USE OF CORRUGATED PIPE HEAT EXCHANGERS IN WASTE HEAT RECOVERY STEAM GENERATORS

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Abstract. Heat exchangers are used for a variety of purposes in various domestic, commercial & industrial applications. Some common examples are sensible heating & sensible cooling in chemical processes, condensation and steam generation in power plants, and recovery of waste heat etc. Heat exchangers performance enhancement can lead to more economical design of heat exchanger which increases energy, material & cost savings related to heat exchange process. Making corrugation on pipe surface is one such method. Numerous works has been carried out in this area, where parameters such as inside diameter of pipe, corrugation pitch, corrugation depth, temperature and phase of liquid has been studied. Corrugation of tubes can increase both the inner and outer surface convective heat transfer coefficient without any considerable rise in pressure drop. It also results in increase in mixing of fluid, flow turbulence, unsteadiness by limiting the fluid boundary layer growth adjacent to the tube surface.

Keywords: Heat exchanger, Corrugation pitch, Corrugation depth, Pressure drop etc.

1. INTRODUCTION
With population growth, the demand for electricity has increased rapidly. This has motivated the researchers to look for more efficient and sustainable technologies for power generation thereby reducing the cost of electric power generation and reducing emissions containing harmful pollutants. Some of these technologies are power plants with waste heat recovery systems consisting of a gas cycle and steam cycle, or a steam power plant linked by a Heat Recovery Steam Generator (HRSG). The HRSG produces steam at a suitable temperature and pressure by recovering heat from the flue gases coming out of the gas turbine.

1.1. Origin of the project
Heat exchangers are used in different purposes ranging from conversions, utilization & recovery of thermal energy. Some common examples are sensible heating & sensible cooling in chemical processes, condensation and steam generation in power plants, and recovery of waste heat etc. The techniques used for increasing the heat exchanger performance are also referred as heat transfer augmentation techniques or Heat transfer intensification techniques. Heat transfer through convection is improved by decreasing the thermal
resistance in the heat exchanger. Heat transfer augmentation techniques lead to increase in heat transfer coefficient but at the same time leads to increase in pressure drop. Thus, the heat transfer rate and pressure drop are the main parameters which are to be considered during design of heat exchanger using any of these techniques. There are three main categories into which Existing enhancement techniques can be classified: Passive Method, Active Method, and Compound Method.

1.2. Objectives
The Objective of the project includes the following:
- Comparative study of corrugated pipe and plain pipe heat exchangers in HRSG.

1.3. Literature review
Use of corrugated tube can produce superior heat exchanger performance. Corrugation of tubes can increase both the inner and outer surface convective heat transfer coefficient without any considerable rise in pressure drop. Increase in mixing of fluid, flow turbulence, unsteadiness by limiting the fluid boundary layer growth adjacent to the tube surface. To reduce the size of industrial shell and tube heat exchangers corrugated tubes are chosen. The use of corrugated tubes is still a new method for enhancing the performance of heat exchangers and a lot of work is going on in this field.

Rainieri and Pagliarini conducted experiments to study the heat transfer characteristics of tubes with corrugation. Transverse and helical corrugations were made on the smooth surface of the pipe with different pitches with ethylene glycol as the fluid used. Among the smooth tube, helical corrugated tubes and transverse corrugated tubes, results showed that helical transverse corrugated tubes were having the highest heat transfer enhancement.

Laohalertdecha and Wongwises investigated the behavior of heat transfer coefficient and pressure drop due to change in pitch in condensation and evaporation process in corrugated tubes with R-134a as working fluid. For all experiments performed, corrugated tube is better than the smooth pipe in terms of heat transfer coefficient and pressure drop.

Targanski and Cieslinski studied evaporation inside a smooth copper pipe, smooth stainless steel pipe and two enhanced pipes for R-407C/oil mixtures and pure R-407C as the working fluids. Results obtained were in favour of corrugated tubes.

1.4. Methodology
The design of HRSG involves following steps:
- Determination of design input parameters.
- Working Fluid selection.
- Selection of working pressure.
- Determination of temperature profile.
- Determination of Convection Heat Transfer Coefficient.
- Heat exchanger Area calculation.
- Determination of number of pipes.
- Calculation and Comparison of Heat Exchanger Performance.

2. Mathematical Modeling
Representation of the processes by a mathematical model helps in simulating the system

2.1. Assumptions
In the development of mathematical model certain assumptions were made. These are as follows:

- Water and flue gas are in steady flow within the the system.
- Unfired HRSG.
- No loss of thermal energy from the heat exchangers.
- To prevent condensation, the temperature of flue gases in the chimney is assumed to be more than 100°C.
- Staggered arrangement of tubes is selected for better heat transfer.

| Table 1. Parameters |
|----------------------|
| Sr. No. | Design Input Parameters | Value |
| 1. | Flow rate of water at the tube inlet, $m_w$ | 1 kg/s |
| 2. | Flow rate of flue gas at the shell inlet, $m_g$ | 1.573 kg/s |
| 3. | Water temperature at the tube inlet, $T_{c1}$ | 242°C |
| 4. | Flue Gas temperature at the shell inlet, $T_{h1}$ | 852°C |
| 5. | Flue gas temperature at the shell outlet, $T_{h2}$ | 420°C |
| 6. | Tube inside Dia, $D_i$ | 25.4 mm |
| 7. | Tube outside dia, $D_o$ | 31 mm |
| 8. | Shell dia | 838 mm |
| 9. | Pressure inside tube | 160 bar |
| 10. | Pressure outside tube | 1 bar |

3. Temperature Profile
The determination of temperature profile is the first step in the design of HRSG. The inlet and outlet temperatures of the flue gases and working fluid are calculated by energy balance method. The calculation should be done by using a suitable pinch point (Difference between the gas temperature at the evaporator outlet and the temperature of the fluid at the evaporator inlet). Equating the heat lost to the heat gained

$$Q_{cold} = Q_{hot}$$

$$m_w(h_{cold,o} - h_{cold,in}) = m_g(h_{hot,in} - h_{hot,o})$$

$$m_{cw}(T_{c,2} - T_{c,1}) = m_{cg}(T_{h,1} - T_{h,2})$$

Substituting the values from Tables 2.1, 2.3 in the above equation the water temperature at the tube outlet is found to be 347.3°C.

3.1. Overall Heat Transfer Coefficients
Calculation of the overall heat transfer coefficient (U) can be done as shown below:
Thin steel tubes having high thermal conductivity were used in the heat exchanger. Therefore, the second term in the above equation can be neglected. Therefore the equation for $U$ gets simplified to the form:

$$\frac{1}{UA} = \frac{1}{h_n A_n} + \frac{\ln \left( \frac{D_0}{D_i} \right)}{2\pi k l} + \frac{1}{h_i A_i} \quad (2)$$

To calculate the inner heat transfer coefficient for steady state incompressible flows inside a tube with uniform cross-sectional area, the flow intensity can be calculated using the Reynolds number:

$$Re_D = \frac{4m_w}{\pi D_i \mu} \quad (4)$$

The present calculation shows that the Reynold number is $9.13 \times 10^3$, thus the flow is turbulent. Now, for smooth circular pipe with turbulent flow the Nusselt number ($Nu_D$) is given by DittusBoelter Equation

$$Nu_D = 0.023 Re_D^{0.8} Pr^n \quad (5)$$

For cooling, $n = 0.3$, and for heating, $n = 0.4$. The value of $Nu_D$ was found to be 33.34. The heat transfer coefficient for convective heat transfer inside the pipe is calculated as:

$$h_i = Nu_D \frac{K}{D_i} \quad (6)$$

The value of internal heat transfer coefficient was found to be 721.3 W/mK.

Next, the heat transfer coefficient for the convective heat transfer outside the bank tubes is calculated. The heat transfer in flow over tube bundles depends largely on the flow pattern and the degree of turbulence, which in turn are functions of the velocity of the fluid and the size and arrangement of the tubes. The tube bundles can be in-line or staggered arrangements. The bundle geometry is characterized by the Transverse pitch $S_T$ and the longitudinal pitch $S_L$ between the tube centers. The diagonal pitch $S_D$ is sometimes used for the staggered arrangement. To define the Reynolds number for flow through the tube bank, the flow velocity is based on the minimum flow area. The Reynolds number is

$$Re_{D,\text{max}} = \frac{V_{\text{max}} D}{\nu} \quad (7)$$

In the case of staggered configuration, the maximum velocity could be at either the diagonal plane $A_2$ or the transverse plane $A_1$ of Figure. $V_{\text{max}}$ will be at $A_2$ if the spacing between the rows is such that such that

$$2(S_D - D) < (S_T - D)$$

$$S_D = \sqrt{(S_T^2 + \left( \frac{S_T}{2} \right)^2)} < \frac{S_T + D}{2} \quad (8)$$
In our case $S_T=S_L=3D=93$ mm substituting these values in equation 8 it was found that $V_{\text{max}}$ occurs at $A_1$ and is given by

$$V_{\text{max}} = \frac{S_T}{(S_T - D)} V$$

(9)

The $V_{\text{max}}$ was found to be 12.08 m/s. Putting this value in equation 7, the value of $Re_{D,\text{max}}$ is 3780. Zukauskas developed a correlation for average Nusselt number for flow over a bank of tube

$$\bar{N}_u D = C_1 Re_{D,\text{max}}^{m} Pr^0.36 \left( \frac{Pr}{Pr_w} \right)^{0.27}$$

(10)

where $n_R$ is the number of tube rows, all properties are at the bulk fluid temperature (average of the fluid outlet $T_o$ and inlet $T_i$ temperatures) except $Pr_w (\frac{Pr}{Pr_w}=0.94)$ which is evaluated at the tube wall temperature, and the constants $m$ and $C_1$ are listed in Table below.

Table 2. Values of Constants

| Configuration       | $Re_{D,\text{max}}$ | $C_1$  | $m$   |
|---------------------|----------------------|--------|-------|
| Staggered($S_T/S_L<2$) | $10^3 - 2 \times 10^5$ | $0.35(S_T/S_L)^{0.2}$ | 0.60  |

If the number of rows of tubes, $n_R$, is less than 20 there is a decrease in the average heat transfer coefficient, thus a correction factor is incorporated such that

$$\bar{Nu}_D(n_R<20) = C_2 \bar{Nu}_D(n_R\geq20)$$

(11)
where $C_2$ is given in Table below.

| $n_R$ | 1   | 2   | 3   | 4   | 5   | 10  | 13  | 16  |
|-------|-----|-----|-----|-----|-----|-----|-----|-----|
| Aligned | 0.70 | 0.80 | 0.86 | 0.90 | 0.92 | 0.95 | 0.97 | 0.98 | 0.99 |
| Staggered | 0.64 | 0.76 | 0.84 | 0.89 | 0.92 | 0.95 | 0.97 | 0.98 | 0.99 |

$N_{uD}$ value was found to be 40.58. The convection heat transfer coefficient $h_o$ can then be calculated using

$$h_o = N_{uD} \frac{K}{D_n}$$

(12)

$h_o$ was found to be 100.69 $W/m^2K$.

Table 3. Fluid Properties

| Properties                      | Water | Flue gas |
|--------------------------------|-------|----------|
| Density, $\rho$ (kg/m$^3$)     | 641   | 0.384    |
| Dynamic Viscosity, $\mu$ (Ns/m$^2$) | $57 \times 10^{-6}$ | $39.26 \times 10^{-6}$ |
| Specific heat capacity, $c$ (KJ/KgK) | 8     | 1.226    |
| Thermal conductivity, $k$ (W/mk) | $25.4 \times 10^{-3}$ | $78.4 \times 10^{-3}$ |
| Prandtl number, Pr              | 0.96  | 0.615    |

From equation 3, 6, 12 the overall heat transfer coefficient $U$ is calculated to be 88.35 $W/m^2K$.

3.2 Area Calculation

The total heat transfer is given by

$$Q = FUoAo(\Delta T_{LM})_{counterflow}$$

(13)

where

$$\Delta T_{LM} = \frac{(T_{b2} - T_{c1}) - (T_{b1} - T_{c2})}{\ln \frac{T_{b2} - T_{c1}}{T_{b1} - T_{c2}}}$$

The value of correction factor $F$ was determined using the plot of $F$ for single pass cross-flow heat exchanger (one fluid mixed and other unmixed). Using equation 13 the outer area required for the heat transfer is calculated. The values of $Q$, $U$,($\Delta T_{LM})_{counterflow}$ are 842.4 KW, 88.35 $W/m^2K$ and 313.5K respectively. Then, the length of tubes inside the shell was to be 2.71 m as the number of tubes and standard tube diameters have already been assumed (57 and 31 mm respectively).

4. Design of Corrugated Pipe
In mathematical modeling the analysis was done by assuming the cross-section of tubes inside the shell to be plain. By using the same length and inlet conditions for a corrugated pipe the heat transfer can be enhanced, which increases the outlet temperature of water. In design view, mainly 3 parameters has to be specified for the cross-section of corrugated pipe. They are inner diameter ($d_i$), pitch($p$), height($e$).

Figure 2. Schematic diagram of a corrugated pipe

Figure 3. Design considered for analysis

The Fig.3 shows the design, which we have considered by taking the parameters as $d_i = 25.4$ mm, $p = 20$ mm, $e = 2.085$ mm.

5. Results and Discussion
Figure 4. Temperature profile along the axis of plain pipe

Figure 5. Temperature profile along the axis of corrugated pipe
Figure 6. Temperature distribution at inlet

Figure 7. Temperature distribution at outlet
Figure 8. Effect on velocity vector of fluid due to corrugation

Figure 9. Turbulent kinetic energy corrugated pipe
Fig. 10. Temperature distribution in corrugated pipe

Fig. 4- Fig. 10 provided above are the results obtained from the analysis done with given inlet conditions. The values obtained for different parameters after the analysis is mentioned in the table below:

Table 4. Resulting Outlet Conditions

| Parameter                  | Plain Tube | Corrugated Tube |
|----------------------------|------------|-----------------|
| Tube Outlet Temperature    | 285°C      | 294.3°C         |
| Shell Outlet Temperature   | 759°C      | 740°C           |
| Pressure drop in the tube  | 4413 Pa    | 192678 Pa       |

6. Conclusions
The effect of corrugation on the heat transfer and the pressure drop in the pipe was investigated numerically. It has been seen that the temperature rise is more in the case of corrugated pipe as compared to plain pipe due to enhanced heat transfer but the pressure drop is also increased. Hence, care must be taken that the heat transfer must be economical and not at the cost of significant increase in pumping power.

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