ANALYSIS COMPARING PERFORMANCE OF A CONVENTIONAL SHELL AND TUBE HEAT EXCHANGER USING KERN, BELL AND BELL DELAWARE METHOD

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Abstract

The transfer of heat to and from process fluids is an essential part of most of the chemical processes. Therefore, heat exchangers (HEs) are used extensively and regularly in process and allied industries and are very important during design and operation. The most commonly used type of HE is the shell and tube heat exchanger. In the present study, a comparative analysis of a water to water STHE wherein, hot water flows inside the tubes and cold water inside the shell is made, to study and analyze the heat transfer coefficient and pressure drops for different mass flow rates and inlet and outlet temperatures, using Kern, Bell and Bell Delaware methods. This paper purely aims at studying and comparing different methods of STHE and bringing out which method is better for adopting in shell side calculations.

Keywords: *STHE, Heat transfer coefficient, shell &Tube heat exchanger, Pressure Drop, TEMA, Bell Delaware method, Kern Method & Bell method.*

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1. INTRODUCTION

Shell-and-tube heat exchangers _STHXs_ are widely used in many industrial areas, and more than 35–40% of heat exchangers are of this type due to their robust geometry construction, easy maintenance, and possible upgrades. Besides supporting the tube bundles, the baffles in shell-and-tube heat exchangers form flow passage for the shell-side fluid in conjunction with the shell. The most-commonly used baffle is the segmental baffle, which forces the shell-side fluid going through in a zigzag manner, hence, improves the heat transfer with a large pressure drop penalty. This type of heat exchanger has been well-developed and probably is still the most-commonly used type of the shell and tube heat exchangers [1].

Heat exchangers are one of the most important devices of mechanical systems in modern society. Most industrial processes involve the transfer of heat and more often, it is required that the heat transfer process be controlled. According to Oko (2008), a heat exchanger is a device of finite volume in which heat is exchanged between two media, one being cold and the other being hot. There are different types of heat exchangers; but the type widely used in industrial application is the shell and tube [2]. Mass velocity strongly influences the heat-transfer coefficient. Thus, with increasing mass velocity, pressure drop increases more rapidly than does the heat-transfer coefficient. Consequently, there will be an optimum mass velocity above which it will be wasteful to increase mass velocity further. The construction geometry and thermal parameters such as mass flow rate, heat transfer coefficient etc are strongly influenced by each other.[3]. There are design charts such as E-NTU (Effectiveness- Number of Transfer Unit) curves and LMTD (Logarithm Mean Temperature Difference) correction factor curves for the analysis of simple types of exchangers. Similar design charts do not exist for the analysis of complex heat exchangers with multiple entries on the shell side and complex flow arrangements (Ravikumaur et al, 1988).

In this way, the design of shell and tube heat exchangers is a very important subject in industrial processes. Nevertheless, some difficulties are found, especially in the shell-side design, because of the complex characteristics of heat transfer and pressure drop [4].

The flow in the shell side of a shell-and-tube heat exchanger with segmental baffles is very complex. The baffles lead to a stream inside the shell, which is partly perpendicular and partly parallel to the tube bank. The gaps between the tubes and the holes in the baffles and the gap between a baffle and the shell cause leakage streams, which may modify the main stream significantly. Since the tubes of the heat exchanger cannot be placed very near to the shell, bypass streams S, may be formed, which influence also the main stream. The flow direction of the main stream relative to the tubes is different in the window sections created by the baffle cut from that in the cross flow sections existing between the segmental baffles. This necessitates the use of different equations to calculate the pressure drop in the window sections to those used in the cross flow sections. The spacing between the tube plates and the first and the last baffle, which is mostly dictated by the diameter of the inlet and oulet nozzles, differs in many cases from the spacing between two adjacent baffles and some of the aforementioned streams are not present in the first and in the last heat exchanger sections. This adds to the complexity of the problem [5].

In designing shell and tube heat exchangers, to calculate the heat exchange area, different methods were proposed such as Kern Method, Bell, Bell Delaware etc [6].

The present paper employs all the three methods mentioned above to study and analyze various parameters of a shell and tube heat exchanger and compare the results to identify which method gives the best result.

2. KERN METHOD

The kern method was based on experimental work on commercial exchangers with standard tolerances and will give a reasonably satisfactory prediction of the heat-transfer coefficient for standard designs. The prediction of pressure drop is less satisfactory, as pressure drop is more affected by leakage and bypassing than heat transfer. The shell-side heat transfer and friction factors are correlated in a similar manner to those for tube-side flow by using a hypothetical shell velocity and shell diameter. As the cross-sectional area for flow will vary across the shell diameter, the linear and mass velocities are based on the maximum area for cross-flow: that at the shell equator. The shell equivalent diameter is calculated using the flow area between the tubes taken in the axial direction (parallel to the tubes) and the wetted perimeter of the tubes. The method used by D.Q. Kern is simple and more explanative. All the parameter related to heat exchanger are obtained in well manner and brief without any complication as compared to other method, the calculation process is quite and simple detailed.

Among all the methods, the Kern method provided a simple method for calculating shell side pressure drop and heat transfer coefficient. However, this method cannot adequately account the baffle to shell and tube to baffle leakage.

3. BELL METHOD

In Bell's method the heat-transfer coefficient and pressure drop are estimated from correlations for flow over ideal tube-banks, and the effects of leakage, bypassing and flow in the window zone are allowed for by applying correction factors. This approach will give more satisfactory predictions of the heat-transfer coefficient and pressure drop than Kern's method; and, as it takes into account the effects of leakage and bypassing, can be used to investigate the effects of constructional tolerances and the use of sealing strips. The procedure in a simplified and modified form to that given byBell (1963), is outlined below. The method is not recommended when the by-pass flow area is greater than 30% of the cross-flow area, unless sealing strips are used[5].

Bell (1978) has proposed a graphical method based on the operating lines in stagewise process design, to estimate the value of N. This procedure utilizes the inlet and outlet temperatures of both hot and cold streams in which N is about 3. In this work, it has been found that for N > 3, Bell's method frequently cannot be used to predict feasible designs of multipass exchangers. Specifically, by following the procedure used in the development of the Kremser equation in stagewise process design (McCabe and Smith, 1976)[5]

4. BELL DELAWARE METHOD

Shell side flow is complex, combines crossflow and baffle Window flow, as well as baffle-shell and bundle-shell bypass streams and other complex flow patterns

In a baffled shell and tube heat exchanger, only a fraction of the fluid flow through the shell side of a heat exchanger actually flows across the tube bundle in the idealized path normal to the axis of the tubes. The remaining fraction of the fluid flows through bypass areas. The fluid seeks the flow path of less resistance from the inlet to the outlet of the exchanger.

In the Bell Delaware method, the fluid flow in the shell is divided into a number of individual streams A through F as shown in FIG 1.

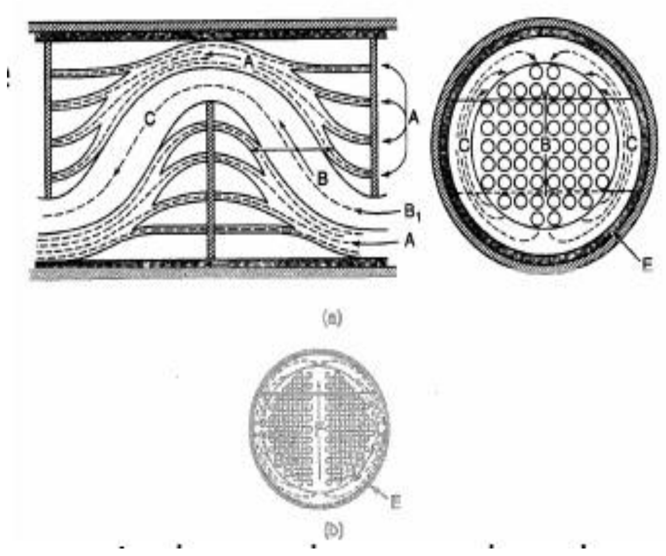

Fig- 1 cross section of shell and tube heat exchanger

Each of the streams from A to F introduces a correction factor to the heat transfer correlation for ideal cross-flow across a bank of tubes such as J_c, J_L, J_b, J_s, J_r .

In the present study, efforts have been made to study Kern, Bell & Bell Delaware method and apply these methods in calculating heat transfer coeffient, Reynold's number, pressure drops, overall heat transfer coefficient etc for a heat exchanger which has been designed and fabricated for our experimental investigations and compare the results and identify which method is more efficient in calculating the shell side parameters. The heat transfer fluid used is water. Hot water flows inside the tubes and cold water flows inside shell.

4.1 Fluid Properties Considered

Shell side fluid properties:

ρ_s = 1000 kg/m^3

μ_s = 0.00088 N-s/ m^2

C_{ps} = 4.187kJ/kg'K

K_s = 0.00098 kJ/s-m'K.

Tube side fluid properties:

ρ_t = 1000 kg/m^3

μ_t= 0.00086 N-s/ m^2

C_{pt} = 4.187kJ/kg'K

K_t = 0.00098 kJ/s-m'K.

5. HEAT EXCHANGER SPECIFICATIONS:

In the present study, a stainless steel shell and tube heat exchanger is used to study the various parameters of the heat exchanger such as heat transfer coefficient, Reynolds's number, pressure drop, Overall heat transfer coefficient etc using water as a heat transfer medium.

Specifications of the heat exchanger are as follows:

Shell diameter (Ds)	0.2m
Tube inside diameter (Di)	0.016m
Tube outside diameter (Do)	0.01924m
Pitch (Pt)	0.03m
Length of shell (Ls)	0.8m
Length of tube (Lt)	0.825m
Length of baffle (Lb)	0.2m
Number of baffles (Nb)	4
Number of tubes (Nt)	18
Number of shell passes (ns)	1
Number of tube passes (nt)	2
Clearance (C)	0.01076m
Bundle to shell diametrical clearance (Δb)	0.028m
Shell to baffle diametrical clearance (Δsb)	0.0254m
Tube to baffle diametrical clearance (Δtb)	0.0005m

6. NOMENCLATURES

S_m = Area of the shell side cross flow section (m^2).
P_t = Tube pitch (m).
D_o = Tube outside diameter (m).
D_i = Tube inside diameter (m).
D_s = Shell inside diameter (m).
L_b = Baffle spacing (m)
L_s= Length of shell (m).
L_t = Length of tube (m).
t_b = Tube thickness (m).
G_s= Shell side mass velocity (kg/ m^2-s).
G_t = Tube side mass velocity (kg/ m^2-s).
U_s= Shell side linear velocity (m/s).
U_t = Tube side linear velocity (m/s).
m_s= Mass flow rate of the fluid on shell side (kg/s).
m_t= Mass flow rate of the fluid on tube side (kg/s).
ρ_s = Shell side fluid density (kg/m^3).
ρ_t = Tube side fluid density (kg/m^3).
R_{es}= Shell side Reynolds number.
R_{et} = Tube side Reynolds number.
P_{rs}= Shell side Prandtl number.
P_{rt} = Tube side Prandtl number
μ_s = Shell side fluid Viscosity (N-s/ m^2).
μ_t = Tube side fluid viscosity (N-s/ m^2).
μ_w = Viscosity a wall temperature (N-s/ m^2).
C_{ps} = Shell side fluid heat capacity (kJ/kg'K).
C_{pt} = Tube side fluid heat capacity (kJ/kg'K).
K_s = Shell side fluid thermal conductivity (kJ/s-m'K).
K_t = Tube side fluid thermal conductivity (kJ/s-m'K).
h_o= Shell side heat transfer coefficient (W/ m^2'K).
h_i = Shell side ideal heat transfer coefficient (W/ m^2'K).
N_b= Number of baffles.
N_t = Number of tubes.
f = Friction factor.
ΔP_s= Shell side pressure drop (Pa).
n_p = Number of tube passes.
C = Clearance between tubes.
Δ_b = Bundle to shell diametrical clearance.
Δ_{sb}=Shell to baffle diametrical clearance.
Δ_{tb}=Tube to bundle diametrical clearance.
N_{ss}/N_c=Sealing strips per cross flow row.
$D_{otl}=D_s - \Delta_b$
$\Theta = \{D_s-(2*L_c)\}/ D_{otl}$
F_c=Fraction of total number of tubes in a crossflow section.
J_c=Correction factor for baffle cut and spacing.
S_{sb}=Shell to baffle leakage area (m^2).

S_{tb}=Tube to baffle leakage area (m^2).
F_{bp} = Fraction of the crossflow area available for bypass flow.
S_w= Window flow area (m^2).
N_c = Number of tube rows crossed in one crossflow section.
N_{cw}=Effective number crossflow rows in window zone.
ΔP_c=Ideal cross flow pressure drop through one baffle space (Pa).
ΔP_w= Window zone pressure drop (Pa).
R_L= correction factor for baffle leakage effect on pressure drop
R_b= correction factor on pressure drop for bypass flow.

Experimental study is done on the shell and tube water / water heat exchanger and various parameters are calculated for different mass flow rates and at varying inlet and outlet temperatures. Calculations shown below are made using Kern, Bell and Bell Delaware methods for a mass flow rate of .0354Kg/s and further readings are shown for different flow rates and comparison graphs are drawn.

6.1 Calculation of Shell Side Heat Transfer Coefficient Using Kern Method:

$A_s = \{(P_t - D_o)*D*L_b\} / P_t$
$= \{(0.03-0.01924)*0.2*.2\}/(0.03)$
$= 0.01435\ m^2$

$G_s = ms / A_s$
$= 0.0354/0.01435$
$= 2.47\ Kg/m^2\ sec$

$D_e = [4*\{(P_t^2*\surd 3)/4\} - \{(\pi * D_o^2)/8\}] / [(\pi * D_o)/2]$
$=[4\{(0.03^2*\surd 3/4\}\{(\pi*0.01924^2)/8]/[(\pi*0.0924)/2]$
$= 0.0325m$

$R_{es} = (G_s * D_e) / \mu_s$
$= (2.47/0.0325)/0.00088$
$= 91.2$

$P_{rs} = (C_{ps}*\mu_s) / K_s$
$= (4.187*0.00088)/0.00098$
$= 3.76$

$h_s = 0.36*(K_s / D_e)* (R_e\text{^}0.55)*(P_r\text{^}0.33)*\{ (\mu_s / \mu_w)\text{^}0.14\}$
$=0.35*(0.00098/0.0325)*(91.2^{0.55})*(3.76^{0.33})*1$

$\boxed{\mathbf{h_s = 0.201\ W/m^{2o}K}}$

6.2 Calculation of Shell Side Heat Transfer Coefficient Using Bell Method:

Area of the Shell As = P_T – OD/P_T x Ds x P_B
=[(0.03-0.01924)/(0.03)]*0.2*0.2
= $0.0143m^2$

Calculate the shell –side mass velocity
Gs = Ms/ As
= 0.0354/0.01432
=2.46 Kg/m^2 Sec

Calculate the Reynolds number on shell side
Res = Gs do/ μ
= (2.46*0.01924)/(0.008)
=53.7645

Calculate the Prandtl number
Pr = Cp μ / K
=(4.187*0.00088)/(0.00098)
= 3.76

Ideal heat transfer co-efficient is given by

h_{oc}* do/ K = Jh Re $Pr^{1/3}$ (μ/ μw) $^{0.14}$

(h_{oc}*0.01924)/(0.00098)=1*53.97*($3.76^{1/3}$)*1

h_{oc}= 3.445$W/m^{2\ o}K$

Now Jh is calculated, from the fig (2) at Reynolds no
Jh=1

Fn tube row correction factor
Tube vertical pitch, P't = 0.87 x P_T (for triangular pitch)
= 0.87*0.03
=0.026

Baffle cut height, Hc = Ds x Bc
=0.2*0.25
=0.05

Height between Baffle tips = Ds – 2 x (Hc)
= 0.2-(2*0.05)
= 0.1

Ncv = HbT/ P't
=0.1/0.026
=3.8

Now from the figure (3) at Ncv, we get Fn, i.e, (tube row correction factor)
Fn=1

Window Correction factor Fw:-
Height of the baffle chord to the top of the tube bundle, Hb is given by
Hb = Db/ 2 – Ds (0.5 – Bc)
=(0.172/2)-0.2(0.5-0.25)
= 0.036

Db = Ds – ΔPbs
=(0.2-0.028)

=0.172

Bb = Hb / Db
=0.036/0.0172
=0.209

Now from fig (4) at the cut of i.e, (Bb) we get Ra'=0.14

Now, the number of tubes in a window zone is given by
Nw = Nt x Ra'
=18*0.14
=2.54

Nc = Nt - 2Nw
=18-(2*2.52)
=12.96

Rw = 2Nw / Nt
=2*2.52/18
=0.28

From the figure (5) at Rw we get the value of Fw=1.08

Bypass correction Fb
Fb = exp [-α Ab/ As (1 – {2Ns / Ncv }$^{1/3}$)]
=exp [-1.5*(0.0344/0.0143)(1-(2*0.2/3.8)$^{1/3}$)]
=0.1496

Ab = [Ds – Db] x Bp
=[0.2-0.028]*0.2
=0.0344

Leakage correction factor FL.
FL = 1 – β_L {Atb + 2 Asb / A_L}
=1-0.5[0.011883+2*0.000305/0.01218]
=0.4874

AL = total leakage area = [Atb + Asb]
=[0.011883+0.000305]
=0.01218

Atb = Ct π do / 2 (Nt – Nw)
= 0.02548*π*0.01924/2(18-2.52)
=0.011883

Asb = CsDs / 2 (2π – θb)
=0.0005*0.2/2(2π-0.18)
=0.000305

Shell – side heat transfer co-efficient is given by
hs = hco x Fn x Fw x Fb x FL .
=3.445*1*1.08*0.1496*0.4875

$\boxed{\textbf{hs =0.271W/m}^{2o}\textbf{K}}$

6.3 Calculation of Shell Side Heat Transfer Coefficient Using Bell Delaware Method:

STEP 1: Calculate the shell side area at or near the centre line for one cross flow section S_m,

$\mathbf{S_m = L_b*[(D_s – D_{otl}) + \{(D_{otl} – D_o)*(P_t -D_o)\}/P_t]}$
S_m=.2*[(.2 – .172) + {(.172 – .01924)*(.03 -.01924)}/.03]
$\underline{S_m = 0.017\ m^2}$

STEP 2: Calculate shell side mass velocity G_s and linear velocity U_s.

$\mathbf{G_s = m_s / S_m}$
G_s = .0354 / .017
$\underline{G_s = 2.082\ kg/\ m^2\text{-}s}$

$\mathbf{U_s = G_s / \rho_s}$
U_s = 2.082 / 1000
$\underline{U_s = .002082\ m/s}$

STEP 3: Calculate shell side Reynolds number R_{es}.
$\mathbf{R_{es} = (G_s* D_o) / \mu_s}$
R_{es} = (2.082* .01924) / .00088
$\underline{R_{es} = 45.52}$

STEP 4: Calculate shell side Prandtl number P_{rs}.
$\mathbf{P_{rs} = (C_{ps}*\mu_s) / K_s}$
P_{rs} = (4.187*.00088) / .00098
$\underline{P_{rs} = 3.7597}$

STEP 5: Calculate the colburn j factor j_i.
$\mathbf{j_i = a_1*[\{1.33 / (P_t / D_o)\} \wedge a]* (R_{es}\wedge a_2)}$
j_i = 1.36*[{1.33 / (.03 / .01924)} ^ .719]* (45.52^-.657)
$\underline{j_i = 0.0987}$

STEP 6: Calculate the value of the coefficient a.
$\mathbf{a = a_3 / [1+ \{0.14* (R_{es}\wedge a_4)\}]}$
a = 1.450 / [1+ {0.14* (45.52^.519)}]
$\underline{a = 0.719}$

Where, a_1=1.360, a_2 = -.657, a_3 = 1.450 and a_4 = .519 for R_{es} < 100, are the coefficients to be taken from the table given in Kakac book for the obtained value of Reynolds number and pitch and layout.

STEP 7: Calculate the ideal heat transfer coefficient h_i.
$\mathbf{h_i = j_i *C_{ps}*(m_s / S_m)*\{(1/P_{rs})\wedge(2/3)\}*\{ (\mu_s / \mu_w)\wedge 0.14\}}$
h_i = .0987 *4.187*(.0354/ .017)*{(1/3.7597)^(2/3)}*{ (.00088 / .00088)^0.14}
$\underline{\mathbf{h_i = 0.3559\ W/\ m^{2}{}'K}}$

STEP 8: Calculate the fraction of total tubes in crossflow F_c.
Consider, $\mathbf{\Theta = \{D_s\text{-}(2*L_c)\}/ D_{otl}}$

Θ = {.2 - (2*.05}/ .172

$\Theta = 0.581$ rad

$\mathbf{F_c = (1/\pi)*[\pi + (2*\Theta) * \sin\{\cos^{-1}(\Theta)\} - \{2* \cos^{-1}(\Theta)\}]}$
$F_c = (1/\pi)*[\pi + (2*.581) * \sin\{\cos^{-1}(.581)\} - \{2*\cos^{-1}(.581)\}]$
$F_c = 0.695$

Here $L_c = 0.25* .2 = 0.05$ for 25% baffle cut.

STEP 9: Calculate the correction factor for baffle cut and spacing J_c.

The value of J_c can be obtained from the *fig 2.33 of wolverine tube heat transfer data book Page No. 107* for the corresponding value of F_c.

$J_c = 1.05$

STEP 10: Calculate shell to baffle leakage area for one baffle S_{sb}.
$\mathbf{S_{sb} = D_s*(\Delta sb / 2)*[\pi - \cos^{-1}(\Theta)]}$
$S_{sb} = .2*(.0254 / 2)*[\pi - \cos^{-1}(.581)]$
$S_{sb} = 0.00556\ m^2$

STEP 11: Calculate tube to baffle leakage area for one baffle S_{tb}.
$\mathbf{S_{tb} = (\pi*D_o)*(\Delta tb / 2)*N_t*[(1 + F_c)/ 2]}$
$S_{tb} = (\pi*.01924)*(.0005 / 2)*18*[(1 +.695)/ 2]$
$S_{tb} = 0.0002305\ m^2$

STEP 12: Calculate
$\mathbf{(S_{sb} + S_{tb}) / S_m}$

$(.00556 + 0.0002305) / .017$
$(S_{sb} + S_{tb}) / S_m = 0.349$
&

$\mathbf{S_{sb} / (S_{sb} + S_{tb})}$
$0.00556 / (0.00556+0.0002305)$
$S_{sb} / (S_{sb} + S_{tb}) = 0.9601$

STEP 13: Calculate the correction factor for baffle leakage effects J_L.

The value of J_L can be obtained from the *fig 2.34 of wolverine tube heat transfer data book Page No. 108* for the corresponding value obtained in step 12 above.

$J_L = 0.89$

STEP 14: Calculate the fraction of the crossflow area available for bypass flow F_{bp}.
$\mathbf{F_{bp} = (L_b / S_m)*(Ds - D_{otl})}$
$F_{bp} = (.2 / .017)*(.2- .172)$
$F_{bp} = 0.3383$

STEP 15: Calculate the correction factor for bundle bypassing effects due to the clearance between the outermost tubes and the shell and pass dividers J_b.

The value of J_b can be obtained from the *fig 2.35 of wolverine tube heat transfer data book Page No. 109* for the corresponding value of F_{bp}.

$J_b = 0.96$

STEP 16: The correction factors J_s and J_r are equal to 1 for $R_{es}>=100$. But for $R_{es}<100$, J_r can be obtained from *Fig. 2.37 of wolverine tube heat transfer data book Page No. 111*.

$J_r = 0.88$

STEP 17: Calculate the shell side heat transfer coefficient for the exchanger h_o.

$\mathbf{h_o = h_i *J_c*J_L*J_b*J_s *J_r}$
$h_o = 0.3559 *1.05*0.89*0.96$

$\boxed{\mathbf{h_o = 0.32\ W/ m^2 K}}$

Calculation Of Shell Side Pressure Drop Using Kern Method:

$N_b = \{L_s / (L_b + t_b)\} - 1$
$= \{0.800/(0.2+0.00162)\}-1$

$Nb+1= 3.96$
$f = \exp\{0.576 - (0.19*\text{Ln } R_{es})\}$
$= \exp\{0.567-(0.19*\text{Ln}*91.2)\}$
$= 0.754$

$\Delta P_s = [f* G_s^2 * D_s*(N_b+1)] / [2* \rho_s* D_e*\{(\mu_s / \mu_w)^{0.14}\}]$
$=[0.754*(2.47^2)*0.2*3.96]/ [2*1000*0.0325*1]$
$\boxed{\mathbf{\Delta P_s = 0.056Pa}}$

6.4 Calculation of Shell Side Pressure Drop Using Bell Method:

Cross flow zones
$\Delta Pc = \Delta Pi\ F'b\ F'L$
Ideal tube Pr drop (ΔPi)

$(\Delta Pi) = 8Jf\ Ncv\ \rho us^2 / 2\ (\mu/ \mu w)^{-0.14}$
$=8x7.5x3.38x1000x0.002462^2 /2x1$
$=0.6136$

$Res = \rho us\ do/ \mu$
$=1000x0.00246x0.01924/0.00088$
$=53.78$

F 'b bypass correction factor for Pr drop.
$F'b = \exp[-\alpha Ab/ As\ (1 - \{2Ns / Ncc\}^{1/3})]$

$=\exp[(-5x0.0344/0.01434)x(1-(2x0.2)^{1/3}]$
=0.04255

Where, α = 5.0, if Re < 100 for laminar region
α = 4.0 if Re > 100 for turbulent region.
F 'L leakage factor for Pr drop.

F 'L = 1- βL {Atb + 2 Asb/ AL}
=1-0.7(0.0118830+2x0.000305/0.1218)
=0.2820

ΔPc=ΔpiFb'Fl'
=0.6136x0.2820x0.04255
=0.00736

Window zone pressure drop.
$\Delta Pw = F\ 'L\ (2 + 0.6\ Nwv)\ \rho uz^2/2$
$=0.2820(2+0.6x0.833)x(1000x0.00382^2)/2$
=0.005116

Where $U_w = W_s / A_{w\rho}$
=0.0354/0.00596x1000
=0.00593

Geometric Mean velocity $U_z = \sqrt{UwUs}$
= √0.00593x0.00246
=0.0038

$A_w = (\pi/4 \times ID^2 \times Ra) - (Nw \times \pi/4 \times OD^2)$
$=(\pi/4x0.2^2 x0.19)-(2.52x\pi/4x0.01924^2)$
=0.0059

$N_{wv} = H_b / P'_t$
=0.036/0.03
=0.833

End Zone

ΔPe = ΔPi [(Nwv + Ncv/ Ncv)] F 'b
=0.6136(0.833+3.38/3.38)x0.04255
=0.03254

Total shell –side pressure drop
ΔPs = 2ΔPe + ΔPc(Nb – 1) + Nb ΔPw
=2x0.03254+0.00736(4-1)+4x0.005116

ΔPs =0.1074

6.5 Calculation of Shell Side Pressure Drop Using Bell Delaware Method:

STEP 1: Calculate the number of tube rows crossed in one crossflow section N_c.

$\mathbf{N_c = (D_s /P_{tp})*[1 – \{(2*L_c)/D_s\}]}$
$N_c = (.2 /.02598)*[1 – \{(2*.05)/.2\}]$
$N_c = 3.85 = 4$

Where, $\mathbf{P_{tp} = 0.866*P_t}$
$P_{tp} = 0.866*.03$
$P_{tp} = 0.02598$

STEP 2: Calculate the ideal cross flow pressure drop through one baffle space ΔP_b.
$\mathbf{\Delta P_b = [(2*f_s*m_s^2*N_c) / (\rho_s*s_m^2)]* [(\mu_s / \mu_w)^{\wedge}0.14]}$
$\Delta P_b = [(2*.000125*.0354^2*4) / (1000*.017^2)]* [(.00088/ .00088) ^0.14]$
$\Delta P_b = 4.33*10^{-6}$ Pa

STEP 3: Calculate the window flow area S_w.
$\mathbf{S_w=(D_s^2/4)*[\cos^{-1}\Theta-\{\Theta*\sqrt{(1-\Theta^2)}\}]-[(N_t/8)*(1-F_c)*\pi* D_o^2]}$
$S_w=(.2^2/4)*[\cos^{-1}.581-\{.581*\sqrt{(1-.581^2)}\}]-[(18/8)*(1-.695)*\pi* .01924^2]$
$S_w=0.004\ m^2$

STEP 4: Calculate the number of effective cross flow rows in window zone N_{cw}.
$\mathbf{N_{cw} = (0.8*L_c) / P_{tp}}$
$N_{cw} = (0.8*.05) / .02598$
$N_{cw} = 1.54$

STEP 5: Calculate the window zone pressure drop ΔP_w.
$\mathbf{\Delta P_w = [\{(26\ \mu_s\ m_s) /(\ \rho_s\sqrt{(\ S_m\ S_w)}\}*\{(N_{cw} /(\ P_t - D_o)) + (L_b/D_w^2)\}]+[\ m_s^2 / (2\ \rho_s\ S_m\ S_w)]}$

$\Delta P_w = [\{(26* .00088* .0354) /(1000*\sqrt{(.017* .004)}\}*\{(1.54 /(.03- .01924)) + (.2/.022^2)\}]+[.0354^2 / (2*1000*.017* .004)]$
$\Delta P_w = 0.0638$ Pa

STEP 6: Estimate the correction factor on pressure drop for bypass flow R_b.
The value of R_b can be obtained from the fig 2.39 of wolverine tube heat transfer data book for the corresponding value of F_{bp}.

$\mathbf{R_b = 0.78}$

STEP 7: Estimate the correction factor for baffle leakage effect on pressure drop R_L

The value of R_L can be obtained from the fig 2.38 of wolverine tube heat transfer data book for the corresponding value obtained in step 12 above.

$\mathbf{R_L = 0.78}$
STEP 8: Calculate the total pressure drop across shell ΔP_s
$\mathbf{\Delta P_S= [\{(N_b- 1)*\Delta P_b*R_b\} + (N_b*\Delta P_w)]*R_L + [2*\Delta P_b*R_b*\{1+ (N_{cw}/N_c)\}]}$

$\Delta P_S= [\{(4- 1)* 4.33*10^{-6}*.78\} + (4*0.0638)]*.78 + [2*4.33*10^{-6} *.78*\{1+ (1.54/4)\}]$

$\boxed{\mathbf{\Delta P_S= 0.278\ Pa}}$

6.6 Calculation of Tube Side Heat Transfer Coefficient using Gnielinski Correlation:

$A_t = \{(\pi * D_i^2) / 4\}*(N_t / 2)$
$= \{(\pi*0.016^2)/4\}*(18/2)$
$= 0.00180\ m^2$

$G_t = m_t / A_t$
$= 0.0291/0.00180$
$= 16.17$ Kg/m sec

$U_t = G_t / \rho_t$
$= 16.17/1000$
$= 0.01617$ m/sec

$R_{es} = (G_t * D_i) / \mu_t$
$= (16.17*0.016)/0.00086$
$=300.83$

$P_{rt} = (C_{pt} * \mu_t) / K_t$
$= (4.187*0.00086)/0.00098$
$= 3.67$

$f = \{(1.58*\ln R_{et}) – 3.28\} \wedge (-2)$
$= \{(1.58*\ln 300.83)-3.28\}\wedge(-2)$
$= 0.0304$

$N_{ut} = \{(f /2)*(Ret -1000)* P_{rt}\} / \{1+(12.7*\sqrt{} (f /2)*(P_{rt} \wedge (2/3))-1)\}$
$=\{(0.0304/2)*(300.83-1000)*3.67]/\{1+(12.7*\sqrt{} (0.0304/2)*(3.67^{2/3}- 1)\}$
$=-12.4$

$h_i = (N_{ut} * K_t) / D_i$
$= (-12.4*0.00098)/0.016$

$\boxed{\mathbf{h_i = -0.78\ W/m^{2\ 0}K}}$

6.7 Calculation of Tube Side Pressure Drop:

$\Delta P_t = [\{(4* f* L_t* n_p) / D_i\} + (4* n_p)]*[(\rho t*U_t^2) / 2]$
$=[\{(4*0.0304*0.825*2)/0.016\}+(4*2)]*[(1000*0.01617^2)/2]$
$= 2.685$Pa
$.00878^2) / 2]$

$\boxed{\mathbf{\Delta P_t =2.685Pa}}$

7. RESULTS AND DISCUSSION

Shell side results for four sample readings are shown below

	Shell side (Cold Water)	**R1**	**R2**	**R3**	**R4**
1	Mass flow rate (Kg/sec)	0.0257	0.0299	0.035	0.0397
2	Temperature at inlet (^{o}c)	29.7	30.1	30.6	31.2
3	Temperature at outlet (^{o}c)	32.2	33	33.5	33.9
4	Reynolds number (**Kern Method**)	66.93	76.45	91.12	102.4
5	Reynolds number (**Bell Method**)	35.53	45.28	53.97	60.68
6	Reynolds number (**Bell Delaware Method**)	32.96	38.346	44.887	50.915
7	Prandtl number	3.76	3.76	3.76	3.76
8	Heat transfer coefficient (W/ m^2'K). (**Kern Method)**	0.17	0.183	0.201	0.214
9	Heat transfer coefficient (W/ m^2'K).(**Bell Method)**	0.162	0.231	0.271	0.296
10	Heat transfer coefficient (W/ m^2'K).(**Bell Delaware method**)	0.286	0.302	0.32	0.336
11	Pressure drop (Pa) **(Kern Method)**	0.032	0.041	0.056	0.069
12	Pressure drop (Pa) **Bell Method)**	0.065	0.084	0.1075	0.109
13	Pressure drop (Pa **Bell Delaware method)**)	0.169	0.215	0.278	0.343

Tube side results for four sample readings are shown below

Sl.no	Tube side	Result 1	Result 2	Result 3	Result 4
1	Mass flow rate (Kg/sec), Mt	0.020	0.0231	0.0291	0.0364
2	Temperature at inlet (^{o}c), Thi	49.8	52.6	54.4	54.8
3	Temperature at outlet (^{o}c), Tho	34.9	34.7	34.6	34.5
4	Reynolds number, Ret	206.718	238.76	300.775	376.227
5	Prandtl number, Prt	3.674	3.674	3.674	3.674
6	Heat transfer coefficient (W/ m^2'K), hi	-1.106	-1.002	-0.842	-0.693
7	Pressure drop (Pa), ΔPa	1.456	1.836	2.684	3.91

8. GRAPHS

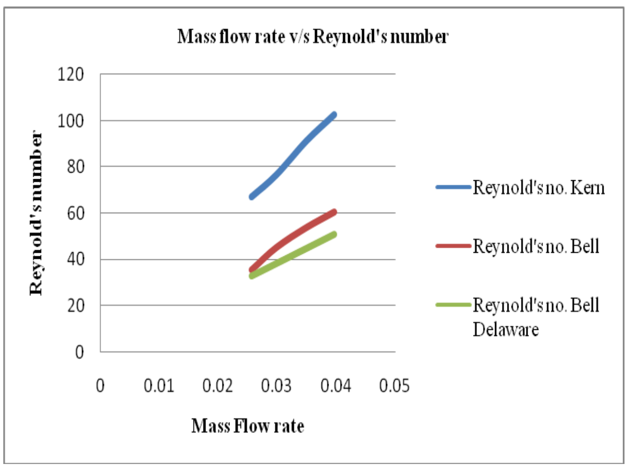

Fig a. Comparison of Variation of Reynold's number w.r.t Flow rate on the Shell side using three methods

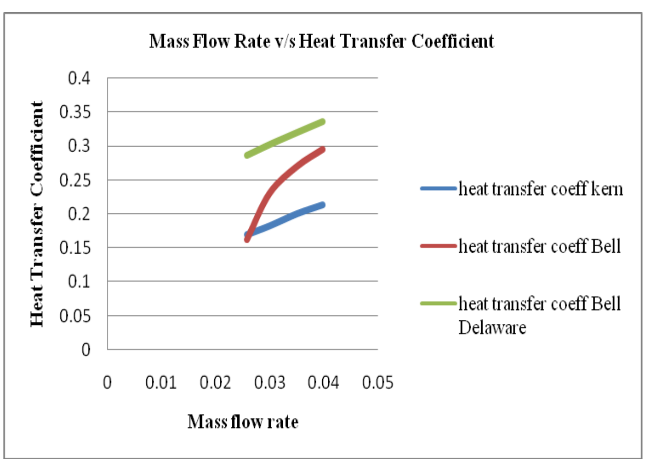

Fig b. Comparison of Variation of Heat Transfer Coefficient w.r.t Flow rate on the Shell side using three methods

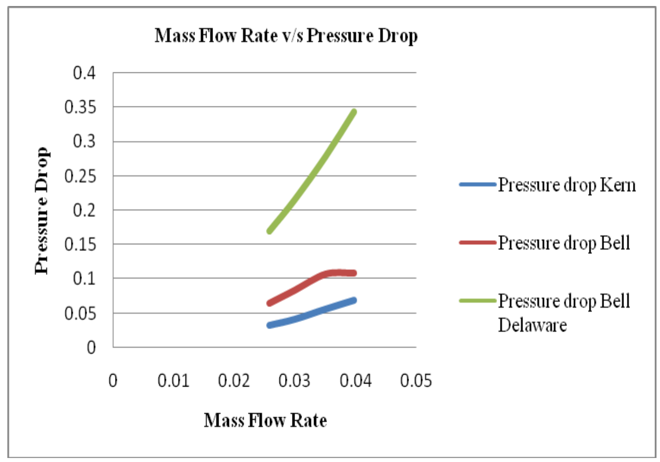

Fig c. Comparison of Variation of Pressuer drop w.r.t Flow rate on the Shell side using the three methods

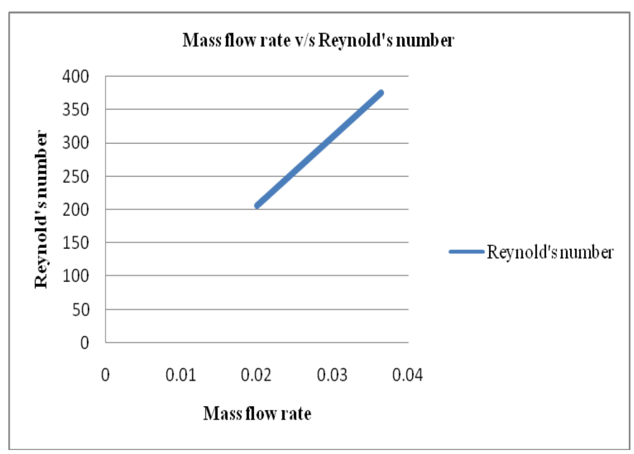

Fig d. Variation of Reynold's number w.r.t Flow rate on the Tube side

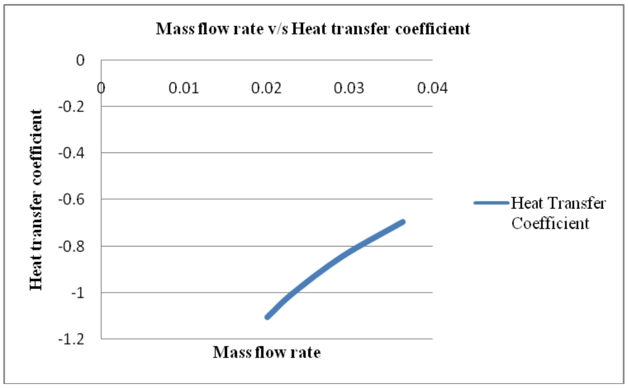

Fig e. Variation of Heat Transfer Coefficient w.r.t Flow rate on the Tube side

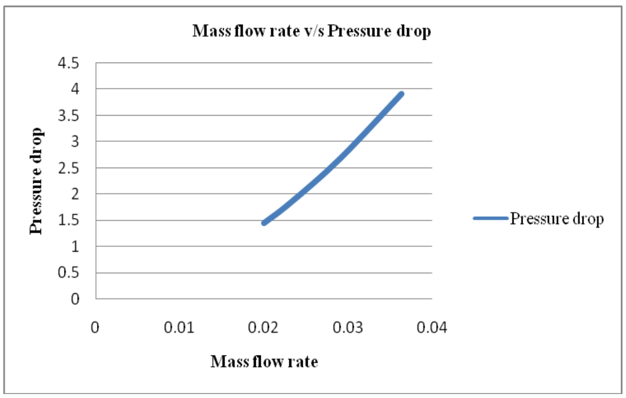

Fig f. Variation of Pressure drop w.r.t Flow rate on the Tube side.

9. CONCLUSIONS

The shell and tube heat exchanger is analyzed using Kern, Bell and Bell Delaware methods and heat transfer coeffient, Reynold's number, pressure drops are calculated for various mass flow rates and the results are shown in the graphs above. We found that, shell side heat transfer coefficient increases with increasing mass flow rate in all the three methods, but the heat transfer given by Bell Delaware method is much more than the other two methods. Also the shell side pressure increase rapidly with increasing flow rate and this increase is again more in Bell Delaware method as compared to others.

Since in a baffled heat exchanger, there is a obstruction to flow, drop in the pressure is definitely more when compared to the heat exchanger without baffles. Kern method does not take in to consideration the obstructions due to baffles in calculating the pressure drops and hence the pressure drop given by Kern method is unrealistic. Whereas, the pressure drops given by Bell and Bell Delaware methods, is more realistic, since these methods consider pressure drop due to bypass and leakage streams caused by the baffles in the heat exchanger.

REFERENCES

[1]. **Jian-Fei Zhang, Ya-Ling He, Wen-Quan Tao1,** A Design and Rating Method for Shell-and-Tube Heat Exchangers With Helical Baffles, School of Power and Energy Engineering, Xi'an Jiaotong University, Xi'an 710049, China

[2]. **Ebieto, CE and Eke G B,** "Performance Analysis of Shell and Tube Heat Exchangers: A case study" Department of Mechanical Engineering, Faculty of Engineering, University of Port Harcourt, Port Harcourt, Nigeria. Journal of Emerging trends in engineering and applied sciences (JETEAS) 3(5):899-903.

[3]. **Sandeep K. Patel, Professor Alkesh M. Mavani** "Shell & tube heat exchanger thermal design With optimization of mass flow rate and baffle spacing" International Journal of Advanced engineering Research and Studies IJAERS/Vol.II/ Issue I/Oct.-Dec.,2012/130-135

[4]. **Mauro A. S. S. Ravagnani[1], Aline P. Silva[1] and Jose A. Caballero[2]** "Optimal Shell and Tube Heat Exchangers Design" State University Of Maringa[1,] University Of Alicante[2], Brazil[1], Spain[2].

[5]. **Edward S. Gaddis a,*, Volker Gnielinski b** "Pressure drop on the shell side of shell-and-tube heat exchangers with segmental baffles1" Institute fir Thermische Verfuhrenstechnik, Technische UniversitZit ClausthaI, LeibnizsraJe 1.5, 38678 ClausthaI-Zellerfeld, Germany b Lehrstuhl ldnd Institut fir Then-n&he Verfahrenstechnik, Universitiit h'arlsruhe TH, I CaiserstraJe 12, 76131 Karlsrtlhe, Germany

[6]. **Sadik kakac**, "Heat Exchangers Selection, Rating and Thermal Design", 2002.

[7]. Sunilkumar Shinde, Mustansir Hatim Pancha "Comparative Thermal Performance Analysis Of Segmental Baffle Heat Exchanger with Continuous Helical Baffle Heat Exchanger using Kern method. International Journal of Engineering Research and Applications(IJERA),July-August 2012

[8]. Nenad Radojkovic1, Gradimir Ilic1, Zarko Stevanovic, Mica Vukic1, Dejan Mitrovic1, Goran Vuckovic1 "Experimental study on thermal and flow processes in shell and tube heat exchangers" Mechanical Engineering Vol. 1, No 10, 2003, pp. 1377 – 1384

[9]. **Yao, Ye; Zhang, Xingyu; and Guo, Yiying,** "Experimental Study on Heat Transfer Enhancement of Water-water Shell-and-Tube Heat Exchanger Assisted by Power Ultrasonic". *International Refrigeration and Air* Conditioning Conference (2010) [11] John E Edwards,

[10]. **Young-Seok Son", J ee-Young Shin "**Performance of a Shell-and-Tube Heat Exchanger with Spiral Baffle Plates" Dong-Eui University, Busan 614-714, Korea KSME International Journal, Vol. 15. No. n. pp. 1555-1562, 2001

[11]. **Tinker t. (1951**) "shell-side characteristic of shell- and-tube heat exchangers", parts ii, iii, and in: proceedings of general discussion on heat transfer, institute of mechanical engineers and american society of mechanical engineers, london, new york, pp. 89.

[12]. **B.V. Babua, S.A. Munawar,** Differential evolution strategies for optimal design of shell-and-tube heat

exchangers,Department of Chemical Engineering, Birla Institute of Technology and Science (BITS), Pilani

[13]. Comparative Analysis of Finned Tube and Bared Tube Type Shell and Tube Heat Exchanger **Shiv Kumar Rathore, Ajeet Bergaley**, International Journal of Engineering and Innovative Technology (IJEIT) Volume 2, Issue 1, July 2012, ISSN: 2277-3754 ISO 9001:2008 Certified.

[14]. **M. Siddique, A.-R. A. Khaled, N. I. Abdulhafiz, and A. Y. Boukhary** "Recent Advances in Heat Transfer Enhancements" Hindawi Publishing Corporation International Journal of Chemical Engineering Volume 2010