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Thermal Designing of Air Preheater Used in Continuous Caustic Fusion Plant

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Abstract

Thermal designing of an air preheater typically include the determination of heat transfer coefficient, Fin parameter, Colburn factor, fanning friction factor, heat transfer area, overall heat transfer coefficient, its type and size. The Continuous Caustic Fusion plant is a plant to produce export grade caustic soda Flakes from caustic soda solution. Design of heat exchangers has a greater role in thermal engineering field. An air preheater is a type of heat exchanger device designed to heat air before another process. Here the air preheater is a plate fin heat exchanger. Some problems are involved in the working of Air preheater used in the Continuous Caustic Fusion plant. Heat transfer is not properly taking place inside the Air Preheater. The outlet temperature of heat exchanger cannot be increased to a required value because of fouling produced inside the heat exchanger. So the air preheater is to be redesigned, to overcome this situation. In this paper finding the fouling factor and redesigning of plate fin heat exchanger are demonstrated.

Keywords: Thermal designing, Air preheater, Continuous Caustic Fusion Plant, Heat transfer coefficient, Over-all heat transfer coefficient, fouling factor.

Nomenclature

A	heat transfer surface area
A_o	minimum free-flow area
C^*	heat capacity rate ratio
C_{min}	smaller heat capacity rate
Cp	specific heat
D_h	hydraulic diameter
F	fouling factor

f	flow friction
G	mass velocity
h	heat transfer coefficient
h'	thickness
j	heat transfer factor or Colburn factor
k	thermal conductivity
L	flow length
l	fin length
l_c	fin conduction length
m	mass flow rate
M_f	fin parameter
NTU	number of transfer units
p_f	fin pitch
P_r	prandtl number
q	Actual heat transfer rate
q_{max}	max heat transfer rate
Re	Reynolds number
R_w	wall resistance
s	fin spacing
T_1	temperature of hot fluid inlet
T2	temperature of hot fluid outlet
t_1	temperature of cold fluid inlet
t2	temperature of cold fluid outlet
U	overall heat transfer coefficient
UA	overall thermal conductance
ε	effectiveness
μ	viscosity
δ	fin thickness
δ_w	plate thickness
η_f	fin efficiency
η_o	extended surface efficiency

Introduction

Heat exchangers have greater application in the field of oil industries, chemical industries, etc. In this case an air preheater which is provided in a Continuous Caustic Fusion plant is considered for the work. It seems that there are some problems involved in the working of an air preheater. That is, the heat transfer is not properly taking place inside the air preheater. Expected temperature is not generated at the air preheater of a Continuous Caustic Fusion Plant. This causes, decrease the efficiency of air preheater and increase in power consumption. One of the reasons for that, the fouling inside the air preheater. So that the work aims to find out the fouling factor and redesign the air preheater in economical manner by considering the fouling inside air preheater. This particular air preheater has air as the primary fluid and flue gas as the secondary fluid. Based on the present situation, it is decided that ε- NTU method can be applied for the design verification. Then the effect of fouling in the air preheater is determined and a design modification to the air preheater is suggested based on ε- NTU method.

Plate heat exchangers with fin type are widely used in aeronautical and cryogenic engineering fields. It is very important to correct the design and quality of heat exchangers is essential for proper functioning of such systems. The relations between heat transfer properties and design techniques are discussed in well-known text book "Fundamentals of Heat Exchanger Design" by Ramesh K Shah and Dusan P Seculic [1]. It explained an excellent introduction to fin geometry. Kays and London [2] detailed a different design data for plate fin heat exchangers fin surfaces. The variation of f and j with different Re for different surface configuration is given in graphical form and tabular form as well. London and Shah [3] explained performance of offset strip-fin non-dimensional geometrical parameters. Dimensionless parameters include fin thickness, fin pitch, Colburn factor, friction factor etc. Shah and London [4] provided laminar flow analytical results for rectangular offset ducts. Joshi and Webb [5] discussed the goodness factor comparison for different fin surfaces for plate fin surfaces.

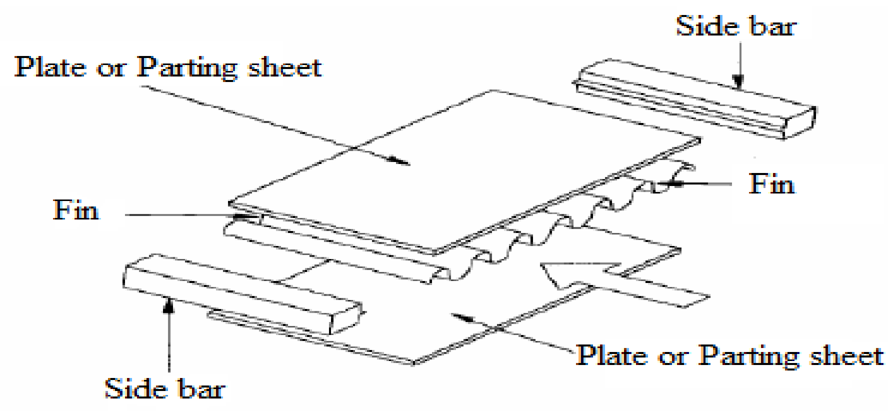

Fig. 1. Basic components of plate-fin heat exchanger [6]

Shah and Sekulic [6] present in detailed the design factors of two fluid operations. They have designed the heat exchanger with fundamental steps involved. They have presented ε-*NTU* method for cross flow plate fin heat exchangers. Manson [7] proposed correlation equations using a database of different geometries. Weiting [8] obtained *j* and *f* correlations for the laminar and turbulent flow regions. Nuntaphan et al. [9] discuss the effect of, spacing of fin on the air side performance at low Re for both staggered and inline arrangements. They find the correlations to predict the arrangements. Manglik and Bergles [10] carried an experimental research on offset rectangular fins. They find the effects of fins heat transfer and pressure drop for different offset rectangular fins.

ε – NTU Method of Air preheater Designing

In the ε -NTU method [1], the heat transfer rate from the hot fluid to the cold fluid in the exchanger is expressed as

$$q = \varepsilon\, C_{min}\, (T_1 - t_1) \quad (1)$$

The heat exchanger effectiveness sometimes referred to as the thermal efficiency. The effectiveness ε is non-dimensional parameter, and it can be shown that in general it is dependent on NTU, C*, and the flow arrangement for a direct-transfer type heat exchanger. Effectiveness is defined as the measure of thermal performance of a heat exchanger [1]. It is referred to as, for a given heat exchanger of any flow arrangement as a ratio of the actual heat transfer rate from the hot fluid to the cold fluid to the maximum possible heat transfer rate.ie,

$$\boldsymbol{\varepsilon} = \frac{q}{q_{max}} \quad (2)$$

NTU is defined as a ratio of the overall thermal conductance to the smaller heat capacity rate.

$$NTU = \frac{UA}{C_{min}} \quad (3)$$

NTU indicate the non-dimensional heat transfer size and therefore it is a design parameter in the plate fin heat exchanger. NTU provides a compound measure of the heat exchanger size through the product of heat transfer surface area A and the overall heat transfer coefficient U [1].The design method for heat exchangers developed is based on the ε -NTU theory. This method allows relatively a straight solution of the corresponding design problems. Such design theory is used in many applications. For the designing of plate fin heat exchangers, use the ε-NTU method. Because it is the most common method used in industries. Here explains a step-by-step method for the designing of a plate fin cross flow heat exchangers. The basic steps involved in the analysis of heat exchanger explains the following parameters: surface geometrical properties, fluid physical properties, Reynolds numbers, heat transfer coefficients and fin efficiencies, wall thermal resistance and overall thermal conductance, heat transfer rate, outlet temperatures.

1. Find out the surface geometrical properties on each fluid side .It includes the minimum free-flow area A_o, heat transfer surface area A(both cold and hot), flow length L, hydraulic diameter D_h, fin geometry(s, δ, h',l) for fin efficiency calculations.

$$s = p_f - \delta \quad (4)$$

$$h' = \delta_w - \delta \quad (5)$$

$$d_h = \frac{2(s-\delta)h'l}{h'l + sl + h'\delta} \quad (6)$$

$$A_o = \frac{dh * 72}{4} \quad (7)$$

$$A = (2h'l) + (2sl) + (2h'\delta) \quad (8)$$

2. Determine the fluid mean temperature and fluid thermo physical properties on each fluid side. The fluid properties needed for the designing are μ, C_p, k, and P_r.

3. Compute the mass velocity using mass flow rate and Reynolds number using fluid properties for both cold and hot fluid

$$G = \frac{m}{Ao} \quad (9)$$

$$Re = \frac{G \times Dh}{\mu} \quad (10)$$

Subsequently, compute heat transfer and flow friction characteristics of heat transfer surfaces on each fluid side of the exchanger.

$$j = 0.6522(Re)^{-0.5403}\left(\frac{s}{h'}\right)^{-0.1541}\left(\frac{\delta}{l}\right)^{0.1499}\left(\frac{\delta}{s}\right)^{-0.0678}\left[1 + \left(0.00005269(Re)^{1.34}\left(\frac{s}{h'}\right)^{0.504}\left(\frac{\delta}{l}\right)^{0.456}\left(\frac{\delta}{s}\right)^{-1.055}\right]^{0.1} \quad (11)$$

$$f = 9.6243(Re)^{-0.7422}\left(\frac{s}{h'}\right)^{-0.1856}\left(\frac{\delta}{l}\right)^{0.3053}\left(\frac{\delta}{s}\right)^{-0.2653}\left[1 + \left(0.00000007669(Re)^{4.429}\left(\frac{s}{h'}\right)^{0.920}\left(\frac{\delta}{l}\right)^{3.767}\left(\frac{\delta}{s}\right)^{0.236}\right]^{0.1} \quad (12)$$

4. From j, compute the heat transfer coefficients for both fluid streams from the following equation.

$$h = \frac{j \times G \times C_p}{P_r^{2/3}} \quad (13)$$

5. Using heat transfer coefficient, fin geometry parameters and thermal conductivity of material determine the fin parameter.

$$M_f = \left(\frac{2h}{K_f \times \delta}\left(1 + \frac{\delta}{l}\right)\right)^{1/2} \quad (14)$$

Also find out the fin efficiency η_f and the extended surface efficiency η_o from fin parameter for both cold and hot fluid.

$$\eta_f = \frac{\tanh(M_f l_c)}{M_f l_c} \quad (15)$$

$$\eta_o = 1 - (1 - \eta_f) \times 0.785 \quad (16)$$

6. Compute the overall thermal conductance UA

$$\frac{1}{UA} = \left(\frac{1}{\eta_o hA}\right)_{\text{hot fluid}} + R_w + \left(\frac{1}{\eta_o hA}\right)_{\text{cold fluid}} \quad (17)$$

7. Finally calculate the fouling factor

$$f = \left(\frac{1}{UA}\right)_{\text{ideal}} - \left(\frac{1}{UA}\right)_{\text{actual}} \quad (18)$$

Design Validation for Air preheater

Table 1 show that the design calculation for the air preheater in a continuous caustic fusion plant using ε-NTU method. In ideal case calculations, use the ideal conditions in the continuous caustic fusion plant. In air preheater, the hot fluid is flue gas and cold fluid is air. The ideal case calculation implies that a perfectly designed heat exchanger operate properly in a plant. In actual case calculations, the only difference from the ideal case calculation is that the temperature variation due to fouling effect. The outlet temperature of air preheater is reduced to 230°c to 185°c. Due to this temperature change, the heat transfer coefficient, fin parameters, efficiency of fins are reduced. Also fouling is increased to 1.288 k/w. In re-design calculations, for the same fouling effect reduced the fin thickness up to 0.185mm. So that the fins pacing increased to1.61 mm. Also the heat transfer coefficient, fin parameter, efficiency is increased from actual case. In this conditions got the required temperature range for the air preheater.

Table 1. Design calculation of air preheater in a continuous caustic fusion plant

No:	Parameter	Ideal Case		Actual Case		Re-design Case	
		Hot Fluid	Cold Fluid	Hot Fluid	Cold Fluid	Hot Fluid	Cold Fluid
1	δ (mm)	5	5	5	5	5	5
2	δ_w(mm)	0.2	0.2	0.2	0.2	0.185	0.185
3	p_f(mm)	1.795	1.795	1.795	1.795	1.795	1.795
4	l (mm)	6	6	6	6	6	6
5	h' (mm)	9.3	9.3	9.3	9.3	9.315	9.315
6	l_c (mm)	0.0045	0.0045	0.0045	0.0045	0.0045	0.0045
7	R_w(k/w)	4.7E-7	4.7E-7	4.7E-7	4.7E-7	4.7E-7	4.7E-7
8	m (kg/s)	2.249	2.14	2.249	1.27	1.27	1.27
9	C_p(kj/kg)	1.151	1.014	1.151	1.0106	1.151	1.03
10	T_i (^{o}c)	447.4	25	447.4	25	447.4	25
11	T_o (^{o}c)	271	230	271	185	271	225
12	μ (kg/ms)	3E-05	2E-05	3E-05	2.2E-05	2.6E-05	2.67E-05
13	s (mm)	1.595	1.595	1.595	1.595	1.61	1.61
14	K_f(w/mk)	18	18	18	18	18	18
No:	Parameter	Ideal Case		Actual Case		Re-design Case	
		Hot Fluid	Cold Fluid	Hot Fluid	Cold Fluid	Hot Fluid	Cold Fluid
15	P_r	0.731	0.688	0.731	0.692	0.731	0.68
16	A (mm^2)	0.0001	0.0001	0.0001	0.0001	0.000135	0.000135
17	G(Kg/m^{2}s)	54.089	51.46	54.089	30.54	53.51	30.22
18	D_h(mm^2)	0.0023	0.0023	0.0023	0.00231	0.00237	0.002368
19	Re	4754	5201	4821	3281	4821	2680
20	j	0.0071	0.0068	0.007	0.00835	0.00703	0.009124
21	f	0.0285	0.0277	0.0285	0.03182	0.4106	0.0328
22	h (W/m^{2}k)	543.88	456.16	546	333	533.4	367.1
23	Mf	558.77	511.73	560.2	437.4	574.7	476.7
24	F(K/W)	-		1.288		1.288	

Effect of Fouling on the Performance of the Air Preheater and the Design Modification

The outlet temperature of the air preheater cannot be increased to the required value because of the fouling produced inside the air preheater. Therefore, it is necessary to determine the effect of fouling inside the air preheater. By using ε-NTU method, the fouling factor is 1.288K/W. This fouling factor is greatly affected by the performance of air preheater. The fouling will be increases, the temperature will be reduce.

Figure 2 shows that the relation between heat transfer coefficient and temperature. Heat transfer coefficient is nearly linear with temperature. That is when temperature increases the heat transfer coefficient also increases. But the fouling reduces the temperature and heat transfer coefficient. Figure 3 shows that the relation between overall efficiency and fouling factor. It shows that, the efficiency of plate fin heat exchanger reduces with respect to the increase of fouling factor.

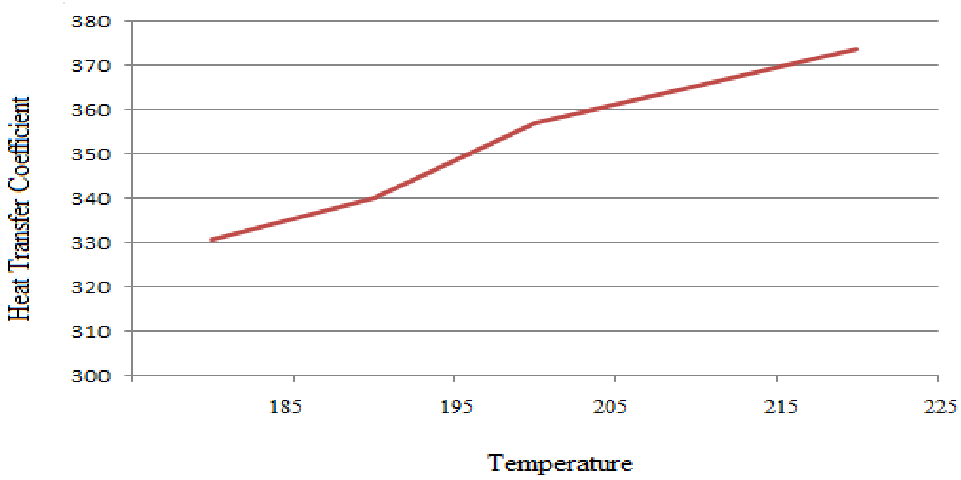

Fig. 2. Heat transfer coefficient vs Temperature

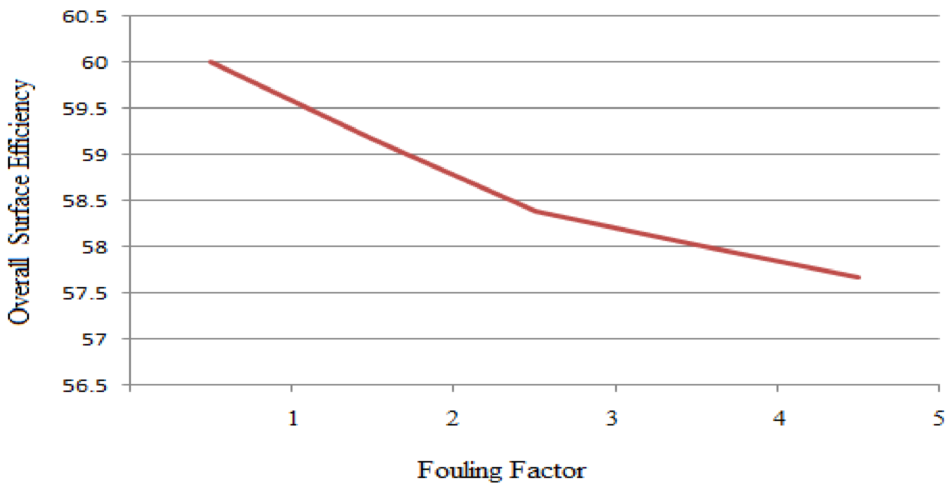

Fig. 3. Overall efficiency vs Fouling Factor

The maximum value of fouling factor is 4.239(k/W) at 225°c.Therefore corresponding to the maximum value of the fouling factor generated, the air preheater needs to be redesigned in order to overcome this situation. This can be done by using the fin characteristics. Where the main fin character is fin thickness (δ). In order to compensate this situation, the area should be change. The maximum value of fin thickness required for the fin is 0.185 mm. That means the fin with this much amount of fin thickness is capable of with desired effects under such circumstances. In order to considering the easiness of removal of the fouling, the thickness value of the fin can be approximated to 0.185 mm.

From redesign calculation, it is found that a better design with fin thickness of 0.185mm will be able to overcome the fouling effect. Also the fin spacing will be increases to 0.015 mm. The cleaning of the air preheater will be better if the modified design is applying. The fin thickness will reduce to 0.015mm will be able to working at the same fouling effect.

Conclusion

The outlet temperature of heat exchanger can be increased to a required value, by reducing the fin thickness. At the same fouling effect, the value of fin thickness reduced to 0.015 mm. that is the value of δ become 0.185mm. So the redesigned value can overcome the situation.

References

1. Ramesh K Shah and Dusan P Seculic, "Fundamentals of Heat Exchanger Design"
2. W. M. Kays and A. L. London, "Compact heat exchangers," 2nd edition, New York: McGraw-Hill, 1963, pp. 605-645.
A. L. London, R. K. Shah, "Offset rectangular plate fin surfaces – heat transfer and flow friction characteristics," ASME J. Eng. Power 90 (Series A) pp. 218–228, 1968.
3. R. K. Shah, A. L. London, "Laminar Forced Convection in Ducts," Supplement I to Advances in Heat Transfer, Academic Press, New York, 1978.
4. H. M. Joshi and R. L. Webb, "Heat transfer and friction in the offset strip-fin heat exchanger," International Journal of Heat Mass Transfer, vol.30, no. 1, pp. 69-84, 1987.
5. R. K. Shah and D. P. Sekulic, "Fundamentals of heat exchanger design," 3rd edition, John Wiley & Sons, New York, 1998.
6. S. V. Manson, "Correlations of heat transfer data and of friction data for interrupted plate fins staggered in successive rows," NACA Tech. Note 2237, National Advisory Committee for Aeronautics, Washington, DC, 1950.
A. R. Wieting, "Empirical correlations for heat transfer and flow friction characteristics of rectangular offset-fin plate-fin heat exchangers," ASME, Int. J. Heat Transfer, vol. 97, pp. 480-490, 1975.
A. Nuntaphan, T. Kiatsiriroat and C. Wang, "Air side performance at low Reynolds number of cross flow heat exchanger using crimped spiral fins," International Communications in Heat and Mass Transfer, vol. 32, pp. 151-165, 2005
7. R. M. Manglik and A. E. Bergles, "Heat transfer and pressure drop correlations for the rectangular offset-strip-fin compact heat exchanger," Experimental Thermal and Fluid Science, vol. 10, no. 2, pp. 171-180, 1995.