Design of Economizer 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. Economiser 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, Economizer

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 Economizer

The feed water from the high pressure heaters enters the economizer and picks up heat from the flue gases after the low temperature super heater. Many types of economizer are designed for picking up heat from the flue gas. These can be
classified as an inline or staggered arrangement based on the type of tube arrangement. The staggered arrangement is compact and occupies less volume for the same amount of heat transfer when compared to the inline arrangement. Economizers are also designed with plain tube and fined tubes. The fins can be longitudinal or spiral. All these types are suitable for clean fuels like gas, oil, and low ash coals. For high ash coals, only the plain tube inline arrangement is used. This is mainly to reduce ash erosion and thus reduce erosion failures. These economizers pick up about 50 to 55 degrees centigrade in a large capacity boiler, which will reduce the flue gas temperature by about 150 to 170 degree centigrade. The boiler designers always keep the economizer water outlet temperature to about 25 to 35 degrees below the drum saturation temperature. This is done to mainly avoid steaming in the economizer. A steaming economizer generally is less reliable. As a rule of thumb, for every one degree pick up of economizer water temperature, there will be a drop of about 3 to 3.5 degrees.

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
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°C
Rate of steam production: 96x3 ton/day
Flue gas velocity: 8 to 10 m/s
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III. DESIGN CALCULATION

Assuming no heat loss by radiation.
Energy absorbed by water = Energy loss by flue gas.
An Economizer 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%).

The temperature of flue gas out was recalculated using the following equations:

\[ M \cdot c_p \cdot (T_{ho} - T_{ho}) \] (1)

Where:
- \( M \): Mass flow rate of Flue gas out
- \( c_p \): Specific Heat Capacity of furnace oil
- \( T_{ho} \): Temperature of water out of the economizer
- \( T_{ho} \): Temperature of water in the economizer

Enthalpy of flue gas at inlet = Heat supplied by furnace / flow of flue gas

The appropriate average temperature difference is a log mean temperature difference, \( \Delta T_{lm} \).

Considering a counter flow heat exchanger:

\[ q = UA \Delta T_{lm} \]

Where,
- \( q \): Energy absorbed
- \( A \): Surface area
- \( U \): Overall heat transfer coefficient.

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

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

\[ \Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{\ln \left( \frac{\Delta T_1}{\Delta T_2} \right)} \] (3)

\[ q = \frac{\sqrt{(R^2 + 1) \ln \left( \frac{1 - S}{1 - R} \right)}}{(R - 1) \ln \left( \frac{2 - S[(R^2 + 1)/2 - S]^{1/2}}{2 - S(R^2 + 1/2 - S)^{1/2}} \right)} \] (7)

Where,
- \( \Delta T_{lm} \): Logarithmic mean temperature difference
- \( T_1 \): Hot fluid temperature at inlet
- \( T_2 \): Hot fluid temperature at outlet
- \( t_1 \): Cold fluid temperature at inlet
- \( t_2 \): Cold fluid temperature at outlet

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.

\[ \Delta T_{lm} = F_t \Delta T_{lm} \] (4)

Where,
- \( F_t \): 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.

\[ R = \frac{T_{ho} - T_{co}}{T_{hi} - T_{ci}} \] (5)

\[ S = \frac{t_2 - t_1}{T_{hi} - T_{ci}} \] (6)

\[ F_t = \frac{\sqrt{(R^2 + 1) \ln \left( \frac{1 - S}{1 - R} \right)}}{(R - 1) \ln \left( \frac{2 - S[(R^2 + 1)/2 - S]^{1/2}}{2 - S(R^2 + 1/2 - S)^{1/2}} \right)} \] (7)

\[ \Delta T_{lm} = \frac{(T_{hi} - t_1) - (T_{ci} - t_1)}{\ln \left( \frac{T_{hi} - t_1}{T_{ci} - t_1} \right)} \] (8)

Where,
- \( \Delta T_{lm} \): Logarithmic mean temperature difference
- \( T_{hi} \): Temperature of flue gas in the economizer
- \( T_{ci} \): Temperature of flue gas out of the economizer
- \( T_{co} \): Temperature of water out of the economizer
- \( T_{ho} \): Temperature of water in the economizer

The assumptions are:

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.

The calorific value of fuel:

\[ \text{Calorific value} = 9800 \text{ to } 10100 \text{ kcal/kg} \]

The density of fuel:

\[ \text{Density of fuel} : 0.97 \text{ to } 1.07 \text{ g/cc} \]

The specific heat capacity of furnace oil:

\[ \text{Cp: Specific Heat Capacity of furnace oil} \]

The rate of steam production:

\[ \text{Rate of steam production} : 96x3 \text{ ton/day} \]

The boiler operation time:

\[ \text{Boiler operation time} : 8760 \text{ Hrs/year} \]

The boiler type:

\[ \text{Boiler type} : \text{Fire tube boiler} \]

The equipment used for:

\[ \text{Equipment used for} : \text{Production of NaOH flakes} \]

The fuel used:

\[ \text{Fuel used} : \text{Furnace oil} \]

The density of fuel:

\[ \text{Density of fuel} : 0.97 \text{ to } 1.07 \text{ g/cc} \]

The specific heat capacity of fuel:

\[ \text{Specific heat capacity of fuel} : 2.09 \text{ KJ/KgK} \]

The specific heat capacity of flue gas:

\[ \text{Specific heat capacity of flue gas} : 1.3 \text{ KJ/KgK} \]

The boiler operation time:

\[ \text{Boiler operation time} : 8760 \text{ Hrs/year} \]

The flue gas temperature:

\[ \text{Flue gas temperature} : 260 \text{ to } 270°C \]

The rate of steam production:

\[ \text{Rate of steam production} : 96x3 \text{ ton/day} \]

The flue gas velocity:

\[ \text{Flue gas velocity} : 8 \text{ to } 10 \text{ m/s} \]

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3.1 Fouling Factor

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.

3.2 Tube Length

Shell-diameter-to-tube-length ratio should be within limits of 1/5 to 1/15. In an 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. 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.

\[
\text{Tube length} = (10 \times \text{diameter of shell}) = 10 \times 1.067 = 10.67 \text{ m}
\]

3.3 Area of one tube

Assumption: Tube outside diameter = 20 mm
Wall thickness = 2.6 mm
Inner diameter = 16 mm = 0.016 m

\[
\text{Area} = \pi \times \text{Do} \times L
\]

Area = \[\pi \times 0.02 \times 10.67 = 0.67041587 \text{ m}^2\]  

3.4 Number of tube

Total heat transfer area required for an economizer at full load condition is varies between 1487.567 to 1793.6785 m².

Assuming area A = 1687 m²

\[
\text{n} = \frac{\text{A}}{\pi \times \text{Do} \times L}
\]

n = 1687/(\pi \times 0.02 \times 10.67) = 2517.7 = 2518

3.5 Bundle diameter

\[
\text{Db} = \frac{\text{Nd}^{0.5}}{\text{Ks}^{0.33}}
\]

Where,
\[\text{Nd} = \text{Number of tubes}
\]
\[\text{Db} = \text{Bundle diameter}
\]
\[\text{Do} = \text{Tube outside diameter}
\]
For 1 shell two pass \(Ks = 0.156\) and \(n = 2.291\)
Db = 0.02 x (2518/0.156)^0.5/2.291 = 1.373259 m = 1.37 m

3.6 Tube arrangement and Tube pitch

Tube pitch, \(T_p\) = 1.25 \(D_o\)
= 1.25 x 0.02 = 0.025 m

3.7 Tubes in the middle

\[
= \frac{\text{Db}}{T_p} = \frac{1.373259}{0.025} = 54.93036 = 55
\]

3.8 Tube cross section area

Total cross section area inside tube = \[\pi \text{D}_i^2/4\]

\[= \left(\pi \times 0.01982 \right)^4 = 0.0003079075 \text{ m}^2\]

3.9 Tube per pass

Tube per pass = number of tubes / total number of pass
2518/2 = 1259

3.10 Total flow area

Total flow area = No. of tubes per pass x Total area inside the tube
= 1259 x 0.0003079075 = 0.387655543 m²

3.11 Water mass velocity

Water mass velocity = Mass flow rate / Total flow area
Mass flow rate of water for an economizer can be between 15 to 85 kg/s
Assume mass flow rate = 40 kg/s
= 39.283 (kg/s) /0.387655543 m² = 101.344079 kg/s.m²

3.12 Water linear velocity

Water linear velocity = water mass velocity / density of water
Assuming density of water = 945.7139 kg/m³
=101.344079/945.7139 = 0.107151653 m/s

3.13 Tube side coefficient

\[
\frac{h_i d_i}{k_f} = j_s Re Pr^{0.33} \left(\frac{\mu}{\mu_w}\right)^{0.14}
\]

Where,
\[h_i: \text{The tube side coefficient}
\]
\[d_i: \text{The inside diameter}
\]
\[Re: \text{Reynold's number}
\]
\[Pr: \text{Prandlt Number}
\]
\[\mu: \text{Viscosity of water}
\]

3.14 Reynold's Number

\[Re = \frac{\rho u d_i}{\mu}
\]

Assume \(d_i = 0.0198\) m
= (945.71391138317 x 0.107151653 x 0.0198)/ 0.8 x 10⁻³
= 2508.03652

3.15 Prandlt number

\[Pr = \frac{C_p \mu}{k_f}
\]

Where:
\[C_p: \text{The specific heat of water (4.2 KJ/ kg °C)}
\]
\[K_f: \text{Thermal conductivity of water}
\]
\[Pr = (4.2 \times 103 \times 0.8 \times 10⁻³)/ 0.59
\]

=5.7

Viscosity correction is neglected. The L/ D ratio is calculated to be 666.875.
$$h_i = jhRePr^{0.33} \frac{k_f}{d_i} = 0.0018 \times 2508.03652 \times 5.7033^{0.33} \times 0.59/0.016 = 295.6494616 \text{ W/m}^2\text{oC}$$

### 3.16 Shell Diameter

The economizer is a u-tube with a shell bundle diameter of 1.37 m, the bundle diametrical clearance is found to be 18 mm from figure.

$$D_s = 1.37 + 0.018 = 1.388 \text{ m}$$

The shell diameter was also calculated using the excel flowsheet designed by Mr Prem Baboo and was found to be 1.445 m.

### 3.17 Baffle spacing

$$= D_s/5 = 1.4/5 = 0.28 \text{ m}$$

### 3.18 Cross-flow area

$$\text{Cross flow area} = \frac{\text{Tube pitch – Outer diameter}}{\text{Tube pitch}} \times D_s \times B_s$$

### 3.19 Maximum Mass Flow rate of Flue gas

Mass velocity, $$G_s = \frac{\text{Mass flow rate}}{\text{Cross flow area}} = \frac{47.04 \text{ kg/s}}{0.0784 \text{ m}^2} = 600 \text{ kg/s.m}^2$$

Density of flue gas was previously calculated as 0.0298 lb/ft$^3$ = 0.4773485 kg/m$^3$

Flue gas linear velocity = Flue gas mass velocity / Density of flue gas

$$= 600 / 0.4773485 = 1256.943 \text{ m/s}$$

### 3.20 Equivalent diameter

$$D_e = \frac{\alpha x - \beta x}{\pi d_a} = \frac{1.27 (P_r - 0.785 d_o)}{d_a}$$

$$= (1.27/0.02) \times (0.025^2 - 0.785 \times 0.02^2) = 0.0197 \text{ m}$$

### 3.21 Re

$$Re = \frac{G_s d_a}{\mu}$$

Where

- $$G_s$$: the Mass velocity of the flue gas
- $$D_e$$: Equivalent Diameter
- $$\mu$$: Viscosity of Flue gas

$$Re = (600 \times 0.01097) / 0.0000248 = 265,403.2258$$

### 3.22 Prandlt number

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

Where

- $$C_p$$: Specific heat of flue gas
- $$\mu$$: Viscosity of flue gas
- $$K_f$$: Thermal conductivity of flue gas

$$= (1.388 \times 0.0000248) / 0.0363211 = 9.477246 \times 10^{-4}$$

A baffle cut of 25% is chosen; the value of $j_b$ is calculated by extrapolation from the following graph and is found to be 0.13.
3.23 Overall transfer coefficient

\[
\frac{1}{U_o} = \frac{1}{h_o} + \frac{1}{h_{ad}} + \frac{d_o \ln \left( \frac{d_o}{d_i} \right)}{2 \lambda_o} + \frac{d_o}{d_i} \times \frac{1}{h_{ad}} + \frac{d_o}{d_i} \times \frac{1}{h_i} \tag{21}
\]

Where

- \( U_o \): Overall heat transfer coefficient based on the outside area of the tube, W/m²°C
- \( h_o \): Outside fluid film coefficient, W/m²°C
- \( h_i \): Inside fluid film coefficient, W/m²°C
- \( h_{ad} \): Fouling factor, W/m²°C
- \( h_{id} \): Inside dirt coefficient, W/m²°C
- \( k_w \): Thermal conductivity of tube wall material, W/m²°C
- \( d_i \): Tube inside diameter, m
- \( d_o \): Tube outside diameter, m

\[ h_o = 103,882.0343 \text{ W/m}^2\text{C} \]
\[ h_i = 295.6494616 \text{ W/m}^2\text{C} \]
\[ D_o = 0.02\text{m} \]
\[ D_i = 0.016\text{m} \]
\[ h_{ad}, \text{ Fouling factor of flue gases} = 2000 \text{ W/m}^2\text{C} \]
\[ h_{id}, \text{ Fouling factor of river water} = 3000 \text{ W/m}^2\text{C} \]

Thermal conductivity of material = 50 W/mK (stainless steel)

\[ \frac{1}{U_o} = 5.1989 \times 10^{-3} \text{ W/m}²\text{C} \]
\[ U_o = 192.3 \text{ W/m}²\text{C} \]

IV. REFERENCES