ISSN: 2277-3754
ISO 9001:2008 Certified
International Journal of Engineering and Innovative Technology (IJEIT)
Volume 2, Issue 1, July 2012

Comparative Analysis of Finned Tube and Bared Tube Type Shell and Tube Heat Exchanger

Shiv Kumar Rathore, Ajeet Bergaley

Abstract*— The aim of this paper is to identify the advantages of low-finned tube Heat Exchangers over Plain tube (Bare Tube) units. To use finned tubes to advantage in this application, several technical issues were to be addressed. (1) Shell side and tube side Pressure, (2) Cost, (3) Weight and (4) Size of Heat Exchanger, Enhanced tubular heat exchangers results in a much more compact design than conventional plain tube units, obtaining not only thermal, mechanical and economical advantages for the heat exchanger, but also for the associated support structure, piping and skid package unit, and also notably reduce cost for shipping and installation of all these components. A more realistic comparison is made on the basis of respective cost per meter of tubing divided by the overall heat transfer coefficient for the optimized units, which gives a cost to performance ratio. This approach includes the entire thermal effect of internal and external heat transfer augmentation and fouling factors in the evaluation. This is typically quite close to reality and easy for the thermal designer to evaluate himself. The results of this analysis shows that the finned tube heat exchanger is more economical than Conventional Bare tube Exchanger, The tube side pressure drop and fluid velocity is higher than the conventional bare tube exchanger, which prevent fouling inside the tubes, The shell side pressure drop is some lesser but fluid velocity is higher than the conventional heat exchanger. Which safe the outer surface of tubes from fouling creation and fluid transfer time The shell diameter of finned tube Exchanger is lesser than Conventional bare tube heat exchanger, which saves sheet material and reduces the size of the shell, which helps to easily installation in the plant.

Keywords — **Low Finned Tube , Bare Tubes, Shell and Tube type Heat exchanger, Kerns Method, Heat Transfer Coefficient, Heat Transfer rate, Bells Method, Pressure Drop, Baffles, Tube Pitchl, Tube Patterns.**

I. INTRODUCTION

Heat exchangers have always been an important part to the lifecycle and operation of many systems. A heat exchanger is a device built for efficient heat from one medium to another in order to carry and process energy. Typically one medium is cooled while the other is heated. They are widely used in petroleum refineries, chemical plants, petrochemical plants, natural gas processing, air conditioning, refrigeration and automotive applications.

Fig 1: 1:2 pass shell and tube Heat Exchanger

The use of integral finned tube shell and tube heat exchanger design contributes greatly to the development of compact design. Enhanced tubular heat exchangers results in a much more compact design than conventional plain tube units, obtaining not only thermal, mechanical and economical advantages for the heat exchanger, but also for the associated support structure, piping and skid package unit, and also notably reduce cost for shipping and installation of all these components. The compact enhanced designs also greatly reduce the quantities of the two fluids resident within the exchanger, sometimes an important safety consideration. There are many thermal advantages of utilizing augmentation that must be weighed against their higher cost relative to plain tubing and their economic benefit on plant operation. For many small increases to production capacity (10 to 30%), the purchase and installation of completely new exchangers are the “bottleneck” of a unit operation, and then augmentations may be the right solution. In order to develop a compact shell and tube heat exchanger, the basic principles of heat transfer must be considered. The amount of heat transferred is a function of the heat exchanger geometry and configuration of the unit affects all of these parameters and must be varied until an optimum design is reached. The standard geometry and configuration option include, but are not limited to, baffle and support plate arrangement, placement of fluids, number of passes, tube pattern and tube enhancements. All of these should be considered when designing a heat exchanger for cost savings. As explained by Polly et al. [8] one way of enhancing heat transfer and consequently reducing exchanger size to pass the fluid through the tube side of exchangers fitted with appropriate type of tube inserts. An alternative means of reducing the size of a unit is the use of low-finned tubes to augment the shell side surface area. Low finned tubes might be significantly more expensive than the plain tube. Therefore, their use has to be justified by a significant exchanger size reduction and identification of other potential benefits in simplifying fabrication of a heat exchanger unit. As explained by investigators such as Fraas A.P. and Ozisik M.N. [10], comparing the heat transfer coefficients for shell side with those for flow through the inside of round tubes, it is apparent that it is advantageous to place the more viscous fluid, tending to give the lower Reynolds number, on the shell-side and the less viscous fluid, giving the Reynolds number, on the tube-side. In this way advantage can be taken of the higher heat transfer coefficients at lower Reynolds numbers given by cross flow conditions, and the two heat transfer coefficients can be made more nearly the same to give a well proportioned heat exchanger. Using low finned tubes to enhance the heat transfer in cross-flow and to provide a greater surface area per unit

ISSN: 2277-3754
ISO 9001:2008 Certified
International Journal of Engineering and Innovative Technology (IJEIT)
Volume 2, Issue 1, July 2012

volume than plain tubes further reduces the size of the unit. Size reduction is not always the sole reason why the use of low-finned tubes results in more cost effective heat exchanger designs. Finned tubes are now widely used in industrial shell and tube units used as boiler, economizers, in water heaters and in air cooler heat exchangers. The application of finned tubes is not limited to single phase flow. They can also be used in condensation and boiling applications. In all application their use provides heat exchangers, which are compact than a plain tube. The recent study of authors, Jafari Nasr M.R. and Polly G.T. [9], showed that mechanical constraints play a significant role in the design of shell and tube heat exchangers. For example, it is normal practice in exchanger design to restrict the length of the tubes to less than 6 meters. This restriction can lean to the use of multi passing of tubes, which usually introduces the need to examine temperature cross considerations, a reduction in the effective mean temperature difference and therefore a need for increase in surface area. If the use of a single shell results in too low a value of F_t (temperature correction factor) the designer should move to a multiple shell in design. The potential benefits of low finned tubes than plain tube units have already been investigated by Jafari Nasr M.R. [11]. The results clearly showed that using low finned tube could provide a significant area reduction for a given duty.

I. PERFORMANCE TERMS AND DEFINITIONS: [16]

Overall heat transfer coefficient, (U): -
Heat exchanger performance is normally evaluated by the overall heat transfer coefficient U that is defined by the equation

$$Q = U \times A \times MTD \qquad [16](1)$$

Where,
Q = Heat transferred rate in k Cal/hr = watt
U = Overall heat transfer coefficient, k Cal/hr/m^2/℃ or $\frac{W}{m^2}$℃
A = Heat transfer surface area in m^2
MTD = Mean temperature Difference, ℃ = LMTD x F
LMTD = logarithmic mean temperature difference, ℃
F = Correction Factor
When the hot and cold stream flows and inlet temperature are constant, the heat transfer coefficient may be evaluated using the above formula. It may be observed that the heat pick up by the cold fluid starts reducing with time.
Nomenclature,
A typical heat exchanger is shown in figure 2.1 with nomenclature.
Heat duty of the exchanger can be calculated either on the hot side fluid or cold side fluid as given below

Heat Duty for Hot Fluid

$$Q_h = W_h \times C_{ph} \times (T_i - T_o) \qquad (2)$$

Heat Duty for Cold Fluid

$$Q_c = W_c \times C_{pc} \times (t_i - t_o) \qquad (3)$$

Where, W_h = Mass flow rate of hot fluid,
W_c = Mass flow rate of Cold fluid,
C_{ph} = Mass flow rate of hot fluid,
C_{pc} = Mass flow rate of Cold fluid,
T_i , T_o = Inlet and outlet temperature of hot fluid,
t_i , t_o = Inlet and outlet temperature of Cold fluid,

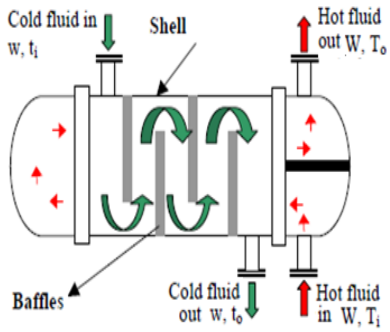

Fig 2. Typical Shell and Tube Heat Exchanger [16]

If the operating heat duty is less than design heat duty, it may be due to heat losses, fouling in tubes, reduced flow rate (hot or cold) etc. Hence for simple performance monitoring of exchanger, efficiency may be considered as factor of performance irrespective of other parameter. However, in industrial practice, fouling factor method is more predominantly used.

II. PROBLEM STATEMENT & METHADOLOGY

A. ***Problem Statement:*** Identification of advantages of low-finned tube Heat Exchangers over Plain tube (Bare Tube) units has been developed. To use finned tubes to advantage in this application, several technical issues were to be addressed. (1) Shell side and tube side Pressure, (2) Cost, (3) Weight and (4) Size of Heat Exchanger

B. Methodology Of Heat Exchanger Performance Assessment

Determination of overall heat transfer coefficient, U: - [2]
The overall heat transfer coefficient, U, including fouling, shall be calculated as follows:

$$U = \frac{1}{\left[\left(\frac{1}{h_o} + r_o\right)\left(\frac{1}{E_f}\right) + r_w + r_i\left(\frac{A_o}{A_i}\right) + \frac{1}{h_i}\left(\frac{A_o}{A_i}\right)\right]} \qquad [2], (4)$$

Where
U = Overall heat transfer coefficient (fouled), (external surface) $\frac{W}{m^2}$℃
h_o = Film Coefficient of fluid outside tubes, (external surface) $\frac{W}{m^2}$℃

h_i = Film coefficient of fluid inside tubes, (internal surface) $\frac{w}{m^2}$℃

r_o = fouling resistance on outside of tubes, (External surface) m^2℃$\frac{}{w}$

r_i = Fouling resistance on inside of tubes (internal surface) m^2℃$\frac{}{w}$

R_w = Resistance of tuber wall referred to outside surface, including extended if present (External surface) surface if present, (External surface) m^2℃$\frac{}{w}$

A_o = Effective External surface, m^2

A_i = Effective internal surface, m^2

E_f = Fin efficiency (equal one for bare tubes and less than one for finned tubes)

***Tube Wall Resistance*: [2]**

(A) For Bare Tubes: -

$$r_w = \frac{d_o}{24\,k}\left[\ln\left(\frac{d_o}{d_o - 2t}\right)\right]$$

[2], (5)

(B) Integral Circumferentially Finned Tubes: -

$$r_w = \frac{t}{12\,k}\,\frac{[d_o + 2\,N\,Z\,(d_o + Z)]}{(d_o - t)}$$

[2], (6)

Where

d_o = O.D. of bare tube or root diameter of fin, m

z = Fin height, m

t = Tube wall thickness, m

N = Number of fins per meter

K = Tube wall thermal conductivity, $\frac{w}{m}$℃

C. Mean Temperature Difference: - [2]

MTD = LMTD x F = Mean Temperature difference, ℃

Where

LMTD = Logarithmic mean temperature difference, ℃

(A) LMTD for Parallel Flow: -

$$\text{LMTD} = \frac{[T_i - t_i] - [T_o - t_o]}{\ln\left[\frac{T_i - t_i}{T_o - t_o}\right]}$$

[2], (7)

(B) LMTD for True Counter Flow: -

$$\text{LMTD} = \frac{[T_i - t_o] - [T_o - t_i]}{\ln\left[\frac{T_i - t_o}{T_o - t_i}\right]}$$

[2], (8)

Where

T_i , T_o = Hot fluid inlet, outlet temperature, ℃

t_i , t_o = Cold fluid inlet, outlet temperature, ℃

And **F** = the LMTD correction factor to adjust for deviation from true counter flow, is a function **R** and **S** and can be obtained from Equations as Follows:

$R = \frac{T_i - T_o}{t_o - t_i}$ is equal to the shell-side fluid flow rate times the fluid mean specific heat; divided by the tube-side fluid flow rate times the tube-side fluid specific heat.

And $S = \frac{t_o - t_i}{T_i - t_i}$ is a measure of the temperature efficiency of the exchanger

For a 1 shell: 2 tubes pass exchanger, the correction factor is given by:

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

[3], (9)

The derivation of this equation is given by Kern (1950). The equation for a 1 shell: 2 tube pass exchanger can be used for any exchanger with an even number of tube passes.

D.Design Methods [3]:

Procedure for calculating the shell-side heat transfer coefficient:

(A) **Kerns Method:** [3]

The method was based on experimental work on commercial exchangers with standard tolerances and will give a responsibly 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-section area for flow will vary across the shell diameter, the linear and mass

ISSN: 2277-3754
ISO 9001:2008 Certified
International Journal of Engineering and Innovative Technology (IJEIT)
Volume 2, Issue 1, July 2012

velocities are based on the maximum area for cross-flow: that at the equator. The shell equivalent diameter is calculated using the wetted perimeter of the tubes: Figure 3.5

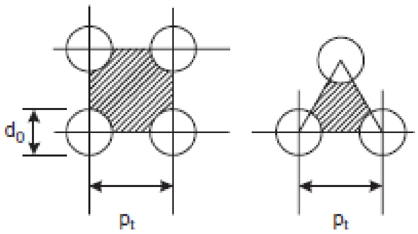

Fig 3. Equivalent Diameter, Cross-Sectional Areas and Wetted Perimeters [3]

Shell–side "J_h" and "J_f" factors for use in this method are given in Appendix 9 and 10 for various baffle cuts and tube arrangements. These figures are based on data given by Kern (1950) and Ludwig (2001).

The procedure for calculating the shell-side heat transfer coefficient and pressure drop for a single shell pass exchanger is given below:

Procedure

1. Calculate the area for cross-flow "A_s" for the hypothetical row of tubes at the shell equation, given by :

$$A_s = \frac{(P_t - d_o) D_s \, \ell_B}{P_t}$$

[3], (10)

Where, P_t = tube pitch,
d_o = tube outside diameter,
D_s = shell inside diameter, m
ℓ_B = baffle spacing, m.

The term $\frac{(P_t - d_o)}{P_t}$ is the ratio of clearance between tubes and the total distance between tube centers.

2. Calculate the shell-side mass velocity "G_s" and the linear velocity "U_s" [3]

$$G_s = \frac{W_s}{A_s} \quad \& \quad U_s = \frac{G_s}{\rho}$$

Where, W_s = fluid flow-rate on the shell-side, kg/s,
ρ = shell-side fluid density, kg/m^3.

3. Calculate the shell-side equivalent diameter (hydraulic diameter), for a square pitch arrangement:

$$d_e = \frac{4\left(\frac{P_t^2 - \pi d_o^2}{4}\right)}{\pi d_o} = \frac{1.27}{d_o}\left[(P)_t^2 - 0.785\, d_o^2\right]$$

[3], (11)

For an equilateral triangular pitch arrangement:

$$d_e = \frac{4\left(\frac{1.74 P_t^2 - 0.5\pi d_o^2}{4}\right)}{\frac{\pi d_o}{2}} = \frac{1.10}{d_o}\left[(P)_t^2 - 0.917\, d_o^2\right]$$

[3], (12)

Where,
d_e = equivalent diameter, m.

4. Calculate the shell-side Reynolds number and Prantal number, given by:

$$\text{Reynolds numbers} = R_e = \frac{G_s d_e}{\mu} = \frac{U_s d_e \rho}{\mu}$$

$$\& \text{ Prantal number} = P_r = \frac{C_p \mu}{K_f}$$

[3], (13)

5. For the calculate Reynolds number, read the value of "J_h" from Appendix 9 for the selected baffle cut and tube arrangement, and calculate the shell-side heat transfer coefficient "h_s" from:

$$N_u = \frac{h_s d_e}{k_f} = J_h \times R_e \times P_r^{\frac{1}{3}} \times \left(\frac{\mu}{\mu_w}\right)^{0.14}$$

[3], (14)

The tube wall temperature can be estimated using the method given for the tube-side.

6. For the calculated shell-side Reynolds number, read the friction factor from (Appendix10) and calculate the shell-side pressure drop from:

$$\Delta P_s = 8\, J_f \left(\frac{D_s}{d_e}\right)\left(\frac{L}{\ell_B}\right)\frac{\rho U_s^2}{2}\left(\frac{\mu}{\mu_w}\right)^{-0.14}$$

[3], (15)

Where, L = tube length,
ℓ_B = baffle spacing.

The term $\left(\frac{L}{\ell_B}\right)$ is the number of times the flow crosses the tube bundle = (N_b + 1),
Where "N_b" is the number of baffles.

***Procedure for Calculating the Shell-Side Pressure Drop*: [3]**

(A) Bell's Method:

The pressure drops in the cross-flow zones and window zones are determined separately, and summed to give the total shell-side pressure drop. In Bell's Method the heat-transfer coefficient and pressure drop are estimated from correction's

ISSN: 2277-3754
ISO 9001:2008 Certified
International Journal of Engineering and Innovative Technology (IJEIT)
Volume 2, Issue 1, July 2012

for flow estimated for flow over ideal tube banks, and the effects of leakage, by passing 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 by passing, can be used to investigate the effects of construction tolerances and the use of sealing trips. The procedure in a simplified and modified form to that given by Bell (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.

1. Cross flow zone pressure drop ΔP_c:

The pressure drop in the cross-flow zone between the baffles tips is calculated from corrections for ideal tube banks and the corrected for leakage and by-passing.

$$\Delta P_c = \Delta P_i \times f_b \times f_L \qquad [3], (16)$$

Where,
ΔP_c = the pressure drop in a cross-flow zone between the baffle tips, corrected by passing and leakage.
ΔP_i = the pressure drop calculated for an equivalent ideal tube bank.
f_b = by pass correction factor,
f_L = leakage correction factor.

ΔP_i Ideal tube bank pressure drop: The number of tube rows has little effect on the friction factors little effect on the friction factors and is ignored. Any suitable correlation for the cross-flow friction factor can be used for that given in Appendix 11. The pressure drop across the ideal tube bank is given by:

$$\Delta P_i = 8\, J_f \times N_{cv} \times \frac{\rho_s U_s^{\,2}}{2} \left(\frac{\mu}{\mu_w}\right)^{-0.14} \qquad [3], (17)$$

Where,
N_{cv} = number of tubes rows crossed (in the cross-flow region).
U_s = shell-side velocity based on the clearance area at the bundle equator,
J_f = friction factor [3] at the appropriate Reynolds number,

f_b Bypass correction factor for Pressure drop:
Bypassing will affect the pressure drop only in the cross-flow zones. The correction factor is calculated from the equation used to calculate the bypass correction factor for heat transfer equation

$$f_b = \exp\left[-\alpha \frac{A_b}{A_s}\left(1 - \left(\frac{2N_s}{N_{cv}}\right)^{\frac{1}{3}}\right)\right] \qquad [3](18)$$

But with the following values for the constant "α"
Laminar region, $R_e < 100, \alpha = 5.0$
Transient and turbulent region, $R_e > 100, \alpha = 4.0$
The correction factor for exchangers without sealing strip is shown in Appendix 4

f_L Leakage factor for pressure drop:
Leakage will affect the pressure drop in both the cross-flow and window zones. The factor is calculated using the equation for the heat-transfer leakage correction factor,
Equation with the values for the coefficient "β_L" taken from the Appendix 5

$$f_L = 1 - \beta_L \left[\frac{(A_{tb} + 2A_{sb})}{A_L}\right] \qquad [3], (19)$$

Where, β_L = a factor obtained from Appendix 5,
A_{tb} = the tube to baffle clearance area, per baffle,
A_{sb} = shell-to-baffle clearance area, per baffle,
A_L = total leakage area = $(A_{tb} + A_{sb})$

2. **Window zone pressure drop ΔP_w :**

Any suitable method can be used to determine the pressure drop in the window area; see Butterworth (1977). Bell's used a method proposed by Colburn corrected for leakage the window drop for turbulent flow is given by:

$$\Delta P_w = f_L\,(2 + 0.6 N_{wv}) \frac{\rho\, U_z^{\,2}}{2} \qquad [3], (20)$$

Where, $U_z = \sqrt{U_w U_s}$ = the geometric mean velocity,
U_w = The velocity in the window zone, based on the window area less the area occupied by the tubes A_w,

$$U_w = \frac{W_s}{\rho A_w}$$

3. **End Zone pressure Drop ΔP_e :**

$$\Delta P_e = \Delta P_i \left[\frac{N_{wv} + N_{cv}}{N_{cv}}\right] f_b \qquad [3], (21)$$

4. **Total shell side Pressure Drop ΔP_s :**

Summing the pressure drops over all the zones in series from inlet to outlet gives:

$\Delta P_s = 2$ end zones $+ (N_1 b - 1)$ cross flow zone + N_b window zones

$$\Delta P_s = 2\,\Delta P_e + (N_1 b - 1)\,\Delta P_1 c + N_b\,\Delta P_w \qquad [3], (22)$$

Where N_b is the number of baffles = $\left[\left(\frac{L}{\ell_B}\right) - 1\right]$

ISSN: 2277-3754
ISO 9001:2008 Certified
International Journal of Engineering and Innovative Technology (IJEIT)
Volume 2, Issue 1, July 2012

An estimate of the pressure loss incurred in the shell inlet and outlet nozzles must be added to that calculated by equation of ΔP_s.

E. Conventional Bare Tube Heat Exchanger Detail:

Table.1 Detail of Bare tube Heat Exchanger

1:2 passes Shell and Tube Heat Exchanger which we have considered	
Component Name	Parameter
Shell	M.S. 10.00 mm thick
Surface Area	200 m^2
Mass flow rate of hexane (hot fluid), W_h	33.33 kg/s
Disk	M.S. 8.00 mm thick
Bolts	M.S. 20.00 mm dia.
Gasket	3.00mm packing seat
Tubes	S.S., 19.05 mm dia., 5.426 m length, seamless
No. of Tubes	616
Tube type	Fixed type
Tube Layout	Triangular pitch
Tube Pitch	1.25% of 19.05 = 23.8125
Head	Floating type
Flange Thickness	10 to 12mm

Table .2 Detail of Hot Fluid (Shell Side Fluid)

Hexane Vapor Properties Detail	
Inlet Temperature, T_i	65℃
Outlet Temperature, T_o	40℃
Density "ρ_h"	659 kg/m^3 at 25℃
Viscosity μ_h	0.3000 m Ns/m^2 at 25℃
Specific heat Capacity "C_{ph}"	2.27 kJ/kg℃
Thermal Conductivity "K_h"	0.1279 W/m℃

N-hexane is used in the extraction of vegetable oil from seeds such as safflower, soybean, cotton, and flax (HSDB, 1995). It is also used as an alcohol denaturant and as paint diluents. The textile, furniture and leather industries use n-hexane as a cleaning agent. Many petroleum and gasoline products contain n-hexane.

Table .3 Detail of Cold Fluid (Tube-Side Fluid)

Water Properties Detail	
Inlet Temperature, t_i	30℃
Outlet Temperature, t_o	40℃
Specific Heat Capacity, C_{pc}	4.2 kJ/kg ℃
Density, ρ_w	995 kg/m^3
Viscosity, μ_w	0.8 m Ns/m^2
Thermal Conductivity K_w	0.59 W/m℃
Weight of the Bare tube	0.719 kg/m for 19.05 O.D.

Table .4 Bare Tube Detail

Detail of Bare Tube Heat Exchanger	
Tube Length, L	5.426 meter
Tube outer diameter, d_o	19.05 mm
Tube internal diameter, d_{Bi}	16.104mm
Tube wall thickness, t_B	1.473 mm
Tube material	Austenitic stainless steel
Thermal conductivity of tube, k	$16 \frac{w}{m}$℃
Number of tube	N_{Bt} = 616 No's
Number of tube Passes	2
Number of Shell Passes	1
Total surface area of the Bare tube Bundle, $A_{Bo,Bundle}$	200.034 m^2

Outer surface area of a bare tube

$$A_{Bo} = \pi\, d_o\, L = \pi \times 0.01905 \times 5.426 = 0.32473\ m^2$$

Total area of bare tube Bundle $A_{Bo\,Bundle} = A_{Bo}$ x No's of tubes = 0.32473 x 616 = 200.034 m^2 (23)

F. Finned Tube Detail According To the Project Heat Exchanger: [1]

The code number of finned tubes is taken from HPT DATA BOOK, is HPT# 285035, Type2 for Smooth Bore
Where, 28 fins per inches = 1102 fins per meter,
5 root diameter in 8^{th} of an inches = 19.05 mm outer meter dia. of tubes
.035 wall thickness under fin in 1000^{th} of an inches = 0.889 mm wall thickness under fins condition

Table 5 Finned Tube Detail which we have considered

No. of fines	1102 fins per meter
Outer diameter of fines tubes	d_{Fo} = 19.05mm
Outer diameter of blank space	d'_{Fo} = 17.272mm
Inner diameter of finned tubes	d_{Fi} = 15.494mm
Wall thickness of finned tubes	t' = 0.889mm

ISSN: 2277-3754
ISO 9001:2008 Certified
International Journal of Engineering and Innovative Technology (IJEIT)
Volume 2, Issue 1, July 2012

Fin height	Z = 0.889mm
Fin thickness	t_f = 0.305mm
Length of one finned tube	L = 5.426 m
Weight of a finned tube HPT#285035	0.538 kg/m
Fin and tube material	Austenitic stainless steel
Thermal conductivity of fin material	16 $\frac{W}{m}$ ℃

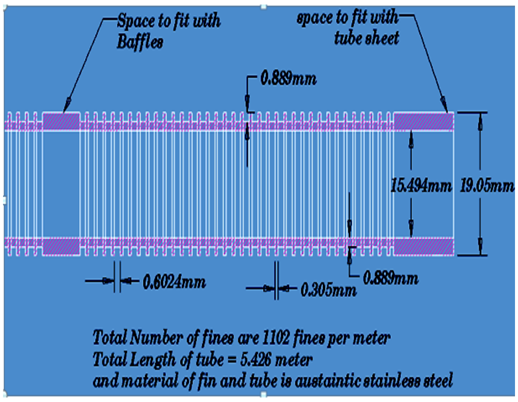

Fig 4. Finned tube Detail according to Project

If we apply this finned tube in this working Heat-Exchanger as it is, instead of bare tube (without change No. of tubes) then we will calculate these entire factors in this way:

Calculation of surface Area of Finned Tube: [15]

Area of the one Fin surface,

$$A_{fin} = 2\pi\left(\left(\frac{d_{Fo}}{2}\right)^2 - \left(\frac{d'_{Fo}}{2N}\right)^2\right) + 2\pi\left(\frac{d_{Fo}}{2}\right)t_f$$

[15] (24)

$$= 2\pi\left(\left(\frac{19.05}{2}\right)^2 - \left(\frac{17.272}{2}\right)^2\right) + 2\pi\left(\frac{19.05}{2}\right)0.305$$

= 119.696276 mm^2

Area of the one un-fin surface, $A_{unfin} = \pi\, d'_o\, f_s$,

Where, f_s = gap between two fins = 0.607 mm

d'_o = space diameter of fin tubes = 17.272mm

$A_{unfin} = \pi \times 17.272 \times 0.607$ = 32.93678 mm^2

(25)

Then the total area of a unit fin with blank space,

$A_{unit\,fin} = A_{fin} + A_{unfin}$ = 152.633 mm^2

[15] (26)

But 1102 fins are applied in one meter of tubes,

Then, the total surface area of one meter finned tube equal to:

= No. of fins per meter x $A_{unit\,fin}$

= 168201.63 mm^2 = 0.1682 m^2

But the total length one tube = 5.426 meter

Then the total surface Area of one Finned Tube = $A_{fo,tube}$

$A_{fo,tube}$ = 5.426x 0.168201 = 0.91266 m^2

(27)

Where the total numbers of tubes are using here are: = 616

Then the total surface area of the fin tube Bundle,

$A_{fo,Bundle}$ = No. of tubes x $A_{f,total}$

$A_{fo,Bundle}$ = 616 x 0.91266 m^2 = 562.1998 m^2

(28)

According to the HPT DATA BOOK the weight of the finned tube is = 0.538 kg/m [1]

Therefore the weight of a finned tube

M_{ft} = 0.538 x L = 0.538 x 5.426 = 2.9192 kg [1]

(29)

G. Calculation of Overall Heat Transfer Rate of Bare Tube Heat Exchanger:

From the equation No. 1, [16]

$$Q_B = U_B \times A_{Bo\,Bundle} \times MTD$$

Where,

Q_B = Heat transfer rate of bare tube heat exchanger, watt =?

U_B = 785.58 $\frac{W}{m^2}$ ℃

$A_{Bo\,Bundle}$ = 200.034 m^2

MTD = 13.1942 ℃

Now substitute these values in the equation of "Q_B",

Q_B = 2073372.34 watt = 2073.37 KW

The heat transfer rate of one Bare tube = Q_B /No. of tubes = Q_B /616 = 2073372.34/616 = 3365.86 watt/tube

H. Calculation Of Overall Heat Transfer Rate Of Finned Tube Heat Exchanger: [16]

$$Q_F = U_F \times A_{Fo\,Bundle} \times MTD$$

Where, Q_F = Heat transfer rate of finned tube heat exchanger in watt =?

U_F = 346.78 $\frac{W}{m^2}$ ℃

$A_{Fo\,Bundle}$ = 562.1998 m^2

ISSN: 2277-3754
ISO 9001:2008 Certified
International Journal of Engineering and Innovative Technology (IJEIT)
Volume 2, Issue 1, July 2012

MTD = 13.1942 ℃
Now substitute these values in the equation of "Q_F",
Q_F = 2572336.5 watt = 2572.34kw
And
The heat transfer rate of one Bare tube = Q_F /No. of tubes
= 2572336.57/616 = 4175.87 watt/tube (30)
If equate the heat transfer rate of Finned tube heat exchanger to the Bare tube heat heat-exchanger.
Then required No's of tubes = Q_B /heat transfer rate of one finned tube = 2073372.34/4175.87 = 496.51 ≅ 497 tubes
Therefore required only 497 tubes to achieve the same heat transfer rate as working bare tube heat exchanger
Then total No's of saved Finned tubes as compare to the Bare tubes Heat-Exchanger = 616 497= 119 tubes (31)
Hence there is saving 119 tubes to achieve same heat transfer rate

I Cost Analyses of Heat Exchanger

Cost of Bare tube Bundle of Heat Exchanger:

The total number of Bare tubes using = 616 No's
Current market price of Bare tubes = 280 Rs/kg
Weight of Bare tube = 0.719kg/meter for 19.05 outer diameter
Length of Bare tube = 5.426 meter
Weight of the one Bare tube = 0.719 x 5.426 = 3.901294 kg
Weight of the Bare tube Bundle = 3.901294 x no. of tubes = 3.901294 x 616 = 2403.197 kg
Total price of the Bare tube Bundle = 2403.197kg x 280 Rs/kg = 672895.16 Rs

Cost of finned tube Bundle of Shell and Tube Heat Exchanger:

The total number of finned tubes using = 497 No's
Current market price of finned tubes = 385 Rs/kg
Weight of finned tube = 0.538kg/meter for HPT#285035
Length of finned tube = 5.426 meter
Weight of the one finned tube = 0.538 x 5.426 = 2.919188 kg
Weight of the finned tube Bundle = 2.919188 x no. of tubes = 2.919188 x 497 = 1450.836 kg
Total price of the finned tube Bundle = 1450.836436 kg x 385 Rs/kg = 558572.0279 Rs

Comparative Cost Analysis & Result

Bare tube bundle price = 672895.16 Rs
Finned tube bundle price = 558572.0279 Rs
Saving cost only for tube Bundle = 672895.16 Rs − 558572.0279 Rs = 114323.13 Rs

Table 6.Comparison table of bare tube and finned tube heat Exchanger:

Performance Parameter	Bare tube heat Exchanger	Finned tube heat Exchanger
Number of tubes	616	497
Tube bundle weight	2403.197 kg	1450.84 kg
Market Price	672895.16 Rs	558572.03 Rs
Tube bundle diameter	D_{Bb} = 656.824	D_{Fb} = 595.94 mm
Shell Diameter	D_{Bs} = 750.8237 ≅ 800 mm	D_{Fs} = 687.94mm ≅ 730 mm
Tube side Pressure Drop	ΔP_{Bt} = 7.5 kPa	ΔP_{Ft} = 12.7 kPa
Shell side pressure Drop	ΔP_{Bs} = 2959.81 N/m^2	ΔP_{Fs} = 2743.58 N/m^2
tube-side fluid velocity	U_{Bt} = 0.722 m/s	U'_{Ft} = 0.965 m/s
Shell side fluid velocity	U_{Bz} = 0.626 m/s	U_{Fz} = 0.704 m/s

III. RESULTS AND DISCUSSIONS

The comparative results are summarized in table 3.6
This work has resulted in several things:
1. The finned tube heat exchanger is more economical than Conventional Bare tube Exchanger
2. The tube side pressure drop and fluid velocity is higher than the conventional bare tube exchanger, which prevent fouling inside the tubes,
3. The shell side pressure drop is some lesser but fluid velocity is higher than the conventional heat exchanger. Which safe the outer surface of tubes from fouling creation and fluid transfer time,
4. The shell diameter of finned tube Exchanger is lesser than Conventional bare tube heat exchanger, which saves sheet material and reduces the size of the shell, which helps to easily installation in the plant,
5. There using fewer tubes than conventional heat exchanger to achieve same heat transfer rate, which benefit to save cost as well as total weight of Heat exchanger, to easy transfer here and there.
6. There is small inner diameter and less No. of tubes than conventional Heat exchanger to achieve same heat transfer rate so there using less amount of water to transfer heat from hot fluid.

IV.CONCLUSION

From this paper it can be seen that the use of finned tube can contribute greatly in the development of compact shell and tube heat exchanger designs. The benefits of compact shell and tube heat exchanger designs is that they lead to cost saving either in original equipment and installation cost due to reduced size, or in increased production economics due to increase capacity. The augmentation may not only reduce the cost of the tubing, but also those of the heads, shell, baffles and tube sheets (smaller diameters, smaller wall thicknesses, fewer tube holes to drill, less alloy cladding material, etc). Even for conventional Heat exchangers, if the entire cost of the heat exchanger is included as it should be its total cost to

ISSN: 2277-3754
ISO 9001:2008 Certified
International Journal of Engineering and Innovative Technology (IJEIT)
Volume 2, Issue 1, July 2012

plant, a more compact, lighter weight enhanced shell-and-tube unit can greatly reduce the cost of shipping and installation

REFERENCES

[1] "Fine-Fin, Heat Exchanger Tubing, Engineering Data Book", HPT (HIGH PERFORMANCE TUBE, INC) 792 Chimney Rock Rd. Martinsville, NJ 08836.

[2] "HEI (Heat Exchanger Institute Incorporated) standards for Power Plant Heat Exchangers", Fourth Edition, Thermal Engineering international (USA) INC. 5701 South Eastern Avenue, Suite #300 Loss Angeles, CA 90040, p1-29

[3] Ray Sinnott and Gavin Towler, "Chemical Engineering design, Fifth Edition" (Coulson & Richardson's Chemical Engineering Series) CHAPTER 12. Heat-transfer Equipment.

[4] Daniel R. Lewin "Heat Exchanger design Lecture 7", Department of Chemical Engineering Technion, Haifa, Israel,

[5] WOLVERINE (1984) "Wolverine Tube Heat Transfer Data Book Low Fin Tubes", (Wolverine Division of UOP Inc.).

[6] WOLVERINE (1984) Chapter 2 "Design Considerations for Enhanced Heat Exchangers", (Wolverine Tube, Inc.). Engineering Data Book III

[7] G.V. Harshe Dy. (Director, NPTI (WR) NAGPUR) "Condenser and Circulating Water System" National Power Training Institute (Ministry of Power, Government of India NAGPUR-440022, India. March, 2004, p 9-10

[8] Polley, G.T., Terrenova, Jafari Nasr, M.R., "Potential Benefits of Heat Transfer Enhancement", 10th International Heat Transfer Conference, Brighton, UK, (1995).

[9] M.R. Jafari Nasr, and polley, G.T., "Part A: Derivation of charts for the Approximate Determination of the Area Requirements of Heat Exchangers using Plain and Low Finned Tube Bundles", CE and T, 23, 1, (2000), 46-54.

[10] Fraas, A.P., Ozisik, m.N., "Heat Exchanger Design", John Wiley and Sons, Inc., New York, London, Sydney, (1965).

[11] Jafari Nasr, M.R. and Polley, G.T., "Part B: Extension of Rapid Sizing Algorithm or Shell-and-Tube Heat Exchanger with Tube side pressure Drop Constraint and Multi-passes", CE and T, 23, 2, (2000), 141-150.

[12] ESDU (Engineering Science Data Unit), "Low-Finned Staggered Banks, Heat Transfer and Pressure loss for Turbulent Single Phase Cross Flow", ESDU Number 84016, (1984).

[13] M.R. Jafari Nasr and A.T. Zoghi "Full Analysis of Low Finned Tube Heat Exchangers" Research Institute of Petroleum Industry (RIPI), National Iranian Oil Company (NIOC) (2001-2003)

[14] Rene Hofmann, Friedrich Frasz, Karl Ponweiser "Experimental Analysis of Enhanced Heat transfer and Pressure-Drop of Serrated Finned-Tube Bundles with different Fin Geometries" Institute of Thermodynamics and Energy conversion Vienna University of Technology A-1060 Vienna, Getreid market 9/E302 Austria.

[15] NTPL Lectures, Module 3 "Heat Transfer from Extended Surfaces", 26-Dec-2011.

[16] "Energy Performance Assessment For Equipment & Utility Systems",Vol.4, a Guide book for National Certification Examination for Energy Managers and Energy Auditors Chapter 04 Heat Exchangers p 56-57,

Author's profile

Shiv Kumar Rathore is M.E. student at Mechanical Engg. Deptt. of Institute of Engineering & Technology, Devi Ahilya University Indore, India. He received his B.E. Degree in 2008 in Mechanical Engineering from the University Institute of Technology Berkatullah University Bhopal (M.P.), India. He is currently working on finned tube Exchanger design instead of Bare tube Conventional heat Exchanger.E-mail address: shiv.seenu@gmail.com

Mr. Ajeet Bergaley obtained his B.E. degree in Mechanical Engineering from T.I.T Bhopal (M.P.), India. He received his M. TECH. in" Computer Integrated Manufacturing "from S.G.S.I.TS., Indore, India. He is presently working as Assistant Professor in Mechanical Engineering Department, Institute of Engineering & Technology, Devi Ahilya University Indore, India. His areas of interest are fields related to CAD/CAM, CAE, CIM, Finite Element Analysis, Production Engineering and Manufacturing Sciences
E-mail address: ajeet_bergaley@yahoo.com