RESEARCH ARTICLE OPEN ACCESS

Remaining Life Analysis of Boiler Tubes on Behalf of Hoop Stresses Produced During Operation of Power Plant

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ABSTRACT
Boiler tube material plays an important role in efficient power generation from a fossil fuel power plant. In order to meet out the gap between fluids to increase heat available per unit mass flow of steam. Waste heat utilization phenomenon is a big challenge on fossil fuel power plants as after use of high grade coal in thermal power plants the efficiency of power plants is not at the level of required value. Clean and efficient power generation with economical aspects is the basic need of growing power generation plants to justify the quality of power and clean power generation. Life analysis technique to calculate remaining life of boiler tubes at critical zones of high temperature requires much attention and is an important hypothesis in research field. Generation of repetitive and fluctuating stress during flow of high temperature and pressure fluid require proper attention on the methodology to be used to calculate the efficiency of system and absorption efficiency of tube material. In this paper complete mathematical analysis of boiler tubes is conducted for calculation of remaining life of boiler tubes, Hoop stress values are calculated and used with mathematical tool to calculate the efficiency. Hoop stress based calculation of efficiency is more reliable and may give more accurate and practical aspects based results.
Keywords- Absorption efficiency, Heat availability, Hoop stress analysis, Life analysis, Clean power, Tube material characteristics.

I. INTRODUCTION

The function of boiler is to convert the water into the superheated steam, which is then delivered to turbine to generate electricity. Pulverized coal is the common fuel used in boiler along with preheated air. The boiler consists of different critical components like economizer, water wall, super heater and reheater tubes. Platen super heater is one of the critical components of drum type utility boiler and located in the furnace zone as pendent coil. These coils are directly exposed to radiant flux zone for superheating the steam for desired temperature and pressure.

Thermal power plant boiler is one of the critical equipment for the power generation industries. In the present situation of power generation, pulverized coal fired power station are backbones of industrial development in the country, thus necessitation their maximum availability in terms of plant load factor (PLF). At the same time reliability and safety aspect is also to be considered. The major percentage of the forced shutdown of the power stations is from boiler side. So it is necessary to predict the probable root cause of the forced outages and also the remedial action to prevent the recurrence of similar failure in future. A drum type utility boiler for thermal power generation typically consists of different pressure parts tubes like water wall, economizer, super heater and reheater. Different damage mechanism like creep, fatigue, corrosion and corrosion are responsible of the different pressure parts tube failure. This paper documents one such failure investigation to identify the probable cause/causes of frequent failure of boiler tubes used for power generation.

The maximum hoop stress varied as a function of location, with the peaks in the range of 55-60 ksi generally at the uppermost tie welds and the outer-loop tubes, where the tube to tube temperature differences were highest. Corrosion results in Varying hoop stress caused by temperature fluctuations. Cyclic hoop stress can cause tube expansion resulting in creep-like longitudinal water wall tube cracking. Thinner walls and thinner walls result in higher hoop stresses which increases the potential for cracking. Such a component fails in since when subjected to an excessively high internal pressure. While it might fail by satiated along a path following the circumference of the tube. This suggests that the hoop stress is significantly higher than the axial stress. Hoop stresses do not vary through the wall. The only stress in the lateral direction (see Figure 1) is contributed by the hoop stress.

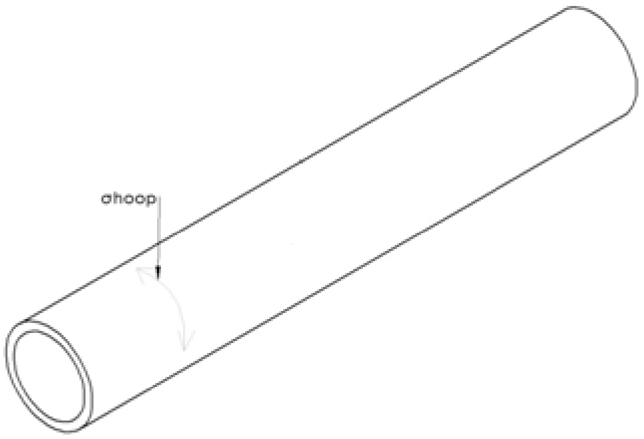

Figure 1 – Stress from internal pressure

The hoop stress in the lateral direction is the largest contributor to the maximum stress.

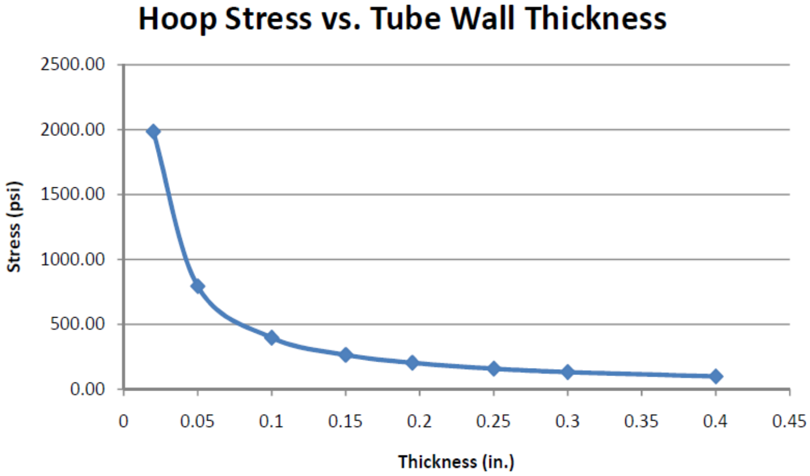

Figure 2: Hoop stress as a function of tube wall thickness

Figure 2 shows the relation between tube thickness and hoop stress. It shows that as the Wall thickness of tube is increased the hoop stress decreases. This is important to the strength of the tubes as the hoop stress is the main role to the maximum stress. While increasing the tube thickness gives better strength to the tubes, the thermal resistance and hence the combined tube wall and heat transfer coefficient would decrease greatly.

When we calculate the efficiency the using longitudinal stress the value of efficiency more than 100% which is in practical and shows artificial result.

II. FAILURE CAUSES

With increasing the life of thermal power plant there are several causes of failure occur due to stress concentration, flow rate, heat transfer, corrosion, erosion, fatigue, and creep etc.

the following are shown in tabular form as:

➢ Creep(long –term overheating)	23.4%
➢ Fatigue(thermal 8.6%, corrosion 5.3%)	13.7%
➢ Ash corrosion (coal 8.1%, oil 1.4%, refuse 2.5%)	12.2%
➢ Hydrogen damage	10.6%
➢ Weld failure	8.0%
➢ High temperature (short-term overheating)	8.8%
➢ Erosion	7.5%
➢ Oxygen pitting	5.6%
➢ Caustic attack	3.5%
➢ Stress corrosion cracking	2.6%

III. Calculation

3.1 Definitions

σ_m = membrane hoop stress(N/mm^2)
$\sigma_{1,2}$= Peak stress (N/mm^2)
$Kt_{1,2}$=Stress concentration factor (SCF) due to peaking
D = Mean diameter
δ = inward or out ward peaking determined in accordance
t = Boiler shell thickness (mm)
P_1= Normal operating pressure range (N/mm^2)
P_2= Partial pressure range (N/mm^2)
N_1 = Allowable number of fatigue cycles due to pressure range P_1
N_2 = Allowable number of fatigue cycles due to pressure range P_2
$C_{1,2}$ = Pressure cycles

3.2 Boiler specification

- External diameter=1800mm
- Shell thickness=10.6 mm
- Safety valve setting=150 psi(1.034N/mm^2)
- Measured peaking using a bridge gauge=6 mm

Burner cuts in at 90 psi (N/mm^2) and cuts out at 120 psi (0.83N/mm). The boiler operates 12 hours/day and cycles twice per hour as partial pressure cycles, it operates for 5 day/week and the factory is shut down for 2 week per year.
The boiler has one full pressure cycle per working day.
D= 1789.4mm
T=10.6mm
P_1=0.83N/mm^2
P_2=0.21N/mm^2
C_1=250 cycles
C_2=6000 cycles

3.3 peaking

δ = 6+1mm=7mm
δ=6+0.6mm=6.6mm
δ=7mm

3.3 Membrane hoop stress

Membrane hoop stress σ_{m1}

$$\sigma = \frac{Pd}{2t}$$

$$\sigma_{m1} = \frac{0.83\times1789.4}{2\times10.6} = 70 \text{ N/mm}^2$$

Membrane hoop stress σ_{m2}

$$\sigma_{m1} = \frac{0.21\times1789.4}{2\times10.6} = 17.7\text{N/mm}^2$$

3.4 Calculation of the stress concentration factor (Kt_1)

$$\beta = 0.0075\sqrt{\frac{1789.4\times7\times70}{10.6^2}}$$

β=0.66
tanh β_1/β_2 = 0.875

$$Kt_1=1+\frac{6\times7}{10.6}\times0.875=4.47$$

3.5 Calculation of peak stress for P_1

$\sigma\rho_1$ = 4.47×70= 312.9 N/mm^2
20% of P_1=. Therefore P_2>0.2 P_1

3.6 Calculation of the stress concentration factor (Kt_2)

β_2 =0.0075√1789.4×7×70/10.6^2
β_2 =0.33
tanh β_1/β_2 = 0.965

$$Kt_2=1+\frac{6\times7}{10.6}\times0.965=4.82$$

3.7 Calculation of peak stress for P_2

$\sigma\rho_2$ =4.82× 17.7= 85.3N/mm^2

3.7.1 Calculation of fatigue life(N)

N1=1500 cycles(measured from graph)
N1=1485 cycles(calculated)

3.7.2 Calculationof fatigue life(N)

N2=74000 cycles(measured from graph)
N2=73310 cycles(calculated)

3.8 Calculation of ultrasonic inspection interval

$$\text{inspection interval} = \frac{1}{\frac{C_1}{N_1}+\frac{C_2}{N_2}}=3.99 \text{ years}$$

3.9 Resultant Hoop Stress

$$\sigma_m= \sqrt{70^2 + 17.7^2} =72.20 \text{ N/mm}^2$$

3.10 Hoop Stress

$$\sigma_n=P\left(\frac{r_i+\frac{t}{2}}{t}\right)$$

$$= 1.04\left(\frac{894.7+5.3}{10.6}\right)$$

$$=88.30\text{N/mm}^2$$

3.11 Stress concentration factor (Kt)

$$\beta= 0.0075\sqrt{\frac{1789.4\times7\times88.30}{10.6^2}} = 0.66$$

$$\text{Maximum Shear Stress} = \frac{Pd}{8t}$$

$$=\frac{1.04\times1789.4}{8\times10.6}$$

$$=21.94 \text{ N/mm}^2$$

For longitudinal :-

$$\sigma=\frac{P\times d}{2t\times\eta}$$

by using σ_m=72.20 N/mm^2

$$72.20=\frac{1.04\times1789.4}{2\times10.6\times\eta}$$

$$\eta=\frac{1.04\times1789.4}{1530.64}=1.21(121\%)$$

by using σ_n= 88.30 N/mm^2

$$88.30=\frac{1.04\times1789.4}{2\times10.6\times\eta}$$

$$\eta=\frac{1860.976}{1871.96}=0.99(99\%)$$

3.12 Calculation of the stress concentration factor (Kt_1)

$$\beta_1=0.0075\sqrt{\frac{1986\times10\times929}{14^2}}$$

β_1=0.73
tanh β_1/β_1=0.86

$Kt_1 = 1+\frac{6\times10}{14}\times 0.86 = 4.69$
Calculation of peak stress for P_1
$\sigma\rho_1 = 4.69 \times 92.9 = 435.7$ N/mm^2

2.1 A further example using the following boiler specification

- External diameter = 2000mm
- Shell thickness = 14mm
- design pressure= 1.38N/mm^2
- Maximum working pressure = 1.31N/mm^2

Measured peaking using a needle gauge = 10mm
Burner cuts in at 1.1 N/mm^2 and cuts out at 1.31 N/mm^2. The boiler operates 24 hours/day for 5 days each week and cycles twice per hour as partial pressure cycles.
The boiler has one full pressure cycle per working day
D=1986mm
T=14mm
P_1=1.3 1N/mm
P_2=0.2 N/mm
C_1=52 cycles
C_2=17500cycles

2.2 Peaking:-

As the peaking was measured using a needle gauge there is no need to use a correction factor.

δ = 10mm

2.3Membrane hoop stress:-

$\sigma_{m1}=\frac{1.31\times1986}{2\times14} = 92.9$ N/mm^2
$\sigma_{m2}=\frac{0.2\times1986}{2\times14} = 14.2$N/mm^2

2.3Calculation of the stress concentration factor (Kt_1):-

$\beta1 = 0.0075\sqrt{\frac{1986\times10\times92.9}{14^2}}$

$\beta1 = 0.73$

$\tanh\beta1 / \beta1 = 0.86$

$K_{t1}=1+\frac{6\times10}{14}\times 0.86 = 4.69$

2.4Calculation of peak stress for P_1:-

$\sigma\rho1 = 4.69 \times 92.9 = 435.7$ N/mm^2

This peak stress is greater than 330 N/mm^2 so the ultrasonic interval cannot be determined

and the boiler should not be operated until one of the options in 5.11 has been implemented.

2.5 Resultant Hoop Stress:-

$\sigma_m \sqrt{92.9^2+14.2^2} = 93.97$ N/mm^2

Hoop Stress

$\sigma_n=P(\frac{r_i+\frac{t}{2}}{t})$
$= 1.51(\frac{993+7}{14}) = 107.85$ N/mm^2

For longitudinal:-
$\sigma=\frac{P\times d}{2t\times\eta}$
By using σ_m=93.97 N/mm^2
$93.97=\frac{1.51\times1986}{2\times14\times\eta}$
$\eta=\frac{1.51\times1986}{2\times14\times93.97}$ =1.13(113% A hypothetical data)
By using σ_m=107.85 N/mm^2
$107.85=\frac{1.51\times1986}{2\times14\times\eta}$
$\eta=\frac{1.51\times1986}{2\times14\times107.85}$ = 0.99(99%)

IV. CONCLUSION

The calculation of remaining life of boiler tubes on behalf of longitudinal thermal stress may give feasible result bit on behalf of efficiency calculation the result obtained through longitudinal stress are hypothetical and for the reliability of the operation of boiler tube at higher temperature and flow rate the remaining life analysis must give logical result in comparison with efficiency calculation. The calculation of efficiency on behalf of hoop stress value give more accurate result of efficiency on behalf of this paper the hoop stress values and formulas are to be used for calculation of efficiency for safe and reliable operation of modern thermal power plant.

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