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A Numerical Model For Thermal-Hydraulic Design of a Shell And Single Pass Low Finned Tube Bundle Heat Exchanger

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Abstract:
This investigation represents a computerized model for the thermal-hydraulic design of a single shell – single pass of enhance tube bundle heat exchanger using the step by step technique (SST). The design procedure suggested in this study is also suitable for multi-tube passes using the most familiar methods of design of shell and tube heat exchanger such as Kern and Bell-Delaware. The (SST) was considered as a basic in order to incorporate the effect of the physical property change due to temperature variation along the heat exchanger. The model basic design depends on the selection of the low finned tube characteristics. The use of such surface will have the advantage of avoiding the space restrictions of the equipment layout in the industrial applications and reduction in the cost of manufacture machining as well. The model was intended to be a choice for the lubricating oil cooling system of a gas turbine power station in Debis-Kirkuk plant. The verification of the model showed that using such enhanced tubes in the cooling system will improve the operating conditions especially during summer season in Iraq.

Key words: Heat Exchangers Design**,** Single Phase**,** Shell and Tube**,** Enhanced Surfaces**,** Low Finned Tube

نموذج عددي للتصميم الحراري - الهايدروليكي لمبادل من نوع القشرة وممر واحد لحزمة الأنابيب المزعنفة

الخلاصة

يقدم هذا البحث نموذج حاسوبي للتصميم الحراري – الهايدروليكي لمبادل من نوع القشرة وحزمة الأنابيب لممر واحد بأستخدام الأنابيب المطورة بإعتماد طريقة الخطوة-خطوة (SST). التصميم المقترح لهذه الدراسة يلائم المبادلات التي تحتوي على اكثر من ممر في جانب الأنابيب بأستخدام الطرق المعتمدة في تصميم هذه المبادلات(Bell-Delaware & Kern).تم اعتماد طريقة الخطوة-خطوة كأساس للتصميم ولضمان تأثير تغير المواصفات الفيزياوية للموائع نتيجة تغير درجة الحرارة على طول المبادل الحراري.
يعتمد التصميم على اختيار المواصفات الفيزياوية (الأبعاد) للأنابيب المزعنفة.أن استخدام مثل هذه الأنابيب له فائدة كبيرة بتجاوز المحددات الخاصة بالحيز والناتجة من توزيع المعدات في التطبيقات الصناعية وكذلك لخفض الكلفة التصنيعية لهذه المبادلات.
النموذج المقترح قد تم انجازه ليشكل جزء رئيسي من منظومة تبريد زيت المحركات المستخدمة في الوحدات التوربينية العاملة في محطة كهرباء (دبس-كركوك) لإنتاج الطاقة الكهربائية. وقد بينت نتائج النموذج المقترح باستخدام الأنابيب المزعنفة تحسن أداء هذه الوحدات وخصوصاً خلال فصل الصيف في العراق.

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11- Nomenclature:

A: Surface Area (m^2)
B: Baffle Spacing (m)
C_p: Specific Heat (kJ/kg Cº)
d: Tube Diameter (m)
d_f: Finned Tube Outside Diameter, figure (2.b), (m)
d_h: Hydraulic Diameter Defined by eq. (14.e)
d_r: Tube Diameter Measured to the Fins Root, figure (2.b), (m)
D: Diameter (m)
f: Friction Factor (dimensionless)
F: Fouling Resistance (m^2 Cº/W)
G: Mass Velocity (kg/m^2 s)
h: Heat Transfer Coefficient (W/m^2 Cº)
H: Fluid Enthalpy (kJ/kg)
j: Colburn Factor (dimensionless)
k: Thermal Conductivity (W/m Cº)
L: Heat Exchanger Length (m)
L_f: Fin Height (m)
LMTD: Logarithmic Mean Temperature Difference (Cº)
m: Mass Flow Rate (kg/s)
n:Number of Tubes per Pass
N_f: Number of Fins per Unit Length
N_p: Number of Passes
N_r: Number of Tubes in Central Row of Bundle
n_t: Number of Vertical Rows in Heat Exchanger
N_t: Total Number of Tubes
Nu: Nusselt Number (dimensionless)
P_f: Fin Pitch (m)
P_p: Pumping Power (kW)
Pr: Prandtle Number (dimensionless)
PR: Tube Pitch Ratio (dimensionless)
P_L: Longitudinal Tube Pitch (m)
P_T: Transverse Tube Pitch (m)
ΔP: Pressure Drop (Pa)
Q: Heat Exchanger Load (kW)
Re: Reynolds Number (dimensionless)
S_f: Fin Gap (m)
St: Stanton Number (dimensionless)
t_f: Fin Thickness (m)
t: Tube Side Fluid Temperature (Cº)
T: Temperature (Cº) or Shell Side Fluid Temperature (Cº)
ΔT_m: Corrected Mean Temperature Difference (Cº)
u_t: Tube Side Fluid Velocity (m/s)

U_o: Heat Exchanger Overall Heat Transfer Coefficient (W/m^2 C°)

Greek Symbols:

δ_f: Parameter Defined by eq. (17.b)
ρ: Fluid Density (kg/m^3)
μ: Fluid Dynamic Viscosity (N/m^2 s)
η_f: Fin Temperature Efficiency (dimensionless)
η: Total Surface Temperature Efficiency (dimensionless)

Subscripts:

b: Bundle
e: Exit or Edge of Baffle Spacing
f: Finned or Fin
i: Inside
Inc: Increment
o: Outside
out: Outlet Value
in: Inlet Value
r: Fin Root Value
S: Shell Side Value
t: Tube Side Value
tot: Total Value
w: Wall Value

Introduction:

The shell and tube heat exchangers are widely used in process industry, power plant, medical applications and many other practical fields. These heat exchangers are capable of handling a quite high load at moderate size with good thermal and hydraulic efficiency in the moderate range of industrial applications. Therefore, this type of heat exchangers occupied a large area of research and investigation to establish the more easily and efficient procedure of design with optimization in its characteristics and cost. For some applications, the size of equipments of the working unit such as heat exchangers is of a vital importance in the site layout due to the space limitations available for both vertical and horizontal orientations. Although, an inclined orientation of heat exchanger may be used, it is not always the available option for the unit layout of equipments due to the above established reasons. The thermal-hydraulic design of the shell and tube heat exchanger according to the original verified standards such as (TEMA) code (1968) [1] was well established due to its highly accurate assessment of the thermal and hydraulic design. Although, many codes was presented for the design of this type of heat exchangers, but it is rarely to find out details about these codes in the open literature.

Butterworth (1973) [2] outlined a general procedure for the design of shell and tube heat exchangers where the overall heat transfer coefficient varies along the heat exchanger. For the case where it does not change in phase from pass to pass, the tube side temperature can be estimated from the enthalpy in the form:

$$dh_{II} / dh_I = -(T - t_{II}) / (T - t_I) ...(1.a)$$

$$H = H_{in} - (m_t / m_S)(h_{II} - h_I) ......(1.b)$$

With the temperature-enthalpy curve of both streams, the temperature can be estimated at each section. A detailed procedure for the calculation of equation (1) is presented by Butterworth and Cousins (1979) [3]. Kern (1950) [4] and Bell-Delaware (1980) [5] presented a procedure for the global thermal and hydraulic design of the shell and tube bundle heat exchanger. There is no available complete approach for the thermal-hydraulic design of the shell and tube heat exchanger where enhanced surfaces are used in the open literature. However, there are comprehensive correlations for the prediction of the heat transfer coefficient on the finned tube surface. Yusur (1997) [6] presented a procedure for the prediction of the thermal design of a horizontal process condenser. In his study a plain and low finned tube surface were used for the design prediction of condensers applicable in the vapor compression refrigeration systems. Mohammed (2005) [7] investigated experimentally and theoretically the thermal-hydraulic design of shell and tube heat exchanger using the step by step technique for a single tube pass. The method described here, depends on experimental data collected for a single tube pass heat exchanger applying only smooth tube bundle. His results showed a good agreement between the experimental data and the predicted parameters of the smooth tube bundle heat exchanger design.

In this investigation, the design of a single shell single tube pass was studied when using low finned tubes in a horizontal heat exchanger without phase change on both streams, shell and tube sides. For this object, an enhanced tube surface, low finned, is suggested for such investigation where its detailed physical characteristics are known and available for the design approach. The enhanced tubes are usually used whenever; the difference in heat transfer coefficient between the two fluids is quite large.

1- **Theoretical Model:**

The model suggested in this study can be described for both tube and shell sides with full details of the step by step numerical technique application.

2-1- **Thermal Analysis:**

2-1-1 **Tube Side:**

The tube side single phase heat transfer coefficient may be estimated by a variety of available correlations, empirical and semi-empirical, suggested by several workers. The pioneer work of Petukhov-Kirillov (1973) [8] postulated an equation for the turbulent flow region, $Re_t > 2100$, in the form:

$$Nu_t = \frac{(f/2)\,Re_t\,Pr_t}{1.07 + 12.7(f/2)^{1/2}(Pr_t^{\,2/3} - 1)} ..(2.a)$$

where the friction factor is defined by:

$$f = (1.58 \ln \mathrm{Re}_t - 3.28)^{-2} ....(2.b)$$

It may be also estimated by the use of Dittus-Boelter (1930) [9] equation applied for the turbulent stream flow as:

$$Nu_t = 0.0243 \mathrm{Re}_t^{\ 0.8} \mathrm{Pr}_t^{\ n} ......(3)$$

where n= 0.4 for heating and n=0.3 for cooling of the fluid stream.
In the laminar flow region where, $\mathrm{Re}_t <$ 2100, the Sieder and Tate (1936) [10] is suggested to be used in the form:

$$Nu_t = 1.86(\mathrm{Re}_t \mathrm{Pr}_t)^{1/3}(d_i / L)^{1/3} ..(4)$$

The heat transfer coefficient is deduced from the Nusselt number by:

$$h_t = Nu_t \frac{k_t}{d_i} ....(5)$$

The Reynolds number, Re_t, is referred to

$$\mathrm{Re}_t = \frac{r_t u_t d_i}{m_t} ....(6.a)$$

and the Prandtle number, Pr_t, is defined by:

$$\mathrm{Pr}_t = \frac{m_t cp_t}{k_t} ....(6.b)$$

The total number of tubes for a heat exchanger mainly depends on the flow requirements on the tube side, number of passes and tube inside diameter in the form:

$$N_t \frac{4 m_t N_P}{r_t u_t p d_i^{\ 2}} ....(7)$$

2-1-2- **Shell Size:**

The shell diameter is usually estimated for smooth tubes from the total number of tubes, tube outer diameter, number of tubes, and tube layout and pitch arrangement. There is no available method for shell size estimation where enhanced tube bundle heat exchanger is used. Since all the above controlling parameters are not a function of the tube condition, the shell size may be estimated using the smooth tube bundle controlling equations with some attention paid to the tube characteristic condition. It is suggested to estimate the shell size by using one of the following methods:

a- **Tube Count Approach:**

In this method, a scale drawing should be prepared depending on the knowledge of:

1- Total number of tubes, N_t, and tube outside diameter, d_o.
2- Tube arrangement and layout, pitch and shape of pattern (triangular or square).

a. Number of passes, N_P, number of tubes per pass, n, and the inside shell diameter clearance, C_S.

From the scale drawing, the shell inside diameter, D_S, is measured from the outer tube bundle diameter, D_b, and the clearance between the extreme outer tube and the inside shell diameter clearance, C_S, in the form:

$$D_S = D_b + C_S ....(8)$$

The clearance, C_S, in (mm) may be estimated according to the shell type from figure (1), [11], or by using the following prepared formula deduced from the above figure in the present work as:

1- Fixed tube sheet and (U) tube bundles:

$$C_S = 10 D_b + 8 ....(9.a)$$

2- Outside Packed Head:

$$C_S = 38 ......(9.b)$$

3- Split-Ring Floating Head:

$$C_S = \frac{1}{9}(400 + 250 D_b) ....(9.c)$$

4- Pull Through Floating Head:

$$C_S = 6.25(14 + D_b) ......(9.d)$$

b- **Mathematical Approach:**

Another method is suggested by the present work to use the same mathematical form of formulae used for the smooth tube by replacing the outside diameter of the low finned tube, d_f , in the form of (d_f=d_r +2 L_f), figure (2.b), in all of the relevant formulae. The following represents this approach describing the shell size estimation.

The number of tubes is calculated by taking the shell circle and dividing it by the projected area of the tube layout, Schlünder (1989) [12]:

$$N_t = z \frac{p D_S^{\ 2}}{4 A_t} ......(10.a)$$

where the projected area for one tube is obtained from:

$$A_t = y P_T^{\ 2} .....(10.b)$$

and the tube count calculation constant values for different tube passes are:

$z = 0.93$ for one tube pass

$z = 0.9$ for two tube passes

$z = 0.85$ for more than two tube Passes and the layout constant values are:

$y = 1.0$ for (90°) and (45°) tube pattern

$y = 0.87$ for (30°) tube pattern

The tube pitch ratio, P_R , is

$$P_R = \frac{P_T}{d_f} ....(10.c)$$

Combination of eqs.(10.a, 10.b), and eq.(10.c) results:

$$N_t = \frac{p z D_S^{\ 2}}{4 y P_R^{\ 2} d_f^{\ 2}} .....(10.d)$$

Therefore, the shell inside diameter has the form:

$$D_S = 0.637 \sqrt{\frac{y}{z}} \left(\frac{A_{of} P_R^{\ 2} d_f}{L}\right)^{1/2} ...(11.a)$$

where (A_{of}) refers to the outside heat transfer surface area based on the outside diameter of the low finned tube, (d_f), and can be calculated from the following formula:

$$A_{of} = p d_f L N_t ....(11.b)$$

2-1-3- **Shell Side Heat Transfer Coefficient:**

The single phase heat transfer coefficient correlation for integral low finned tube bundle in cross flow, given by Briggs and Young (1963) [13], can be applied to make an approximation of the fin heat transfer coefficient in the form:

$$h_f = 0.134 \ \mathrm{Re}^{0.681} \ \mathrm{Pr}^{0.33} \left[\frac{p_f - t_f}{L_f}\right]^{0.2} \left(\frac{P_f}{t_f}\right)^{-0.1134} \left(\frac{k}{d_f}\right) ....(12.a)$$

The Reynolds number is calculated at the root of the tube as a bare tube without fins

$$\mathrm{Re} = G_S d_r / m .....(12.b)$$

Another correlation is given by Ginielinski presented in (1983) [11] to estimate the low finned tube heat transfer coefficient from:

$$h_f = 0.155(\text{Re}')^{0.6}\,\text{Pr}^{0.33}\left(\frac{m}{m_w}\right)^{0.14}\left(\frac{k}{d'}\right) ....(13.a)$$

where

$$\text{Re}' = \frac{G_S d'}{m} ...(13.b)$$

In which

$$G_S = \frac{m_S}{NB[(P_T - d_r) - (d_f - d_r)t_f N_f]} .....(13.c)$$

$$N = \frac{p}{4}\frac{D_S}{P_T} ...(13.d)$$

$$d' = (d_f^{\,2} - d_i^{\,2})^{0.5} ....(13.e)$$

The recommended tube pitch is (1.25) times the outside diameter of the tube and this will be normally used unless process requirements dictate otherwise, that is

$$P_T = 1.25 d_f$$

Figure (2) shows the physical features of a low finned tube presented by Wolverine industrial catalogue (2001) [14].

Shah (1981) [15] recommended a correlation for the single phase heat transfer coefficient presented by Rabas which has the form:

$$j = St\,\text{Pr}^{2/3} ....(14.a)$$

where (j) is the Colburn factor defined by the following equation:

$$j = 0.29\,\text{Re}^n\left(\frac{P_f}{d_r}\right)^{1.115}\left(\frac{P_f}{L_f}\right)^{0.257}\left(\frac{t_f}{P_f}\right)^{0.666}\left(\frac{d_f}{d_r}\right)^{0.473}\left(\frac{d_r}{t_f}\right)^{0.772} ....(14.b)$$

where the exponent (n) on Reynolds number is given by:

$$n = -0.415 + 0.0346\ln\left(\frac{d_f}{P_f}\right) ...(14.c)$$

(St) is Stanton number given by:

$$St = \frac{Nu}{\text{Re}\,\text{Pr}}$$

and

$$j = \frac{Nu}{\text{Re}\,\text{Pr}^{1/3}}$$

Then

$$h_f = j\,\text{Re}\,\text{Pr}^{1/3}\left(\frac{k}{d_h}\right) ...(14.d)$$

where Reynolds number is based on the hydraulic diameter defined by:

$$\text{Re} = \frac{G_S d_h}{m}$$

And the hydraulic diameter is expressed by:

$$d_h = 4B[\frac{(P_T - d_r) - (d_f - d_r)t_f N_f}{p[(\frac{d_f^{\,2} - d_r^{\,2}}{2} + d_f t_f)N_f + (1 - t_f N_f)d_r]}]..............(14.e)$$

2-1-4- **Overall Heat Transfer Coefficient:**

The overall heat transfer coefficient for integral low finned tube is bundle can be calculated on the basis of total surface area [14], as follows:

$$U_o = [F_i + \frac{1}{h_t(\frac{A_i}{A_{tot}})} + \frac{d_f \ln(\frac{d_f}{d_i})}{2k_w} + F_o + \frac{1}{h_f h}]^{-1}.....................(15)$$

where (η) is the surface efficiency and may be estimated from:

$$h = 1 - \frac{A_f}{A_{tot}}(1 - h_f)..(16.a)$$

(A_f) is the area of fins exposed to the fluid stream, and calculated in the present work by the form:

$$A_f = L[\frac{p}{2}(d_f^{\,2} - d_r^{\,2})N_f + pd_f t_f N_f]....(16.b)$$

(A_r) is the base area, calculated from:

$$A_r = pd_r(1 - t_f N_f)L....(16.c)$$

The total area, A_{tot} , is:

$$A_{tot} = A_f + A_r...(16.d)$$

(η_f) is the fin efficiency given in graphical form. By using the least square method, the second order polynomial was fitted, with standard deviation of (0.9 %) [6] as:

$$h_f = 1 - 0.235d_f - 0.117d_f^{\,2}.(17.a)$$

where

$$d_f = Z_c^{\,1.5}(\frac{h_f}{k_w A_G})...(17.b)$$

$$A_G = t_f Z_c....(17.c)$$

$$Z_c = \frac{d_f - d_r}{2} + \frac{t_f}{2}..(17.d)$$

2-2- **Hydraulic Analysis:**

2-2-1- **Tube Side:**

The pressure drop encountered by the fluid making (N_P) passes through a heat exchanger plus the additional pressure drop introduced by the change of direction in the passes are multiplied by the kinetic energy of the fluid flow. Therefore, the tube side pressure drop is calculated, [11], by the formula:

$$\Delta P_t = 4N_P[f\frac{L}{d_i} + 1]\frac{r_t u_t^{\,2}}{2}...(18.a)$$

where

$$f = (1.58\ln \mathrm{Re}_t^{\,-3.28})^{-2}..(18.b)$$

2-2-2- **Shell Side:**

The pressure drop for low finned tube bundle can be calculated from the correlation developed by Rabas et.al. (1980) [16] in the expression:

$$\Delta P_f = 2f\frac{G_S{}^2}{r}N_r \,..(19.a)$$

where

$$N_r = \frac{D_S}{P_T} \,...(19.b)$$

The finned tube friction factor is presented by the following expression:

$$f = 3.805\,\mathrm{Re}^{-0.2336}(\frac{S_f}{d_f})^{0.2512}(\frac{L_f}{S_f})^{0.7593}(\frac{d_r}{d_f})^{0.7292}(\frac{d_r}{P_T})^{0.709}(\frac{P_T}{P_L})^{0.3791} \,...(19.c)$$

The above equation is applied for the following limitation conditions:

$$1000 \le \mathrm{Re} \le 25000$$

$P_L \le P_T$ and $n_t \ge 6$

The expression developed by Chai (1988) [17] may also be applied to the pressure drop estimation; the friction factor is formulated by:

$$f = 1.748\,\mathrm{Re}^{-0.233}(\frac{L_f}{S_f})^{0.552}(\frac{d_r}{P_T})^{0.599}(\frac{d_r}{P_L})^{0.1738} \,...(20)$$

This correlation is valid for the following conditions:

$895 \le \mathrm{Re} \le 713000$ $\quad 20^\circ \le T_{LA} \le 40^\circ$

$(P_T / d_r) \le 4$ and $n_t \ge 4$

The Engineering Science Data Unit, ESDU (1985) [18] developed a pressure drop correlation for low finned tube heat exchangers where the friction has the form:

$$f = 4.71\,\mathrm{Re}^{-0.286}(\frac{P_T}{d_r}-1)^{-0..36}(\frac{L_f}{S_f})^{0.51}(\frac{P_T-d_r}{P_L-d_r})^{0.536} \,....(21)$$

This equation is valid for the following conditions:

$1000 \le \mathrm{Re} \le 800000$ and $n_t \ge 10$

The above equations (19.c), (20) and (21) are applied with Reynolds number based on root diameter, d_r , and maximum velocity, v_{max} . The latter occurred at minimum flow area, A_{min} , where maximum mass velocity takes place in the form of eq.(13.c) and

$$\mathrm{Re} = \frac{G_S d_r}{m} \,...(22)$$

The above equations were included in the model presented in this study to allow the designer to have a variety of options to be selected to suit heat exchanger design. Further, the calculation of the pressure drop of both streams will be reflected on the power consumption by the system circulation pumps. The power consumption was estimated in this work using the following formula:

$$P_p = \frac{m\Delta P}{r h_p} \,.....(23)$$

This equation can be applied for both side of the heat exchanger, shell and tube, by using the appropriate values of the involved variables.

2-3- **Step By Step Approach:**

As mentioned earlier in this study, the model presented here is devoted to the single shell-single tube pass heat exchanger allowing the use of enhanced surface of the low finned tube type. Figure (3) represents the scheme of heat exchanger division to a number of increments, equally sized, for the whole baffle spacing of the heat exchanger. Noting that the mathematical model presented here depends on the following assumptions:

1. The heat transfer mechanism follows one dimension and the existence of forced convection region for the fluid flow.
2. All of the thermal physical properties of both of streams, pure fluid or mixture, are in direct variation with temperature only along the heat exchanger.
3. The heat exchanger is fully insulated from the surroundings; therefore, the heat rejected out of one stream would be absorbed by the other.

An energy balance for one increment is described as follows:

a- The energy balance on both sides of heat exchanger is

$$Q_t = m_t cp_t (t_{in} - t_{out})_{Inc} \, ..(24.a)$$

$$Q_S = m_S cp_S (T_{out} - T_{in})_{Inc} \, ...(24.b)$$

For counter flow and cooling on the tube side, the inlet temperature (t_{in}) is higher than (t_{out}) and therefore (T_{out} > T_{in}) and the opposite is true.

b- The key of the calculation procedure starts from the knowledge of two temperatures, one on each stream side. Therefore, the assumption of one exit temperature from the increment will ease the calculation and the fourth value of temperature will be obtained from the above energy balance, eq.(24). The assumed temperature should be consequently proved to be correct.

c- The energy balance obtained above should satisfy the heat transfer rate equation:

$$Q_{Inc} = [UA\Delta T_m]_{Inc} \, ...(25)$$

In this equation, the (RHS) of the statement is known, since:

i- The value of $(\Delta T_m)_{Inc}$ is defined by the four temperature values described above.

ii- The increment surface area is defined by the full dimensions of (d_f , d_r , L_f , t_f , N_f) and the length of increment is specified by the designer.

iii- The overall heat transfer coefficient of increment, U_{Inc} , is calculated from the knowledge of (h_i), (h_f), and other parameters related to the existence of fins, eq.(15).

d- Comparing the energy balance at eq.(24) with that obtained by eq.(25) will produce the limit of accuracy for the assumed temperature at exit of the increment. Repeating the calculation until acceptable satisfaction of accuracy is obtained for the assumed temperature of the increment. This temperature value will be considered as the entering temperature for the next increment. This scheme will be repeated until the final process temperature at exit for the heat exchanger is obtained according to the thermal design requirements. The calculation fashion covers the hydraulic requirements as well. The design procedure was terminated when the absolute temperature difference between the predicted and calculated values of the process fluid fell within acceptable limit. A value of this limit was chosen to be equal or less than a value of (0.1 %).

4- Case Study:

For verification of the model presented in this work, it is suggested to apply the model to an industrial field cooling process problem in the electricity power production. The case considered is related to gas turbine power plant cooling oil system. There is a restriction for the working temperature of the oil of these units. When the engine oil temperature exceeds (75 C˚), the ability of these units for power production will undergo a severe restrictions leading to shut down the engine. Therefore, a cooling system was designed by the manufacturers companies to construct a closed loop of using air cooled system. This cooling system mode suffers from restriction of summer condition due to temperature rise of the environment which in turn has a great draw back to the generation unit efficiency. Further, the air cooled system is inefficient for cooling down the oil temperature to acceptable level of about (55 C˚). It is postulated for such units working in severe summer conditions such as that existing in Iraq to design a cooling system using a shell and tube heat exchanger with open circulating system for the cooling water side. This cooling source is withdrawn from a water source such as a river. This method can also be used in a closed circulating system in connection with a cooling tower to cool down the cooling water.

The above cooling system may also be used in connection with the air cooled system, Figure(4), to allow a reliable option in the choice of the cooling media, air or water systems. For the conditions of circulating oil shown in table (1), the heat exchanger will be designed for both cases of plain and enhanced surfaces for the verification of the technique presented in this research.

5- Results And Discussion:

5-1- Overall Design Review:

The design requirements for the circulating oil conditions, Table(1), will be used for the design of a low finned tube heat exchanger. The characteristics of both tubes, smooth and low finned surfaces are shown in Table (2). The final design requirements for both surfaces are shown in Table (3) to satisfy the acceptable range of oil conditions. The data shown in this table was deduced from the use of Rabas et.al [16] correlation for the pressure drop estimation and the correlation of Briggs and Young [13] for the heat transfer coefficient for the low finned tube surface. It is obvious from the results of this table that the low finned surface heat exchanger is shorter than that of the smooth tube surface for the same heat exchanger load. For the step by step technique, the reduction in the heat exchanger length is (25%) and about

(18%) for the application of Kern procedure when compared with that of the plain tube. Further, the present design model exhibited a large reduction in the pressure drop for both sides of the heat exchanger. The percentage of the reduction in heat exchanger pressure drop on the shell side (process fluid) is about (25-38) % for the (SST) and (19%) for the Kern method when compared with the smooth tube heat exchanger. The operating conditions of both of streams, tube and shell sides, are presented in table (4). In this table it is suggested to use river water, since the gas turbine power plant considered in this study, Debis-Kirkuk is established nearby a river. However, the suggested cooling unit may use a cooling tower water or any other source.

5-2- Evalu\ation of Design Correlations:

Table (5) is prepared to perform an evaluation for the various correlations presented in the present work for the estimation of the heat transfer coefficient and the pressure drop exhibited by the low finned tube. It is obvious that the correlation presented by Ginielinski [11] predicts higher values for the heat transfer coefficient on the low finned tube than the other methods. The predicted value is about three times the predicted values by Rabas [15] and Briggs and Young [13] which increases the overall heat transfer coefficient and reduces the heat exchanger size, Table (5.a). In fact, the predicted heat transfer coefficient of the low finned surface by [15] and [13] are almost the same which in turn produces contiguous heat exchanger length and shell side pressure drop, table (5.b). Further, the (SST) method predicts lower heat exchanger length than the Kern method constructed in this work by (9%). The power consumption saving of the circulating system due to the application of the enhanced surface when compared with that of the plain tube heat exchanger is ranged between (25-38)% according to the hydraulic correlation used for the shell side pressure drop estimation. The power consumption of the circulating pumps is based on an efficiency of (80%) for both sides of the heat exchanger.

5-2-1- Low Finned Surface Heat Transfer Coefficient:

The correlations formulated by Rabas [15] and Briggs and Young [13] are strongly recommended for the estimation of the low finned surface heat transfer coefficient as it is seen for the results of this work. Figure(5) shows the variation of the overall heat transfer coefficient of the low finned tube heat exchanger with the baffle spacing along the flow direction of the tube side service fluid for both of the above methods. The results deduced from the present model for the (U_o) values are shown in Table (6.a). It is obvious that the (U_o) values increase in the direction of the water flow direction where corresponds to the entering shell side process fluid (oil) to these baffle spacing sections. That is, the water performs a complete counter flow scheme with the hot entering oil. However, the prediction of the correlation of Briggs and Young [13] when compared with that of Rabas [15] shows a higher values of (U_o) by (9-13) % along the heat exchanger in spite of having the same trend of distribution. Applying these correlations in the design of heat exchanger with Kern method reveals a contiguous value of the mean overall heat transfer coefficient when compared with those predicted by the (SST) model. The predicted values of the fouled (U_o) for Kern method by using Briggs and Young [13] and Rabas [15] correlations are 138 (W/m^2 Cº) and 125.5 (W/m^2 Cº) respectively. The respective values of (U_o) for the (SST) model are 139 (W/m^2 Cº) and 127 (W/m^2 Cº) for the above correlations.

5-2-2- **Low Finned Surface Pressure Drop:**

The pressure drop estimated by the (ESDU) [18] method shows a high values for the shell side and much higher than those estimated by Rabas et.al [16] and Chai [17] correlations. This could be due to the application conditions imposed on its usage, the Reynolds number and other parameters. The results of the expressions of [16] and [17] are contiguous in their predictions, table (5.b).

5-2-3- **Low Finned Surface Temperature Distribution:**

The temperature distribution of both of streams, tube and shell sides, predicted for the above suggested methods, [11], [15], and [13] are shown in table (6.b). Figure (6) shows the variation of the temperature on both sides of the heat exchanger predicted in the present (SST) model. The data for this figure is deduced from Rabas [15] correlation for the low finned tube heat transfer coefficient. The results show a smooth temperature distribution scheme in the heat exchanger.

5-3- **Effect of Baffle Spacing:**

It is very interesting to show the flexibility of the present model and its response to the different parameters encountered in the thermal-hydraulic design of the heat exchanger. One of the most important parameters in the shell and tube heat exchanger design is the baffle spacing on the shell side for a given specified heat duty and fluid streams conditions. For this purpose, it is suggested to use different baffle spacing which leads to different number of baffles fixed in the heat exchanger in order to examine the present model for a given duty requirements.

This choice will show the criteria of the heat exchanger relevant to the overall heat transfer coefficient and shell side pressure drop. In turn, this will affect the design optimization directly through the power consumption on the shell side and total surface area required for the given job. Since, these factors are related to the size of the equipment, the manufacturing process, and space requirements.

To clarify the above mentioned argument, a choice should be made for the methods of the prediction of the heat transfer coefficient and pressure drop on the shell side to perform the comparison for baffle spacing effect. The heat transfer coefficient will be predicted by Rabas [15] correlation, eq.(14.a), since it showed a moderate values, table (5.a), among the rest of methods outlined in this work. Rabas et.al. [16] correlation, eq.(19.c), predicted the highest pressure drop on the shell side among those presented in table (5.b). Therefore, it will be selected for this purpose to show the effect of baffle spacing on the performance of the heat exchanger. Table (7) was prepared to show the effect of baffle number and baffle spacing on the considered parameters to describe the performance of the heat exchanger. This data were obtained by using the operation conditions of the case study shown in table (4).

The overall heat transfer coefficient showed a smooth decrease in its value as the baffle spacing was increased, figure (7). This of course was due to the reduction in the fluid flow velocity on the shell side which reduces the heat transfer coefficient obtained from the finned bundle. The results also revealed that increasing the baffle spacing by (60 %) causes a decrease in the heat exchanger overall heat transfer coefficient of about (33 %). On the other hand, this exhibited a sharp and great reduction in the shell side fluid stream pressure drop and power consumption of about (60 %), figure (8). However, it is not recommended to design a heat exchanger with a small spacing between the baffles to avoid the penalty of high pressure drop and consequently a higher pumping power and cost. The results of this figure showed that the gradient of the pressure drop with baffle spacing was steeper (higher) for the baffle spacing above (750 mm) then becomes flatter there after.

The above arguments conclude that a compromise should be made between heat exchanger size and cost of different parameters affecting the economic design of the heat exchanger.

6- Conclusion:

The results of the design model presented here showed that:

1- A thermal and hydraulic design of the single shell pass and enhanced tube bundle heat exchanger design model is presented with a powerful tool to be used in the gas turbine power plant in Iraq at high efficiency.
2- A quite large saving in the power consumption of the heat exchanger circulating system requirements when compared with that of the smooth tube. This is mainly due to the reduction in the run followed by the fluid passing there, and the reduction in the pressure drop in the shell and tube sides when was compared with that of the plain tube heat exchanger.
3- From the results of the presented design model in this work, it is strongly recommended to apply the Rabas [15] and Briggs and young [13] correlations for the low finned tube heat transfer coefficient estimation. Further, the (ESDU) [18] method for the pressure drop estimation for the low finned tube predicts very high pressure drop and it is not favorite for the design application of the present work.
4- Avoiding the problem of space restrictions when using the low finned tube surface which in turn offer a reliability in equipments layout of the unit.

5- Although, the cost requirements is not considered in the present model, but it is expected to be lower for the enhanced tubes from the point of view of workshop machining of the heat exchanger than that of the plain tube one. This of course does not include the cost of enhanced surfaces.

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Table (1): The Operating Conditions of Circulated Oil in Gas Turbine Power Plant of (Debis- Kirkuk).

Entering Temp. (C°)	Leaving Temp. (C°)	Flow Rate (m^3/hr)	Working Pressure (bar)	Allowable ΔP (bar)
75	55	283	6.5	1

Table (2): The Characteristics of Tube Surfaces Considered for the Model Verification.

Tube Surface	O.D. (mm)	I.D. (mm)	FPI (Fin/in.)	Fin Height (mm)	Fin Thick. (mm)	Fin Pitch (mm)	Fin Gap (mm)
Low-Finned	18.8	13.74	19	1.5	0.38	1.39	1.01
Smooth	19.0	15.75	---	---	-----	----	----

Table (3): A Comparison for the Thermal Design of the Heat exchanger Between the Present Work and the Smooth Surface Application for (Debis-Kirkuk) Plant.

Dimensions	Smooth Tube	Low-Finned Tube Kern Method	Low-Finned Tube Step By Step (SST)
Tube O.D. (mm)	19	18.8	18.8
Tube Length (m)	6.0	4.8	4.35
No. of Tubes/Pass	257	1214	1214
No. of Passes	4	1	1
Total No. of Tubes	1028	1214	1214
Tube Layout And Pitch	Δ at 24 (mm)	Δ at 23.5 (mm)	Δ at 23.5 (mm)
Shell Inside Diameter (m)	1.0	0.9	0.9
Baffle Spacing (mm)	500	750	750
Shell Side Nozzle (mm)	150	150	150
Tube Side Nozzle (mm)	200	200	200
Shell Side ΔP (kPa)	97	69	62
Tube Side ΔP (kPa)	22	0.6	0.5
Tube Material	Galv. C.St.	Galv. C.St.	Galv. C.St.
Shell Material	C.St.	C.St.	C.St.

Table (4): The Operating Conditions of the Heat Exchanger Requirements.

Characteristics	Shell Side	Tube Side
Circulated Fluid Stream	Oil (4003)	River Water
Entering Fluid Temperature (C°)	75.0	35.0
Leaving Fluid Temperature (C°)	55.0	50.0
Fluid Flow Rate (kg/hr)	244000	162000
Operating Pressure (bar)	6.5	4.0
Fouling Resistance ($m^2.C^o/W$)	0.0007	0.00034
Material of Construction	Carbon Steel	Galvanized Carbon Steel

Table (5.a): Evaluation of the Different Correlations for Heat Transfer Coefficient Estimation of the Low Finned Tube Heat Exchanger with the (SST) Model.

Method	L_{tube} (m)	h_{tube} (W/m^2 C°)	h_{Shell} (W/m^2 C°)	U_o (W/m^2 C°)	A_{total} (m^2)	$Q_{H.E.}$ (kW)
Ginielin.[11]	1.95	1969	450	323	412.5	2799
Rabas [15]	4.8	1950	144.5	127	1014.3	2801
Briggs And Young [13]	4.35	1943	160.3	139	919.4	2801

Table (5.b): Evaluation of the Different Correlations for Pressure Drop Estimation of the Low Finned Tube Heat Exchanger with the (SST) Model.

H.T.C. Method	ΔP_{Shell} Method	ΔP_{Shell} (kPa)	Pump Power (kW)	Power Saving (%)
Ginielinski [11]	Rabas et.al.[16]	31.0	3.0	---
	Chai [17]	30.0	3.0	---
Rabas [15]	Rabas et.al.[16]	72.0	7.0	25
	Chai [17]	70.0	6.9	28
Briggs And Young [13]	Rabas et.al.[16]	62.0	6.1	36
	Chai [17]	60.6	6.0	38

Table (6.a): The Variation of the Overall Heat Transfer Coefficient with Baffle Spacing Along the Heat Exchanger of the Present Work.

Ref.No.	B.No. 1 U_o	B.No. 2 U_o	B.No. 3 U_o	B.No. 4 U_o	B.No. 5 U_o	B.No. 6 U_o	B.No. 7 U_o
[11]	307.7	321.7	338.0				
[15]	112.4	115.6	119.5	124.3	130.4	138.8	146.8
[13]	124.4	128.5	133.5	139.9	148.5	159.7	------

Table (6.b): The Temperature Distribution Along the Heat Exchanger for the Present Work.

Ref.No.	B.No. 1 T_e (C°)	B.No. 2 T_e (C°)	B.No. 3 T_e (C°)	B.No.4 T_e (C°)	B.No. 5 T_e (C°)	B.No. 6 T_e (C°)	B.No. 7 T_e (C°)
[11] S	62.4	70.74	75.0				
T	40.5	46.70	49.97				
[15] S	57.60	60.40	63.40	66.56	69.99	73.74	75.00
T	36.94	39.00	41.20	43.60	46.20	49.00	50.00
[13] S	57.90	61.00	64.40	67.98	71.95	75.00	
T	37.15	39.46	41.95	44.66	47.66	50.00	

S: Shell Side T:Tube Side

Table (7): The Effect of Baffle Spacing and Baffle Number on the Heat Exchanger Performance.

Baffle Spacing (mm)	L_{tube} (m)	h_{Shell} (W/m^2 C°)	U_o (W/ m^2 C°)	ΔP_{Shell} (kPa)	Pump Power (kW)
500	3.4	215	179	148	14.6
600	4.0	180	154	107	10.5
700	4.5	155	135	81	8
750	4.8	145	127	66.5	6.5
800	5.1	136	120	64	6.3

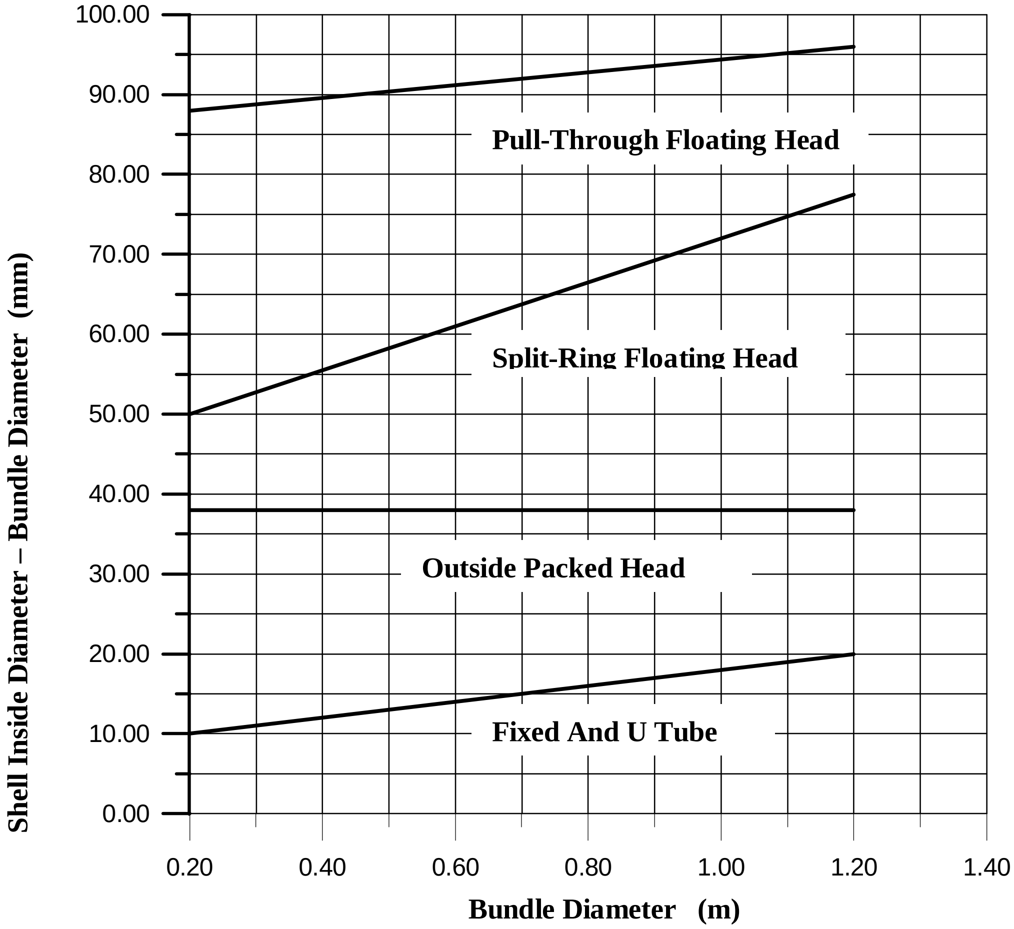

Figure (1): Shell Inside Diameter Clearance Estimation

Figure (2.a): A Low Finned Tube Structure Photograph

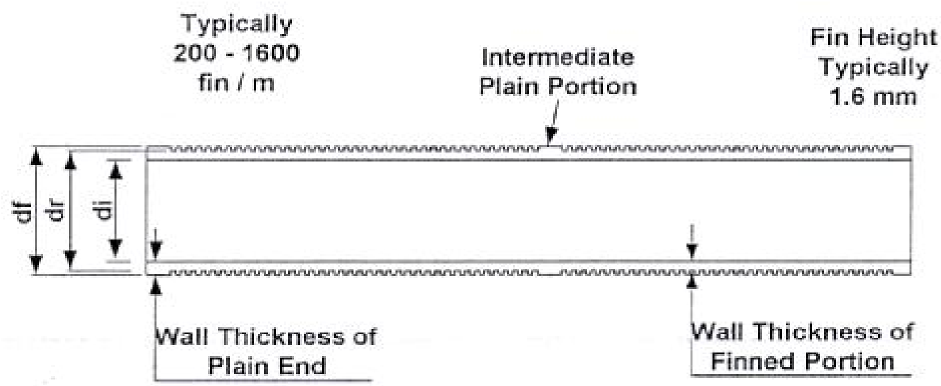

Figure (2.b): Schematic Diagram of Main Features of Low Finned Tube

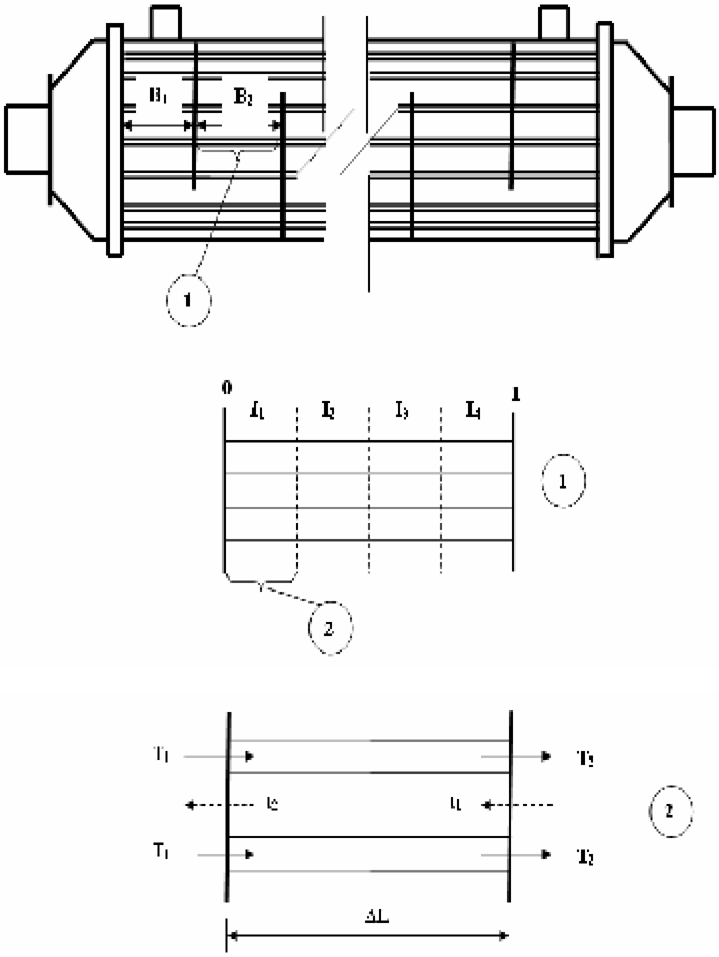

Figure (3) Schematic Diagram Showing the Assembly of Step by Step Method for Single Shell – Single Tube Pass Heat Exchanger.

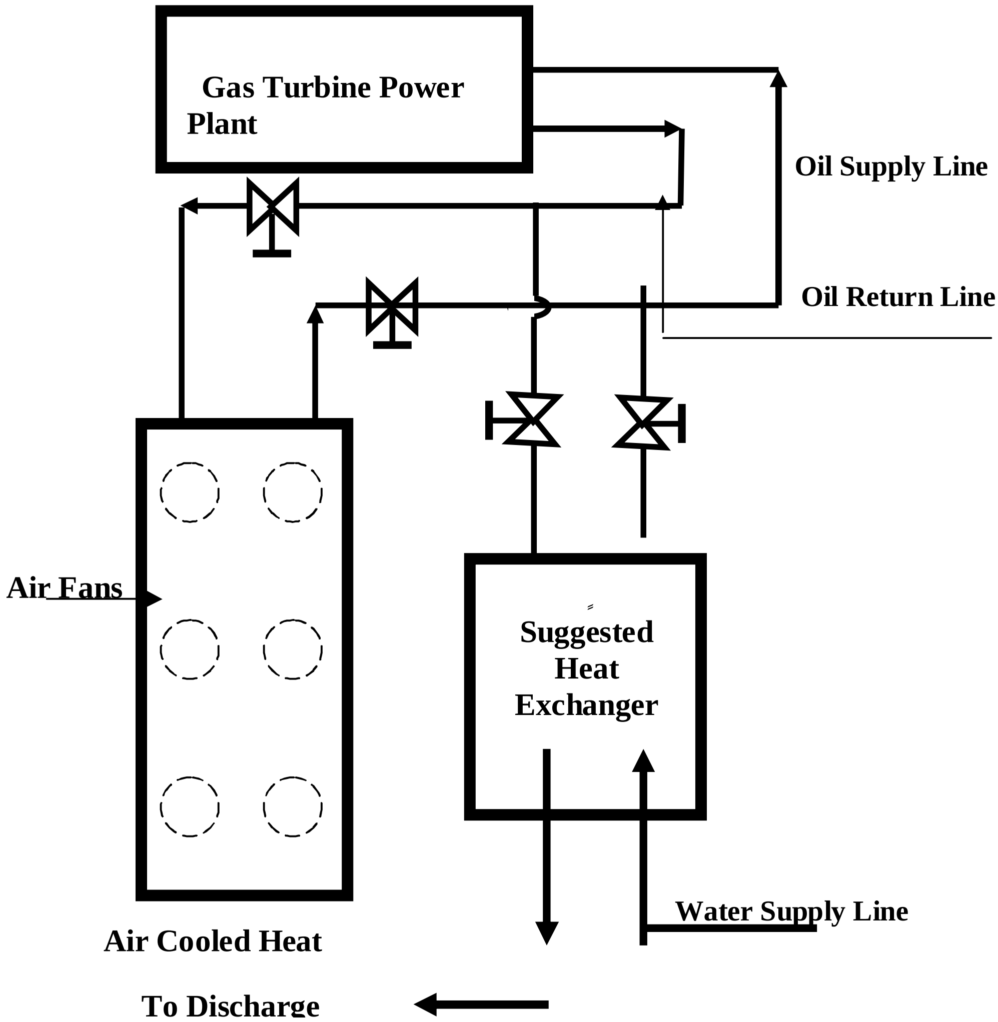

Figure (4): Schematic Diagram Showing the Suggested Cooling

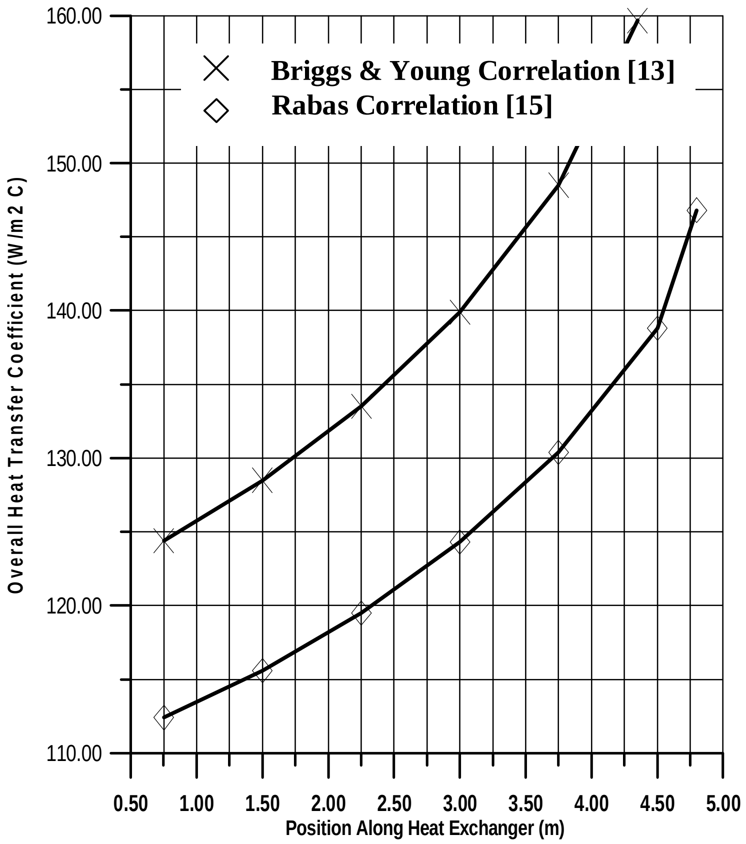

Figure (5): The Variation in Overall Heat Transfer Coefficient for Different Methods of Low Finned Tube Heat Transfer Coefficient Prediction.

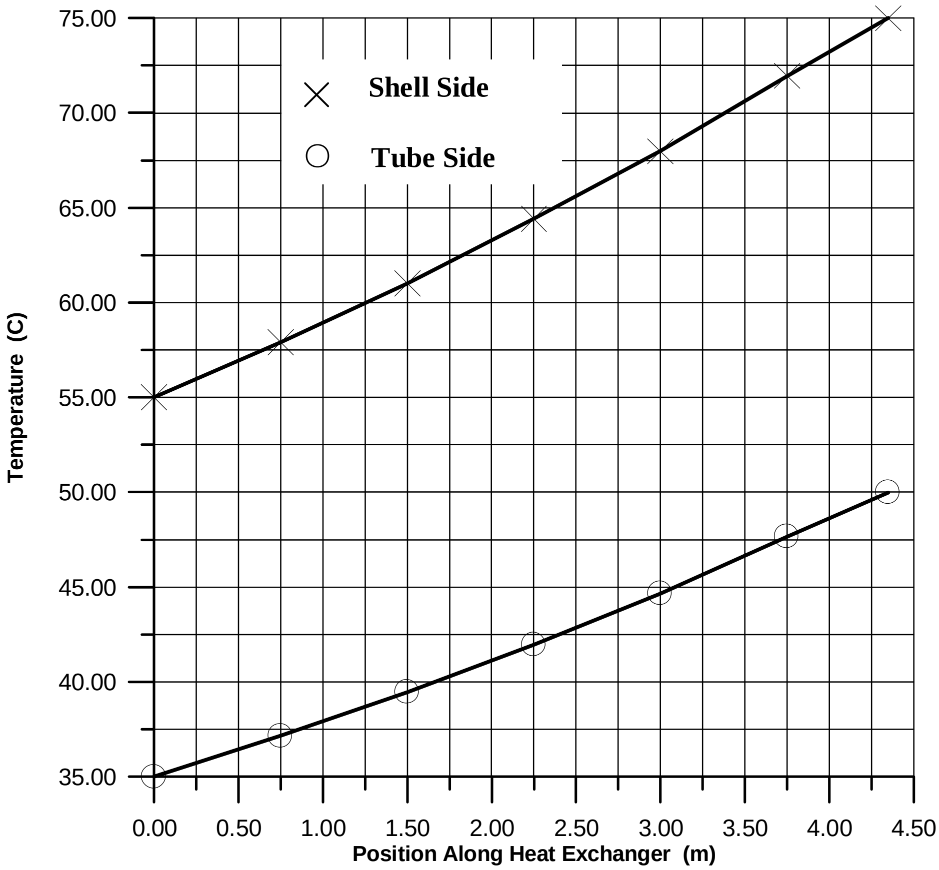

Figure (6): The Distribution of Temperature Along The Heat Exchanger for Both Sides Applying Rabas [15] for Low Finned Tube Heat Transfer Coefficient Prediction.

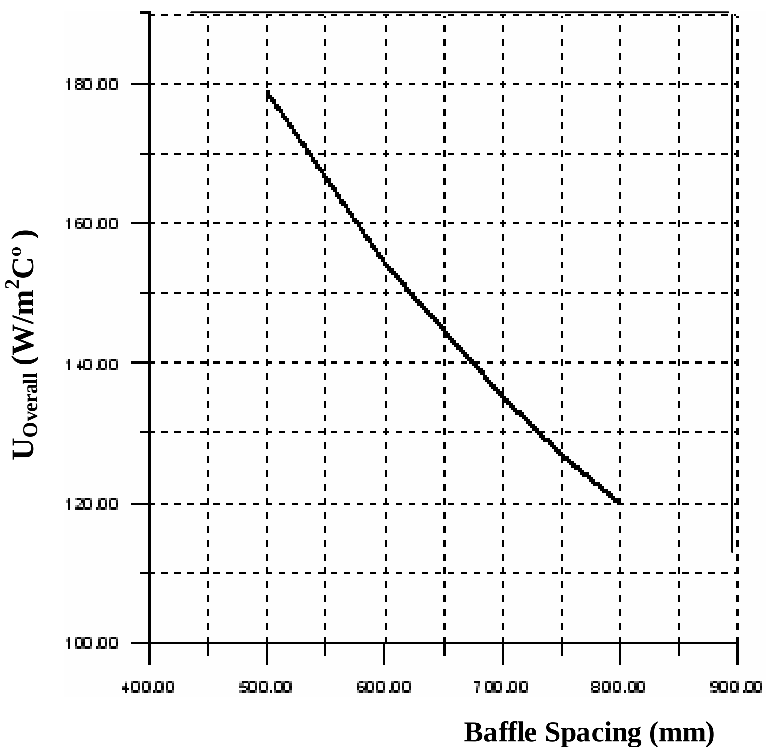

Figure (7): The Variation of the Overall Heat Transfer Coefficient with Baffle Spacing of Finned Tube Bundle Heat Exchanger.

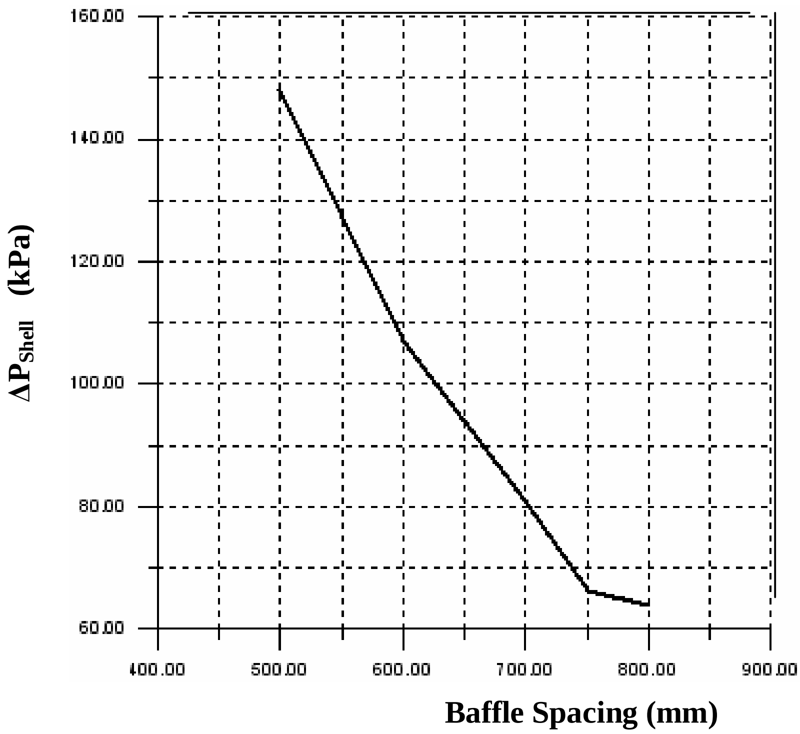

Figure (8): The Variation of the Shell Side Pressure Drop for Finned Tube Bundle with Baffle Spacing of the Heat Exchanger.