MODIFICATION AND DESIGN OF FINNED TUBE HEAT EXCHANGER

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Abstract : Finned tube heat exchangers are most widely used in chemical industries where tube side fluid is either liquid or gas and fin side fluid is gas. As gas have very low heat transfer coefficient fins are provided on gas side. The heat exchanger used in this paper is in sugar industry where it is required to modify the heat exchanger from single pass to multi-pass heat exchanger and change its application from heating of air to cooling of air. Existing heat exchanger have fin side fluid air while on tube side it is hot steam. Calculations have been made to modify the existing finned tube heat exchanger.

Keywords **- finned tube heat exchanger; heat transfer rate; L type fin; staggered arrangement.**

I. INTRODUCTION

Optimization of a heat exchanger depends upon the material saving. Hence, cost of material should be consider while designing any heat exchanger. Widening the temperature difference between surface of tube and fluid, increasing the surface area of heat transfer are some of the factors which are considered to increase the convection coefficient i.e. to increase the heat transfer rate in a heat exchanger.

To enhance the heat transfer rate between a surface and the fluid (preferably gases) extended surfaces are used either inside the tube or outside the tube. Extended surface may be in the form of longitudinal fins or transverse fins.

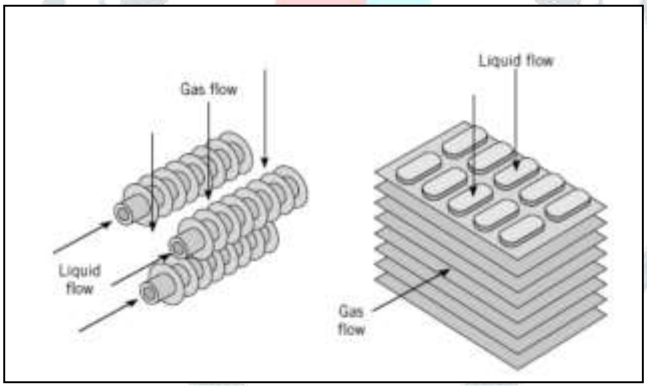

Figure 1 Types of finned tube heat exchanger

Fins are commonly used in extended surface exchangers. Conventional fin-tube exchangers often characterize the considerable difference between liquids' heat transfer coefficients. In a gas-to-liquid exchanger, the heat transfer coefficient on the liquid side is generally higher than that on the gas side.

The fin geometry has become as increasingly important factor in the design of a plate-and-fin heat exchanger. The high performance offset strip, wavy and louver fins provide quite high heat transfer coefficients for gases and two-phase applications. Enhanced surface geometries are widely used with liquids for cooling electronic equipment. The typical extended surfaces used for the plate-and-fin heat exchangers are: Plain Fin, Wavy Fin, Offset Strip Fin, Louvered Fin, Perforated Fin, etc

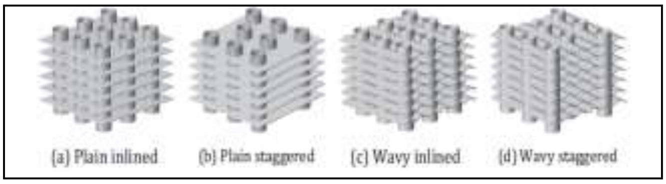

Figure 2 Types of fin arrangements

Based on the tube arrangement, these types of heat exchangers are divided in two different groups such as staggered and in-lined tube arrangement. Figure 2 shows some typical finned-tube heat exchanger designs especially for plain and wavy structure.

II. SPECIFICATION OF EXISTING HEAT EXCHANGER

Table 1: Specificaiton Of Existing Heat Exchanger

Description	Dimension	Material
Fin (L-type cross section)	0.45 mm (thickness) x 13 mm (height), 10 Fins per inch (FPI)	Alluminum
Tube	12.5 mm nominal diameter (21.34 mm actual)	Seamless stainless steel
Heating surface area	743.35 m^2	---

III. CALCULATIONS FOR EXISTING HEAT EXCHANGER

3.1 Heat Transfer Rate

In existing heat exchanger steam is used as the fluid on tube side to heat the air for drying the sugar in sugar industry. Calculating the heat transfer rate we get,

$\dot{Q}$ = ms Cs (Tis-Tos), kW (1)

m_s= mass flow rate of steam , kg/sec

Cs= Specific heat of steam, kJ/kg K

T_{is}= Inlet temperature of steam, ^{o}C/K

T_{os}= Outlet temperature of steam, ^{o}C/K

$\dot{Q}$ = 23 x 0.47 x (126-56) = 756.7 kW

TBALE 2 Data for calculation of existing heat exchanger

Temperature	Tube Side Flow (Steam)	Fin side flow (Air)
Inlet (^{o}C)	126	30
Outlet (^{o}C)	56	50
Specific Heat (kJ/kg K)	0.47	1.007
Mass Flow Rate (kg/sec)	23	37.63

3.2 Calculation for number of tubes for existing heat exchanger

Total Heat transfer Area = 743.35 m^2

Tube OD, d_o = 21.3 mm

Fin Height = 13 mm

Fin OD, D_o = 47.3 mm

Number of fins per inch = 10 FPI (fins per inch) or Fin spacing (s_f) = 2.54 mm

Fin thickness (t_f) = 0.45 mm

Type of fin = Circular wounded L-type cross section

Length of tube, L_t = 2040 mm

Effective Length of tube (finned tube), L =1940 mm

Length of heat exchanger tube sheet, W = 3100 mm

Calculation for Number of tubes in existing heat exchanger:

Total finned tube area per tube, $A_{f1} = \frac{\frac{\pi}{2}(D_o^2 - d_o^2 + 2D_o t_f)L_t}{s_f} = 2.19\ m^2\ per\ tube$

Total Bare tube area per tube $A_{b1} = \pi d_o \left\{L\left(1 - \frac{t_f}{s_f}\right)\right\} = 0.11 m^2 per\ tube$

Total heat transfer area = 743.35 m^2 = $\left(A_{f1} + A_{b1}\right) \times number\ of\ tubes$

Number of tubes in existing heat exchanger = 323 tubes

IV. DESIGN OF MODIFIED HEAT EXCHANGER

4.1 Heat transfer rate

To cool the water the modification made was by changing the fluid from steam to water.

Temperature	Tube Side Flow (Water)	Fin side flow (Air)
Inlet (°C)	8	50
Outlet (°C)	24	25
Specific Heat (kJ/kg K)	4.197	1.007
Mass Flow Rate (kg/sec)	14.11	37.63

TABLE 3 DATA FOR MODIFIED HEAT EXCHANGER

$Q = m_w C_w (T_{ow}-T_{iw})$, kW (2)

m_w= mass flow rate of steam , kg/sec

Cw= Specific heat of steam, kJ/kg K

T_{iw}= Inlet temperature of steam, oC/K

T_{ow}= Outlet temperature of steam, oC/K

$\dot{Q}$= 14.11 x 4.197 x (24-8) = 947.7 kW

So, we can achieve increase in heat transfer if we change the fluid in heat exchanger from steam to chilled water for required application

4.2 Air side calculations

1. Fin area per unit length of tube,

$$A_f = \frac{\pi}{4}\left(D_o{}^2 - d_o{}^2\right) \times 2 \times \text{No. of fins per one meter of tube length } (n_f) \quad (3)$$

$$= \frac{\pi}{4}(47.3^2 - 21.3^2) \times 2 \times \frac{10}{25.4} \times 1000$$

$$= 1.1 \frac{m^2}{\text{m of tube length}}$$

2. Bare Tube area, $\boldsymbol{A_b}$

$$A_b = \pi\, d_o \times 1 \text{ tube} - \pi\, d_o t_f n_f \quad (4)$$

$$= \pi \times 0.0213 \times 1 - \pi \times 0.0213 \times 0.45 \times \frac{10}{25.4} \times 1000$$

$$= 0.055 \frac{m^2}{\text{m of tube length}}$$

3. Projected Perimeter (Pe)

$$Pe = 2 \times (D_o - d_o) \times n_f + 2 \times (1 - t_f n_f) \quad (5)$$

$$= 2 \times (0.0413 - 0.0213) \times \frac{10}{25.4} \times 1000 + 2 \times \left(1 - 0.45 \times \frac{10}{25.4}\right)$$

$$= 22.12 \frac{m}{\text{m of tube length}}$$

4. Equivalent Diameter $\mathbf{d_e} = \frac{\mathbf{2\times(A_f+A_b)}}{\boldsymbol{\pi}\times \mathbf{P_e}}$ = 0.0261 m

5. Air Side Flow area , $\boldsymbol{a_s}$

$$a_s = (WL) - n_p \times d_o \times L - n_p t_f n_f L (D_o - d_o) \quad (6)$$

W(length of heat exchanger)=3100 mm , L (Effective length of tube) = 1940 mm

$a_s = 3.048\ m^2$

6. Air Side Mass velocity, $\mathbf{G_s}$

$$G_s = \frac{\dot{m}_a}{a_s} = 12.34 \text{ kg/s m}$$

7. Air side Reynolds number (Re)

$$Re = \frac{d_e\, G_s}{\mu} = 16516.61$$

μ = Viscosity of air at 50 oC = 1.95 x 10-5 kg/m.s

8. Air Side Prandtl Number (Pr)

$$Pr = \frac{\mu\, c_p}{k}, \quad (7)$$

k= Thermal conductivity of air = 0.02735 W/m ℃
c_p = Specific heat of air = 1007 J/kg K (At 50 oC)
Pr = 0.7179

9. Clean heat transfer coefficient, for 100 % fin efficiency

$$h_f = J_{ha} \frac{k}{d_e} Pr^{\frac{1}{3}} \; \frac{W}{m^2 K} \tag{8}$$

J_{ha} = Factor for heat transfer Coefficient = 0.0852072 Re0.7324
J_{ha} = 104.64
$\therefore h_f = 98.18 \frac{W}{m^2 K}$

10. Dirty Fin side Heat transfer coefficient (100% fin efficiency) ($\boldsymbol{h'_f}$)

$$\frac{1}{h'_f} = \frac{1}{h_f} + \frac{1}{h_{do}} \tag{9}$$

h_{do}=Dirty fin side heat transfer coefficient (Selected from Table no)

$$h'_f = 96.46 \frac{W}{m^2\, ℃}$$

11. Dirty fin side heat transfer coefficient (Efficiency ≠ 100 %)

$$h_{ft} = (\Omega \times A'_f + A'_o) \times \frac{h'_f}{A_i} \tag{10}$$

Where, A'_f = Total fin area = $n_t A_f L = 1.1 \times n_t L$
A'_b = Total bare tube area = $n_t A_b L = 0.055 \times n_t L$
$A_i = n_t \pi d_i L = 0.05378 \times n_t L$

Determining **Ω**:

$$\text{Function} = (r_e - r_b)\sqrt{\frac{h'_f}{k_m\, y_b}} \tag{11}$$

Where, km = Thermal conductivity of fin material (aluminum) = 205 W/m ℃
yb = $\frac{t_f}{2} = \frac{0.45}{2}$ = 0.225
r_e = radius of finned tube = 0.0473/2
r_b = radius of bare tube = 0.0213/2

$$Function = (0.02365 - 0.01065)\sqrt{\frac{98.18 \times 1000}{205 \times 0.225}} = 0.6$$

$$\frac{r_e}{r_b} = \frac{0.0473/2}{0.0213/2} = 2.2$$

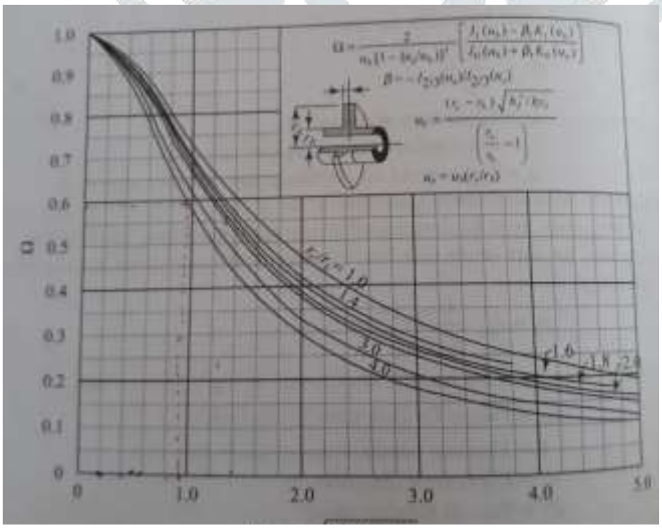

Figure 3 Efficiency of annular fins with constant thickness

From graph, fin efficiency (Ω) = 0.85

$\therefore h_{ft}$ = 1807.33 W/m^2°C

4.3 Tube side calculations

1. Inner tube area

Total number of tubes per pass/bank = 59

a_t = 59 x $\frac{\pi}{4}$ $d_i{}^2$ = 0.01358 m^2

d_i = 17.12 mm

2. Tube mass velocity,

   $G_t = \frac{\dot{m}_w}{a_t}$ = 1039.1 kg/s m

3. Tube velocity, $u_t = \frac{G_t}{\rho}$ = 1.039 $\frac{m}{s}$
4. Reylonds number, $Re_t = \frac{d_i G_t}{\mu_w}$ = 20734.33 μ_w at 8℃ $= 1.308 \times 10^{-3} \frac{kg\, m}{s^2}$
5. Pr = $\frac{\mu_w c_{pw}}{k_w}$ = 9.47 , k_w at 8℃ $= 0.575 \frac{W}{m\,K}$, c_{pw} at 8℃ $= 4.195 \frac{kJ}{kg\,K}$
6. According to Dittus – Bolter Equation, Internal heat transfer coefficient $\boldsymbol{h_i}$

   Nu = 0.023 $Re^{0.8}\, Pr^{0.33} = \frac{h_i d_i}{k_w}$ (12)

   h_i = 4607.22 $\frac{W}{m^2\,K}$
7. Overall Heat transfer Coefficient, ***U***

   $\frac{1}{U} = \frac{1}{h'_{ft}} + \frac{1}{h_i} + \frac{1}{h_{id}}$, (13)

   $h_{id} = fouling\ factor\ coefficient = 6000\ \frac{W}{m^2K}$

   U = 1090 $\frac{W}{m^2K}$

4.4 LMTD for cross flow

Logarithmic mean temperature difference is given by,

$$\Delta T_{lm} = \frac{(T_{hi} - T_{ci}) - (T_{ho} - T_{co})}{\ln\frac{T_{hi}-T_{ci}}{T_{ho}-T_{co}}} = 10.96\ {}^oC$$

For cross flow in any heat exchanger there must be considered a correction factor for Logarithmic mean temperature difference (LMTD)

F_t = Correction Factor = 0.92

$\Delta T_m = F_t \Delta T_{lm} = 10.09\ {}^oC$

So, Total internal tube heat transfer area required is,

$$A_{ireq} = \frac{\dot{Q}}{U \Delta T_m} = \frac{947.3 \times 10^3}{1090 \times 10.09} = 37.13\ m^2$$

Heat transfer area per bank = $n_p \pi d_i L = 6.156\ m^2$, Total number of tube passes required = 6, Total number of tubes = 59 x 6 = 354 tubes

V. PRESSURE DROP ON AIR AND TUBE SIDE

5.1 Air side pressure drop

Net Free Volume

$$V_{fv} = WLx' - n_p \times \frac{\pi}{4} d_o^2 L - n_p t_f n_f L \times (D_o^2 - d_o^2) \quad (14)$$

$x' = Distance\ between\ two\ banks, mm = 50 \sin 60^o = 43.30\ mm$

$V_{fv} = 0.2235\ m^3$

Volumetric Equivalent Diameter,

$$D_{ev} = 4 \times \frac{V_{fv}}{(A'_f + A'_b)\left(\frac{n_p}{n_t}\right)} = \frac{826 \times 4 \times 0.2235}{1850.81 \times 59} = 6.76 \times 10^{-3} m$$

Reylonds number for calculating pressure drop,

$$Re = \frac{D_{ev} G_s}{\mu_a} = 4291.73$$

Air side fan friction factor,

$$J_{fa} = 1.08558 \times Re^{-0.128025} = 0.3720$$

Pressure drop

$$\Delta p_s = \frac{J_{fa} G_s^2 L_p}{D_{ev} \rho_a} \left(\frac{D_{ev}}{S_t}\right)^{0.4} \left(\frac{S_l}{S_t}\right)^{0.6} \quad (15)$$

$L_p = Effective\ path\ length\ for\ pressure\ drop = x' \times number\ of\ pass$

$\rho_a = \frac{PM}{RT} = \frac{1 \times 29}{0.0821 \times 323} = 1.09\,\frac{kg}{m^3}$

$\Delta p_s = 2106.77\,\frac{N}{m^2}$

$= 214.21\ mm\ Water\ column < 350\ mm\ Water\ Column\ (allowable)$

5.2 Tube Side pressure drop

$$\Delta p_t = N_P \left(8J_f\left(\frac{L}{d_i}\right) + 2.5\right)\frac{\rho_w u_t^2}{2} \quad (16)$$

$N_p = number\ of\ passes = 14$

$J_f = tube\ side\ friction\ factor = 0.004,\ at\ Re = 20734.33$

$\Delta p_t = 46.27\ kPa$ < maximum allowable pressure drop tube side 68kPa

VI. ACKNOWLEDGMENT

Using Siemens NX 9.0 software a CAD model was prepared and meshing was done accordingly

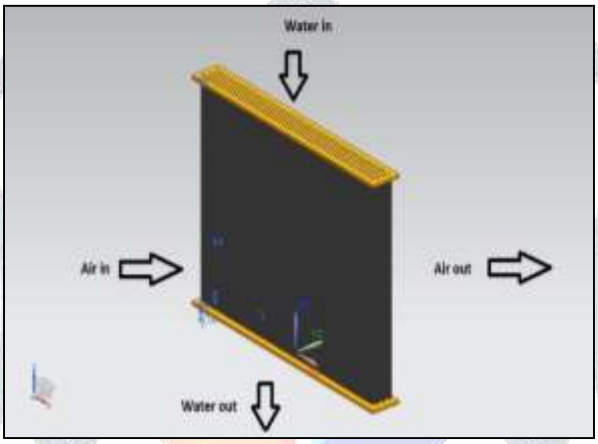

Figure 4 Existing single pass finned tube heat exchanger

A single column of tube is taken to do the analysis of 351 tube heat exchanger, to compare it with the theoretical calculations.

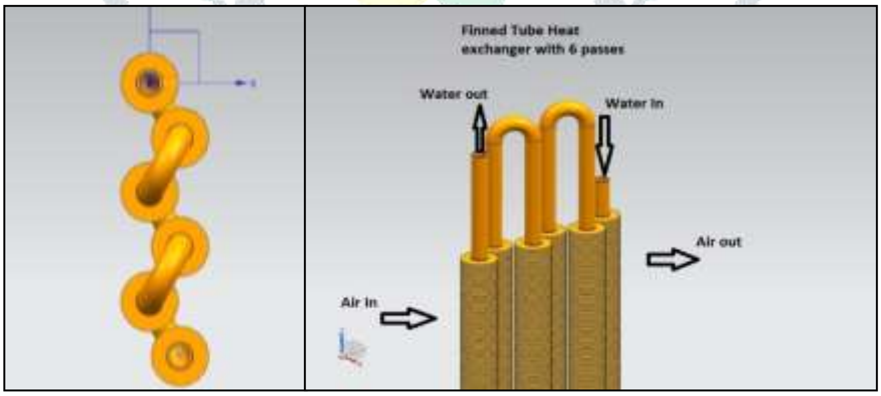

Figure 5 Modified Heat Exchanger with 6 passes

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