

## ***A Mathematical Model for Thermo-Hydraulic Design of Shell and Tube Heat Exchanger Using a Step By Step Technique***

*Dr. Ali Hussain Tarrad*

*Mechanical Engineering Dept., College of Eng.  
Al-Mustansiriya University, Baghdad, Iraq*

*Asst. Lect. Ali Ghazi Mohammed*

*Mechanical Engineering Dept., College of Eng.  
Al-Mosul University, Mosul, Iraq*

### **Abstract**

*In this work experimental and theoretical model predictions for thermal and hydraulic design of shell and tube heat exchanger are presented. The tests were carried out at hot fluid temperature range between (40) to (60) C° at atmospheric pressure for volumetric flow rates ranged between (800) to (1800) l/hr.*

*The model presented for this object was suggested to be accomplished by using the step by step technique. In this model, the heat exchanger was divided into longitudinal increments along the heat exchanger for both tube and shell sides. The output of each increment for both sides of process and service fluids including the thermal and hydraulic parameters are considered to be the input of the next increment and so on until the final temperature and load of the heat exchanger together with the hydraulic requirements were reached.*

*Two methods were applied in the suggested design model, Kern and Bell-Delaware, in addition to the step by step technique. The prediction of heat exchanger performance of the present model well agreed with the above mentioned methods.*

*The results of the present model showed a good agreement with the experimental data obtained during this investigation for the performance parameters considered in the model. The predicted values of the overall heat transfer coefficient showed a divergence ranged between (15%) and (17%) for service fluid flow rates of (850) and (1000) l/hr respectively.*

## الخلاصة

في هذا العمل تم التنبؤ بالتصميم الحراري- الهيدروليكي عملياً وبنموذج رياضي لمبادل من نوع القشرة وحزمة الأنابيب. التجارب العملية تم إجرائها لمدى درجة حرارة تتراوح بين (٤٠ - ٦٠) م° للمائع الساخن عند الضغط الجوي ولمعدل جريان حجمي يتراوح بين (٨٠٠ - ١٨٠٠) لتر/ساعة. النموذج الرياضي المقترح لهذا البحث يعتمد على طريقة الخطوة خطوة. لقد تم تقسيم المبادل إلى مجموعة من الخلايا على طول المبادل ولكلا الجانبين، القشرة والأنابيب. المتغيرات المستحصلة من كل خلية لجانبى الجريان (المائع العملياتي والمائع الخدمي) متضمناً المتغيرات الحرارية والهيدروليكية تعتبر كمدخلات للخلية التالية وهكذا حتى يتم استيفاء متطلبات درجة الحرارة، الحمل الحراري، وهبوط الضغط للتصميم. هناك طريقتان تم تطبيقها في النموذج المقترح للتصميم، (Bell-Delaware & Kern) بالإضافة لطريقة الخطوة خطوة. الأداء الذي تم التنبؤ به للمبادل الحراري خلال هذا البحث متفق تماماً مع الطرق المذكورة في أعلاه والتي اعتمدت بين طيات هذا العمل. وقد بينت النتائج للنموذج الحالي توافق جيد مع النتائج العملية التي تم الحصول عليها خلال هذا البحث للمتغيرات التي اعتمدت كمؤشر للأداء. أقيم النظرية لمعامل انتقال الحرارة الكلي أظهرت انحراف عن تلك المقاسة عملياً لمدى يتراوح بين (١٥%) و (١٧%) لمعدل جريان للمائع العملياتي يتراوح بين (٨٥٠) و (١٠٠٠) لتر/ساعة على التعاقب.

## 1. Introduction

The shell and tube heat exchanger is a well known type of heat exchangers used in the industrial applications. This is due to the simplicity and confidence in its thermal-hydraulic design. Further, it has a wide application in chemical process engineering, electric power station, food industry, air conditioning equipment and many other life fields. However, the design optimization still lacks some information concerning thermal and hydraulic data. In addition, the design of the shell and tube bundle heat exchanger needs the art of thermal and mechanical optimization. Therefore, a model for the design of new heat exchanger or rating an existing one to satisfy the thermal and hydraulic requirements is required to minimize the effort and manufacturing cost.

Patanker and Spalding<sup>[1]</sup>, created a fashion in process equipment design applying a numerical procedure for calculating fluid flow distribution. They computed the velocity distribution in addition to the pressure and temperature through out the heat exchanger. Mikhailov and Ozisik<sup>[2]</sup>, adopted a finite element model of analysis for heat exchanger calculation. Ravikumaur, et. al.<sup>[3]</sup>, presented a finite element model to predict the temperature distribution in a shell and tube heat exchanger. The model can be effectively used to analyze and design heat exchanger. Lorenzini, et. al.<sup>[4]</sup>, presented a modeling of fluid flow and heat transfer in heat exchangers to investigate the effect of different turbulence models on the velocity field and heat transfer coefficient.

Keene, et. al.<sup>[5]</sup>, investigated the effects of baffles length and position on the flow and heat transfer, and choosing an optimal baffle size and position for the shell and tube heat exchanger considered. Vieira, et. al.<sup>[6]</sup>, explored a new design algorithm based on total annual cost optimization for thermal equipment, with mean tube side and shell side flow velocities, constraints, studying also the influence of pumping cost in networks final cost.

Stevanovic, et. al.<sup>[7]</sup>, used a (CFD) technique to carry out thermo-hydraulic calculation and geometric optimization for shell and tube heat exchanger. They calculated the velocity and temperature distribution as well as the total heat transfer rate. Yusur<sup>[8]</sup>, investigated a step

by step method for thermal-hydraulic design of a shell and tube condenser based on Silver method in which the baffle spacing are considered one by one.

In the present work a step by step technique is applied to find out the thermal and hydraulic design of the single pass shell and tube heat exchanger. Although, the work was verified by experimental work, it was also compared with Kern <sup>[9]</sup>, and Bell-Delaware <sup>[10]</sup>, methods. A computer program model was built for this purpose to incorporate the idea of step by step method suggested in this study.

## 2. Experimental Apparatus

An experimental test system was designed and executed prior to the tests depending on available shell and tube heat exchanger. **Figure (1)** shows a schematic diagram of the test system. It is mainly consisting of two circuits for process and service fluid circulation. Two circulating loops are designed for hot and cold streams to pass through the heat exchanger and two reservoirs. One tank is used for cold (service) water circulation in an open loop and the other is used for preparing the hot (process) water passing through the tube side of the heat exchanger. Two pumps were used to establish the flow process together with control panel for heating elements (12 kW) control and instruments. The temperature and pressure of both streams were measured at the entering and leaving positions of the heat exchanger for both sides.

The test section (Heat Exchanger) is shown in **Fig.(2)**. It is of Bowman Type (3739-5) having dimensions of (1000 mm) overall length, (800 mm) tube effective length, which contains (35) copper tubes of (11 mm) inner diameter and (13.5 mm) outer diameter. The tubes are distributed as a triangular (30°) tube pattern. The clearance between two adjacent tubes is (1.5 mm), and the tube pitch is (15 mm). The shell inner diameter is (114 mm). The shell side inlet and outlet nozzles are of (25.4 mm), the tube side inlet and exit connections are of (50.8 mm), and the total tubes surface area is (1.187 m<sup>2</sup>).

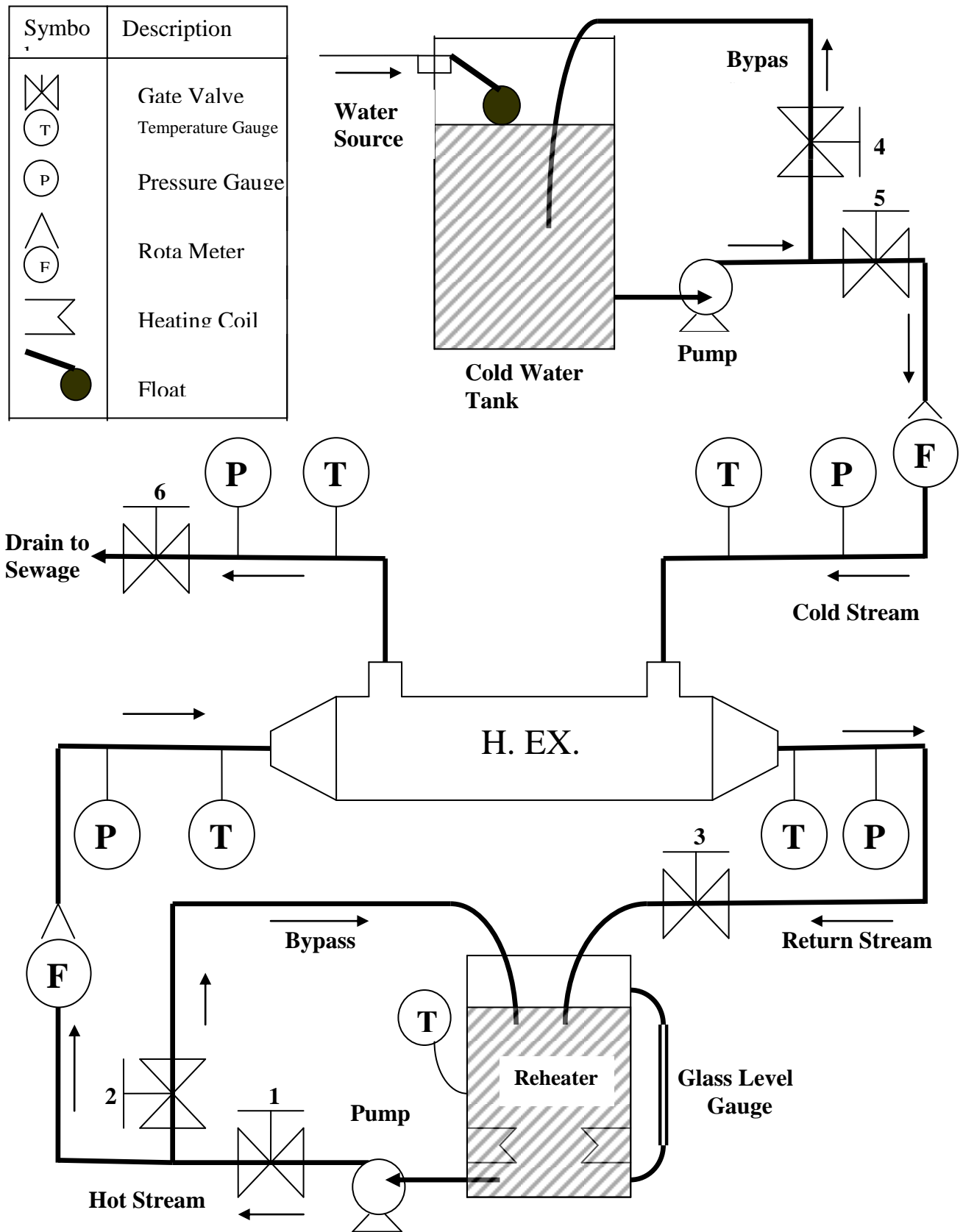


Figure (1) Schematic diagram of the test rig with instrumentation

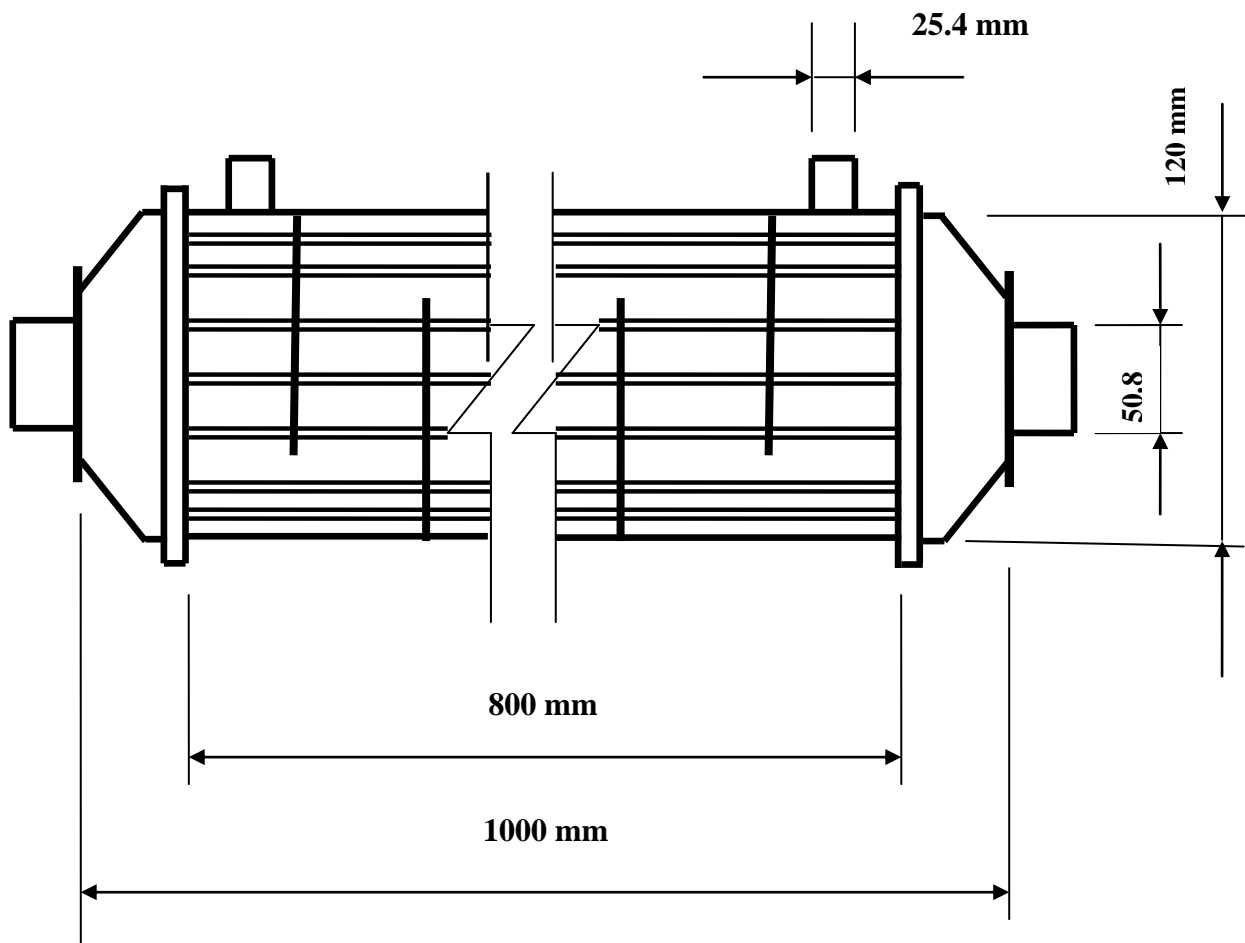


Figure (2) The assembly of test section, shell and tube heat exchanger

### 3. Theoretical Model:

#### 3-1 General

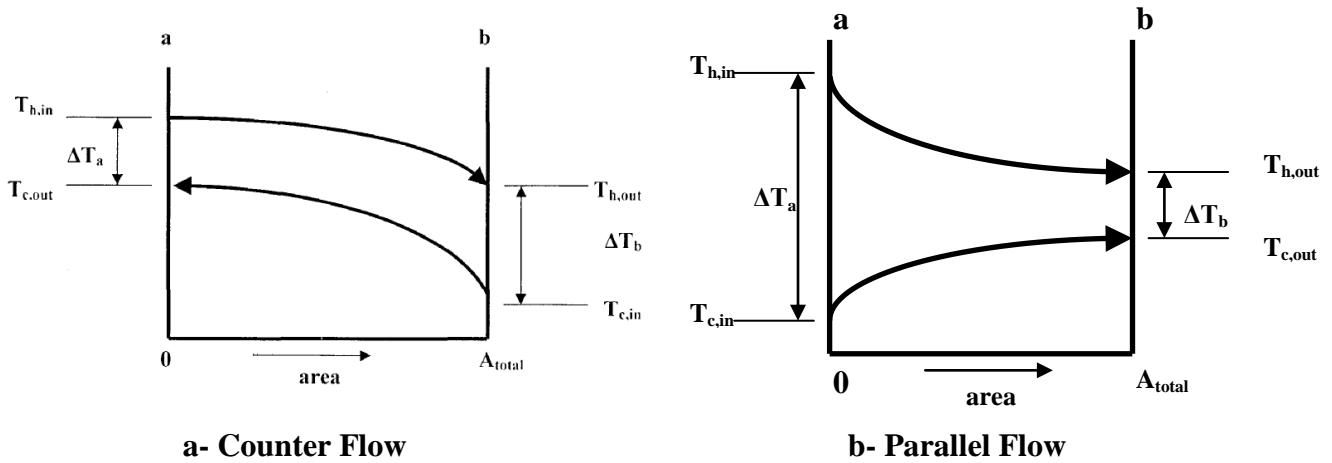
The general equation for heat transfer across a surface is:

$$Q = UA\Delta T_m \dots\dots\dots (1)$$

Before using this equation to determine the heat transfer area required for a given duty, an estimate of the true mean temperature difference ( $\Delta T_m$ ) must be made. **Figure (3)** shows the possible flow direction of both streams in a shell and tube heat exchanger. The following relations may be used for the estimation of the logarithmic mean temperature difference according to flow directions:

For counter flow 
$$LMTD = \frac{(T_{hi} - T_{co}) - (T_{ho} - T_{ci})}{\ln\left[\frac{(T_{hi} - T_{co})}{(T_{ho} - T_{ci})}\right]} \dots\dots\dots (2-a)$$

and for co-current flow 
$$LMTD = \frac{(T_{hi} - T_{ci}) - (T_{ho} - T_{co})}{\ln\left[\frac{(T_{hi} - T_{ci})}{(T_{ho} - T_{co})}\right]} \dots\dots\dots (2-b)$$



**Figure (3) The temperature distribution along a single pass heat exchanger**

The actual temperature difference of a shell and tube heat exchanger is obtained by applying a correction factor (F) to the (LMTD) value to allow for the departure from true counter flow as:

$$\Delta T_m = FLMTD \dots\dots\dots (3-a)$$

where:

$$F = \frac{\sqrt{(R^2 + 1)} \ln(1 - S)/(1 - RS)}{(R - 1) \ln\left[\frac{2 - S[R + 1 - \sqrt{(R^2 + 1)}]}{2 - S[R + 1 + \sqrt{(R^2 + 1)}]}\right]} \dots\dots\dots (3-b)$$

$$R = \frac{(T_{hi} - T_{ho})}{(T_{co} - T_{ci})} \dots\dots\dots (3-c)$$

$$S = \frac{T_{co} - T_{ci}}{(T_{hi} - T_{ci})} \dots\dots\dots (3-d)$$

Equation (3-b) is intended to be for one shell with two tube passes heat exchanger and can be used for any exchanger with an even number of tube passes. The thermal design of heat exchanger is directed to calculate an adequate surface area to handle the thermal duty for the given specifications. Whereas, the hydraulic analysis determines the pressure drop of the fluids flowing in the system, and consequently the pumping power or fan work input necessary to maintain the flow.

### 3-2 Thermal Analysis of Tube Side

The number of tubes depends on the tube side flow rate condition and inner tube size in accordance to the following formula:

$$m_t = \frac{\rho u_t A_c N_t}{N_p} \dots\dots\dots (4-a)$$

where:

$$A_c = \frac{\pi d_i^2}{4} \dots\dots\dots (4-b)$$

The tube side heat transfer coefficient is a function of Reynolds number and Prandtl number. For turbulent flow the following equation developed by Petukhov-Kirillov [11], can be used in the form:

$$Nu_t = \frac{(f/2) Re_t Pr_t}{1.07 + 12.7(f/2)^{1/2} (Pr_t^{2/3} - 1)} \dots\dots\dots (5-a)$$

Where the friction factor is obtained from

$$f = (1.58 \ln Re_t - 3.28)^{-2} \dots\dots\dots (5-b)$$

Dittus and Boelter [12], developed an equation for turbulent flow as:

$$Nu_t = 0.024 Re_t^{0.8} Pr_t^n \dots\dots\dots (6-a)$$

where n=0.4 for heating and n=0.3 for cooling of the flowing fluid. The value of Reynolds number gives an indication for the condition of the flow region. For Reynolds number greater than (2100), the flow will be considered as turbulent region for flow inside tubes, in which:

$$Re_t = \frac{\rho_l u_t d_i}{\mu_l} \dots\dots\dots (6-b)$$

and,

$$Pr_t = \frac{\mu_l c_{p_l}}{k_l} \dots\dots\dots (6-c)$$

For laminar flow, the Sieder and Tate<sup>[13]</sup>, correlation can be used:

$$Nu_t = 1.86(Re_t Pr_t)^{1/3} (d_i / L)^{1/3} (\mu_b / \mu_w)^{0.14} \dots\dots\dots (7)$$

The heat transfer coefficient for the tube side is expressed as follows:

$$h_i = Nu_t \frac{k_l}{d_i}$$

### 3-3 Thermal Analysis of Shell Side

The shell diameter is estimated by Schlünder<sup>[14]</sup>, in the form:

$$D_S = 0.637 \sqrt{\frac{CL}{CTP} \left[ \frac{A_o (PR)^2 d_o}{L} \right]^{1/2}} \dots\dots\dots (8-a)$$

where (A<sub>o</sub>) is the outside heat transfer surface area based on the outside diameter of the tube and can be calculated by:

$$A_o = \pi d_o L N_t \dots\dots\dots (8-b)$$

The tube count calculation constant values for different tube passes are:

- CTP= 0.93                      For one tube pass
- CTP= 0.9                      For two tube passes
- CTP= 0.8                      For more than two tube passes

and the layout constant values are:

CL= 1.0 for (90° and 45°)      and      CL= 0.87 for (30°)

The tube pitch ratio is:

$$PR = \frac{P_T}{d_o} \dots\dots\dots (8-c)$$



The equivalent diameter of the shell side can be estimated from:

For square pitch  $D_e = \frac{4(P_T^2 - \frac{\pi d_o^2}{4})}{\pi d_o} \dots\dots\dots (9-a)$

For triangular pitch  $D_e = \frac{4(\frac{P_T^2 \sqrt{3}}{4} - \frac{\pi d_o^2}{8})}{\pi d_o / 2} \dots\dots\dots (9-b)$

The shell side Reynolds number based on the equivalent diameter and the velocity on the cross flow area at the diameter of the shell, where:

$$Re_S = \left(\frac{m_S}{A_S}\right) \frac{D_e}{\mu} \dots\dots\dots (10-a)$$

where the bundle cross flow area is:

$$A_S = \frac{D_S}{P_T} CB \dots\dots\dots (10-b)$$

And the tube clearance is defined by:

$$C = P_T - d_o \dots\dots\dots (10-c)$$

From the above expressions, the shell side heat transfer coefficient may be estimated by:

$$h_o = 0.36 Re_S^{0.55} Pr_S^{1/3} \left(\frac{k_S}{D_e}\right) \dots\dots\dots (11)$$

The above equation is applied for the whole range of Reynolds on shell side stream.

### 3-4 Final Design

The overall heat transfer coefficient for fouled heat exchanger on both sides is expressed by:

$$U_f = \left[ \frac{1}{h_o} + F_o + \frac{d_o \ln(d_o / d_i)}{2k_w} + \frac{d_o}{d_i} \left( \frac{1}{h_i} + F_i \right) \right]^{-1} \dots\dots\dots (12)$$

The length of the heat exchanger based on the fouled conditions is calculated from:

$$L_f = \frac{A_f}{\pi d_o N_t} \dots\dots\dots (13-a)$$

where:

$$A_f = \frac{Q}{U_f \Delta T_m} \dots\dots\dots (13-b)$$

### 3-5 Hydraulic Analysis

The pressure drop encountered by the fluid making  $N_p$  passes through heat exchanger plus the additional pressure drop introduced by the change of direction in the passes are multiplied by the kinetic energy of the flow. The tube side pressure drop is calculated <sup>[15]</sup> by:

$$\Delta P_t = 4N_p \left( f \frac{L}{d_i} + 1 \right) \left( \frac{\rho \ell u_t^2}{2} \right) \dots\dots\dots (14)$$

The pressure drop of the stream flowing on the shell side can be estimated by:

$$\Delta P_s = \frac{f_s G_s^2 (N_b + 1) D_s}{2 \rho_s D_e \phi_s} \dots\dots\dots (15-a)$$

Where the shell side friction factor is obtained from:

$$f_s = \exp(0.576 - 0.19 \ln Re_s) \dots\dots\dots (15-b)$$

And the viscosity correction factor is:

$$\phi_s = \left( \frac{\mu_b}{\mu_w} \right)^{0.14} \dots\dots\dots (15-c)$$

And the wall temperature can be calculated from

$$T_w = \frac{1}{2} \left( \frac{T_{ci} + T_{co}}{2} + \frac{T_{hi} + T_{ho}}{2} \right) \dots\dots\dots (15-d)$$

## 4. Step by Step Method

The heat transfer coefficient along the heat exchanger is considered as a variable due to the change in the streams properties in accordance with the temperature variation from section

to another in the heat exchanger. For this simple change in heat transfer coefficient, it is decided to use a stepwise calculation procedure to calculate the local heat transfer rates and the pressure drop along the heat exchanger, Mohammed [16].

The heat exchanger is divided into a number of baffle spacing, and each baffles spacing is subdivided into a number of increments as shown in Fig.(4). The scheme suggested for this purpose was based on an iterative process to satisfy the energy and hydraulic balance throughout the heat exchanger. The output of each increment, usually the temperature and pressure drop on both sides; will be considered as the input to the next step and so on. The scheme of the mathematical approach will be terminated when the exit temperature of the process fluid was equal to or within acceptable accuracy to the required value of the design.

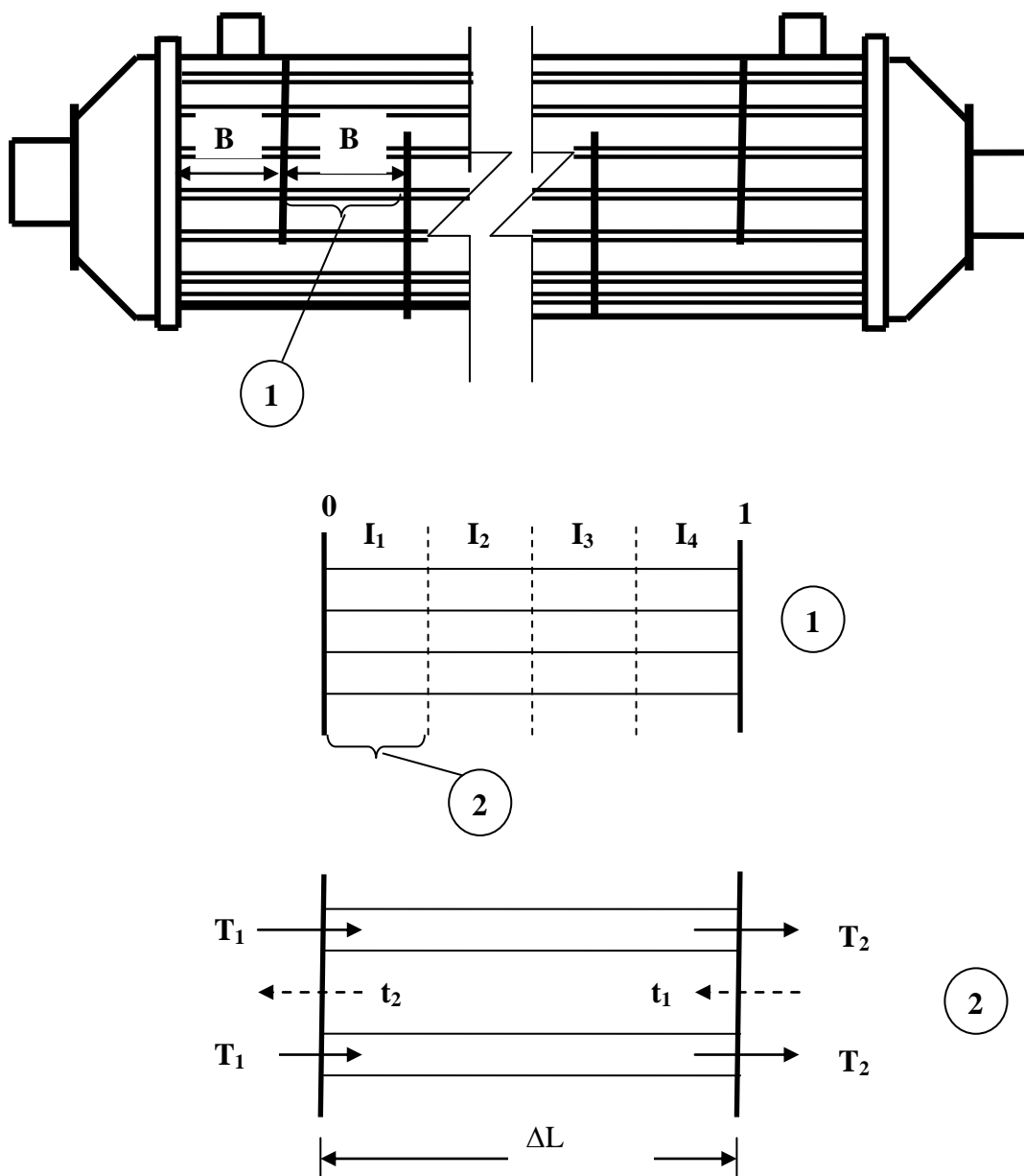


Figure (4) Schematic diagram showing the assembly of step by step method

## 5. Results and Discussion

The experimental data obtained for an entering process fluid range between (40 C°) and (60 C°) with a variety of service fluid flow rates corresponding to (700), (850) and (1000) l/hr. For comparison and verification of the theoretical model, the results obtained at (50 C°) will be used for this object. The fouling resistance for which the theoretical model was based on is corresponding to a value deduced from the literature. A value of (0.00034 m<sup>2</sup> K/W) for the fouling resistance was used on each side of the heat exchanger [15].

The comparison of the theoretical results of this model for Kern with Delaware methods showed a good agreement and a very close to each other at the corresponding values of the flow rate of the process fluid, **Fig.(5)**. Therefore, it is suggested to use Kern method for a comparison with the present model of this study for all of the considered variables of the heat exchanger performance. The overall assessment can be established as follows:

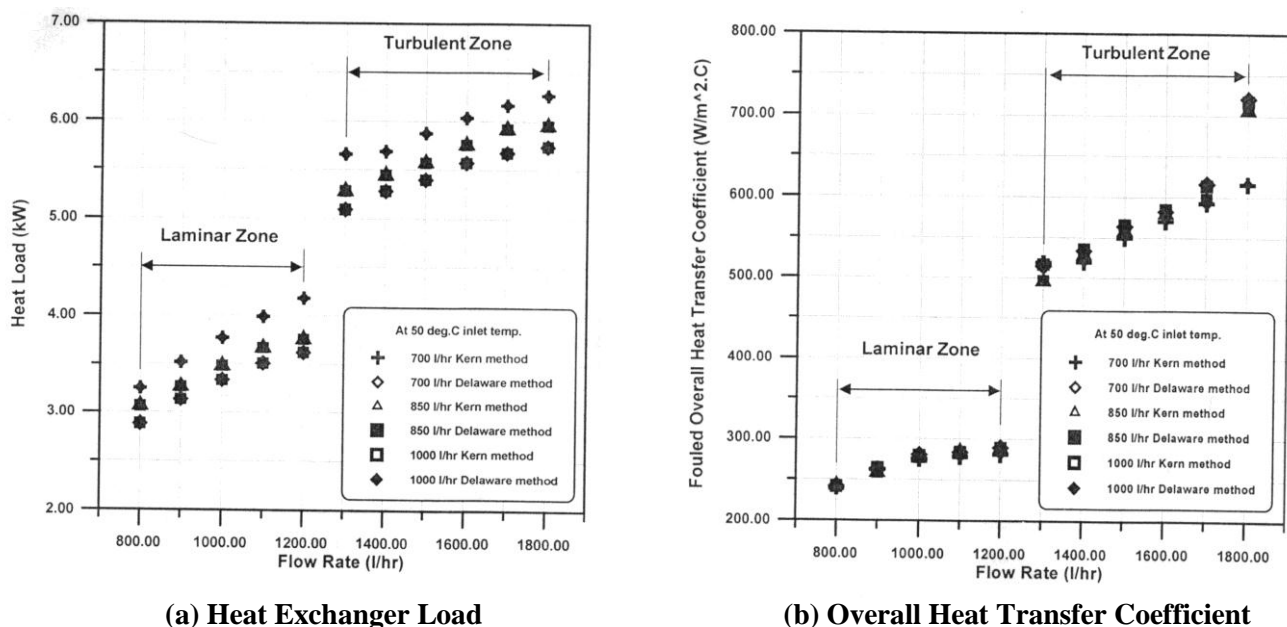
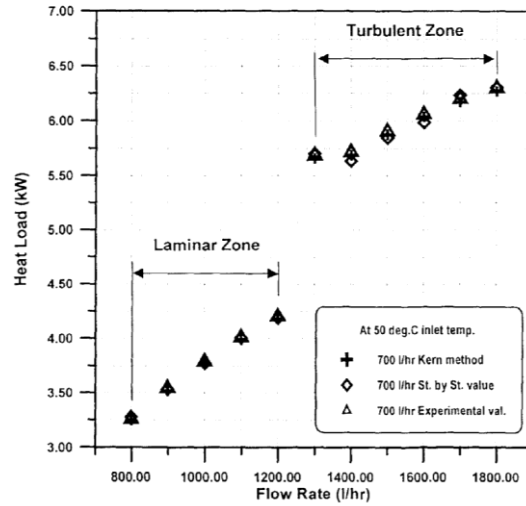


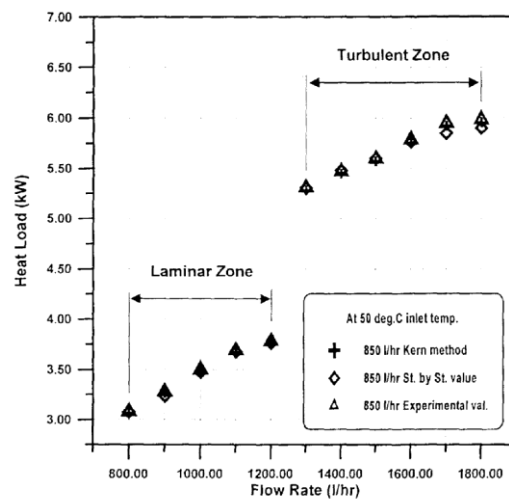
Figure (5) Comparison between Kern and Delaware methods

### 5-1 Heat Exchanger Load

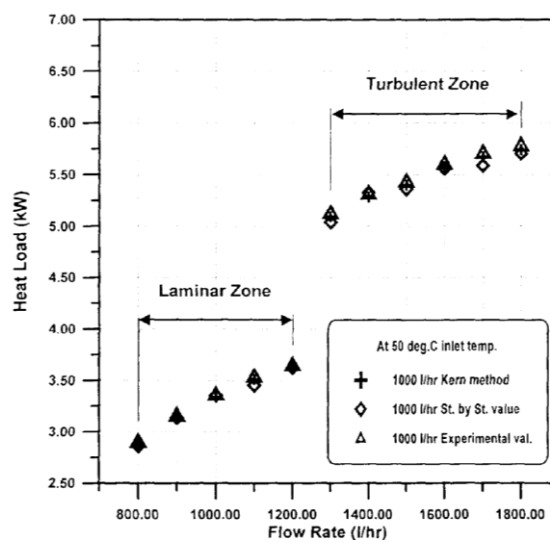
Figures (6.a), (6.b) and (6.c) show a comparison between the experimental and theoretical predictions for the heat exchanger load with the flow rate of the process fluid. These figures were adopted for three different service flow rates; they are (700), (850) and (1000) l/hr respectively. It is obvious that an excellent agreement between the predicted and the experimental data obtained in this study. For specified process fluid flow, such as (1400) l/hr, the discrepancy between theoretical predictions and the experimental data is (0.48%), (0.49%) and (0.48%) for service fluid of (700), (850) and (1000) l/hr respectively. This shows that the divergence of the experimental data from those of theoretical values is almost negligible.



(a)  $V_{Serv.} = 700$  l/hr



(b)  $V_{Serv.} = 850$  l/hr



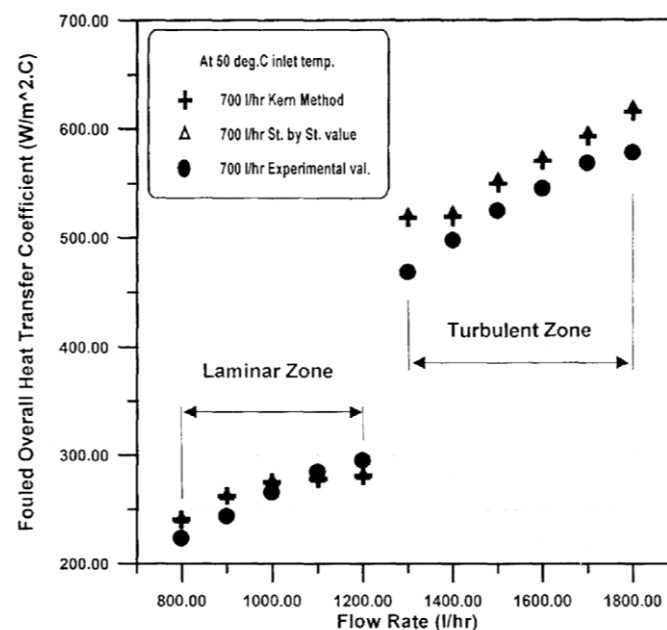
(c)  $V_{Serv.} = 1000$  l/hr

Figure (6) Comparison between the experimental and theoretical predictions of the heat exchanger load with the flow rate of the process fluid

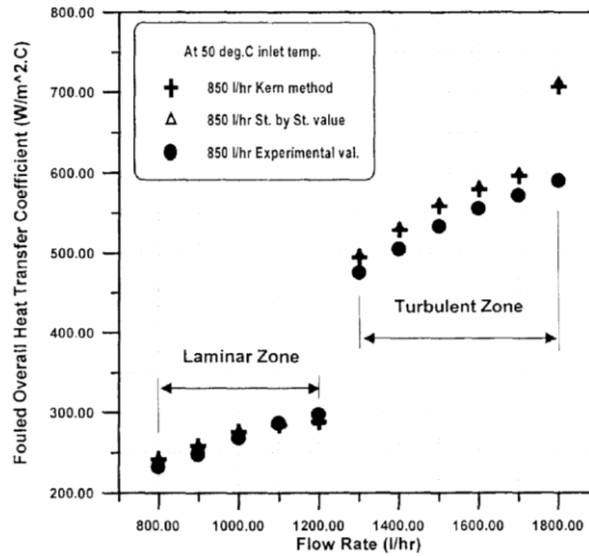
## 5-2 Overall Heat Transfer Coefficient

A comparison between the experimental and theoretical predictions of the mean fouled overall heat transfer coefficient with the flow rate of the process fluid is shown in **Figs.(7.a), (7.b)** and **(7.c)** for the service fluid flow rates at (700), (850) and (1000) l/hr respectively. The trend of these curves shows that, increasing the flow rate of the process and service fluids causes an increase in the mean overall heat transfer coefficient and this trend can be seen in the two regions, the laminar and turbulent zones.

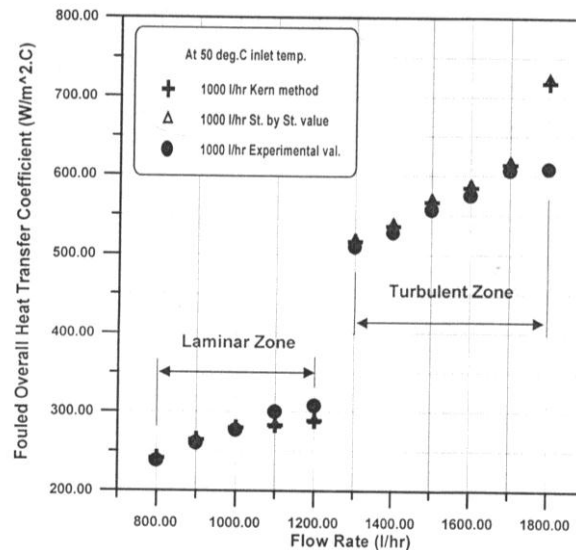
For specified process fluid flow at (1200) l/hr, the discrepancy between the theoretical prediction and the experimental data is (4.6 %), (2.8 %) and (5.9 %) for service fluid flow rate of (700), (850) and (1000) l/hr respectively. The fouled overall heat transfer coefficient of the theoretical predictions were in the range (240 to 616), (242 to 707) and (241 to 718)  $W/m^2 K$  for service fluid flow rate of (700), (850) and (1000) l/hr respectively. The gradient of the overall heat transfer coefficient with the flow rate of the process fluid of the (850) l/hr service fluid flow rate as estimated for these lines are (0.23) and (0.42) for turbulent zone experimentally and theoretically respectively. The corresponding values for laminar region are (0.16) and (0.12).



(a)  $V_{Serv.} = 700$  l/hr



(b)  $V_{Serv.} = 850$  l/hr



(c)  $V_{Serv.} = 1000$  l/hr

Figure (7) Comparison between the experimental and theoretical predictions of the mean fouled overall heat transfer coefficient with flow rate of the process fluid

**5-3 Step by Step Results:**

It is interesting to show the powerful feature of the step by step model presented in this study by setting up the results of the theory in a distribution manner along the heat exchanger. For this object, it is suggested to furnish the heat exchanger load, overall heat transfer coefficient and temperature variation along the heat exchanger with baffle spacing in the direction of process fluid flow.

### 5-3-1 Heat Exchanger Load

Figure (8) shows the heat transfer rate distribution and its variation with baffle spacing along the heat exchanger at (1800) l/hr process fluid flow rate. The trend of this variation (heat load increasing) is due to the increase in the temperature difference of the specified baffle spacing. Increasing the service fluid flow rate in the shell side causes an increase in the baffle spacing load. The percent of increases were in the ranges (3.1 to 9%) and (2.2 to 5.3%) for service fluid flow rates of (850) and (1000) l/hr respectively. Further, this increase causes an increase percent for the baffle load with respect to (700) l/hr are (19.2 %) and (21.5 %) for service fluid flow rates of (850) and (1000) l/hr respectively.

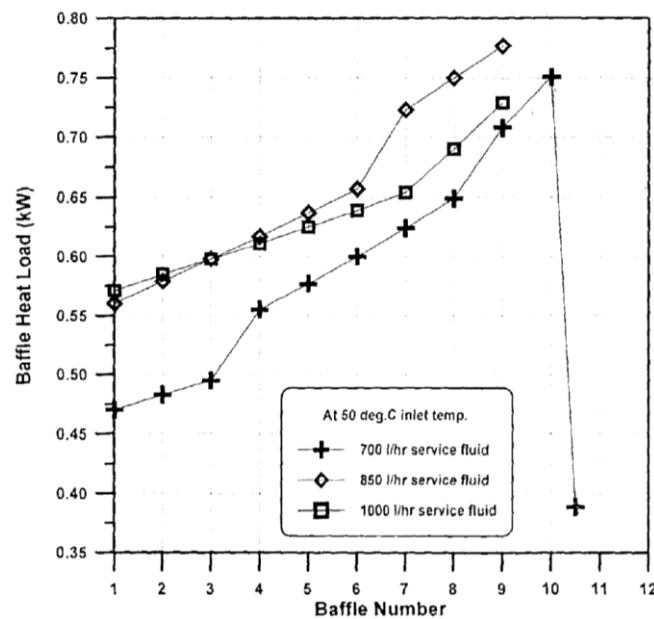
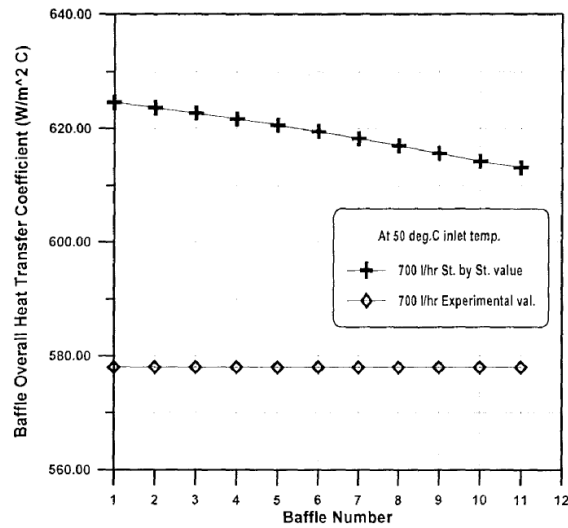


Figure (8) Heat exchanger load distribution with baffle spacing number along the heat exchanger at 1800 l/hr process fluid flow rate

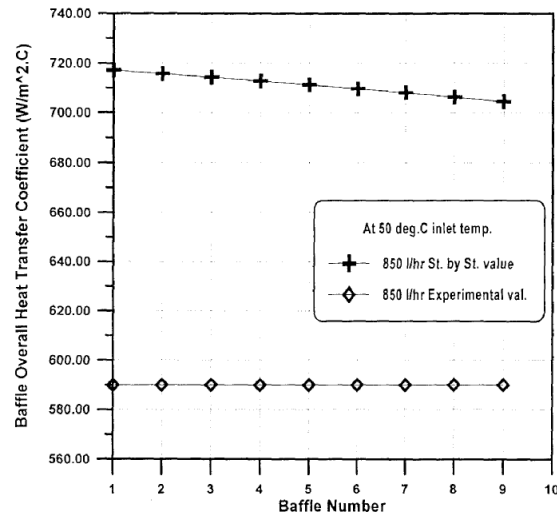
### 5-3-2 Overall Heat Transfer Coefficient

Figures (9.a), (9.b) and (9.c) represent a comparison for the fouled heat transfer coefficient between the theoretical prediction and experimental data with the baffle number for service fluid flow rates (700), (850) and (1000) l/hr respectively. Here, the experimental results represented by a horizontal line since it refers to a mean value along the heat exchanger. From these figures, it is obvious that the overall heat transfer coefficient values decrease with the direction of the process fluid flow. That is from the tube side inlet (shell side exit) towards tube side outlet (shell side inlet). This is because of the physical property variation which in turn reflects the effect on the flow criteria (thermal and hydraulic). The predicted values by the step by step model were higher than those of the experimental data. The discrepancy percentage is ranged between (15 %) and (17 %) for service fluid flow rates of (850) and (1000) l/hr respectively.

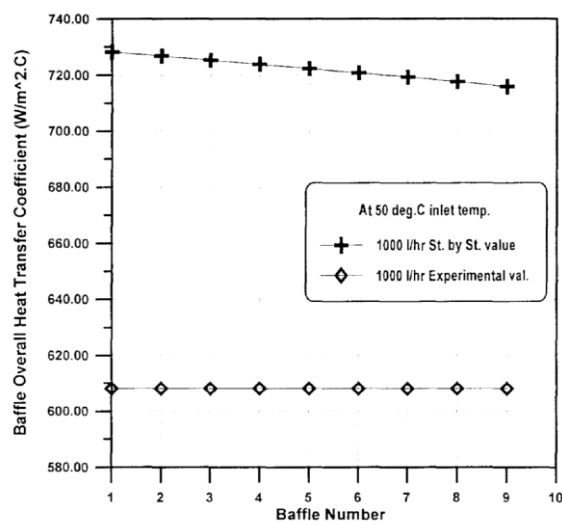




(a)  $V_{Serv.} = 700$  l/hr



(b)  $V_{Serv.} = 850$  l/hr

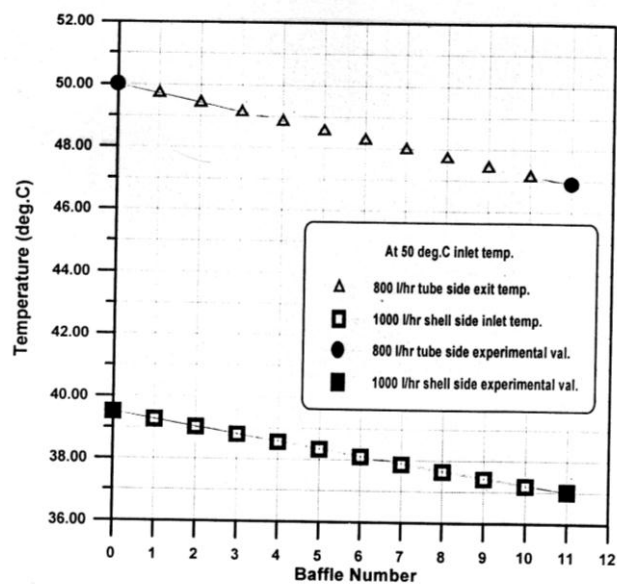


(c)  $V_{Serv.} = 1000$  l/hr

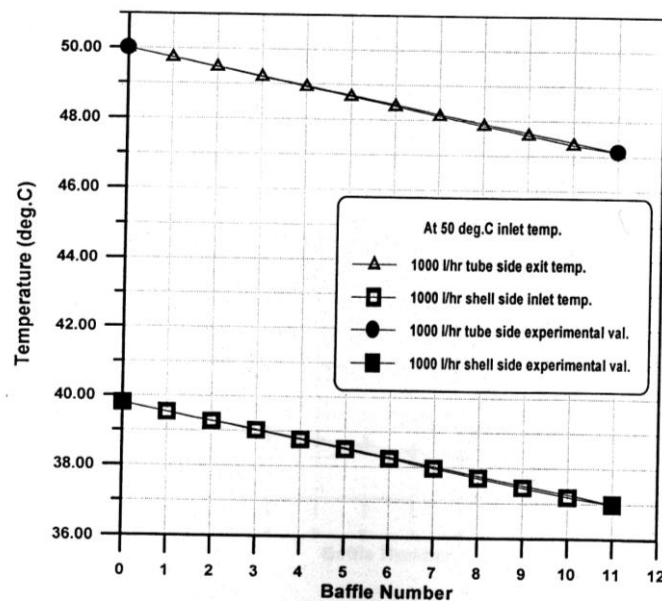
Figure (9): A Comparison between the experimental and theoretical data of the overall heat transfer coefficient along the heat exchanger at 1800 l/hr process fluid flow rate

### 5-3-3 Temperature Distribution

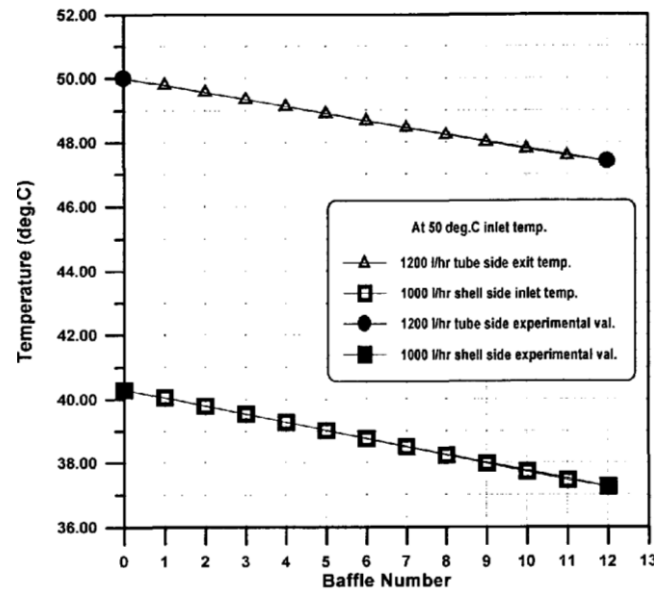
The results of the temperature distribution of the present model are compared with the tests data in Figs.(10.a), (10.b) and (10.c) for (800), (1000) and (1200) l/hr of the process fluid flow rates respectively. The trend of these curves shows that the temperature of the shell side increases with the direction of the process fluid flow for counter flow distribution until the tube process fluid proceeds to the exit temperature from the heat exchanger. The results of the present model, the step by step technique, exhibited a smooth temperature variation along the heat exchanger. The terminal temperatures on both stream sides are well agreed with those of experimental data.



(a)  $V_{Proc.} = 800$  l/hr



(b)  $V_{Proc.} = 1000$  l/hr



(c)  $V_{Proc.} = 1200$  l/hr

**Figure (10) Comparison between the experimental and theoretical predictions of the temperature variation with the baffle number**

#### 5-3-4 Shell Side Pressure Drop

**Figure (11)** shows the variation of the shell side pressure drop per baffle with the baffle number for baffle spacing of (0.08 m). The curves show that the pressure drop per baffle increases with the direction of the process fluid flow. That is from tube side inlet (shell side exit) towards tube side outlet (shell side inlet). The shell side pressure drop values per baffle were in the range of (245 to 250), (345 to 350) and (466 to 472) Pa, for the service fluid flow rate of (700), (850) and (1000) l/hr respectively. This slight increase of the pressure drop per baffle was due to the physical property variation with temperature predicted in this research. Further, the pressure drop is proportional directly to the flow rate or (velocity) of the stream when all other parameters are considered to be constant as shown in figure (11).

#### 5-3-5 Tube Side Pressure Drop

The variation of the tube side pressure drop per baffle with the baffle number is shown in **Fig.(12)**. The trend of the results shows that the tube side pressure drop per baffle exhibited a slight increase with the direction of the process fluid flow. This was due to the temperature variation which affects the thermal properties of the fluid stream throughout the tube side of the heat exchanger. It is obvious that there is convergence between the corresponding values for different service flow rates. The discrepancy percentage of the tube side pressure drop per baffle for baffle number (4) are (0.1 %) and (0.12 %) for (850) and (1000) l/hr service fluid flow rate with respectively. The tube side pressure drop values per baffle were in the range of (3.7 to 3.8), (3.3 to 3.8) and (3.7 to 3.8) Pa, for the service fluid flow rate of (700), (850) and (1000) l/hr respectively.

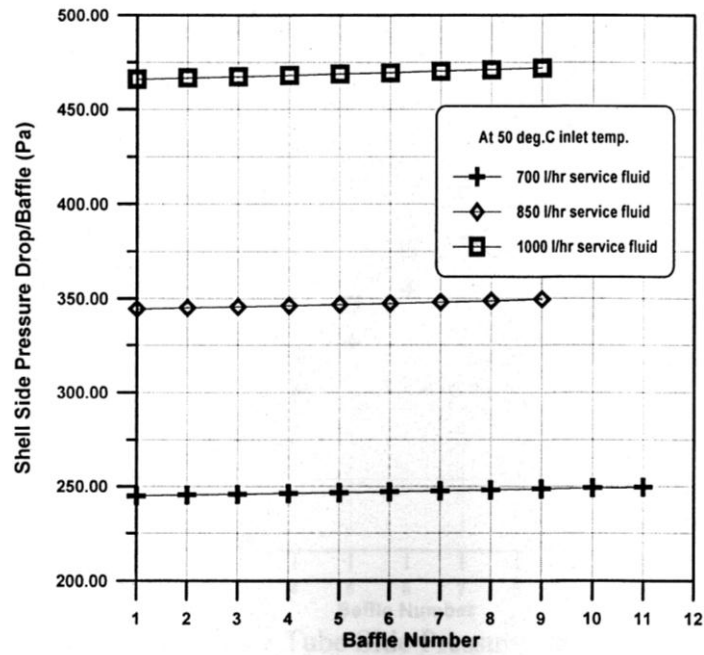


Figure (11) Variation of shell side pressure drop per baffle number at 1800 1/hr process fluid flow rate

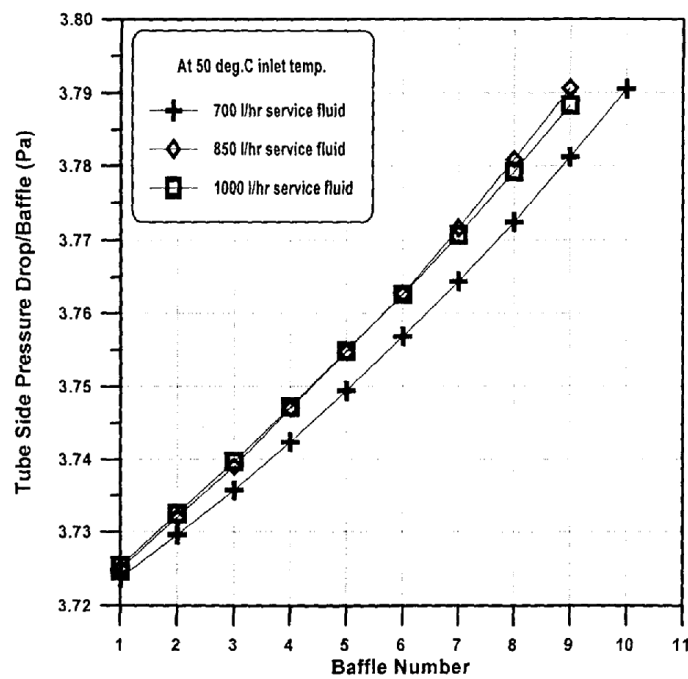


Figure (12) Variation of the tube side pressure drop per baffle with baffle number at 1800 l/hr process fluid flow rate

## 6. Conclusions

A software computer program model has been built in this study can be used for the thermal and hydraulic design of a shell and tube heat exchanger having single pass on the shell and tube sides. For the counter flow shell and tube heat exchanger investigated in this research, the followings have been concluded:

1. Increasing the tube side flow rate (process fluid) causes an increase in the heat exchanger load, the overall heat transfer coefficient, its pressure drop and its heat transfer coefficient with a decrease in temperature difference along the heat exchanger.
2. Increasing the tube side inlet temperature and the shell side fluid flow rate causes an increase in the temperature difference, the heat exchanger load and the overall heat transfer coefficient.
3. For the optimization purpose of the thermal and hydraulic design of the shell and tube heat exchanger a combination of the following different variables should be considered:
  - a) The inlet and outlet streams temperature.
  - b) Flow rates on both sides of the heat exchanger.
  - c) The space and layout limitations of the equipments, and
  - d) Power consumption
4. The computer program implemented in the design object has helped in saving time and efforts needed for the design process. Also, it gives the designer a variety of options which may be considered for optimization of the thermal-hydraulic design and cost consideration.

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## ***Nomenclature***

A:	Area ( $m^2$ )
$A_S$ :	Cross Flow Area at Bundle Center Line ( $m^2$ )
B:	Baffle Spacing (m)
$c_p$ :	Specific Heat at Constant Pressure (kJ/kg K)
$D_e$ :	Shell Equivalent Diameter (m)
$D_S$ :	Shell Inside Diameter (m)
d:	Tube Diameter (m)
F:	Temperature Correction Factor, eq.(3.b)
$F_i$ :	Tube Side Fouling Resistance ( $m^2$ K/W)
$F_o$ :	Shell Side Fouling Resistance ( $m^2$ K/W)
f:	Flow Friction Factor
k:	Thermal Conductivity (W/m K)
L:	Tube Length (m)
LMTD:	Logarithmic Mean Temperature Difference ( $C^\circ$ )
m:	Mass Flow Rate (kg/s)
$N_P$ :	Number of Tube Passes
$N_t$ :	Total Number of Tubes
Nu:	Nusselt Number
PR:	Tube Pitch Ratio
$P_T$ :	Tube Pitch (m)
Q:	Heat Transfer Rate (W)
R:	Temperature Ratio, eq.(3.c)
Re:	Reynolds Number
S:	Temperature Ratio, eq.(3.d)
T:	Temperature ( $C^\circ$ )
U:	Overall Heat Transfer Coefficient ( $W/m^2$ K)
u:	Fluid Velocity (m/s)
V:	Volumetric Flow Rate (l/hr)

## ***Greek Symbols***

$\Delta P$ :	Pressure Drop (Pa)
$\Delta T$ :	Temperature Difference (K)
$\rho$ :	Density ( $kg/m^3$ )
$\mu$ :	Dynamic Viscosity (kg/m s)
$\phi$ :	Viscosity Correction Factor, eq.(15.c)

***Subscript:***

b:	At Bulk Condition Value
c:	Cold Stream, Cross Section
f:	Fouled Condition
h:	Hot Stream, Hydraulic
i:	Inside
ℓ :	Liquid
m:	Mean
o:	outside
Proc.:	Process Fluid
S:	Shell Side
Serv.:	Service Fluid
t:	Tube Side
w:	At Wall Condition Value