HEAT TRANSFER ENHANCEMENT IN SUPER HEATER TUBE USING GEOMETRIC MODIFICATION

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Abstract

Super heater Tubes are made up of ASTM SA213 T22 Material at Thermal Power Station, sikka. For heat transfer enhancement, geometric model of tube is being prepared and imported in ANSYS. Convective film coefficients for steam and flue gases have been computed from given boundary condition.

After that, by keeping thickness 8mm constant and varying outer and inner diameter, three models were prepared and analyzed. For sake of convenience only partial tube was taken for analysis. From simulation results, heat flow can be calculated for all models. As heat flow through pipe increases, temperature gain for steam increases. For one of the model, heat flow value increases from original value. Using those geometric parameters, Geometric model of whole tube was made and analyzed. From results, it was found that temperature of steam has increased as compare to original temperature of steam. So that heat transfer enhancement is done by geometric modification.

Keywords*— *heat transfer enhancement, super heater, Geometric modification.

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1. INTRODUCTION

The primary elements of a boiler are the heat transfer surfaces, which transfer the heat from the flue gases to the water/steam circulation. The objective of the boiler designer is to optimize thermal efficiency and economic investment by arranging the heat transfer surfaces and the fuel-burning equipment. Heat transfer surfaces in modern boilers are furnaces, evaporators, super-heaters, economizers and air pre-heaters. The surfaces cover the interior of the boiler from the furnace to the boiler exhaust. Here two types of super heaters are used . I have analyzed convective super heater in which major heat transfers via convection, so that all equation related to convection can be applied.

2. LITERATURE REVIEW

To improve the performance of heat exchanging devices for reducing material cost and surface area and decreasing the difference for heat transfer thereby for reducing external irreversibility, lot of techniques have been used. Among different passive means to increase heat transfer coefficient, twisted tape inserts are promising. The secondary flow (swirl flow) generated by twisted tape effects fluid flow across the tape-partitioned tube, promotes greater mixing and higher heat transfer coefficients [1].

In this paper we explore the opportunity to maximize the production of power in steam-turbine power plant by properly configuring the hardware at the interface between the stream of hot gas produced by the furnace and the steam that circulates through the power producing cycle. The interface consists of four heat exchangers, super heaters and re heaters two in parallel flow other two in counter flow. The search for better configurations is based on constructional design, and consists of searching for the distribution of heat exchanger surface (number of heat exchangers, types, sizes) such that the total power output of the turbines is maximum, subject to fixed total size for the heat transfer surface, and fixed maximum allowable steam heat exchanger wall temperature[2].

A numerical investigation of annular finned tubes was performed using the computational fluid dynamics (CFD) software ANSYS FLUENT. The effects of fin spacing, fin height, fin thickness, and fin material on the overall heat transfer and pressure drop were determined for a single row of finned tubes in cross-flow. As fin height (hf) increased, the predicted overall heat transfer increased. The magnitude of the increase in overall heat transfer was larger for smaller fin spacing (s). A similar trend was seen with the pressure drop as the fin height increased the pressure drop increased with the magnitude of the increase being more significant for smaller fin spacing. For a fin spacing over tube diameter ratio of greater than one-third, the increase in pressure drop with fin height is very small[3].

When the tube metal is in contact with the steam at elevated temperature over a period of time, it may progress to form a layer of oxide scale. As a result, in the prolonged exposure, this phenomenon would lead to potential creep rupture problems. The improved ferrite steels that have very good creep strength such as grades 91 and 92 have been widely used for power plant applications. For example, a chromium–molybdenum–vanadium steel tube (SA213-T91) has been available in the market for two decades. The SA213-T91 material has better high temperature strength and oxidation resistance than those of the more widely used materials such as T11 and T22.[4]

The high temperature corrosion tests are performed on 20#steel, TP347H and super alloy C22. The high temperature corrosion behaviours of these superheated materials in the synthetic salt been investigated .For comparison, the column diagram has been obtained by mass loss. The scanning electron microscopy (SEM) with energy dispersive spectrometer (EDS) is used to compositions of the corrosion products. The results have shown that the super alloy C22 exhibits the high corrosion resistance. For analysis samples are been prepared for all the materials and the samples are cut into The corrosion resistances of super heater materials depend on the Ni/Cr content, which have positive influence on the corrosion behaviour .Super alloy C22 has the highest resistance, followed by TP347H and 20#steel.[5]

2.1 Nomenclature

Re	Reynolds Number=4m/πdμ
Pr	Prandtl number
Nu	Nusselt number
Q	Heat flow
q	Heat flux
h	Film coefficient
m	Mass flow rate
K	Thermal conductivity
d	Inner diameter of pipe
D	Outer diameter of pipe
µ	Absolute viscosity
c	Specific heat
R	Total thermal resistance in pipe
∆T	Temperature difference
L	Length of pipe
r	Radius of pipe

3. PRE PROCESSING AND FEA

All the dimensions of super heater are given and based on that model was prepared in Pro-engineer v 4.0 and the same is imported in ANSYS v 14.0. Meshing was done with SOLID90 element. Steam is considered forced convection due to turbulent flow inside the tube for cross flow heat exchanger [7] and based on this film co-efficient can computed from equation below,

$$\mathrm{hs} = 0.023\left(\frac{\mathrm{k}}{\mathrm{d}}\right)\mathrm{Re}^{0.8}\mathrm{Pr}^{0.4} \qquad (1)$$

Heat transfer outside the boiler tube is considered as forced convection due to cross flow of the hot flue gas over baretubes [7]

$$\mathrm{hg} = 0.33\left(\frac{12\mathrm{k}}{\mathrm{D}}\right)\mathrm{Re}^{0.6}\mathrm{Pr}^{0.33} \qquad (2)$$

Based on equation above film coefficients have been found out. These coefficients are applied as load in loading condition in ANSYS.

Following assumptions were made while doing analysis:

- Heat transfer is considered steady-state and one-dimensional.
- All the fouling resistance are being neglected.
- Change in material properties due to temperature change is ignored.
- Heat transfer through convection is considered, heat transfers by other modes are neglected.
- All dynamic effect due to change in mass flow rates are being neglected.
- Initial Condition for whole pipe is considered uniform 22 0C.
- Thermal contact Resistance for pipe is neglected.
- Internal heat generation is neglected.
- Conductivity of pipe is assumed to be isotropic.
- Changes in Nusselt number due to surface characteristics are neglected. Pipe is assumed to be smooth for analysis purpose.

3.1 Solution

Here for sake of convenience partial tube of 50 mm length is taken for analysis. Three different standard geometry of pipe is taken for analysis. Conductivity of pipe is considered isotropic as 34.606 W/m ^{0}C.

Various models taken for analysis are given below.
Tube-1:O.D 44.5 mm and I.D 28.5 mm
Tube-2: O.D 48.3 mm and I.D 32.3 mm
Tube-3: O.D 42.4 mm and I.D 26.4 mm

Tube 1 is original tube and tube 2 and 3 is modified standard tube.

For tube 1, based on initial conditions film coefficients were found which are 2861.50 W/m^2 ^{0}C and 269.27 W/m^2 ^{0}C for steam and flue gases respectively.

As area of pipe changes, Velocity for flue gases as well as for Steam changes. Based on updated velocity, Reynolds number changes and hence film co-efficient changes. Here based on modified inner diameter and outer diameter, film co-efficient was calculated by equation 1 and 2.Those modified film co-efficient are applied as load in each specific case.

For tube 2, based on initial conditions film coefficients were found which are 2227.5 W/m^2 ^{0}C and 271.5 W/m^2 ^{0}C for steam and flue gases respectively for steam and flue gases respectively.

For tube 3, based on initial conditions film coefficients were found which are 3266.5 W/m^2 ^{0}C and 268.68 W/m^2 ^{0}C for steam and flue gases respectively. After certain time temperature of pipe, steam and flue gases remains constant so steady state analysis is done.

These coefficients are applied as a load for analysis and based on this Heat flow was found. From simulation it is possible to find heat flux and the relation between heat flow and heat flux is given by,

$$\text{Heat flow} = \text{Heat flux X } 2\pi \text{ r L} \qquad (3)$$

Where,
r= radius of cylinder
L= Length of pipe

4. RESULTS AND DISCUSSION:

4.1 Results for Heat Flow:

Here it is noted that heat flow can be calculated at outer surface as well as at inner surface. Value of heat flow is found to be same at inner and outer surface.

Heat Flow for Tube-1

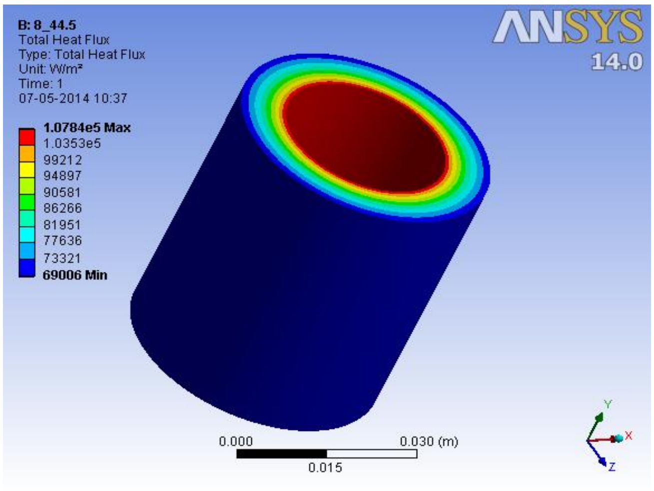

Fig.4.1.1 Heat flux for Tube-1

Based on value of heat flux, heat flow was calculated and it was found to be 482.31 W.

Heat Flow for Tube-2

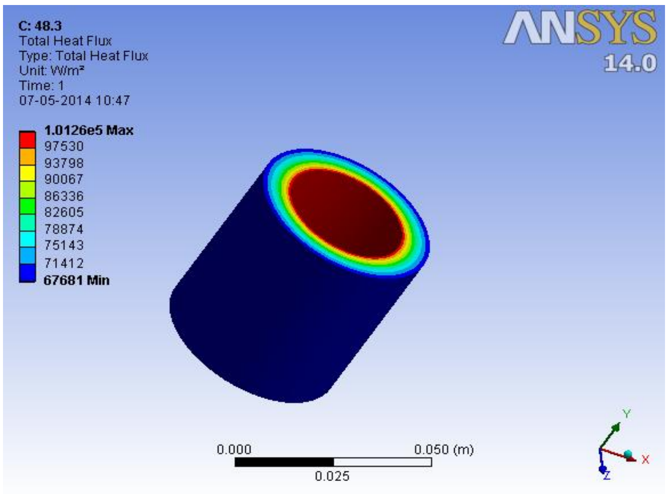

Fig.4.1.2 Heat flux for Tube-2

Based on value of heat flux, heat flow was calculated and it was found to be 515.49 W.

Heat Flow for Tube-3

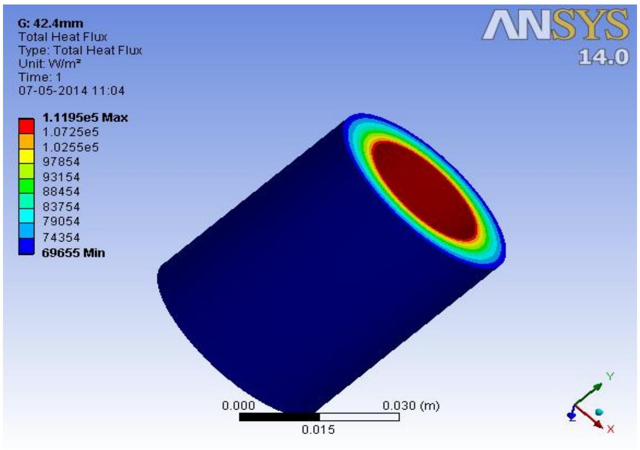

Fig.4.1.3. Heat flux for Tube-3

Based on value of heat flux, heat flow was calculated and it was found to be 463.91 W.

Comparasion:

Table 1 Heat Flow Calculation

Tube No	Heat-flow(Q)(W)	Conclusion
1	482.31	Original
2	515.49	Modified
3	463.91	Modified

As we can conclude from table above it is possible to Enhance heat transfer from original tube we need to change inner diameter of pipe from 28.5 mm to 32.3 mm and outer diameter to 48.3 mm from 44.5 mm.

4.2 Result of Temperature Distribution

Here temperature distribution for pipe-1 and pipe-2 is given.

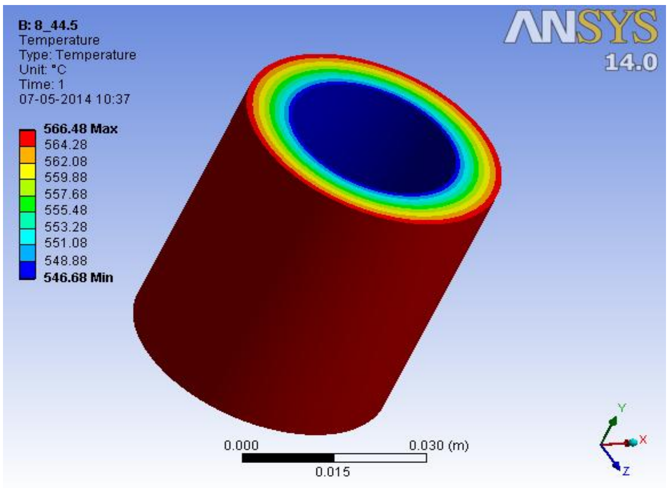

Fig.4.2.1 Temperature distribution for Tube-1

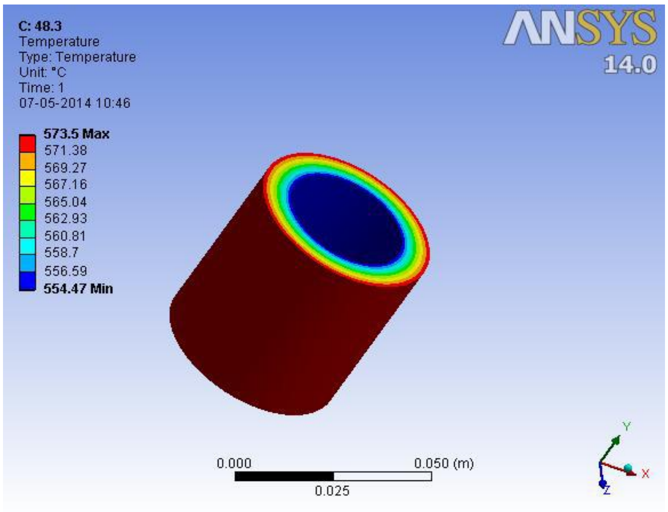

Fig.4.2.2. Temperature distribution for Tube-2

As we can see from temperature distribution diagram, temperature at inner side of tube-2 has increased as compare to temperature at inner side for pipe-1. At inner side of pipe, steam is flowing and its temperature has increased so overall heat transfer is increased.

5. VALIDATION

For original tube validation is given in terms of temperature distribution. As we can see from temperature distribution for pipe-1, inner side temperature of tube is 544.68 ^{0}C. At inner side steam is flowing so if we neglect thermal contact resistance then steam temperature is taken as 544.68 ^{0}C and actual temperature of steam is 540 ^{0}C as observed in plant.

For modified tube, validation is given in terms of heat flow. As we know that Heat flow through cylinders due to conduction and convection [6] is given by following formula,

$$Q = \frac{2\pi L\Delta T}{\frac{1}{hiri}+\frac{1}{horo}+\frac{\ln(r2/r1)}{k}} = \frac{\Delta T}{R}$$

Where, Q = Heat flow through the pipe.
L = Length of Pipe
ΔT = Temperature difference between inner and outer side fluid.
hi =Film coefficient at inner side of the pipe.(Steam)
ho = Film coefficient at outer side of the pipe.(Flue gases)
ri = Inner radius of pipe.
ro = Outer radius of pipe.
k = Thermal conductivity of pipe material.
R= Total thermal resistance.

For modified tube i.e. Tube no. 2, Film coefficients for steam and flue gases were 2227.5 and 271.5 W/m2 ^{0}C and all other parameters were already known. Based on these film Coefficients and using above equation Heat flow was found to be 515.33 W.

Heat flow for tube 2 is already known to us by FEA method and it was 515.49 W. here we can see that heat flow values are almost same. So all the boundary conditions are right and based on those condition, analysis is right.

6. COMPARISON

For the original pipe, average temperature at inner side of pipe was 544.79 0C and for modified pipe average temperature at inner side of pipe is 552.19 0C. Temperature has increased because amount of heat flowing in modified tube is more as compare to original tube. Now I have calculated percentage increases in heat transfer.

$$Q = mC\Delta T$$

Where,
Q = Heat transferred
m= mass flow rate of steam
C= specific heat of steam
ΔT =Temperature difference of super heater outlet and super heater inlet

Here based on value of temperature difference, heat transfer was calculated .Mass flow rate and specific heat assumed to be constant. Based on above equation it was found that heat transfer has increased by 20. 68 %.

7. CONCLUSIONS

By changing geometry of tube heat flow through pipe is increased and hence heat transfer enhanced. In order to increase heat flow, inner diameter of tube should be increased. As inner diameter increases surface area which is in contact has increased and hence heat transfer has increased. By calculation if we increase inner diameter of tubes by keeping thickness constant, then overall resistance in heat flow has decreased and hence heat transfer has increased. If we increase inner diameter then speed of steam will decrease. So steam will take more time to gain its temperature so heat transfer can be increased if we keep mass flow rate constant. By varying inner diameter it is possible to get different heat transfer mechanisms which enhance heat transfer. In same way it is possible to increase heat transfer by changing thickness of pipe but it creates problems because at inner side of the tube pressurised steam is flowing. So it is feasible to enhance heat transfer by keeping thickness constant.

For flue gases, if we change outer diameter of pipe then it will create less effect on heat transfer because it’s film co-efficient is less. So it will not create that much effect as compare to inner diameter.

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