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# Thermodynamic Evaluation of WHRB for it's Optimum performance in Combined Cycle Power Plants

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## ABSTRACT

Combined cycle power plants are being extensively used in view of their capability of offering high specific power output and thermal efficiency for same fuel consumption compared to other thermal power plants. Therefore, the studies for optimization of different systems in combined cycle power plant are of great significance. Waste heat recovery boiler (WHRB) being the interface between the topping cycle and bottoming cycle becomes one of the critical components. Present study is undertaken for thermodynamic analysis of waste heat recovery boiler for design change from spiral fin type to segmented fin type in 663 MW capacity gas/steam combined cycle power plant. Results obtained for combined cycle power plant with segmented fin type WHRB have been compared with the actual plant data of combined cycle power plant with spiral fin type WHRB. The conclusions presented in the study are useful for power plant designers.

**Keywords:** Waste Heat Recovery Boiler, Extended Surfaces, Pinch Point, Approach Point.

**Nomenclature**

Q= Rate of heat transfer, KW
U= Overall heat transfer coefficient, $W/m^2K$
$r_{total}$ = Total resistance available for heat transfer, $m^2K/W$
$r_{in}$= Internal or Tube side resistance available, $m^2K/W$
$r_{out}$ = External or flue gas side resistance, $m^2K/W$
$h_i$= Internal or water/steam side heat transfer coefficient, $W/m^2K$
$h_e$ = External or tube side resistance, $W/m^2K$
$R_e$= Reynolds number
$P_r$= Prandtl number
$a_f$= Fin surface area available per unit length, $m^2/m$
$A_o$= Total outside surface area available per unit length, $m^2/m$
$a_o$= Outside tube surface area available per unit length, $m^2/m$
$h_{iw}$= Internal heat transfer for water phase or tube side heat transfer coefficient, $W/m^2K$
$h_{iv}$= Internal heat transfer for vapor phase or tube side heat transfer coefficient, $W/m^2K$
$\mu_b$= Absolute viscosity at bulk temperature, Ns/m
$\mu_w$= Absolute viscosity at wall temperature, Ns/m
$T_b$= Bulk temperature of vapor; K
$T_w$= Wall temperature of vapor; K
$m_l$= Mass fraction of liquid
$m_v$= Mass fraction of vapor
$C_p$= Heat capacity of fluid at bulk temperature, kJ/kg K

## I. Introduction

Combined cycle based power plants make use of the gas turbine cycle and a steam cycle operating synergistically to achieve, efficient power generation. The gas power cycle has high temperature and it rejects heat with exhaust gases which is used for steam generation in waste heat recovery boiler (WHRB) for steam cycle. WHRB - utilizes the energy of exhaust gas of the gas turbine and produces the steam for steam turbine. It is a critical component of combined cycle power plant. In view of criticality of WHRB its' efficient design and operation is quite essential for improvement of overall system efficiency and power output.

For a given gas turbine exhaust the temperatures at inlet and exit of WHRB determine the amount of energy recovered from the flue gas stream whenever heat exchange occurs between gas/water. The heat transfer coefficient at gas side is lower than at the water side. There are many possibilities to improve a low heat transfer coefficient, increasing the surface area or turbulence. B.V. Reddy and C.J.Butcher [ 15 ] has discussed second law second law analysis of heat recovery steam generator based upon the power generation system..V. Ramprabhu and R.P.Roy[ 14 ] has developed a model for combined cycle power plant. B.Mahmood and B.Rasool [ 18 ] has optimized a fire tube heat recovery steam generator for cogeneration plant through genetic algorithm. E.Martinez [ 20 ] comparative analysis of heat transfer and pressure drop in helically segmented finned tube heat exchangers.

In the present study comparison has been carried out for different types of extended surfaces in WHRB. Detailed mathematical modeling and analysis is presented for segmented fin in the WHRB for its various sub-components. The optimization has been done on the basis of maximum heat recovery with minimum pressure drop for a given heat flow.

## II. Thermodynamic Modeling

In a WHRB the heat transfer rate between the tube and high density water on inside of the tube is far greater than the heat transfer rate between the tube and the low density flue gas passing outside. The outside rate is controlling rate and is responsible for overall heat transfer rate. Therefore in order to increase the rate of heat exchange in WHRB tubes the surface area on outside is extended by adding additional fins or "Finning". Extended surfaces are used in the super heater, evaporator, and economizer and make the WHRB design very compact. The use of fins increases the tube wall and fin tip temperature and heat flux inside the tubes. When tube side coefficient is low, the temperature drop across the tube-side film is high, resulting in high tube wall and fin tip temperature.

The design of compact heat recovery systems requires the knowledge of heat transfer. The empirical relations of Escoa manual [19] are used to analyze WHRB. Segmented finned tubes are used for obtaining compact heat recoveries. The WHRB has been mathematically modeled and presented ahead. The input details of an existing WHRB have been obtained from National Thermal Power Corporation (NTPC) [Table-1]

WHRB is basically an energy recovery heat exchanger that recovers heat from hot exhaust gas from gas turbine and produces steam that can be used to drive the steam turbine.

II.1 **Energy balance** along the WHRB

$$Q=UA\Delta T_m = m_g(Cp_{g1}T_1-Cp_{g2}T_2) = m_s(h_{s1}-h_{s2}) \quad \ldots (1)$$

For $r_{total}$ , total thermal resistance, the **overall heat transfer coefficient** can be-given as

$$U=1/r_{total} \quad \ldots (2)$$

$$r_{total} = r_{in} + r_{out} + r_{wall}$$

$$r_{in}=(1/h_i+R_{fi})(A_o/A_i)$$

**II.2 Inside film (Tube side) heat transfer coefficient**

For water (liquid) flow with Re≥10,000

$$h_{iw} =0.023(k/di)Re^{0.8}Pr^{0..33}(\mu_b/\mu_w)^{0.14}$$

For steam (vapor) flow with Re≥15,000

$$h_{iv} =0.021(k/di)Re^{0.8}Pr^{0.4}(T_b/T_w)^{0.5}$$

for two phase flow,

$$h_i= m_l h_{iw} +m_v h_{iv}$$

**II.3 Outside film (Gas side) heat transfer coefficient**

Effect of radiation heat transfer is neglected due to short tube spacing and the heat transfer coefficient is dependent mainly upon the convection heat transfer.

$$r_{out} =1/h_e$$

$h_e$ is effective heat transfer coefficient

$$h_e = h_o(e*a_f+a_o)/A_o$$

$h_o = 1/(1/ h_c+r_{fo})$, average outside heat transfer coefficient

$$h_c = j\, G_n\, c_p\, (k_b / c_p\, \mu_b)^{0.67}$$

Where j is the Colburn factor.

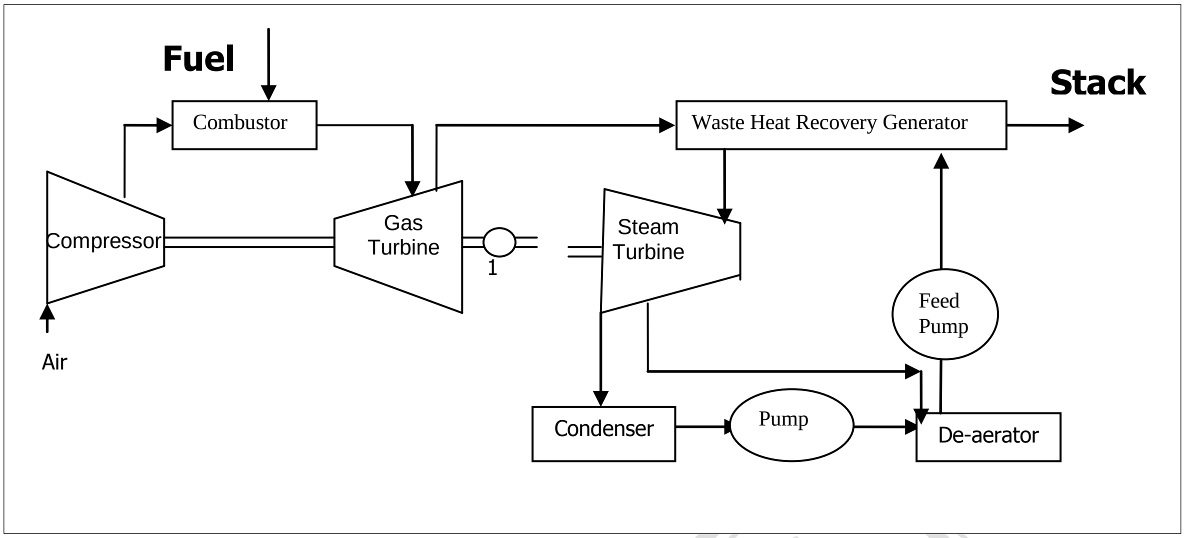


**Figure 1: Simple schematic of a combined cycle gas turbine**

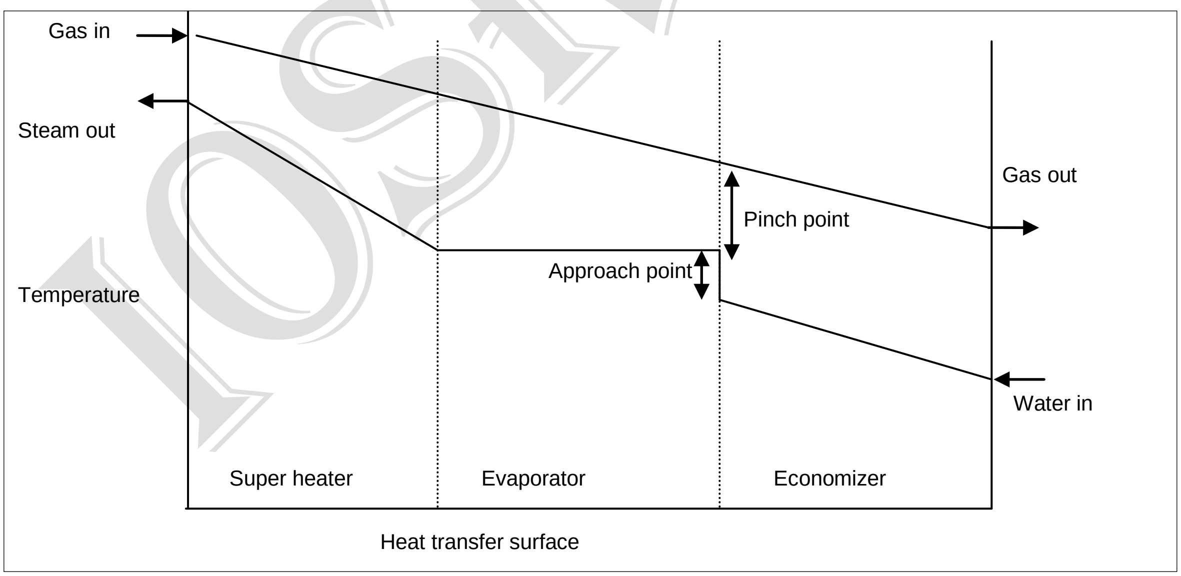


**Figure 2: Temperature profile in a Waste Heat Recovery Boiler**

II.3 **Pressure Drop**

Gas pressure drop can be calculated by the empirical relation given by the Escoa as given below

$$\Delta P_g=(f_o+A)G_o^2N_r/(\rho_b) \qquad \ldots(3)$$

Where $f_o$ and $\rho_b$ are the friction factor and density of gas at average outside temperature, respectively. A is defined as:

$$A= (1+B^2)^* \rho_b/ 4N_r$$

B is the relation between gas free area and cross sectional area calculated on the basis some extended surfaces design for various number of tube wide, length, pitch, number of fins/m, segmented fins and thickness.

## III. Results and Discussion

Thermodynamic modeling has been used for obtaining different parameters of WHRB. The input parameters considered for the study are detailed in TABLE 1 along with the optimized parameter values.

Table 1**:** Input parameters for WHRB design

| **Description** | **Variants of existing plant** | **Variants of optimized plant** |
|---|---|---|
| Diameter of tube (mm) | 31.8 | 50.8 |
| Thickness of the tube (mm) | 2.9 | 3.05 |
| Tube type | Spiral finned | Serrated fins |
| Fin density (number of fin per m) | 158 | 237 |
| Thickness of fin (mm) | 0.8 | 1.25 |
| Height of the fin (mm) | 11 | 19 |
| Tube bundles | Staggered | Staggered |
| Design Temp (deg C) | 495 | 495 |
| Pressure (Bar) | 60 | 60 |

Based on the thermodynamic modeling the WHRB parameters as obtained for optimum performance of WHRB are discussed here.

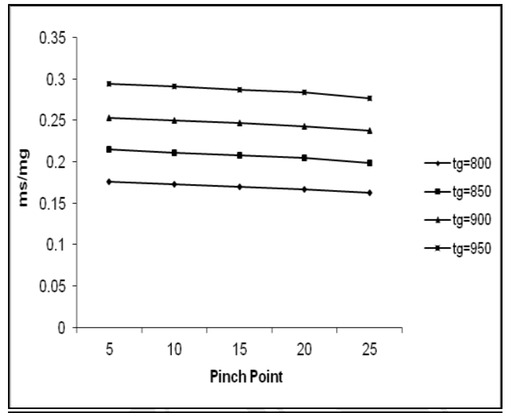


**Figure 3: Variation of steam generation rate for different pinch points at different WHRB inlet temperature**

**Effect of pinch point variation:** Pinch point is the difference between evaporator steam outlet temperature and exhaust gas temperature in WHRB. Pinch point must be as low as possible otherwise it severely affects the heat transfer and affected surface area. Theoretically it is better if this temperature has minimum possible value of zero. From Figure 3. it is evident that as the pinch point increases, steam generation rate reduces. Higher pinch point results in the increased gas temperature entering in the super heater and evaporator and ultimately lowers the steam generation rate.

It is clear from the Fig. 4 that as pinch point increases for a given gas turbine exit temperature the waste heat recovery boiler exit temperature also increases. The reduced heat transfer in economizer and high WHRB exit temperature figure 4. Indicates lower heat recovery in WHRB

At high WHRB inlet temperature the same trend is obtained but, WHRB exit temperature is lowest for highest gas temperature. This is because of the irreversibility associated with them. For better performance low pinch points are favorable. This also results in less irreversibility due to less temperature difference between the gas and steam/water.

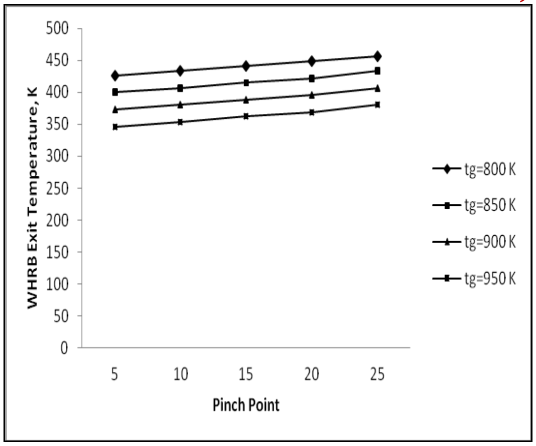


**Figure 4: Variation of WHRB exit temperature for pinch point at various WHRB inlet temperatures**

Reducing the pinch point leads to lower stack losses but this increases cost of WHRB and increases the pressure loss. It is evident that at lower pinch point heat exchange is best and energy loss is also reduced.

Figure 5 shows the variation of steam generation rate for different approach points (temperature difference between the outlet of the water from the economizer and the saturated water in the evaporator) at different gas temperatures. It is evident from fig 5 that the steam generation rate gradually diminishes with increase in approach point due to ineffective utilization of exhaust gas heat at higher approach points compared to lower approach points. For any given approach point the largest steam generation rate is found for highest exhaust gas temperature which can be attributed to the enhanced availability of heat for steam generation.

In single pressure WHRB the decrease in approach point improves its performance. As the steam formation in the economizer imposes the limits on the approach point therefore, this outlet is always kept in sub-cooled region so that, no damage occurs during off design condition. Increase in the WHRB gas inlet temperature improves the steam generation rate.

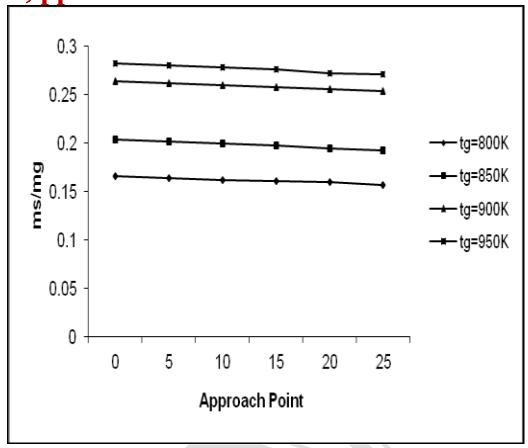


**Figure 5: Variation of steam generation rate with approach points at different WHRB inlet temperatures**

Figure 6 shows the variation of WHRB exit temperature with approach point for different gas turbine exit temperature. As the approach point increases the steam generation rate reduces and WHRB exit temperature increases. This is because a large surface area is required in these systems as a result of low pinch and approach points and the low log mean temperature differences at various heating surfaces.

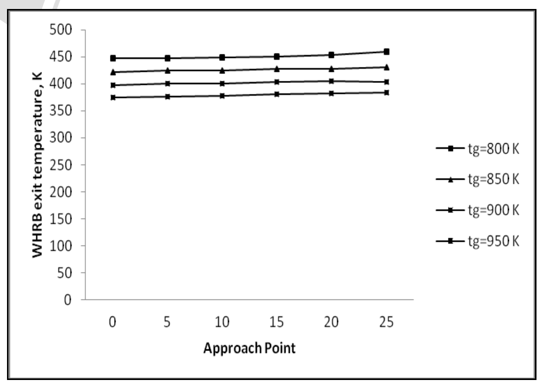


**Figure 6: WHRB exit temperature for various approach points at different WHRB inlet temperatures**

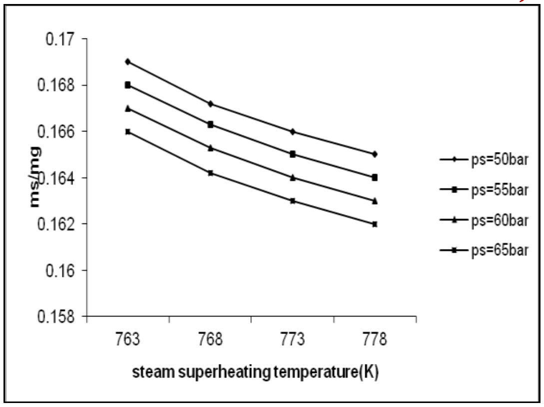


**Figure 7: Steam generation rate vs. steam superheating temperatures for different operating steam pressures**

Figure 7 depicts the steam generation rate variation for increasing superheating steam temperature and steam pressures. Steam pressure and temperature affects the heat exchange between the entering gas & feed water and hence the steam generation.

With the higher steam pressure the system efficiency is increased but limits the total heat recovery from the flue gas in WHRB due to higher saturation temperature. Because gas temperature leaving the evaporator is higher with less steam generation as compared to lower steam pressure therefore, steam generation is less and heat recovery potential is reduced as shown in Figure 7 for a single pressure cycle. Water from the condenser enters the WHRB at the cold end and is then heated to the saturation temperature by flue gases. Following this water enters the evaporator steam drum where the flow is circulated and heat is transferred at constant temperature. Finally steam is superheated by hottest flue gas before passing to the steam turbine. The choice of high steam pressure limits the amount of heat transfer to steam. Therefore in order to generate high pressure 'useful' steam other pressure levels are required.

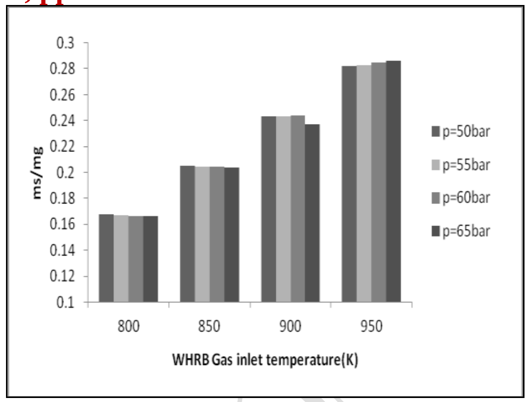


**Figure 8: Variation of steam generation rate with WHRB exit gas inlet temperature for different operating steam pressures**

Figure 8 shows the variation of steam generation rate at different WHRB inlet temperatures. It can be seen that by increasing inlet temperature WHRB steam generation also improves. Here segmented finned tubes analysis is used for compact heat recoveries. In this design few parameters namely tube outer diameter, number of the fins per tube, fin height, fin thickness effect has been analyzed along with the energy recovery for the particular design condition. As the proposed design is focus on the segmented fins for staggered tube bundles. The exposed surface area of serrated fin is larger the solid fins. The parametric design observation is discussed ahead.

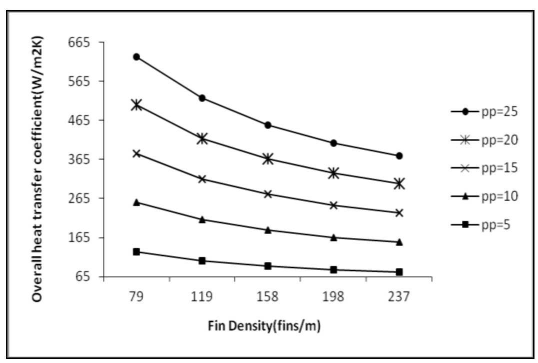


**Figure 9: Fin density and overall heat transfer coefficient variations for different pinch point**

Figure 9 shows that by increasing the fin density on the gas side surface results in lower overall heat transfer coefficient, even though the surface area will become several times greater. The external heat transfer decreases and results in lower overall heat transfer coefficient while water/steam side heat transfer coefficient increases. Ultimately steam generation rate gets improved. A high fin density or a large ratio of outside to inside areas shows poor thermal performance. Therefore use of large fins on evaporator and economizer surfaces is better. The effect of pinch point on particular set of conditions also exhibits the same effect.

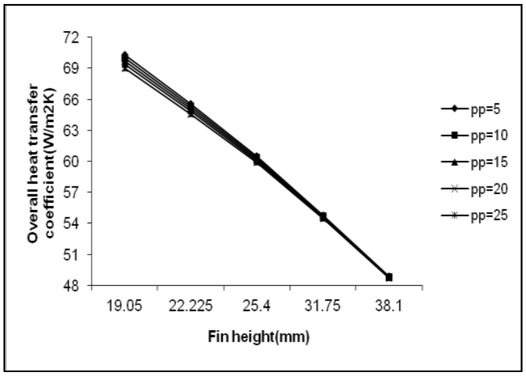


**Figure 10: Variation of overall heat transfer coefficient with fin height for different pinch points**

Figure 10 shows the variation of overall heat transfer coefficient with fin height for different pinch point values. It shows that the lower fin height results in higher overall heat transfer coefficient and these effects are reversed for fin thickness. Higher thickness results in higher heat transfer coefficient as shown in Figure 11.

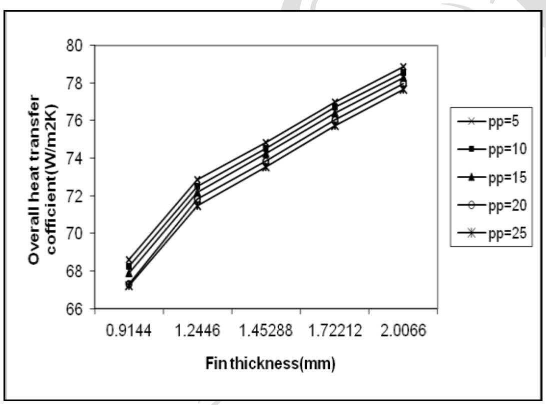


**Figure 11: Fin thickness and heat transfer variations for different pinch points**

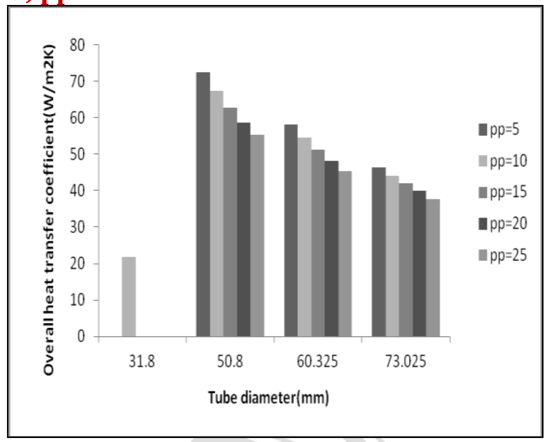


**Figure 12: Variation of overall heat transfer coefficient with tube diameter for different pinch points**

Figure 12 shows that by increasing the tube diameter total surface area increases so it is advisable to use large diameter for the fluid having large heat transfer coefficients. If we compare optimized design results with existing plant as shown in Figure 12 by using serrated and staggered finned tube in the existing design then much heat can be recovered. Therefore, fins have advantage of reducing the total number of tube required and mass of the heat transfer equipment.

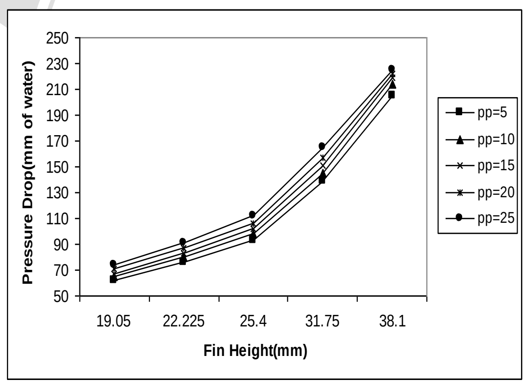


**Figure 13: Variation of pressure drop with fin height for different pinch point**

Figure 13 shows that as the fin height increases the pressure drop increases, it is clear from the Fig. 10 that as the fin height increases the overall heat transfer coefficient decreases which requires more pumping power and operating cost increases.

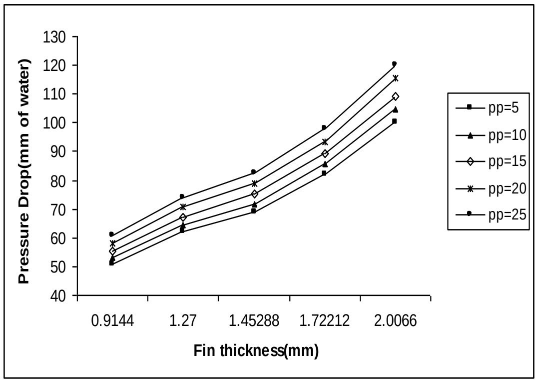


**Figure 14: Variation of pressure drop with fin thickness pinch point**

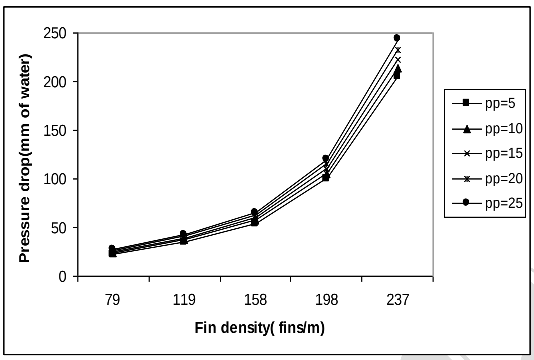


**Figure 15: Variation of pressure drop with fin density for different pinch points**

Graphical pattern shows that as the fin density increases the gas side coefficient decreases resulting in higher pressure drop per row.

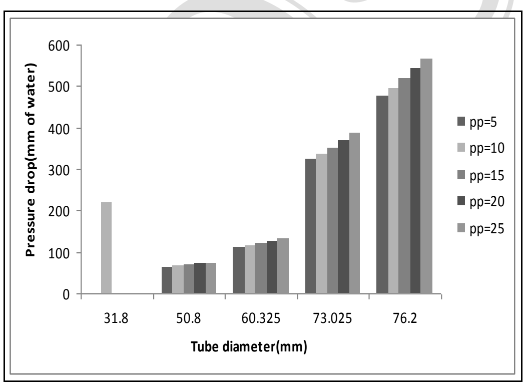


**Figure 16: Variation of pressure drop with tube-diameter and pressure drop for different pinch points**

Similar effects have been seen for fin density, tube diameter and fin thickness on pressure drop. It is evident that gas pressure drop can be adjusted for different surface areas by using proper fin height, fin thickness, fin density and tube diameter and more energy can be transferred optimized area. Fig 16 shows that the pressure drop for solid spiral finned tube is more than serrated staggered arrangement.

## IV. Conclusions

Following conclusions are obtained from the analysis done for the optimum design of WHRB for combined cycle power plants

1. It is desirable to have a lower pinch point for maximum steam generation rate and effective flue gas energy utilization.

2. Segmented fins offer compact heat recoveries because of high gas turbulence and improved exposed contact surface.

3. Fin segmentation improves gas penetration, which improves flow velocity resulting in higher heat transfer.

4. However pressure drop in the gas side increases by increasing tube diameter and offers the operating problem. In existing plant having solid fins with particular diameter pressure drop is large as compared to serrated fins. In actual case the pressure drop is 2.5 times higher for same tube diameter.

5. Higher fin densities for any surface does not make any difference in heat transfer analysis as it acts as solid fin design.

6. Increase in Fin height by 3 units offers the reduction in heat transfer by 20% at its maximum limit.

7. 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 51.8mm.

8. As WHRB 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.

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