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# Modifying the Fin Design and Improving Thermal Efficiency of Super Heater Tube

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*Abstract*— ***Super heaters are among the most important components of boiler and have major importance due to this operation in high temperature and pressure. Fins on the super heater tubes plays a vital role to increase the heat transfer rate. A small unit of improvement in the design of the fins in the super heater gives a large value of positive impact on the performance of super heater in the steam generation system. The core concept of the project is to redesigning the circular shaped fin which is existing in the former super heater tubes to rectangular cross- section helical shaped fin , in order to increase the heat transfer rate (Q) with less fuel (coal) consumption due to increasing the total surface area of the conducting material and as fins. Such redesign is aimed at minimal change in weight and cost of super heater, with maximum positive impact on the overall efficient in the steam power generation.***

***The material used for the fins in our project is SA106GrA which is having a thermal conductivity of 51 w/m k and good thermal properties which are required to increase heat transfer rate. The parametric model is created in 3D modeling software AUTOCAD and the results are calculated by using mathematical methods (Heat transfer Methodology).***

## I. Introduction

Today, most of the electricity produced throughout the world is from steam power plants. However, electricity is being produced by some other power generation sources such as hydropower, gas power, bio-gas power, solar cells, etc. The environmental impact of electricity generation is significant because modern society uses large amounts of electrical power. This power is normally generated at power that converts some other kind of energy into electrical power. Each system has advantages and disadvantages, but many of them pose environmental concerns. It is necessary to reduce energy consumption at a global level in order to solve the problems concerning environment and energy, such as global warming, and most of all, it is required for the developing countries to take energy saving measures, where economic and population present a considerable growth.

Therefore, finding the means of improving the existing system of power generation has become very important project and research for many industries consuming and distributing larger scale of electricity for production processes and to operate modern appliances both at work and at domestic levels. Recently, many industries using their own in-house power generation units, have come up or implemented initiatives or several projects, to increase the power generation within their in-house power generation system, especially from biomass fuels, by installing back -pressure or condensing /extraction steam turbine generators, that take advantage of generating steam at higher pressure and temperature than needed for the process usage.Such projects have increased both the biomass fuel firing capacity and raised the operating steam pressure and temperature of existing boilers to make power generation more economical.

## II. Experimental setup

### *2.1 SUPERHEATER*

A super heater is a device found in steam boilers that is used to convert wet, saturated steam into dry steam. Super heaters are a very beneficial part of the steam cycle, because dry steam contains more thermal energy and increases the overall efficiency of the cycle. Dry steam also is less likely to condense within the cylinders of a reciprocating engine or the casing of a steam turbine. Boiler super heaters can be found in three varieties: radiant super heaters, convection super heaters and separately fired super heaters.

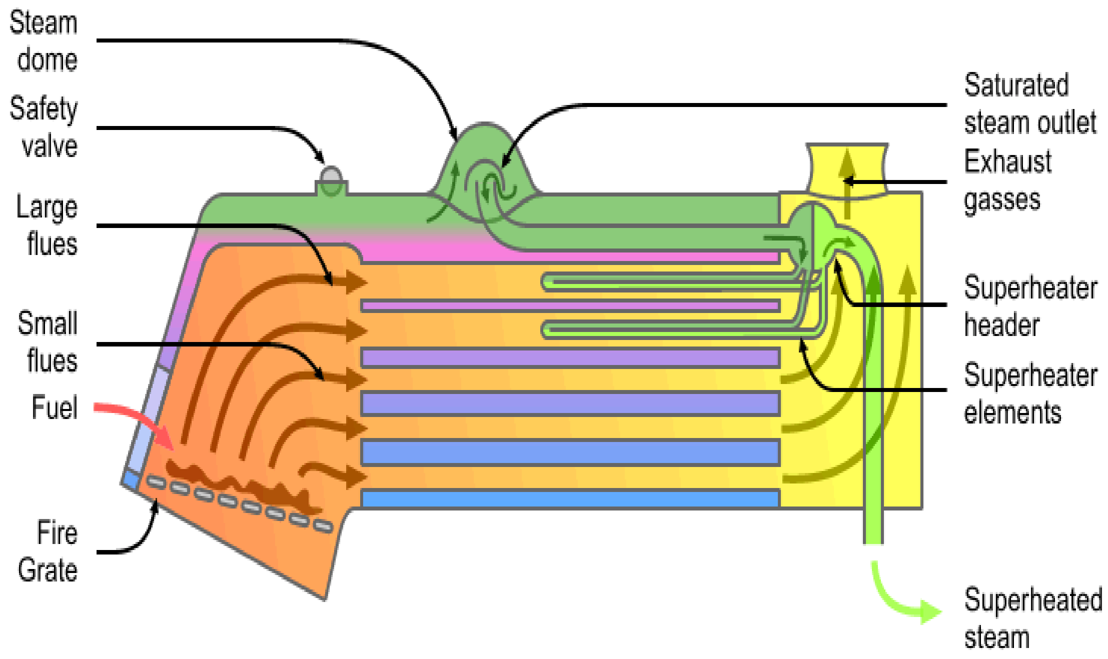



*Fig. 2.1 Superheater location in steam power plant*

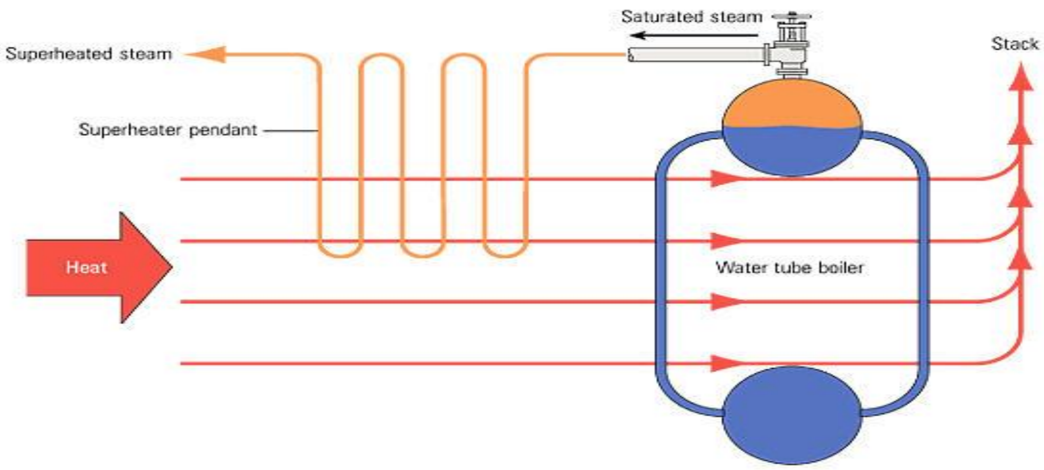



*Fig. 2.2 Steam flow process*

*2.2. Heat Transfer Mechanisms*

Heat transfer mechanisms can be grouped into 3 broad categories:

*2.2.1Conduction*

Regions with greater molecular kinetic energy will pass their thermal energy to regions with less molecular energy through direct molecular collisions, a process known as conduction. In metals, a significant portion of the transported thermal energy is also carried by conduction-band electrons.

*2.2.2 Convection*

When heat conducts into a static fluid it leads to a local volumetric expansion. As a result of gravity-induced pressure gradients, the expanded fluid parcel becomes buoyant and displaces, thereby transporting heat by fluid motion (i.e. convection) in addition to conduction. Such heat-induced fluid motion in initially static fluids is known as free convection.

$$Q=h A_s (T_f - Ts)$$

*2.2.3 Radiation*

All materials radiate thermal energy in amounts determined by their temperature, where the energy is carried by photons of light in the infrared and visible portions of the electromagnetic spectrum. When temperatures are uniform, the radiative flux between objects is in equilibrium and no net thermal energy is exchanged. The balance is upset when temperatures are not uniform, and thermal energy is transported from surfaces of higher to surfaces of lower temperature.

## 1. ANALYSIS

Heat transfer between a solid surface and a moving fluid is governed by the Newton's cooling law:

$$\mathbf{Q = hA(T_s - T_\infty)}$$

whereTs is the surface temperature and $T_\infty$ is the fluid temperature.

The means to increase the heat transfer from the gases to the element in the lower temperature zones are;

1) Increase the temperature difference ($T_s - T_\infty$) between the surface and the fluid.
2) Increase the convection coefficient h. This can be accomplished by increasing the fluid flow over the surface since h is a function of the flow velocity and the higher the velocity, the higher the h. Example: a cooling fan.
3) Increase the contact surface area A. Example: a heat sink with fins.

Many times, when the first option is not in our control and the second option (i.e. increasing h) is already stretched to its limit, we are left with the only alternative of increasing the effective surface area by using fins or extended surfaces. Therefore, the following theoretical analysis is being studied to improve the fin efficiency on the superheater tubes manufactured at BHPV by our project group as detailed here after.

*3.1 Existing Fin Design at BHPV (Circular Fins)*

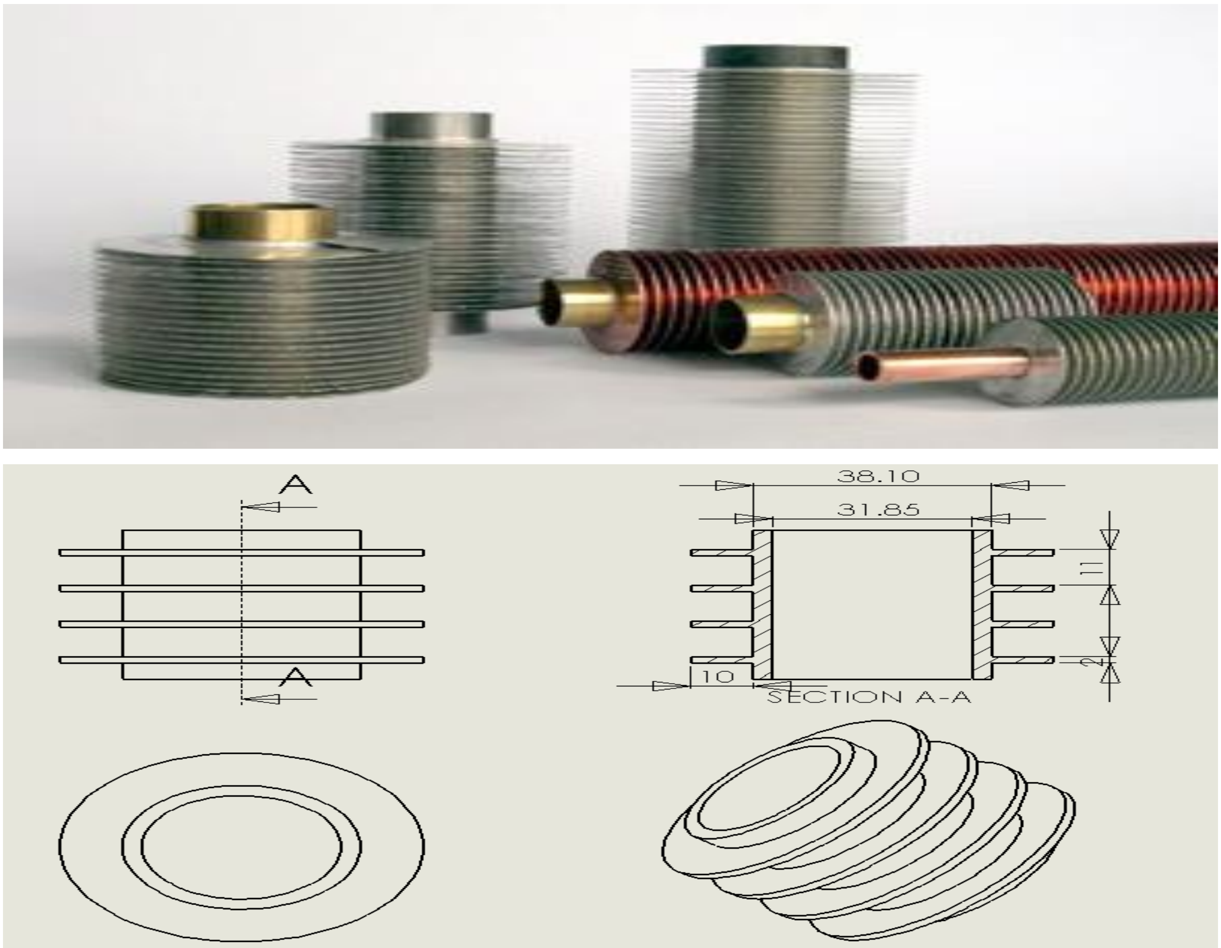



*Fig. 3.1Existing fin (circular plate)*

*3.1.1 Specifications*

Outer Diameter of the Pipe ($D_O$) =38.100 mm

Inner Diameter of the Pipe ($D_I$) =31.850 mm

Thickness of the Pipe (t) =03.125 mm

Thickness of the Fin (t) =02.000 mm

Width of the Fin (w) = 10.000 mm

Pitch of the Fin (p) =11.000 mm

Specimen = SA106GrA

*3.1.2 Properties of SA106GrA*

Thermal conductivity (k) =51W/m.K

Density (ρ) =7815 kg/$m^3$

Thermal Diffusivity (α) =$14.80\times10^{-6}m^2/s$

Specific heat(c) =445 J/kg K

*3.1.3 Working conditions*

Temperature of flue gasses at inlet $T_{(i)\ flue)}=1077^0$ C

Temperature of steam at inlet of super heater $T_{(I)\ Steam}=475^0$ C

Temperature of steam at outlet of super heater$T_{(o)\ Steam}=540^0$ C

Temperature of flue gasses at outlet $T_{(o)\ Flue}=932.4^0$ C

Flow velocity of Flue gasses $v_f$ =10-18 m/s

Flow velocity of steam $v_s$=10-25 m/s

Average velocity of flue gasses $v_{f\ (avg)}$=14 m/s

Average velocity of steam $v_{s\ (avg)}$=17.5 m/s

*3.1.4 Properties of flue gasses at 10770 C temperature*

Density (ρ) =0.261 kg/$m^3$

Thermal Diffusivity (α) =$335.85\times10^{-6}m^2/s$

Specific heat(c) =1319.09 J/kg K

Thermal conductivity (k) =0.1155073 W/m.K

Absolute viscosity (μ) =$50.1518\times10^{-6}$ Ns/$m^2$

Kinematic viscosity (ϑ) =$191.856\times10^{-6}m^2/s$

Prandlt number Pr=0.5723

*3.1.5 Properties of steam at superheater temperature and pressure:*

Density (ρ) =92.80 kg/$m^3$

Thermal Diffusivity (α) =$0.081\times10^{-6}m^2/s$

Specific heat(c) =12351 J/kg K

Thermal conductivity (k) =0.09304 W/m.K

Absolute viscosity (μ) =$25.06\times10^{-6}$ Ns/$m^2$

Kinematic viscosity (ϑ) =$0.272\times10^{-6}m^2/s$

Prandlt number Pr =3.35

In forced convection formulae

**Reynold"s number Re $_D$=** $\dfrac{\rho \times D_0 \times v}{\mu}$

**Nussult number Nu$_D$=** **0.25× (Re $_D$) $^{0.6}$ × (Pr) $^{0.38}$** $\left(\dfrac{\Pr f}{\Pr w}\right)^{0.25}$

(Or)

**Nu$_D$=** $\dfrac{h \times D_0}{k}$ **=C (Re $_D$)$^n$ Pr$^{0.33}$**

Convective heat transfer coefficient **h= Nu$_D$.k/D**

**($h_A$,$h_B$**Will be calculated using above procedure)

**$h_A$**=convective heat transfer coefficient at outside of superheaters

**$h_B$** =convective heat transfer coefficient at inside of superheaters

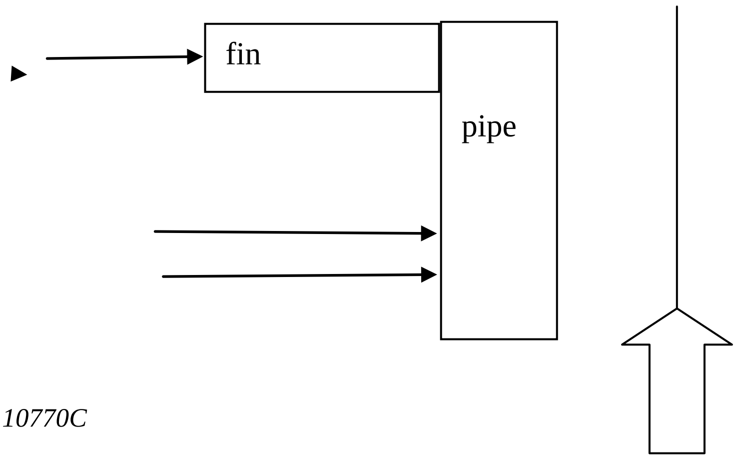
Heat transfer Q = $\dfrac{\Delta T}{R}$

R = Resistance of material

$$\textbf{1)}\quad \textbf{R=}\ \frac{1}{2\pi L}\left[\frac{1}{h_A \times r_1}+\frac{1}{k}\ln\left(\frac{r_1}{r_2}\right)+\frac{1}{k}\ln\left(\frac{r_2}{r_3}\right)+\frac{1}{h_B \times r_3}\right]$$

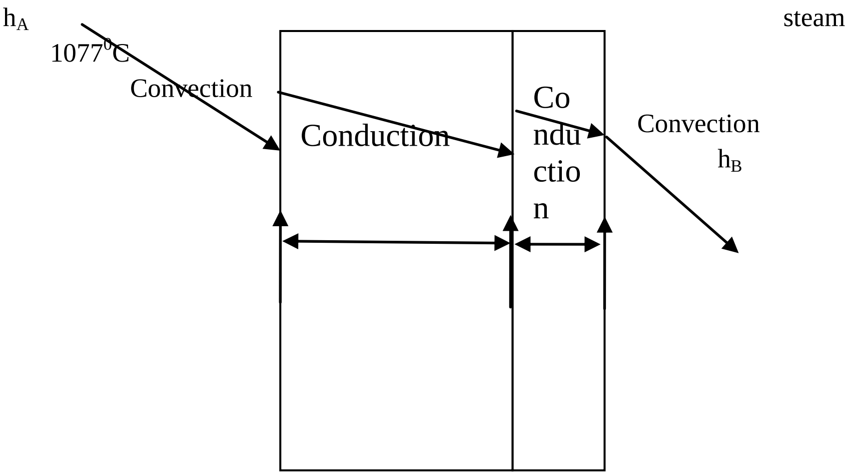
Let's consider the unit length of the super heater

1077$^0$C




*7.1.6 HEAT TRANSFER AT FINNED AREA 10770C*

Heat distribution shown as:



$Re_D = \dfrac{\rho \times D_0 \times v}{\mu} = \dfrac{0.261 \times 14 \times 0.0581}{50 \times 10^{-6}}$ =4233.25

If $Re_D$ is $1000 < Re_D < 200000$ (C=0.193,
n=0.618)

$Nu_D = \dfrac{h \times D_0}{k}$ **=C ($Re_D$)$^n$ $Pr^{0.33}$**

$=0.193 \times (4233.25)^{0.618} \times (0.5723)^{0.33}$

=27.98

$\dfrac{h \times D_0}{k}$ =27.98 =>$h_A$ =55.67 W/㎡ K

Find $h_B$ with above procedure at superheated steam conditions

$Re_D = \dfrac{\rho \times D_{Ipipe} \times v}{\mu} = \dfrac{92.80 \times 17.5 \times 0.03185}{22.50 \times 10^{-6}}$ =1993911.11

If $Re_D$ is $200000 < Re_D < 2000000$ (C=0.076,
n=0.37
m=0.7

$Nu_D = \dfrac{h \times D_{Ipipe}}{k}$ **=C ($Re_D$)$^m$.$Pr^n$.($\dfrac{Pr_\infty}{Pr_w}$)$^{0.25}$** [$Pr_{00}$=3.35 ,$Pr_w$=15.05]

$=0.076.(1993911.11)^{0.7}.(3.35)^{0.37}.(3.35/15.05)^{0.25}$

=2097.73

$\dfrac{h \times D_{Ipipe}}{k}$ =2097.73 =>$h_B = 2097.73 \times \left(\dfrac{0.09304}{0.03185}\right)$

$h_B$=6125.88 W/㎡ K

Resistance of the total section:

R =Resistance of material

**2)** **R=** $\dfrac{1}{2\pi L}\left[\dfrac{1}{h_A \times r_1} + \dfrac{1}{k}\ln\left(\dfrac{r_1}{r_2}\right) + \dfrac{1}{k}\ln\left(\dfrac{r_2}{r_3}\right) + \dfrac{1}{h_B \times r_3}\right]$

$R = \dfrac{1}{6.28 \times 0.002} \times \left[\dfrac{1}{55.67 \times 0.02905} + \dfrac{1}{51}\ln\left(\dfrac{0.02905}{0.01905}\right) + \dfrac{1}{51}\ln\left(\dfrac{0.01905}{0.015925}\right) + \dfrac{1}{6125.89 \times 0.015925}\right]$

R= 47.89

$\dfrac{\Delta T}{R}$ =Q =>Q = 501.8/ 47.89 (Tem. Diff =501.8$^0$C)

Heat transfer in finned area Q=10.889W

*3.1.7 Heat Transfer at un-finned area*

$$h_A = Nu.\frac{k}{D_{OPIPE}} = 21.59 \times \frac{0.1156}{0.0381}$$

$h_A = 65.5$ W/㎡K at outer surface Nu= $21.59$

$h_B$=6125.88W/ ㎡.K

$$R = \frac{1}{2\pi L}\left[\frac{1}{h_A \times r_2} + \frac{1}{k}\ln\left(\frac{r_2}{r_3}\right) + \frac{1}{h_B \times r_3}\right]$$

$R = 17.69[0.8014 + 3.51\times10^{-3} + 0.01025]$

$R = 14.42$

$Q = \frac{\Delta T}{R} = 501.8/14.42 = 34.79$ W

Actual total Heat transfer through unit area is $= (34.79 + 10.889) = 45.68 W$

Actual heat transfer through a module is =45.68×1000×40 =1827200W

**=1.8MW**

*3.2 Proposed Fin Design (Helical fins)*

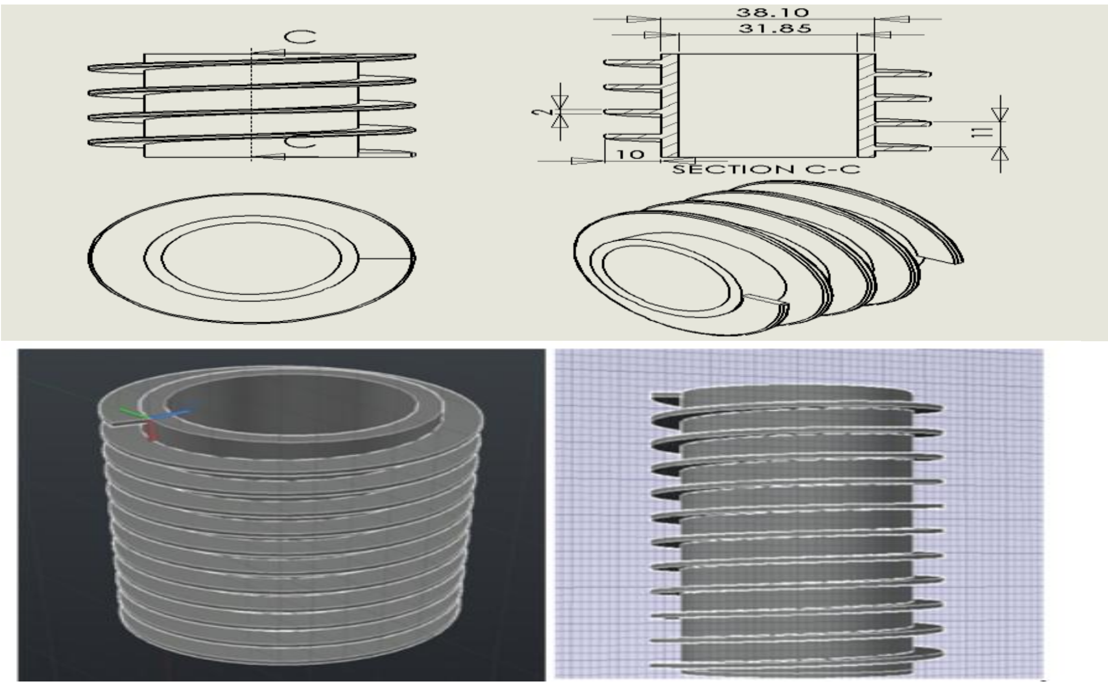



*Fig. 6.2 Proposed fin design (spiral Fin)*

*7.2.1 Specifications*

Outer Diameter of the Pipe ($D_O$) =38.100 mm

Outer Diameter of the Fin ($D_{O\ fin}$)=58.100 mm

Thickness of the Fin (t =02.000 mm

Pitch of the Fin (p =11.000 mm

Specimen = SA106GrA

Width of the Fin ($r_2$-$r_1$) = (L)=10.000 mm

*CALCULATIONS*

$L_c = L + \frac{t}{2} = 10+1 = 11$ mm

$r_{2c} = r_1 + L_c = 19.05+11 = 30.05$ mm

$A_m = (r_{2c} - r_1)\, t = (30.05-19.05) \times 2$

$= 22\ mm^2$

$Q_{MA} = 2\pi h(r_2^2 - r_1^2)\theta$

$Q_{total} = Q_{fin} + Q_{un\ finned}$

$= \eta_f . A_f h\ \theta + (A - .A_f) h\ \theta$

$= h\ \theta[A - (1-\eta_f) A_f]$

A = total area of the fin & un finned surfaces

h = convective heat transfer coefficient

$\Theta = \{(1077+932.4)/2+507.5\}/2 = 501.8^0 C$

A = area of the spiral fin & area of pipe exposed

$$h = \frac{1}{\frac{1}{h_A} - \frac{1}{h_B}}$$

$$\frac{1}{\frac{1}{55.67} - \frac{1}{6125.88}} = 56.18 W/m^2 K$$

$Q_{MA} = h.A\theta = 56.18 \times 0.04516 \times 501.9 = 127.336$ W

$Q_{fin} = Q_{MA} \times \eta f$

*CIRCULAR FINS*

Calculate $L_c^{1.5}\sqrt{\frac{h}{k.A_m}} = 0.0845$ (A = area of fin + area of un finned surface)

$A = \pi \times r^2 + \pi(r_{2c}{}^2 - r^2) = 1079.706 + 3385.548 = 4465.25 \times 10^{-6}\ m^2$

From the efficiency comparison chart for circular fins $\eta_f = 96.52$ %

$Q_{fin} = Q_{MA} \times \eta f = 0.9652 \times 127.33 = 122.89$ W

Heat transfer in total module Q =122.89×1000×40=4915600W

*HELICAL FINS*

$L_c^{1.5}\sqrt{\frac{h}{k.A_m}} = 0.0562$ spiral fin area $A = 4516 \times 10^{-6}\ m^2$

From the efficiency comparison chart for spiral fins $= 98.66$ %

$Q_{fin} = Q_{MA} \times \eta f = 0.9866 \times 127.33 = 125.62 W$

Heat transfer through spiral finned tubes of a module =125.62×1000×40

=5024800W

**ECONOMIC CALCULATIONS**

Amount of extra heat transfer per unit area is = 2.72W

Total heat transfer in a module =1000×40×2.72=108800W

=108.8kw

Cost of production to generate 1KW heat (Rs) =1.28

Amount of money we are saving per unit time for one module

=1.28×108.8

=139.26/-

*RESULT*

TABLE 4.1 COMPARISON OF FIN

| **Description** | **Existing Fin** | **Proposed Fin** |
|---|---|---|
| Fin Type | Circular Fin | Helical fin |
| Heat Transfer Surface area | 0.004465 m2 | 0.004516 m2 |
| Heat flow | Continuous up to one pitch | Continuous to full length |
| Heat Transfer | 122.89W | 125.62W |
| Fin Efficiency | 96.52 % | 98.66% |

It can also be shown that higher the fin to tube surface, the tube wall and fin tip temperatures will be higher as also the gas pressure drop. Higher the fin density, lower the heat transfer coefficient and vice versa as seen by the chart below for heat transfer in finned tubes.

## III. CONCLUSIONS

It is desirable to have a lower pinch point for maximum steam generation rate and effective flue gas energy utilization. Spiral fins offer compact heat recoveries because of high gas turbulence and improved exposed contact surface. Fin spiraling improves gas penetration, which improves flow velocity resulting in higher heat transfer. In the redesigned fin from circular plate to helical plate fin, the increase in fin efficiency is found as 2.14% in a unit pitch of fin, with an increase in heat transfer rate as 125.62w from 122.89w

Higher fin density for any surface does not make any difference in heat transfer analysis as it acts as solid fin design. At definite tube diameter the pressure drop is Minimum and offers best heat recovery under all design conditions. In the present case it is obtained at 38.1m. As waste heat recovery systems are widely used in power, petrochemical and refinery operations as the steam generators therefore there is demand of the finned tube as it offers high volume to surface ratios and high heat transfer coefficients. It fetches nearly Rs140 per unit time in each module.

While using flue gases for heating the super heater requires periodical cleaning of super heater fins with the proper cleaning technology as the flue gases contain carbon dioxide as the main constituent. Other than this, the fin designs can be further improved for obtaining higher thermal efficiency of steam power plants. One of the recommendations can be designing the plain helical fin further to serrated spiral fin, which may yield higher and smoother gas flow rate. The serrated spiral fin design also helps in reducing the overall weight of the system, thereby reducing the cost of fin material too, making the system more economical.

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