Design of Air Pre-Heater to Improve the Efficiency of Boiler in TCC Plant

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Abstract: *The Travancore Cochin Chemicals Limited (TCC) is a Government of Kerala undertaking company. Major product of this company is caustic soda flakes. The company consist large capacity boilers. A Boiler or Steam Generator is a device used to generate steam by applying heat energy to water. Boiler is an integral part of a thermal power plant. Also they uses in industries where steam is required. For the production of large scale steam industries mainly uses two types of boilers: Fire tube Boiler and Water tube Boiler. In Travancore Cochin Chemicals Ltd (TCC) they use Fire Tube Boiler. In TCC, there require large amount of steam for the conversion of caustic soda lye into caustic soda flakes. In the plant boilers works continuously in 24 hours. So large scale of steam generates per day. The fuel used for the working of boilers is Furnace oil. So the requirement of furnace oil is very large. For the production of steam there uses 4 fire tube boilers of capacity 8 ton. For the production 12 ton of steam there requires 1ton of furnace oil. 4 ton of steam produces per hour. So per day TCC produces 96 ton of steam. For the production of this amount of steam per day TCC requires 8 ton of furnace oil. Thus the company has to remit more investment on it. In every boiler plant, a large amount of energy drops through flue gases. Thus the proper extraction of heat energy from flue gas can increase the boiler efficiency considerably. Air preheater can extract heat from the flue gases. So it is possible to minimize the use of furnace oil by installing this device in the plant.*

Keywords: Fire tube Boiler, Water tube Boiler, Air Pre-Heater.

I. INTRODUCTION

A boiler is an enclosed vessel that provides a means for combustion and transfers heat to water until it becomes hot water or steam. The hot water or steam under pressure is then usable for transferring the heat to a process. Many manufacturing processes require steam for heating. Steam is also required for generation of electricity. Steam is generated by heating water in a closed vessel. The equipment where fire or hot gases heat the water in a confined space and generates steam is known as steam boiler. The place where combustion takes place is called as furnace. The flue gases either flow through the tubes or outside the tubes wherein water is on the other side of the tubes. If the flue gases are flowing through the tubes the boiler is known as smoke tube boiler and if water is flowing through the tubes then it is called as water-tube boiler.

1.1 Fire tube Boiler

A fire-tube boiler is a type of boiler in which hot gases from a fire pass through one or (many) more tubes running through a sealed container of water. The heat of the gases is transferred through the walls of the tubes by thermal conduction, heating the water and ultimately creating steam. The fire-tube boiler developed as the third of the four major historical types of boilers: low-pressure tank or "haystack" boilers, flued boilers with one or two large flues, fire-tube boilers with many small tubes, and high-pressure water-tube boilers. Their advantage over flued boilers with a single large flue is that the many small tubes offer far greater heating surface area for the same overall boiler volume. The general construction is as a tank of water penetrated by tubes that carry the hot flue gases from the fire. The tank is usually cylindrical for the most part being the strongest practical shape for a pressurized container and this cylindrical tank may be either horizontal or vertical. This type of boiler was used on virtually all steam locomotives in the horizontal " locomotive" form.

1.2 Water tube Boiler

A water tube boiler is a type of boiler in which water circulates in tubes heated externally by the fire. Fuel is burned inside the furnace, creating hot gas which heats water in the steam-generating tubes. In smaller boilers, additional generating tubes are separate in the furnace, while larger utility boilers rely on the water-filled tubes that make up the walls of the furnace to generate steam. The heated water then rises into the steam drum. Here, saturated steam is drawn off the top of the drum. In some services, the steam will reenter the furnace through a super heater to become superheated. Superheated steam is defined as steam that is heated above the boiling point at a given pressure. Superheated steam is a dry gas and therefore used to drive turbines, since water droplets can severely damage turbine blades. Cool water at the bottom of the steam drum returns to the feed water drum via large-bore 'down comer tubes', where it pre-heats the feed water supply. (In large utility boilers, the feed water is supplied to the steam drum and the down comers supply water to the bottom of the water walls).

To increase economy of the boiler, exhaust gases are also used to pre-heat the air blown into the furnace and warm the feed water supply. Such water tube boilers in thermal power stations are also called steam generating units.

1.3 Air Pre-Heater

Air pre-heaters are provided in boilers to preheat the combustion air. There are two main types: recuperative and regenerative air heaters. Tubular or recuperative air pre-heaters are provided in boilers of medium and small range of steam generation. This type of air pre-heater becomes very large in size if they have to be used in very high capacity boilers like 600 tons/hr of steam production and above. In these cases regenerative air pre-heaters are used. The arrangement of all these air pre-heaters differs with the design and, in large, the way they are combined for very high capacity boilers. Regenerative air per-heaters are compact and can have a stationary or rotating hood. A combination of tubular and regenerative type of air pre-heaters is used in very high capacity boilers. The tubular being used for primary air heating and the regenerative used for the secondary air heating. In case the boiler designers do not want to go for a combination of tubular and regenerative air pre-heater, then they have a choice of tri-sector regenerative air heater. Normally the ambient air is heated to about 300 to 350 degree centigrade. This results in a flue gas temperature drop of around 230 to 250 degree centigrade. So for each degree pick up in air temperature, roughly 0.8 degree drop in flue gas temperature is achieved. Steam coil air pre-heaters are another type. These are used only during start up of the boiler to prevent low temperature corrosion. This air heater does not contribute to improving the efficiency of boilers, but are provided to improve availability. It is seen that during start up the chances of low temperature corrosion is high, and hence the need to provide the steam coil air heaters is evident. Both economizer and air pre-heaters are called heat recovery systems in a boiler. Were it not for these heat recovery systems, present day boilers would be operating at much lower efficiency levels.

II. DATA COLLECTED

Boiler type	: Fire tube boiler
Equipment used for	: Production of NaOH flakes
Fuel used	: Furnace oil
Density of fuel	: 0.97 to 1.07 g/cc
Calorific value	: 9800 to 10100 kcal/kg
Specific heat capacity of fuel	: 2.09 KJ/KgK
Specific heat capacity of flue gas	: 1.3 KJ/KgK
Boiler operation time	: 8760 Hrs/year
Flue gas temperature	: 260 to 270 oC
Rate of steam production	: 96x3 ton/day
Flue gas velocity	: 8 to 10 m/s
Boiler operating pressure	: 2 to 5 Kgf/cm^2

III DESIGN CALCULATION

Energy absorbed by air = Energy loss by flue gas; assuming no heat loss by radiation.
An air pre-heater is generally treated as shell and tube heat exchanger. Does not take into account bypass and leakage streams. Restricted to a fixed baffle cut (25%)

$$q = m_h c_{p,h}(T_{h,i}-T_{h,o})$$

Where :

Th,i : Temperature of flue gas in = 657.35 K
Tc,i = Temperature of water in the pre-heater = 305 K
Tc,o=Temperature of air out of the pre-heater =348 K(assumption)
mf: Mass flow rate of Flue gas = 47.04 Kg/s
ma: Mass flow rate of air = 15.732 Kg/s
Cp_f: Specific Heat Capacity of Flue gas = 1.3 KJ/KgK
Cp_a: specific heat capacity of air = 1.005 KJ/KgK
Th,o: Temperature of Flue gas out = 644.039 K (Calculated)
Therefore q= 679.77195 KJ/s

$$q = UA\Delta Tlm$$

Where,

$$\Delta Tm = \frac{(\Delta T1-\Delta T2)}{\ln(\Delta T1/\Delta T2)} = \frac{(\Delta T2-\Delta T1)}{\ln(\Delta T2/\Delta T1)}$$

q : Energy absorbed
A: Surface area
U: Overall heat transfer coefficient(30-100 W/m2 °C) (Coulson page no.637)
ΔT1 = Th1 - Th2 = Thi – Tco
ΔT2 = Th2 - Tc2 = Tho – Tci
T h,i = Temperature of flue gas in the pre-heater = 657.35 K
Th,o = Temperature of flue gas out of the pre-heater=644.039 K
Tc,I = Temperature of air into the pre-heater = 305 K
Tc,o = Temperature of air out of the pre-heater = 348 K
ΔT1 = 309.35 °C
ΔT2 = 339.039 °C
ΔTlm= 323.967 °C

Assumptions:

1. Negligible heat loss to the surroundings.
2. Negligible kinetic and potential energy changes.
3. Constant properties.
4. Negligible tube wall thermal resistance and fouling factors.
5. Fully developed conditions for the water and flue gas (incropera et al. 2011)

The usual practice in the design of shell and tube heat exchanger is to estimate the 'True temperature difference' from the logarithmic mean temperature(LMTD) by applying a correction factor to allow for the departure from the counter current flow.

$\Delta Tm = Ft \Delta Tlm$

Where,

Ft : The temperature correction factor

The correction factor is a function of the shell and tube fluid temperature and the number of tube and shell passes. It is normally corrected as a function of two dimensionless temperature ratios.

R = (T1 - T2) / (t2 - t1) and S = (t2 - t1) / (T1 - t1)

R is equal to the shell-side flow-rate times the fluid mean specific; divided by the tube-side fluid flow-rite times the tube-side fluid specific heat.

S is measure of the temperature efficiency of the exchanger .

ΔTlm	: Logarithmic mean temperature difference
T1	: Hot fluid temperature at inlet = 657.35 K
T2	: Hot fluid temperature at outlet = 644.039 K
t1	: Cold fluid temperature at inlet = 305 K
t2	: Cold fluid temperature at outlet = 348 K
R	= 0.30955
S	= 0.12203

LMTD correction factor from the graph

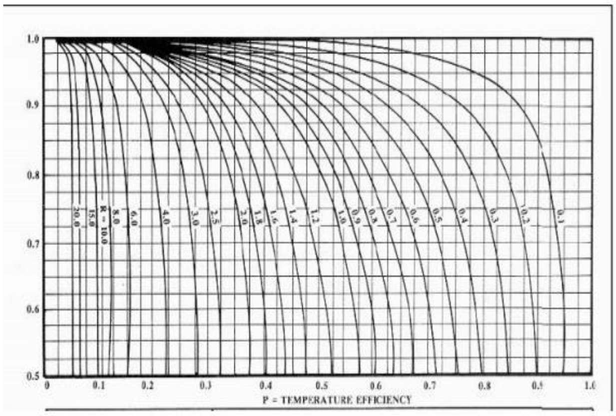

Figure 5.1 LMTD Correction factor

Ft = 0.98

ΔTm = 0.98 x 282.5874 = 276.9356 ^{o}C

The following assumptions are made in the derivation of the temperature correction factor Ft, in addition to those made for the calculation of the log mean temperature difference:

- Equal heat transfer areas in each pass
- A constant overall heat-transfer coefficient in each pass
- The temperature of the shell-side fluid in any pass is constant across any cross section.
- There is no leakage of fluid between shell passes

3.1. Fouling factors

Most process and service fluids will foul the heat-transfer surfaces in an exchanger to a greater or lesser extent. The deposited material will normally have a relatively low thermal conductivity and will reduce the overall coefficient. It is therefore necessary to oversize an exchanger to allow for the reduction in performance during operation. Fouling factors are usually quoted as heat-transfer resistances, rather than coefficients. They are difficult to predict and are usually based on past experience. The Overall Heat Transfer coefficient is determined and is found to be in the range of (30- 100 W/m2oC) when the hot fluid is Flue gases and the cold fluid is air. This range of Overall heat transfer coefficient includes fouling factors. An average value of the mean Overall Heat Transfer coefficient is taken: 65 W/m2 ^{o}C

3.2. Area

$q = UA\Delta Tm$

Area A= 6115.2 / (0.065 x 276.9356) = 339.7179 m2

3.3. Tube Length

Shell-diameter-to-tube-length ratio should be within limits of 1/5 to 1/15 (Subbaraon.d; Kakaç 1991). In his article Design and Rating of Shell and Tube Heat Exchangers John E. Edwards (2008) mentions that the preferred tube length to shell diameter ratio is in the range 5 to 10. Normally the shell diameter is taken to be within the range 150mm to 1067mm (Coulson). For this design of the heat exchanger the initial shell diameter is taken to be 1067 mm and the ratio of L: Ds are taken to be 10.

Tube length = (10 x diameter of shell) = 10 x 1.067 = 10.67 m

3.4. Area of one tube

Assumption: Tube outside diameter = 20 mm

Wall thickness = 1.6 mm (Coulson page no. 645)

Inner diameter = 16 mm = 0.016 m

Area = pie x Do x L

Area = pie x 0.02 x 10.67 = 0.67041587 m^2

3.5. Number of tube

n = A/(pie x Do x L)

= 32.6072 / (pie x 0.02 x 10.67)

= 32.6072 / 0.670415

= 48.6373

= 50

3.6. Bundle diameter

$D_b = d_o (N_t / K_1)^{1/n_1}$

Where ,

N_t = Number of tubes,

D_b = Bundle diameter, mm.

D_o = Tube outside diameter, mm

For 1 shell two pass Ki = 0.156 and n =2.291 (Coulson, 2011 page no: 649)
Db = 0.02 x (50/0.156)^(1/2.291)
= 0.2482 m
= 0.25 m

3.7. Tube arrangement and Tube pitch

Tube pitch, Tp = 1.25 Do = 1.25 x 0.02 = 0.025 m

3.8. Tube cross section area
Total cross section area inside tube = πDi2 /4
= (pie x 0.0162)/4 = 0.00020106 m²

3.9. Tube per pass

$$\text{Tube per pass} = \frac{\text{Number of tubes}}{\text{Total number of pass}}$$

= 50/2 = 25

3.10. Total flow area

Total flow rate = Number of tubes per pass × Total area inside tube

= 25x 0.00020106 = 0.0051526 m²

3.11. Air mass velocity

$$\text{Air mass velocity} = \frac{\text{Mass flow rate}}{\text{Total flow area}}$$

= 15.732/ 0.0051526 = 3053.21585 Kg/sm²

3.12.Air linear velocity

$$\text{Air linear velocity} = \frac{\text{Air mass velocity}}{\text{Density of air}}$$

= 3053.21585 / 1.013 = 3014.03341 m/s

3.13. Tube side coefficient

$$\frac{h_i d_i}{k_f} = j_h Re Pr^{0.33} \left[\frac{\mu}{\mu_w}\right]^{0.14}$$

Thermal conductivity of air, Kf = 0.024 W/mK
Viscosity of air = 1.789 x 10-5 Ns/m²
Where,
hi: The tube side coefficient
di: inside diameter
Re: Reynold's number
Pr: Prandlt Number
µ: Viscosity of air

3.14. Reynold's Number

$$Re = \frac{P\,u\,d_i}{\rho}$$

= (1.013x 3014.03341 x0.016)/ 1.789 x 10^{-5}
= 2730656.988
Where:
ρ: density of air
u: air linear velocity
di: inner diameter of tube
µ: Viscosity of air

3.15. Prandlt number

$$Pr = \frac{C_p\,\mu}{K_f}$$

Where;
Cp: The specific heat of air (1.005 KJ/ kg °C)
Kf: Thermal conductivity of air
Pr = (1.005x 103 x 1.789 x 10^{-5})/0.024
=0.749143

$$\text{Viscosity correction} \left[\frac{\mu}{\mu_w}\right]^{0.14} \text{Is neglected}$$

The L/ Di ratio= 10.67 / 0.016 = 666.875
Jh is read from figure 12.23 from Richardson and Coulson Volume 6 and is read to be 0.0025

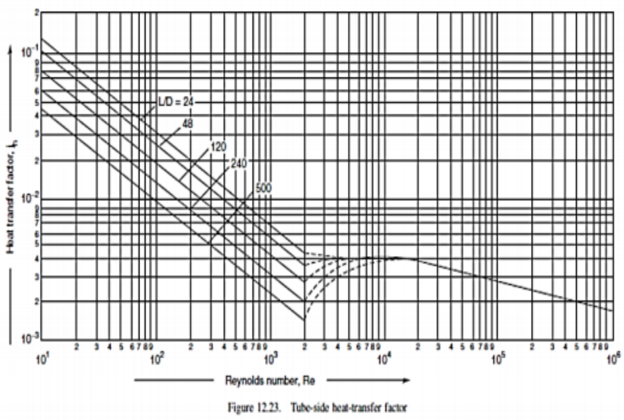

Figure.2 Tube side heat transfer factor

hi = jhRePr0.33kf / di

= 0.0025x 2730656.988 x0.749143 0.33 x 0.024 / 0.016
= 9309.0374 W/m2 °C

3.16. Shell diameter

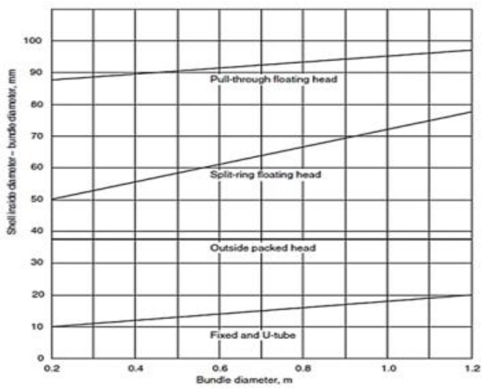

Figure.3 Shell bundle clearance

Given that the preheater is a u-tube with a shell bundle diameter of 0.25 m, the bundle diametrical clearance is read on the Figure 12.10 of Richardson and Coulson and is found to be 18 mm.

Ds = 0.25 + 0.018
= 0.268 m
An average value for shell diameter Ds was hence taken to be 0.27 m for square pitch.

3.17. Baffle spacing

= Ds/5 = 0.27/5
= 0.054 m

3.18. Cross-flow area

The cross flow area is calculated as follows:

$$\text{Cross flow area} = \frac{\text{Tube pitch} - \text{outer diameter}}{\text{Tube pitch}} \times \text{shell diameter} \times \text{baffle spacing}$$

= (0.025– 0.02)/0.025 x 0.27 x 0.054 = 0.002916 m^2

3.19. Maximum Mass Flow rate of Flue gas

The mass velocity is calculated as follows:

$$\text{Mass velocity}, G_s = \frac{\text{Mass flow rate}}{\text{Cross flow area}}$$

= 47.04/ 0.002916 = 16131.6872 Kg/s.m^2
Density of flue gas = 0.4773485 kg/m^3

$$\text{Flue gas linear velocity} = \frac{\text{Flue gas mass velocity}}{\text{Density of flue}}$$

= 16131.6872 / 0.4773485 = 33794.3603 m/s

3.20. Equivalent diameter

$$D_e = \frac{4\left[\dfrac{p_t^2 - \pi d_o^2}{4}\right]}{\pi d_o} = \frac{1.27}{d_o}\left[p_t^2 - 0.785\, d_o^2\right]$$

= (1.27/0.02) x (0.025^2 - 0.785 x 0.02^2)
= 0.0197 m

3.21. Re

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

Where
Gs: the Mass velocity of the flue gas
De: Equivalent Diameter
µ: Viscosity of Flue gas
0.0248 centipoise = 0.0000248 Ns/m^2
Re= (16131.6872 x 0.0197)/0.0000248 = 12814283.78

3.22. Prandlt number

The prandlt number can be calculated as follows:\

$$Pr = \frac{C_p \mu}{K_f}$$

C_p : The specific heat of the flue gas
µ : Viscosity of flue gas
K_f : Thermal comductivity of the flue gas
K= 0.0363211 W/m °C

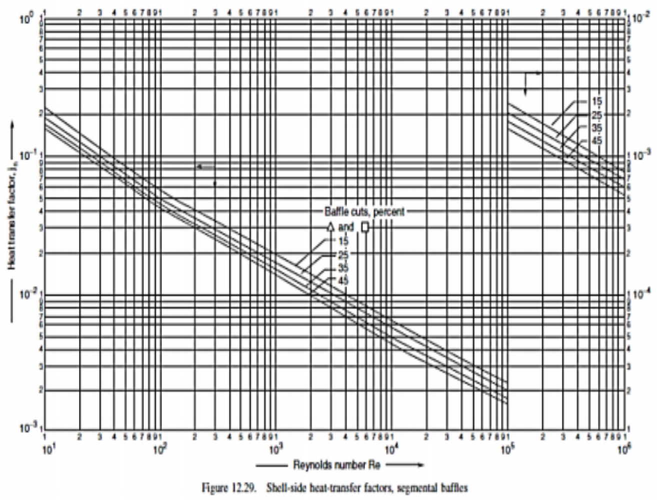

Figure.4Shell side heat transfer factor

Pr = (1.3 x 0.0000248)/0.0363211 = 8.87638 x 10^{-4}

A baffle cut of 25% is chosen; the value of jh is calculated by extrapolation from the following graph and is found to be 0.15.

$$\frac{h_s d_e}{K_f} = j_h Re Pr^{0.33} \left(\frac{\mu}{\mu_w}\right)^{0.14}$$

Neglecting the viscosity correction factor.
hs= (0.15 x 12814283.78 x(8.87638 x 10^{-4}) $^{0.33}$ x 0.0363211) / 0.0197
hs= 348655.226 W/m ^{2} °C

3.23. Overall transfer coefficient

$$\frac{1}{U_o} = \frac{1}{hs} + \frac{1}{fouling\ Factor\ flue\ gases} + \frac{D_o \times \ln\frac{D_o}{D_i}}{2 \times Thermal\ conductivity} + \left(\frac{D_o}{D_i} \times \frac{1}{Fouling\ factor\ of\ air}\right) + \left(\frac{D_o}{D_i} \times \frac{1}{h_i}\right)$$

hs = 348655.226 W/m ^{2} °C
hi = 9309.0374 W/m ^{2} °C
Do = 0.02m
Di = 0.016m

H_{od}, Fouling factor of flue gases = 2000 W/m ^{2} °C
H_{id}, Fouling factor of air = 5000 W/m ^{2} °C (Coulson page no.640)
Thermal conductivity of material = 50 W/mK (stainless steel)
$1/U_o$= 931.7748x 10^{-6} W/m ^{2} °C
U_o= 1073.2206 W/m ^{2} °C

IV. PERFORMANCE EVALUATION OF BOILER

The performance parameters of boiler, like efficiency and evaporation ratio reduces with time due to poor combustion, heat transfer surface fouling and poor operation and maintenance. Even for a new boiler, reasons such as deteriorating fuel quality, water quality etc. can result in poor boiler performance. Boiler efficiency tests help us to find out the deviation of boiler efficiency from the best efficiency and target problem area for corrective action.

4.1 Boiler Efficiency

Thermal efficiency of boiler is defined as the percentage of heat input that is effectively utilized to generate steam. There are two methods of assessing boiler efficiency.

The Direct Method: Where the energy gain of the working fluid (water and steam) is compared with the energy content of the boiler fuel.

The Indirect Method: Where the efficiency is the difference between the losses and the energy input.

4.2 Direct Method

This is also known as 'input-output method' due to the fact that it needs only the useful output (steam) and the heat input (i.e. fuel) for evaluating the efficiency. This efficiency can be evaluated using the formula\

$$\text{Boiler Efficiency} = \frac{\text{Heat output}}{\text{Heat input}} \times 100$$

Parameters to be monitored for the calculation of boiler efficiency by direct method are :

• Quantity of steam generated per hour (Q) in kg/hr.
• Quantity of fuel used per hour (q) in kg/hr.
• The working pressure (in kg/cm2(g)) and superheat temperature (°C), if any
• The temperature of feed water (°C)
• Type of fuel and gross calorific value of the fuel (GCV) in kCal/kg of fuel

$$\text{Boiler Efficiency} = \frac{Q \times (hg - hf)}{q \times GCV} \times 100$$

Where,
hg – Enthalpy of saturated steam in kCal/kg of steam,
hf – Enthalpy of feed water in kCal/kg of water.

4.3 Advantages of direct method

• Plant people can evaluate quickly the efficiency of boilers
• Requires few parameters for computation
• Needs few instruments for monitoring

4.3 Disadvantages Of Direct Method

•Does not give clues to the operator as to why efficiency of system is lower
• Does not calculate various losses accountable for various efficiency levels

4.4 Indirect Method

There are reference standards for Boiler Testing at Site using indirect method namely British Standard, BS 845: 1987 and USA Standard is ASME PTC-4-1 Power Test Code Steam Generating Units'. Indirect method is also called as heat loss method. The efficiency can be arrived at, by subtracting the heat loss fractions from 100. The standards do not include blow down loss in the efficiency determination process.

4.5 Boiler Efficiency Calculation

Boiler Efficiency of Thermax boiler.
Capacity of boiler : 8 ton
Water temperature ,tw : 32 °C
Mass flow rate of steam ms: 4 ton/hr

Mass of fuel mf : 500 Kg/hr
Calorific value of fuel : 9800 Kcal/Kg = 41004.1841 KJ/Kg
Temperature of steam ts = 460 °C
Boiler efficiency = ms(hs-hw)/mf x C.V
Enthalpy of water at 32 °C , hw = hf + X x hfg
(x =0,ie dryness fraction by using steam table)
hw = hf = 134.136 KJ/Kg

Enthalpy of steam at 460°C , hs = 3378.88KJ/Kg
Boiler efficiency = (ms (hs-hw)/ mf x C.V) x 100
= (4000 x (3378.88-134.136)/ 500 x 41004.1841)x 100
= 63.305%

4.6 Boiler Efficiency with Pre-heater

Calorific value of fuel = 41004.1841 KJ/Kg
Heat liberated by the fuel per hour = 41004.1841 x 500
=20502092.05 KJ/hr

Heat liberated by the fuel per second , q = 5695.0255 KJ/s
q = maCpa(T2-T1)
ma = 15.732Kg/s
Cpa = 1.005 KJ/KgK
T1 = 32 °C
5695.0255 = 15.732 x 1.005 x (T2 – 32)
There fore T2 = 392.2016 °C

Reduction in fuel consumption

T1' = Temperature of air out of the preheater = 75 °C
q' = 15.732 x 1.005 x (392.2016 – 75)
q' = 5015.1666 KJ/s
Hence fuel used = (5015.1666 x 3600)/ 41004.1841 =
440.3116 Kg/hr

Reduction in fuel = 500 – 440.3116 = 59.6888 Kg/hr
Increase in Efficiency = (59.6888/500) x 100 = 11.9377%
Efficiency of Boiler with Pre-heater = 63.305 + 11.9377 =
75.2427%

V. RESULT

AIR PRE-HEATER

- Area of heat transfer , A = 32.6072 m^2
- Tube length = 10.67 m
- Area of one tube = 0.67041587 m^2
- Number of tube, n = 50
- Bundle diameter = 0.25 m
- Tube arrangement and Tube pitch, T_p = 0.025 m
- Tube cross section area = 0.00020106 m^2
- Tube per pass = 25
- Total flow area = 0.0051526 m^2
- Air mass velocity = 3053.21585 kg/s.m^2
- Air linear velocity = 3014.03341 m/s
- Shell Diameter = 0.268 m
- Baffle spacing = 0.054 m
- Cross-flow area = 0.002916 m^2
- Maximum Mass Flow rate of Flue gas = 16131.6872 kg/s.m^2
- Equivalent diameter = 0.0197 m
- Overall transfer coefficient, Uo = 1073.2206 W/m^2 °C

VI. REFERENCE

[1]. Coulson and Richardson's Chemical Engineering design hand book_Volume 6, 4th edition.

[2]. Fundamentals of heat and mass transfer_Incropera_Dewitt_Bergman_Lavine_ 6th edition

[3]. Process heat transfer_ Donald Q Kern_ McGRAW-HILL Book company

[4]. Efficiency Analysis and Enhancement of Heat Recovery Steam Generator of a Combined Cycle Power Plant through Incorporation of Additional Bank of Tube in the Economizer_ Sreesankar.J_ International Journal of Scientific Research Engineering & Technology (IJSRET), ISSN 2278 – 0882.

[5]. A Study Analysis and Performance of High Pressure Boilers With its Accessories_ International Research Journal of Engineering and Technology (IRJET) e-ISSN: 2395 -0056 Volume: 02 Issue: 01 | Apr-2015

[6]. Improving boiler efficiency modeling based on ambient air temperature-Jijun Zhou

[7]. A Study Analysis and Performance of High Pressure Boilers With its Accessories J. Suresh babu

[8]. Optimization of Air Preheater Design for the Enhancement of Heat Transfer Coefficient-P. N. Sapkal