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Theoretical and Numerical Analysis of Convective Recuperator for an Oil Fired Water Tube Boiler to Improve the Boiler Performance

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Abstract

Objectives: This paper deals with the theoretical and numerical analysis of a convective recuperator used to recover waste heat from the stack flue gases and use it to pre heat the air before it is fed into the combustion chamber of an oil fired water tube boiler. **Methods/Statistical Analysis:** Three different tube diameters are considered in the present study. Theoretical analysis is carried out to determine the efficiency and other parameters of the boiler with and without recuperator. To support the theoretically obtained results on improved boiler performance by pre heating the air using recuperator, numerical simulation studies is carried out. **Findings:** The study showed that 10" inch (0.254 m) pipe diameter arranged in a 3 x 3 array gave a flow velocity of air at around 15 to 25 m/s. The stack flue gas exit temperature decreased from 200°C to 135.50°C whereas the inlet air temperature after pre heating through the recuperator is found to be 85°C with a rise in temperature of around 50°C. After pre heating the air, the efficiency of the boiler improved from 71.90% to 74.60%. The quantity of furnace oil saved per day is 3.76% with respect to its present consumption level. CFD simulations are carried out using ANSYS CFX. A good agreement is found between the data obtained from simulation and theoretical analysis. Payback period by incorporating the recuperator to the existing boiler is estimated to be 60 working days. **Application/ Improvements:** Optimized performance improvement of existing boiler used in industries is possible without extensive redesign by recovering waste heat leading to energy conservation.

Keywords: Air, Boiler, Convective Recuperator, Flue Gas, Performance, Pre Heat

1. Introduction

Energy is the key input for economic development and growth of all the countries in the world and it is the basic need for all the industrial facilities. Around 35% of total energy consumed in the world is used in industrial sectors and in India it is, around 45%. In the present world scenario, consumption of fossil fuel is increasing rapidly and also this leads to environmental pollution significantly. Demand for fuel is expected to be tripled by the year 2025. The fossil fuel generating capacity has increased at a slower rate in comparison with the increasing demand. Among the fossil fuels, oil is the principle source of commercial energy. So, energy conservation is of greater importance both in economic and environmental aspect.

In industry, energy can be conserved by ensuring optimum utilization of energy. So, the equipments are designed to operate efficiently. For existing machineries,

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efficiency improving modifications can be made without recourse to extensive plant redesign. Boilers find most application in various industries and it is widely used in various processing industries, textile industries and used in steam power plants as a power generation unit. The recovery of heat from exhaust flue gases is one of the most popular energy saving measures[1]. In boilers with an oil fired burner type, reduction in consumption of oil leads to reduction in total cost of production.

A boiler is a closed vessel in which steam is generated due to exchange of heat generated from combustion of fuel. In a typical water tube boiler, fuel is burned in the furnace and hot gases generated by combustion flows over the water tubes and heat the circulating water in the tubes. The heated water converts itself into steam due to sensible and latent heat transfer and its stored in the steam drum as saturated steam. This saturated steam is converted into superheated steam by exposing this steam again to the flue gas path by using a super heater. To increase economy of the boiler, exhaust gases are also used to pre-heat the air blown into the furnace and warm the feed water supply.

2. Factors Affecting the Boiler Performance

Energy saving does not always imply efficiency improvement in boiler. Therefore, it is necessary to understand the factors that influence the efficiency.

2.1 Stack Temperature

Also known as flue gas temperature, the stack temperature above 200°C indicates the potential for recovery of waste heat. By reducing the flue gas temperature by 22°C, the efficiency of the boiler improves by 1%.

2.2 Feed Water Preheating

For an older shell boiler with a flue gas exit temperature of around 260°C, an economizer can be used to reduce the temperature of flue gas to 200°C with an overall improvement in thermal efficiency by 3%. One percent savings in fuel consumption is possible by raising the feed water temperature by 6°C using an economizer.

2.3 Combustion Air Preheating

Combustion air is preheating is an alternative to feed water heating. The thermal efficiency of boiler increases by 1% by raising the combustion air temperature to around 20°C.

2.4 Excess Air

Excess air is the quantity of air available in a furnace which is more than the requirement for combustion to occur. For every 1% reduction in excess air, an improvement of 0.6% rise in efficiency is possible. The optimum excess air level varies depending upon the furnace design, fuel, burner type and other operating process variable. Based on the National Productivity Council report, an excess air of 15 to 20% by weight is permissible for fuel oil.

2.5 Fuel Specification

Different types of fuels like coal, pulverized coal, fuel oil and natural gas are used in the furnace or burner of a boiler and its performance depends upon the type of fuel as well as its composition. Accordingly, the thermal efficiency also varies.

2.6 Ambient Temperature

Ambient air prevailing temperature also affects the performance of a boiler. Boiler efficiency varies by 1% for every 40°C variations in ambient temperature.

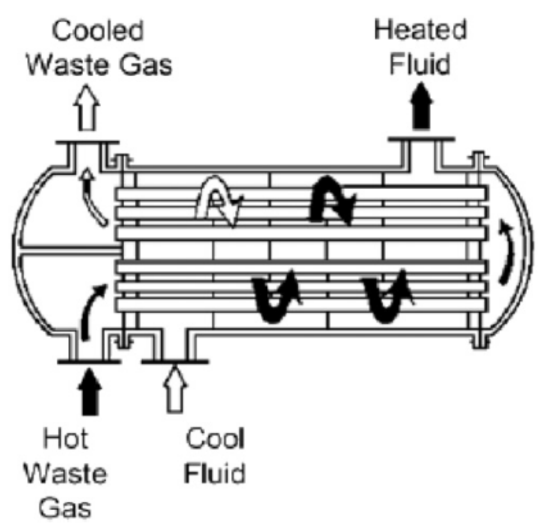

Figure 1. Fluid flow in a typical convective recuperator[23].

2.7 Radiation and Convection Losses

External surfaces of the boiler shell are hotter than the surroundings. These surfaces thus lose heat to the surroundings depending on the surface area and difference in temperature between surface and surroundings. In general, irrespective of the boiler output, the heat loss from the boiler shell is a fixed energy loss. This may be around 1.5% on gross calorific value at full rating which may increase to 6% if the boiler operates at 25% of its output. With proper insulation, these losses through boiler walls and piping can be reduced.

2.8 Reduction of Scaling and Soot Losses

In oil fired boilers, soot built up on tubes acts as an insulator against heat transfer. Elevated stack temperature is an indication of excessive soot built up. When the flue gas temperature rises about 20°C above the temperature is indicator for removing soot deposits. A built up of 3 mm thick soot causes an increase in fuel consumption by 2.5%.

3. Literature Review and Objective of the Present Study

Various heating process involved in Turkish textile industries and steps taken to recover the waste heat using waste heat recovery systems like recuperator was studied. The authors revealed that consumption of energy decreased by using waste heat recovery systems. Exergy based approach and thermodynamic analysis was conducted to optimize the effective working conditions. The payback period was very short[2]. The recovery of flue gas waste heat and water in coal power plant was studied with focus on improving its efficiency[3]. Detailed review was carried out on recovery of waste heat by heat pipes with a view in saving energy consumption and enhancing effectiveness using oscillating heat pipe heat exchanger, two phase closed thermosyphon heat exchanger and conventional heat pipe heat exchanger. Various parameters that influenced the effectiveness of heat exchangers to attain its optimum condition were discussed[4]. Use of various heat transfer equipment and heat exchangers in waste to energy systems and the importance of using CFD with experimental approach in optimum design and operation was highlighted[5]. A variety of newer waste to energy technologies were used in continuous process industries with a view of recovering the heat and energy savings[6].

The heat transfers and pressure drop correction factors for a heat exchanger with an optimized segmental baffle based on the Bell-Delaware method was compared with those for a heat exchanger with helical baffles. The performance of heat exchangers with helical baffles based on test results of various baffles geometries was discussed in detail[7]. Heat transfer analysis of recuperative air pre heater was studied with gas entering the domain in tubes location with temperature around 200°C. The gas entered the tubes domain and passed over the heat exchanger tubes. Baffles were hindering the flue gas path and changed the flow path for effective heat transfer. Baffles were designed such a way that entire length of the tubes had to exchange the heat by conduction and convection process[8,9].

The importance of air pre heater in enhancing the efficiency of thermal power plant using boiler was discussed[10]. Also, the benefits in recovering the heat of flue gas and the implications in lowering the temperature of flue gas and its associated problems were discussed which was again emphasised by latter work[11]. The flow behaviour within an air pre heater with focus on improving the overall heat transfer coefficient was studied numerically[12]. Optimizing the air pre heater design using staggered as well as inline tube arrangement was discussed[13]. The present status of using an air pre heater in extracting energy from the flue gases in power generation was elaborated in brief[14]. Similar studies based on experimental as well as numerical analysis were carried out earlier by several researchers[15–21].

In Kerala Minerals and Metals Ltd., (KMML) in which this study is carried out, the boiler is presently operated without an air pre heating system and huge amount of waste heat through flue gases are released. Huge amount of fuel can be saved by recovering this waste heat. Based on the literature reviews, it is found that for an every 20°C rise in inlet air temperature, the boiler efficiency gets increased by 1%. Also, the reduction of flue gas exit temperature helps in reduction of harmful gases up to certain extent. At the current operating conditions in KMML, the exit flue gas temperature is in the range of 180˚C to 200˚C and the inlet air is fed at an ambient temperature of around 25˚C to 35˚C. Of the various factors stated above, stack temperature loss is considered in this present paper. A convective recuperative type heat exchanger is used to pre heat the air from the waste heat recovery[22,23]. To recover waste heat from flue gases of a boiler / furnace, recuperator is used. Convective recuperator[23,24], as seen in Figure 1, is an effective and compact with multiple tubes making a larger heat transfer area. The hot gases enter the shell surrounding the tubes and passes over the tubes once while the incoming air to be heated is carried through a number of parallel small diameter tubes. Baffles are incorporated along the side walls of the shell which provides hindrance to the flow path of flue gases, so that heat flux and the rate of heat transfer are increased.

The objective of this present study is to improve the boiler efficiency of an oil fired water tube boiler by using the stack temperature of exit flue gas to pre heat the inlet atmospheric air used for fuel combustion in the boiler by a convective recuperator. This ensures more steam generation per litre of furnace oil consumed during combustion. First, the theoretical analysis is carried out followed with CFD simulation. Simulation studies are performed to validate and verify the theoretical results.

4. Results and Discussion

4.1 Theoretical Study

The specification of the boiler taken up for the present study is shown in Table 1. Performance parameters of boiler like efficiency, evaporation ratio depends upon fuel, fuel quality, combustion, water quality, etc. Boiler efficiency test shows the variations in boiler efficiency from the preeminent and helps in locating problematic component / location for taking corrective step. Thermal efficiency of boiler is defined as the percentage of heat input that is effectively utilized to generate steam. By direct method, the boiler efficiency is expressed by the equation as.

$$\text{Boiler efficiency } (\eta) = \frac{Heat_{Output}}{Heat_{Input}} * 100$$

$$= \frac{Q(h_g - h_f)}{m(GCV_{fueloil})} * 100$$

where, h_g and h_f is the enthalpy of saturated steam and feed water respectively in kCal/kg.

Parameters to be monitored for estimating the boiler efficiency by direct method is working pressure (g) in N/m^2, quantity of fuel used per hour (m) in kg/hr, quantity of steam generated per hour (Q) in kg/hr and superheat temperature of steam in ^{0}C, temperature of feed water in ^{0}C, type of fuel and gross calorific value of fuel (GCV) in kcal/kg of fuel. Table 2 shows the properties of air and flue gases at 35^{o}C and 200^{o}C respectively.

Table 1. Water tube boiler specifications

Manufacturer name and type	**ISGEC JOHN THOMSON; Two drum water tube boiler**	
Total heating surface area	1715 m^2	
Quantity of steam generated	17 - 31 TPH	
Maximum working pressure	24 Kg/cm^2(g)	
Casing thickness	5 mm	
Tube thickness	3.251mm	
Flue gas temperature after economizer	160^{o}C to 200^{o}C	
Tube pitching	Parallel to gas flow – 102 mm	
Across gas flow	108/96 mm alternate pitch	
Minimum flow of 4.50 TPH of super heated steam is essential		
	STEAM DRUM	**MUD DRUM**
Internal diameter	1219 mm	991 mm
Thickness	36 mm	32 mm
Length	4.81m	4.45m

Energy balance equation is expressed as,

Heat rejected from flue gas ($Q_{flue\ gas}$) = Heat absorbed by air (Q_{air})

$[mC_p (T_1 – T_2)]_{flue\ gas} = [mC_p (t_1 - t_2)]_{air}$

Based on the above the outlet temperature of flue gas is estimated. Logarithmic mean temperature difference (LMTD) is expressed as $(\theta_2-\theta_1)/\ln(\theta_2/\theta_1)$, where $\theta_1 = T_1-t_2$ and $\theta_2 = T_2-t_1$

The value of overall heat transfer co-efficient (U) for gas to gas heat exchanger varies from 60 to 600W/m^{2}K and presently it is assumed as 60 W/m^{2}K.

Heat transfer, $Q_{air} = [mC_p (t_1 - t_2)]_{air}$

$$\text{Area required, } A = \frac{Q_{air}}{U * LMTD}$$

In this present study, the length of the tube is selected as 5 m. Pitch is selected to be 1.25 times the outer diameter (d_o). The outside diameter, d_o is 10.75 inch (0.274 m) and inner diameter, d_i is10 inch (0.254 m).

Pitch, $P = 1.25 \times d_o$

Surface area of tube (a) = $\pi d_o l$

Number of tubes, $N = A/a$

Flow area, $A_t = N (\pi/4) \times d_i^2$

Table 2. Properties of air and flue gas

Property of air (at 35°C)		**Property of flue gas (at 200°C)**	
Inlet temperature (t_1)	35˚C	Inlet temperature (T_1)	200˚C
Density (ρ)	1.1465 kg/m^3	Density (ρ)	0.7461 kg/m^3
Viscosity (μ)	18.875×10^{-6} Ns/m^2	Viscosity (μ)	22.44×10^{-6} Ns/m^2
Specific heat (C_p)	1.005 kJ/kgK	Specific heat (C_p)	1.097 kJ/kgK
Prandtl number (Pr)	0.685	Prandtl number (Pr)	0.670
Thermal conductivity (k)	27.155×10^{-3} W/m K	Thermal conductivity (k)	40.120×10^{-3} W/m K
Outlet temperature (t_2)	85˚C	Mass flow rate (m)	53,000 kg/hr
Mass flow rate (m)	50,000 kg/hr		

Linear Velocity, V = flow rate of air/ flow area

V = mass flow rate/ (density x flow area)

For air, the linear velocity ranges between 15 to 34 m/s.

Reynolds number, Re = (ρvd)/ μ

Nusselt number, Nu = 0.023 $Re^{0.8}$ $Pr^{0.3}$

Nusselt number, Nu = $(h_t d_i)/k$

h_t = (Nu k)/d_i

Based on the above expression, the tube side heat transfer coefficient is estimated to be 62.3 W/m^{2}K

Overall heat transfer coefficient, $1/U_o = 1/h_t + R_{ft} + [(r_o/r_i) R_{fi}] + (r_o/r_i h_t)$

Based on the above expression, overall heat transfer coefficient (U_o) is estimated to be 58.42 W/m^{2}K. As this value is less than the assumed overall heat transfer coefficient, it is acceptable.

Area required on the basis of U_o is expressed as

$$A = \frac{Q_{air}}{U_o * LMTD}$$

Number of times the fluid crosses the bundle = N + 1 where N is the number of baffles.

N + 1 = tube length/baffle space

The efficiency of the boiler without preheating the air is estimated as

$h_{sat} = h_f + xh_{fg} - h_{fi}$

$h_{sh} = h_g + C_p (t_{sh} - t_{sat}) - (h_f + h_{fg})$

$h_g = h_{sat} + h_{sh}$

$$\text{Efficiency } (\eta) = \frac{Q(h_g - h_f)}{m(GCV_{fueloil})} x100$$

Table 3. Values obtained based on theoretical analysis

Parameters	**Unit**	**Value**
Density of air (ρ_a)	Kg/m^3	1.1465
Air outlet temperature (t_2)	°C	85
Air inlet temperature (t_1)	°C	35
Air mass flow rate(m_a)	Kg/hr	50000
Density of flue gas (ρ_g)	Kg/m^3	0.7461
Flue gas outlet temperature (T_2)	°C	135.50
Flue gas inlet temperature (T_1)	°C	200
Flue gas mass flow rate (m_g)	Kg/hr	53000
Number of tubes (N_t)	-	9
Length of recuperator (L)	m	5.00
Gas side heat transfer coefficient (h_s)	W/m^{2}k	48.612
Air side heat transfer coefficient (h_t)	W/m^{2}k	62.3
Overall heat transfer coefficient (U_0)	W/m^{2}k	58.42

Efficiency of the boiler (ŋ) after preheating the air with recuperator is expressed as

$$\frac{Q(h_g - h_f) + Q_{air}}{m(GCV)_{fueloil}} x100$$

Based on the theoretical study, it is found that the boiler efficiency is 71.79% before installing the recuperator to pre heat the air. The boiler efficiency improved to 74.6% after the air is pre heated and its temperature raised to around 85˚C. Without pre heating the air, quantity of fuel used is 2.1 T/hr and around 50.40 Tonnes/day. After installation of the recuperator to pre heat the air, quantity of fuel used at increased efficiency is expressed as,

$$\frac{\eta_{new}-\eta_{old}}{\eta_{new}} x100$$

By installing the convective recuperator to pre heat the air, the efficiency increased by around 3.76%. The cost of the furnace oil consumed per day without pre heating the air is around Rs.19,75,680/- and cost of the furnace oil saved per day is Rs. 74,286/-. Table 3 shows the consolidated values obtained based on the theoretical analysis.

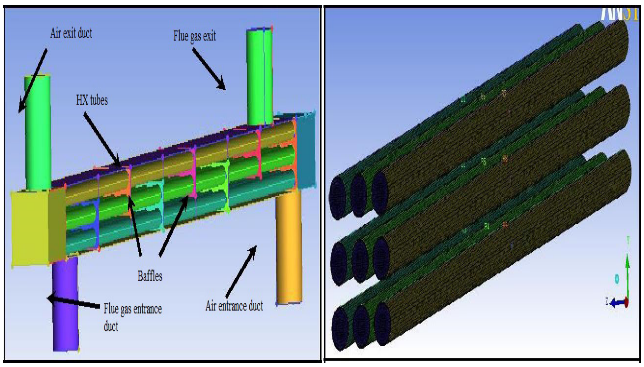

Figure 2. Computational domain of the convective recuperator with flue gas and air flow paths with baffles.

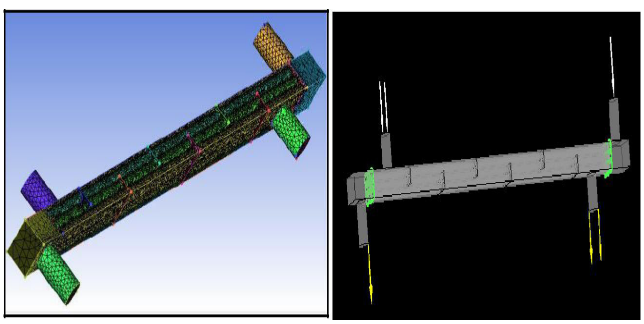

Figure 3. Unstructured mesh generation in a convective recuperator domain.

4.2 Numerical Study

To support the theoretically obtained results on boiler performance by pre heating the air using a convective recuperator, numerical simulation studies is carried out. As the computational domain involves flow of two different gases, the hot flue gas shell domain and air flow tubes domain are modeled and meshed separately. Figure 2 shows the geometric model of the convective recuperator which is created using ANSYS ICEM CFD 12.0. Convective recuperator domain is rectangular in shape and unstructured mesh, as seen in Figure 3 is created through robust volume meshing method. This meshing method generates quadrilateral or triangular volume mesh. The meshes are clustered more at all walls within the convective recuperator to capture the growth of boundary layer near the walls. y^+ value near walls are maintained as per the tuburlence model requirements and the y^+ values are in the range of 0.15 to 1.25. In order to ensure that flow captured is not sensitive to grid distributions, grid refinement study is carried out. Refinement of grid beyond 12,50,676 elements did not vary the result by more than 2.15%. Therefore, result obtained with this grid resolution is found to be optimum to study the flow behaviour within the convective recuperator accurately.

ANSYS CFX 13.0 is used to simulate numerically the heat exchange process in the convective recuperator domain based on counter-flow approach. The hot flue gas enters the shell domain at 200˚C and the air to be pre heated enters the tubes at 35˚C. Baffles provide hindrance to the hot flue gas flow path thereby increasing the heat transfer rate. As the hot flue gas shell domain and air flow tubes domain are modeled and meshed separately, proper interface is introduced in between them. For air flow tubes domain, the inlet boundary condition is mass flow rate and outlet boundary condition is static pressure. The direction of flow at inlet is kept normal to its boundary. For shell domain, the inlet boundary condition is mass flow rate and boundary condition at outlet is static pressure. Again, the direction of flow at inlet of shell domain is kept normal to its boundary. The boundary conditions imposed for this numerical study are shown in Table 4. Stationary domain having non buoyant condition with turbulence at medium intensity level is mentioned as criteria to obtain the solution. At inlet, flue gas and air at 35^0C is specified as the domain fluid. The reference pressure is 1 atm. Wall roughness on walls of baffles, shell as well as tubes domains are neglected and assumed as smooth wall. No-slip condition is assumed at all the walls of baffles, shell as well as tubes domains. Upwind advection scheme is used with automatic timescale specification. Turbulence model used is kω Shear Stress Transport to achieve the closure for all governing equations under steady state conditions. Convergence criteria for all the residuals are set to 10^{-4}.

Velocity streamline distribution is shown in Figure 4. Hot flue gas is entering the shell domain and passes over the tubes. As seen in Figure, presence of baffles increases the velocity of flue gases by acting as hindrance to the flue gas path. This also changes the flow path as seen in the Figure for effective heat transfer. Also, the velocity of flue gas as observed at inlet and outlet are in the desired level for effective heat transfer without any back pressure formation.

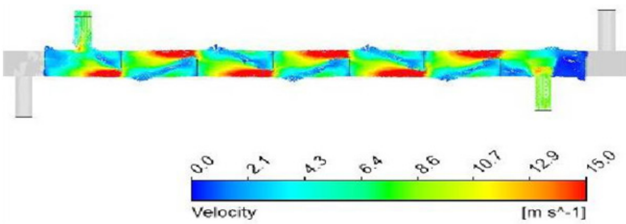

Figure 4. Velocity vector along flue gas flow path.

Figure 5 shows the temperature distribution of the flue gas in the fluid domain. The hot flue gas enters at 200°C and it falls to around 125.50°C near the last baffle position with a drop of around 74.50°C. This fall in temperature indicate the transfer of heat to the air and this data is in the similar range based on theoretical analysis.

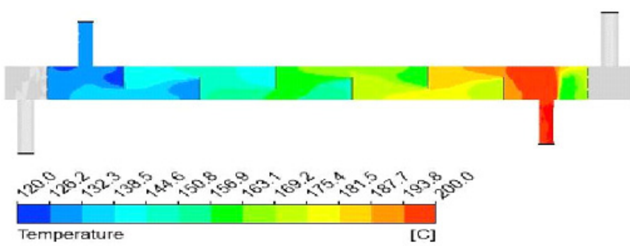

Figure 5. Temperature contours of flue gas flow path.

Figure 6 shows the heat flux over the tubes and maximum heat flux is observed near the wall of the shell domain as the flue gas flows due to presence of baffles. Due to this maximum heat flux, heat transfer rate is enhanced considerably.

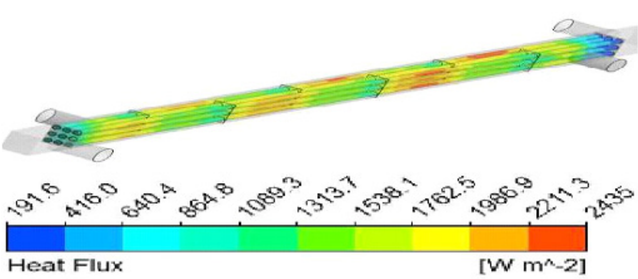

Figure 6. Contours of heat flux over the tubes.

Figure 7 shows the wall temperature distribution on the outer walls of the tubes in a convective recuperator as the hot flue gases flow over it. The contour shows a gradual fall in temperature along the flow direction of hot flue gases in the shell domain of the convective recuperator from inlet side to outlet side indicating the direction in which the heat transfer is taking place.

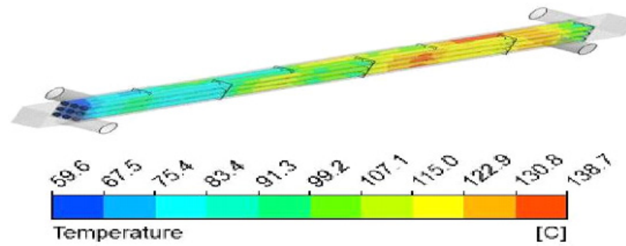

Figure 7. Temperature contours on outer walls of the tubes.

Figure 8 shows the temperature distribution of air passing through the tubes. As seen in the Figure, at inlet the temperature of air is 35°C which increases gradually along the flow direction due to heat transfer to around 87.40°C. The net increase in temperature of air at exit is around 52.40°C which is found to be in similar range based on theoretical analysis.

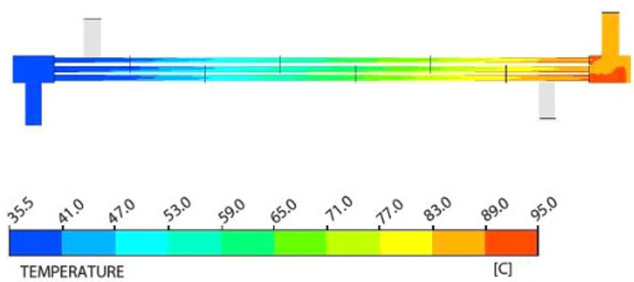

Figure 8. Contours of temperature distribution of air within the tubes.

Figure 9 shows the velocity distribution of air within the tubes. The velocity of air in tubes is observed to be around 19 m/s to 22.50 m/s. Also, this observed velocity of air is close to the estimated theoretical data. No pressure drop as well as back pressure formation is observed within the tubes.

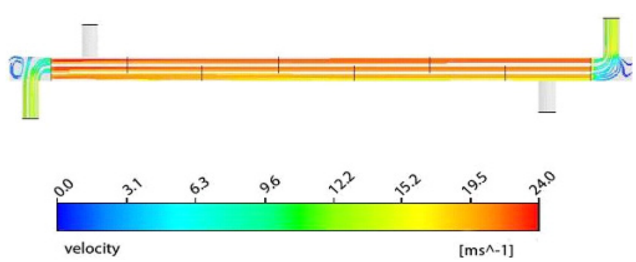

Figure 9. Velocity distribution along the air flow path.

Table 4. Details of the boundary conditions

Domain	Boundary type	Location	Boundary conditions defined
Flue gas flow domain	Outlet	Outlet of flue gas	Static pressure, relative = 0 N/m^2
	Inlet	Inlet of flue gas	Static temperature = 200^{o}C Mass flow rate = 14.722 kg/s
Air flow domain	Outlet	Outlet of air flow	Static pressure, relative = 0 N/m^2
	Inlet	Inlet of air flow	Static temperature = 35^{0}C Mass flow Rate = 13.888 kg/s

A good agreement is obtained on comparing the theoretical data as shown in Table 3 with the results obtained from simulation studies on using a convective recuperator as an additional component to pre heat the air before feeding it to the combustion chamber of the existing boiler. Therefore, the convective recuperator with 10 inch pipes (0.254 m) arranged in an array of 3 x 3 is found to effective in pre heating the air and thereby improving the boiler efficiency by around 3.70%.

The economic analysis is also carried out in this present study. The cost of the recuperator is estimated to be 35 Lakhs in Indian rupees and adding up the transportation, installation and other cost that comes around to 8.20 Lakhs in Indian rupees, the total cost to be incurred in installing and commissioning the recuperator is 43.20 Lakhs in Indian rupees. The likely payback period is estimated to be 60 working days.

5. Conclusion

Based on the theoretical analysis of the recuperator and its subsequent simulation studies using CFD, the following conclusions are made:

- 10" inch (0.254 m) diameter pipe arranged in 3 x 3 array is chosen with a flow velocity of air at around 15 to 25 m/s.
- Temperature of flue gas at outlet decreases from 200˚C to 135.50˚C and inlet air temperature rises from 35˚C to 85˚C after the using a convective recuperator
- Nusselt number and heat transfer coefficient are 565 and 62.3 W/m^{2}K. The boiler efficiency improved from 71.90% to 74.6%.
- Significant amount of energy is saved by the recovery of heat from exhaust flue gases. The quantity of furnace oil saved per day is around 3.76%. The total cost of furnace oil saved per day is around seventy-five thousand in Indian Rupees.
- CFD simulation analysis gives the similar values based on theoretical study
- Payback period is estimated to be 60 working days after installation and commissioning of this recuperator in the existing plant at KKML.

6. Acknowledgement

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