the effect of operation factors.

Figure (8) illustrates the effectiveness of condenser with inlet cooling water temperature for calculated and operational curves. The calculated data obtained from computer program (assume new power plant) and the operational data obtain from the log-sheet for Al-Nassiriyah power plant. The figure shows that the difference between the two curves about 30 percent as a result to the effect of operational conditions and resistances such as fouling factor.

4. Conclusions

From study the performance of Al-Nassiriyah thermal power plant condenser can be noticed:

1. Heat transfer for the lower pass of condenser, where river water first enters, is higher than that for upper pass. While the overall heat transfer coefficient is lower, due to the increase in temperature difference between vapour and cooling water.
2. Comparison of the results shows a difference between the operational data and standard results (new power plant), this attributed to the age of power plant and due to the effect of operational factors.

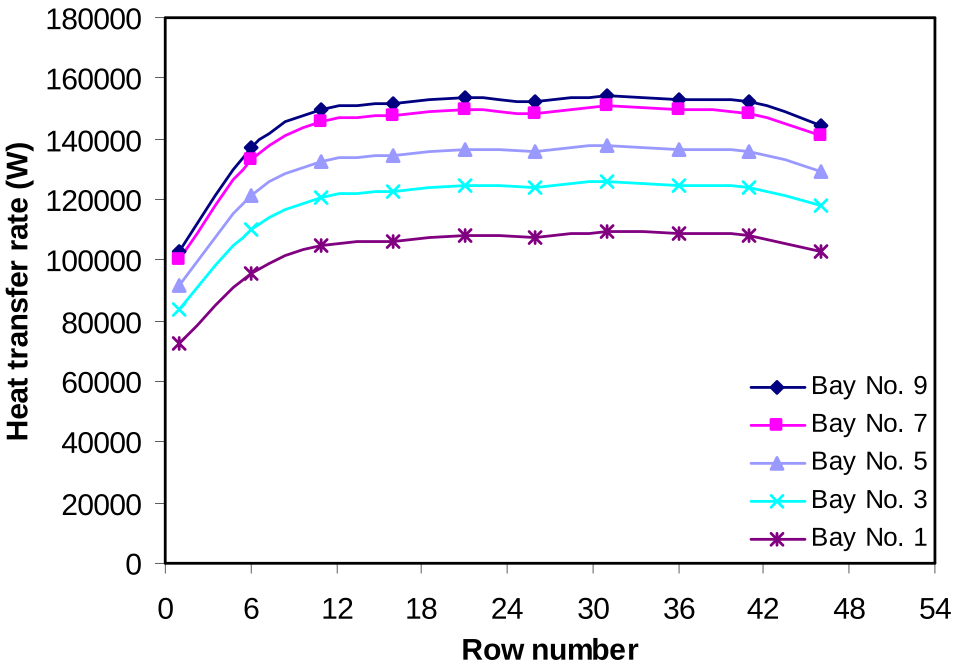

Figure (1). Heat transfer Vs. row number at different bay for upper pass.

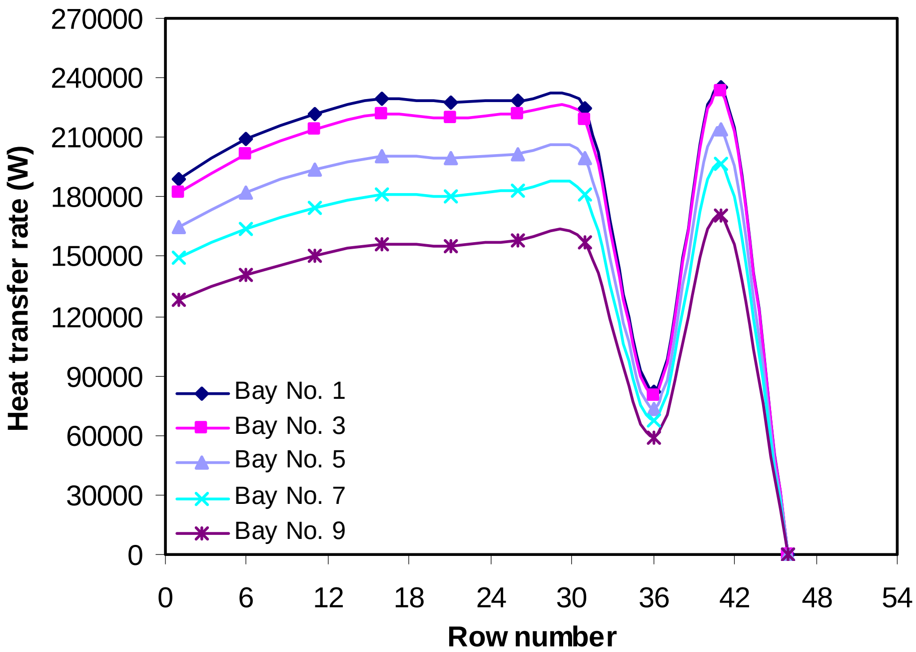

Figure (2). Heat transfer Vs. row number at different bay for lower pass.

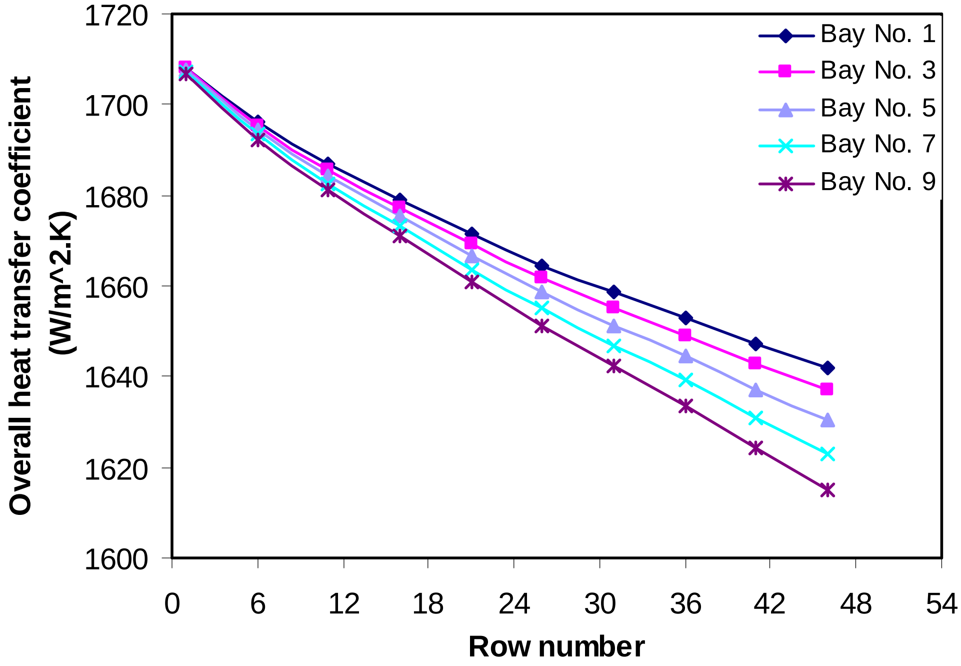

Figure (3). Overall heat transfer coefficient Vs. row number at different bay for upper pass.

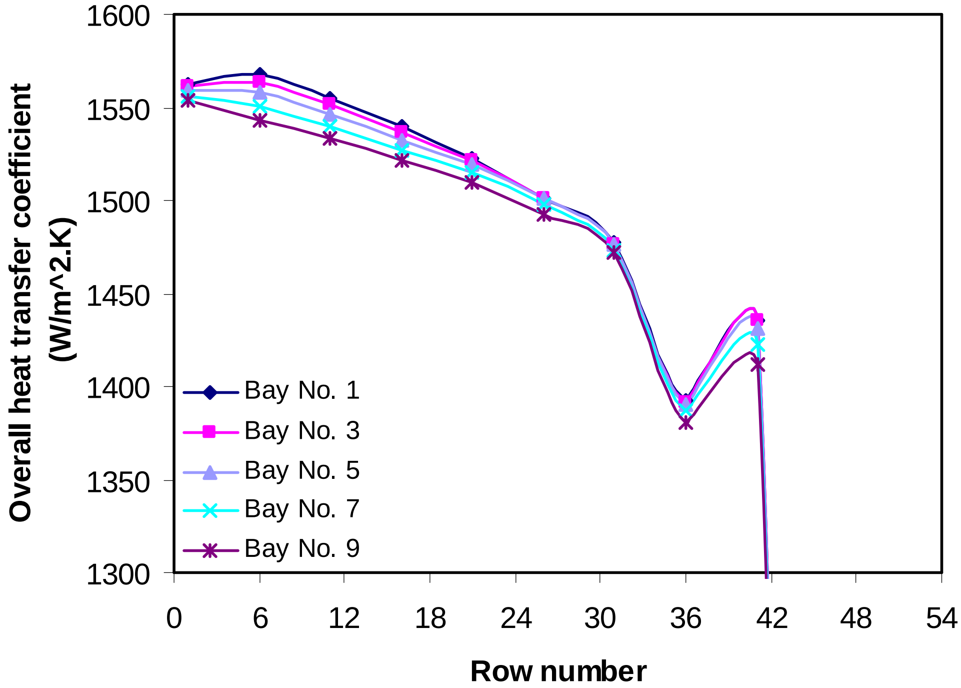

Figure (4). Overall heat transfer coefficient Vs. row number at different bay for lower pass.

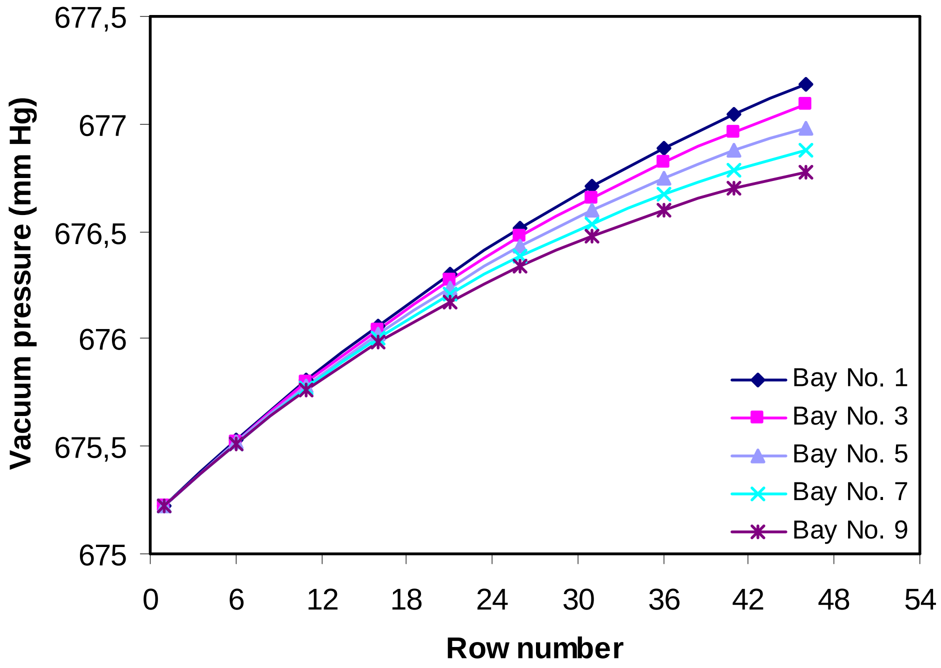

Figure (5). Vacuum pressure Vs. row number at different bay for upper pass.

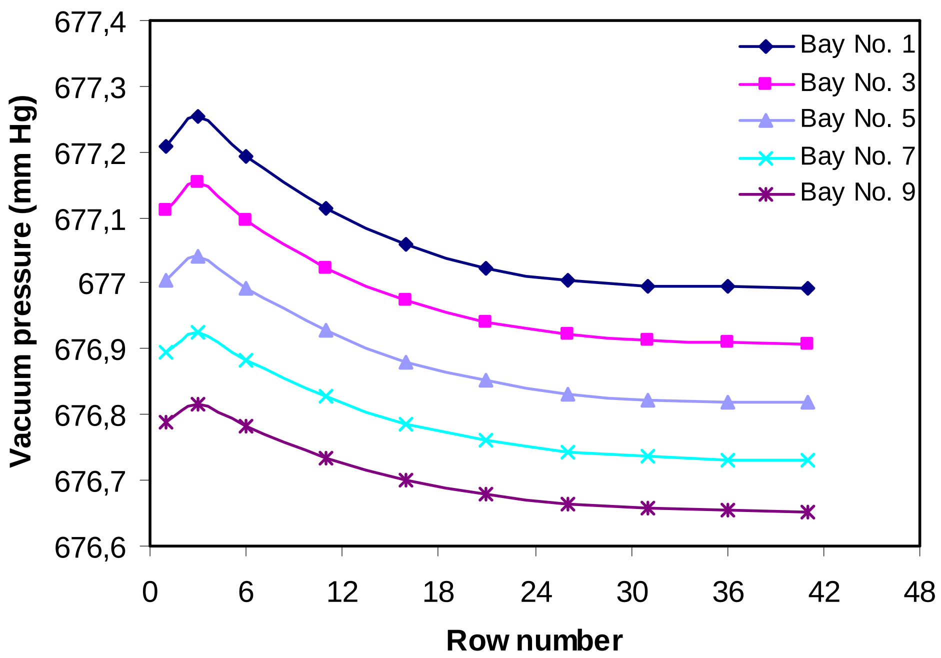

Figure (6). Vacuum pressure Vs. row number at different bay for lower pass.

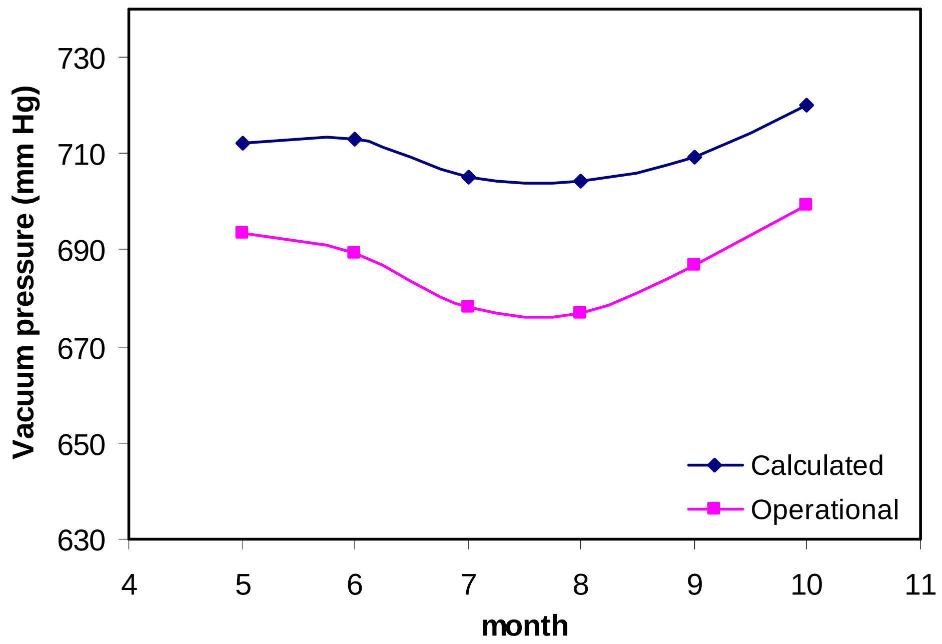

Figure (7). Vacuum pressure Vs. months for calculated and operational data.

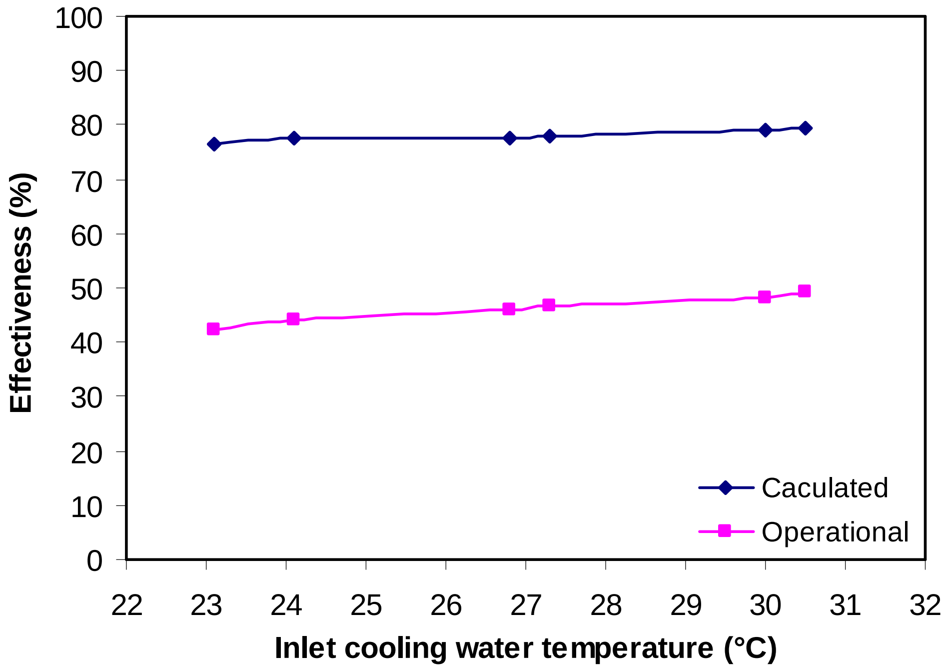

Figure (8). Effectiveness Vs. inlet cooling water temperature for calculated and operational data.

5. References

[1] Ozisik M.N., 1985,"Heat Transfer: A Basic Approach", McGraw-Hill Book Company, International Edition.

[2] Kostyuk A. and Frolov V., 1988,"Steam and Gas Turbines" Mir Publishers Moscow.

[3] Chisholm D., Provan T.F. and Mitchell D., 1965,"Digital Computation Methods for Evaluating Heat Flux in Condensers", Journal of Mechanical Engineering Science, Vol.7, No.2, pp.177-184.

[4] Al-Ka'abiy A.J.H., 2000,"A Study of the Performance of Al-Dura Power Plant Condenser and Its Relation with Water Cooling Problems", M.sc Thesis, University of Al-Mustansiriyah.

[5] AL-Chalaby, A.A. and Rishack, Q.A. , 2001,"The Effect of High Fouling Rates and Subcooling on Surface Condenser", Eng. Technology, Suppl. Of Vol.20, No.4.

[6] Tarrad A.H. and Kamal H.M., 2004,"A Model for Prediction of Surface Condenser Performance in Thermal Power Plants", Journal of Eng. And Development, Vol.8, No.3.

[7] Seungmin Oh. and Revankar S.T., 2005,"Investigation of a Passive Condenser System of Advanced Boiling Water Reactor", 11th, Int. Topical Meeting on Nuclear Reactor Thermal-Hydraulics (Nureth-11) Popes Place Conference Center, University of Purdue.

[8] Manual of Al-Nassiriyah Thermal power plant.

[9] Davidson B.J., 1987,"Condensers for Large Turbines", in Aerothermodynamics of Low Pressure Steam Turbines and Condensers, Eds. Moore, Mj., and Severing, C.h., pp.217-251, Hemisphere.

[10] Macnair E., 1981,"Fouling: Typical Experimental Results and Observations", power condenser heat transfer technology, Eds. By Marto P.J. and Nunn R.H., PP.431-438, Hemisphere.

[11] Slecher C.A. and Rose M.W.,1975,"A Convenient Correlation for Heat Transfer to Constant and Variable Property Fluids in Turbulent Pipe Flow", Int. Journal Heat Mass Transfer, pp.677-683.

[12] Owen R.G., Sardesai R.G., Smith R.A. and Lee W.C., 1983,"Gravity Controlled Condensation on a Horizontal Low-Fin Tube", in Condensers: Theory and Practice, Published by Inst. Chem. Eng. Symposium Series, no.75, pp.415-428.

[13] Collier G.J., 1972,"Convective Boiling and Condensation", McGraw-Hill Book Company.

[14] Rose J.W. , 1984,"Effect of Pressure Gradient in Forced Convection Film Condensation on a Horizontal Tube", Int. J. Heat Mass Transfer, vol.27, pp.39-47.

[15] Chisholm D., 1981,"Modern Developments in Marine Condensers: Non-condensable gases", An Overview, in power condenser heat transfer technology, Eds. By Marto P.J. and Nunn R.H., PP.95-142, Hemisphere.

[16] Lee W.C. and Rose J.W., 1983,"Comparison of Calculation Methods for Non-Condensing Gas Effects in Condensation on a Horizontal Tube", in Condensers: Theory and Practice, Published by Inst. Chem. Eng. Symposium Series, no.75, pp.342-351.

[17] ASHRAE Fundamentals Handbook, ch.5, 1997,"Mass Transfer".

[18] Rose J.W., 1980,"Approximate Equations for Forced Convection Condensation in the Presence of Non-Condensing Gas on a Flat Plate and Horizontal Tube", Int. J. Heat Mass Transfer, vol.23, pp.539-546.

[19] Michael A.G., Lee W.C. and Rose J.W., 1992,"Forced Convection Condensation of Steam on a Small Bank of Horizontal Tubes", Trans. of ASME Journal of Heat Transfer, vol.114, pp.708-713.

[20] Fujii T. , 1983,"Condensation in Tube Banks", in Condensers: Theory and Practice, I. Chem. E. Symp. Series, no.75, pp.3-22, Pergamon Press., London.

6. Nomenclature

A_{mv} : Mean void area in the vapour space (m^2)

A_{vd} : Cross-sectional area of the vapour duct (m^2)

c_p : Specific heat at constant pressure (J/kg.K)

c_r : Parameter defined by equation (19)

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

d_i : Inner tube diameter (m)

d_o : Outer tube diameter (m)

g : Gravitational acceleration (m/s^2)

h : Convective heat transfer coefficient (W/m^2.K)

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

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

k : Thermal conductivity (W/m.K)

L : Length of bays (m)

LMTD : Logarithmic mean temperature difference

M : Molecular weight (kg/kg mole)

m : Mass flow rate (kg/s)

$\ddot{m}_v$: Mass velocity (kg/m^2.s)

NTR : Number of tubes in each row in the bundle

P : Pressure (Pa)

P_{vac}: vacuum pressure of vapour (mmHg)

Pr : Prandtl number

Pt : Tube pitch (m)

Q : Heat flux (W/m^2)

Re : Reynolds number

R_{fou} : Fouling resistance ($m^2.K/W$)

Sc : Schemidt number

Sh : Sherwood number

T : Temperature (K)

U_o : Overall heat transfer coefficient ($W/m^2.K$)

u : Velocity (m/s)

w : Mass fraction

w_{ts} : Width of tube sheet (m)

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

μ : Dynamic viscosity(Pa.s)

ρ : Density (kg/m^3)

η : Parameter defined by equation (3)

ε_{eff} : Effectiveness

Subscript

a : Air

atm: atmosphere

CN : Nusselt

c : Condensate

cr : Condensation rate

cs : Vapour/condensate interface

i : Cooling water inside tube

m : Mean water temperature

mix : Mixture

n : Order of certain row

o: outlet

v : Vapour

vcs : Vapour at vapour condensate interphase

w : Tube wall

∞ : Free stream