International Journal of Applied Research in Mechanical Engineering

Volume 1 | Issue 3 Article 4

January 2012

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Optimization of Air Preheater Design for the Enhancement of Heat Transfer Coefficient

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Abstract - This paper presents an approach for the optimization of air preheater design with inline tube arrangement. The poor performance of an air preheater in the modern power plants is one of the main reason for higher unit heat rate & is responsible for deterioration in boiler efficiency. The main problem of air preheater is the leakage of air to the flue gas side & thereby resulting in poor thermal performance. The higher ash content in Indian coal also adds to the problems associated with tubular air preheater. Air preheaters are designed to meet performance requirements with consideration of highly influencing parameters viz. heat transfer, leakage and pressure drop. In the present work the performance of tubular air preheater is evaluated with the help of Computational Fluid Dynamics (CFD) analysis for In-line tube arrangement for evaluating influence of various parameters viz. Gas flow rate, Gas temperature, tube pitch, etc. The model can also be used while selecting a new type of surface geometry for optimizing the design of air-preheater.

Key words - *Air preheater, Computational Fluid Dynamics (CFD), Efficiency, K-ε model, Tube pitch.*

I. INTRODUCTION

Heating combustion air can raise boiler efficiency about 1% for every 22°C in temperature increase. The most common way to preheat the air is with a heat exchanger on the flue exhaust. There are two types of air preheaters for use in steam generators in thermal power stations One is a tubular type built into the boiler flue gas ducting and the other is a regenerative air preheater. These may be arranged so the gas flows horizontally or vertically across the axis of rotation. Another type of air preheater is the regenerator used in iron or glass manufacture. Many new circulating fluidized bed (CFB) and bubbling fluidized bed (BFB) steam generators are currently incorporating tubular air heaters offering an advantage with regards to the moving parts of a rotary type.

With the increasing price of fuel and technology improvements, the size of a boiler that can be economically equipped with a pre-heater should become smaller. Although still a technology most applicable to large boilers, high energy prices will certainly motivate innovative new applications for economical combustion air pre-heaters on ever smaller boilers. Air heaters can also use extraction steam or other sources of energy depending upon the particular application. The hot air produced by air heaters enhances combustion of all fuels and is needed for drying and transporting the fuel in pulverized coal-fired units. Industrial units fire variety of fuels such as wood, sewage sludge, industrial waste gases as well as coal, oil and natural gas. In the small units tubular, plate and cast iron heaters are widely used.[1]

Fuels fired on stoker grates, such as bituminous coal, wood and refuse, don't require high air temperatures, therefore water or steam coil air heaters can be used. Tubular preheaters consist of straight tube bundles which pass through the outlet ducting of the boiler and open at each end outside of the ducting. Inside the ducting, the hot furnace gases pass around the preheater tubes, transferring heat from the exhaust gas to the air inside the preheater. Regenerative air heaters are relatively compact. Air to gas leakage can be controlled by cold-presetting axial & radial seal plates to minimize gaps at the hot operating conditions, or using sacrificial material.[2]

Computational studies are performed to investigate the effect of various operating parameters on entrained flow through air preheater tubes.CFD is a versatile tool. A large variety of problems with different levels of complexity can be solved.

In a typical tubular air heater, energy is transferred from the hot flue gas flowing inside many thin walled tubes to the cold combustion air flowing outside the tubes. The unit consists of a nest of straight tubes that are roll expanded or welded into tubesheets and enclosed in a steel casing. In the vertical type tubes are supported from either the upper or lower tubesheet while the other (floating) tubesheet is free to move as tubes expand within the casing. An expansion joint between the floating tubesheet and casing provides an air/gas seal. Intermediate baffle plates parallel to the tubesheets are frequently used to separate the flow paths and eliminate tube damaging flow induced vibration. Carbon steel or low alloy corrosion resistant tube materials are used in the tubes which range from 38 to 102 mm in diameter and have wall thicknesses of 1.24 - to 3.05 mm. Larger diameter, heavier gauge tubes are used when the potential for tube plugging and corrosion exists. The most common flow arrangement is counterflow with gas passing vertically through the tubes and air passing horizontally in one or more passes outside the tubes. A variety of single and multiple gas and air path arrangements are used to accommodate plant layouts. Designs frequently include provisions for cold air bypass or hot air recirculation to control cold end corrosion and ash fouling.[3]

A. Pressure drop:

In recuperative air heaters, gas- or air-side pressure drop arises from frictional resistance to flow, inlet and exit shock losses and losses in return bends between flow passes. Pressure drop is proportional to the square of the mass flow rate. [5]

B. Leakage:

Recuperative units may begin operation with essentially zero leakage, but leakage occurs as time and thermal cycles accumulate. With regular maintenance, leakage can be kept below 3%. Approximate air heater leakage can be determined based on gas inlet and outlet oxygen (O_2) analysis (dry basis). [4]

C. Erosion:

Heat transfer surfaces and other air heater parts can suffer erosion damage through impact of high velocity, gas-entrained ash particles. Erosion usually occurs near gas inlets where velocities are highest. The undesirable effects of erosion are structural weakening, loss of heat transfer surface area and perforation of components which can cause air to gas or infiltration leakage. It is controlled by reducing velocities, removing erosive elements from the gas stream, or using sacrificial material. In the design stage, air heaters used with fuels containing highly erosive ash can be sized to limit gas inlet velocities to 50 ft/s (15 m/s). Inlet flues can also be designed to evenly distribute gas over the air heater inlet to eliminate local high velocity areas.

II. PERFORMANCE ANALYSIS OF AIR PREHEATER

Theoretical calculations for evaluating thermal performance of air preheater are mentioned below. For the air heater, the heat transfer rate is determined as follows:

$$q = m_g C_p (T_1 - T_2) \qquad (1)$$

Where,

q = Heat transfer rate, Kcal/hr

m_g = Mass flow rate of flue gas, kg/hr

C_p = Approximate mean specific heat of gas, Kcal/kg °C

T_1 = Gas temperature entering the air heater, °C

T_2 = Assumed air heater exit temperature, °C

For the air side, the temperature rise is:

$$T_2' = T_1' + q/(m_a C_p) \qquad (2)$$

Where,

T_1' = Air temperature entering air heater, °C

T_2' = Air temperature leaving air heater, °C

q = Heat transfer rate, Kcal/hr

m_a = Mass flow rate of air, kg/hr

C_p = Approximate mean specific heat of air, Kcal/kg °C

For cross flow arrangement , the log mean temperature difference is determined as follows:

$$LMTD = \frac{[(T_1 - T_2') - (T_2 - T_1')] \text{ x } F}{[\ln(T_1 - T_2') / \ln(T_2 - T_1')]} \qquad (3)$$

In an air heater, gas & air film heat transfer coefficients (T_f & T_f') are approximately equal. Film temperatures are approximated by the following calculations.[4]

$$T_f = \frac{(T_1 + T_2)}{2} + \frac{LMTD}{4} \qquad (4)$$

$$T_f' = \frac{(T_1' + T_2')}{2} + \frac{LMTD}{4} \qquad (5)$$

The gas mass flux is given by,

$$G_g = \frac{m_g}{A_g} \qquad (6)$$

Where,

m_g = Mass flow rate of flue gas, kg/hr

A_g = Flue gas flow area, m^2

Gas Reynolds number is given by,

$$R_e = K_{Re} \text{ x } G_g \qquad (7)$$

Where,

K_{Re} = Gas properties factor, m^2 hr/ Kg

G_g = gas mass flux, Kg / m^2 hr

Gas film heat transfer coefficient is the sum of the convection heat transfer coefficient from the longitudinal gas flow inside the air heater tubes & a small gaseous radiation component from within the tube. To properly account for the fly ash layer inside the tubes, the gas side heat transfer coefficients will be multiplied by a cleanliness factor in the overall heat transfer calculation.

$$h_{cg} = h_1^{'} \text{ x } F_{pp} \text{ x } F_t \text{ x } \frac{D_i}{D_o} \qquad (8)$$

Where,

h_{cg} = The gas convection heat transfer coefficient, Kcal/m^2 hr °C

$h_1^{'}$ = Basic convection velocity & geometry factor for longitudinal flow

F_{pp} = Physical properties factor

F_t = Temperature factor

D_i = Tube inside dia., mm

D_o = Tube outside dia., mm

Gas side radiation heat transfer coefficient is given by,

$$h_{rg} = h_r' \text{ x } K \qquad (9)$$

Where,

h_{rg} = The gas side radiation heat transfer coefficient, Kcal/m^2 hr °C

h_r' = Basic radiation heat transfer coefficient, Kcal/m^2 hr °C

K = Fuel factor

The air mass flux is calculated as,

$$G_a = \frac{m_a}{A_a} \qquad (10)$$

Where,

m_a = Mass flow rate of air, kg/hr

A_a = Air flow area, m^2

Air Reynolds number is given by,

$$R_e = K_{Re} \text{ x } G_a \qquad (11)$$

Where,

G_a = air mass flux, Kg / m^2 hr

The crossflow convection heat transfer coefficient for air is obtained from following equation:

$$h_{ca} = h_c^{'} \text{ x } F_{pp} \text{ x } F_a \text{ x } F_d \qquad (12)$$

Where,

$h_c^{'}$ = Basic convection velocity & geometry factor for crossflow

F_{pp} = Physical properties factor

F_a = Arrangement factor for in-line tube banks

F_d = Heat transfer depth factor

Assuming negligible wall resistance, the overall heat transfer coefficient is:

$$U = \frac{[F_{FR} \text{ x } (h_{cg} + h_{rg}) \text{ x } h_{ca}]}{[F_{FR} \text{ x } (h_{cg} + h_{rg}) + h_{ca}]} \qquad (13)$$

The cleanliness or fouling resistance factor, F_{FR}, is empirically derived from field test data; 0.9 is representative of bituminous coal. The total heat transfer rate for the air heater is: [5]

$$q = U \text{ x } A \text{ x } LMTD \qquad (14)$$

Where,

A = Heating surface area in m^2

Air heater exit gas temperature is calculated to be:

$$T_2 = T_1 - \frac{q}{(m_g C_p)} \qquad (15)$$

III. INTRODUCTION TO CFD

Computational Fluid Dynamics or CFD is the analysis of systems involving fluid flow, heat transfer and associated phenomena such as chemical reactions by means of computer-based simulation. It works by solving the equation of fluid flow over the region of interest with specified condition on the boundary of that region. [6]

CFD codes are structured around the numerical algorithms that can tackle fluid flow problems. Hence all codes contain three main elements: (i) A pre-processor, (ii) A solver and (iii) A post- processor. [7]

The basic procedure followed for solving any CFD problem is as follows:

1. The geometry (physical bounds) of the problem is defined.
2. The volume occupied by the fluid is divided into discrete cells (the mesh).

3. The physical modeling is defined - for example, the equations of motions + enthalpy + species conservation.
4. Boundary conditions are defined. This involves specifying the fluid behaviour and properties at the boundaries of the problem. For transient problems, the initial conditions are also defined.
5. The equations are solved iteratively as a steady-state or transient case depending on the physical situation.
6. Analysis and visualization of the resulting solution. [8]

IV. LAYOUT AND DESIGN OF SYSTEM

The air heater is the last heat transfer component before the stack. The air heater, when sized properly, will have sufficient surface to provide the required air temperature to the fuel equipment (burners, pulverizers, etc.) and lower the gas temperature to that assumed in the combustion calculations. The most common flow arrangement for tubular air preheater is counter flow with gas passing vertically through the tubes and air passing horizontally in one or more passes outside the tubes. A variety of single and multiple gas and air path arrangements are used to accommodate plant layouts.

Fig.1 shows the assembly of air preheater module considered in order to study the influence of various parameters like tube pitch, flue gas flow etc. on heat transfer coefficient. For this air preheater module design data used is as mentioned below from one of the practical example. Here, air preheater module is last bank before stack in waste heat recovery boiler.

TABLE I: FLUE GAS PARAMETERS

Parameter	Unit	Value
Mass flow rate	Kg/ hr	95850
Inlet temperature	° C	171
Thermal conductivity	KCal/ m h °C	0.0303
Specific heat	KCal/ kg °C	0.2596
viscosity	Kg/ mh	0.0848
Density	Kg/ Nm^3	1.279

TABLE II : AIR PARAMETERS

Parameter	Unit	Value
Mass flow rate	Kg/ hr	12297
Inlet temperature	° C	35
Thermal conductivity	KCal/ m h °C	0.0253
Specific heat	KCal/ kg °C	0.2418
viscosity	Kg/ mh	0.0723
Density	Kg/ Nm^3	1.287

TABLE III: AIR PREHEATER GEOMETRY

Parameter	Unit	Value
Tube side diameter	mm	63.5
Tube thickness	mm	2.04
Tube inside diameter	mm	59.42
Tube material		Carbon steel
Number of tubes wide	Nos.	15
Perpendicular pitch (perpendicular to direction of air flow)	mm	85
Number of tubes deep	Nos.	50
parallel pitch (along the direction of air flow)	mm	85

TABLE IV: BOUNDRY CONDITIONS FOR FLUE GAS

Parameter	Unit	Value
Pressure at outlet	Gauge	0
Velocity at inlet	m/s	15
Wall		No slip & escape
Default interior		Fluid (Flue gas)

FOR AIR

Parameter	Unit	Value
Pressure at outlet	Gauge	0
Velocity at inlet	m/s	6
Wall		No slip & escape
Default interior		Fluid (air)

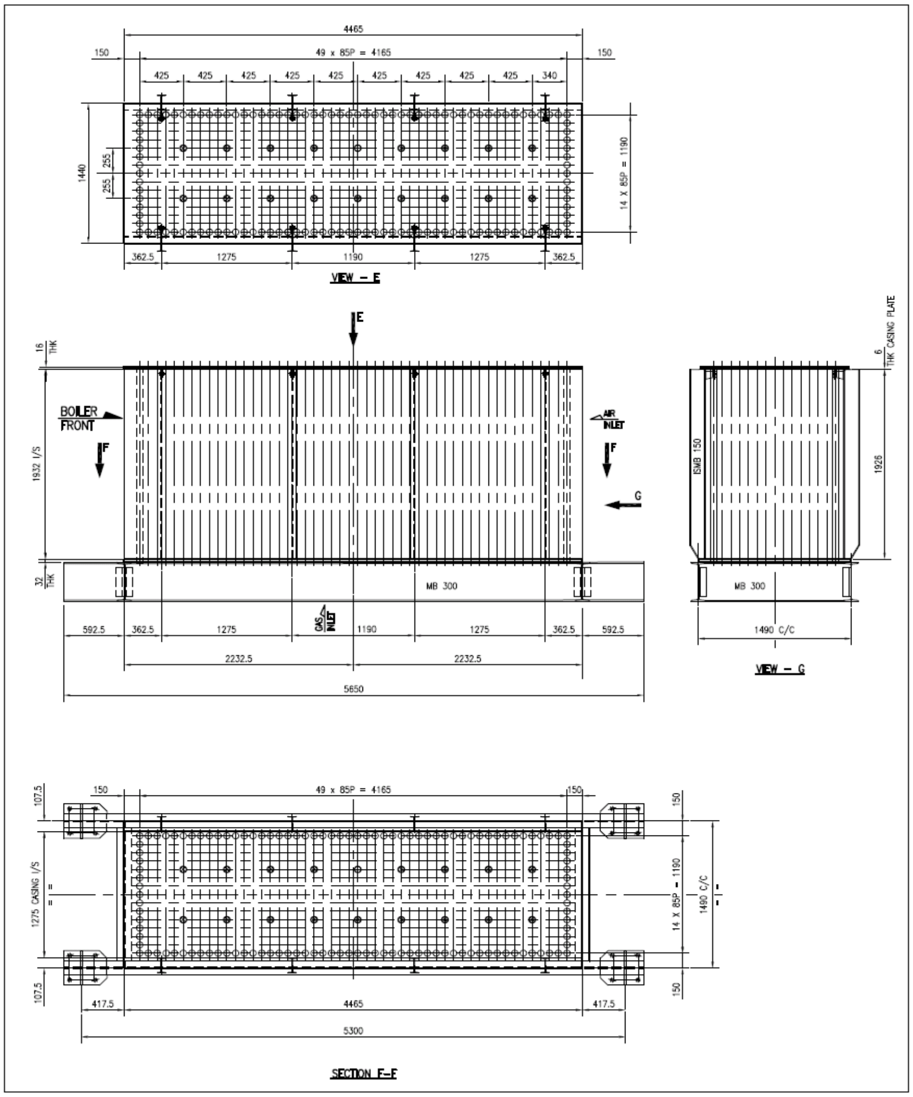

Fig. 1 : Assembly of air preheater module

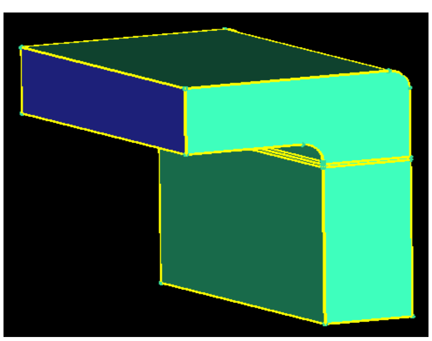

Fig. 2 : Air preheater model with inlet duct

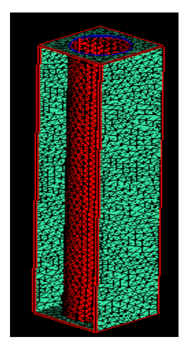

Fig. 3 : Meshed geometry of Air preheater tube

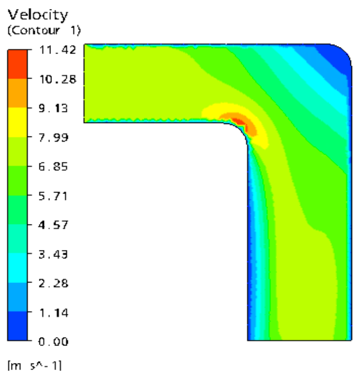

Fig. 4 : Velocity contours across flue gas flow path without perforated plate.

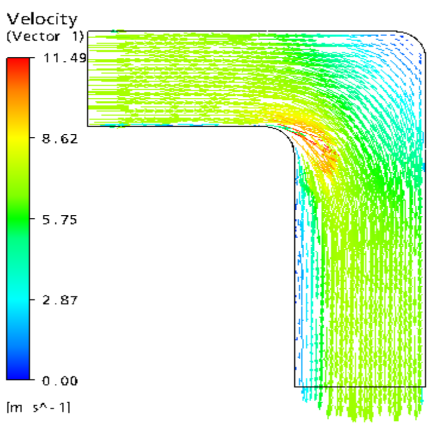

Fig. 5 : Velocity vectors across flue gas flow path without perforated plate

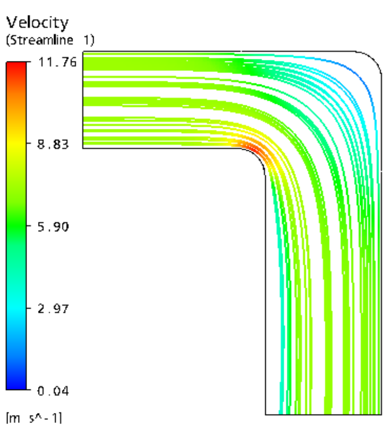

Fig. 6 : Velocity streamlines across flue gas flow path without perforated plate.

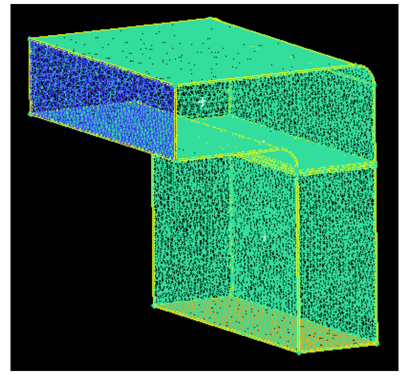

Fig. 7 : Meshed APH model with perforated plate

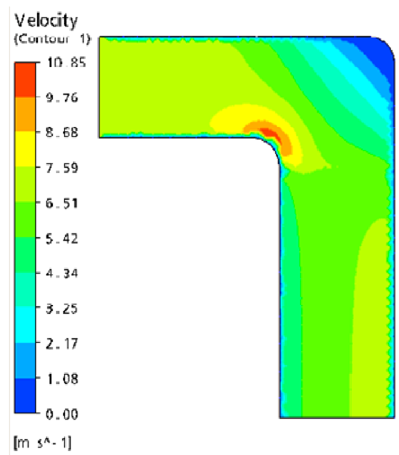

Fig. 8 : Velocity contours across flue gas flow path with perforated plate

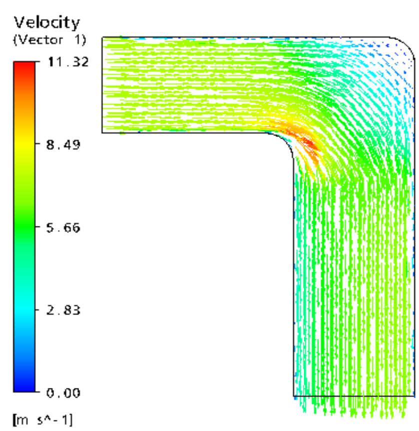

Fig. 9 : Velocity vectors across flue gas flow path with perforated plate.

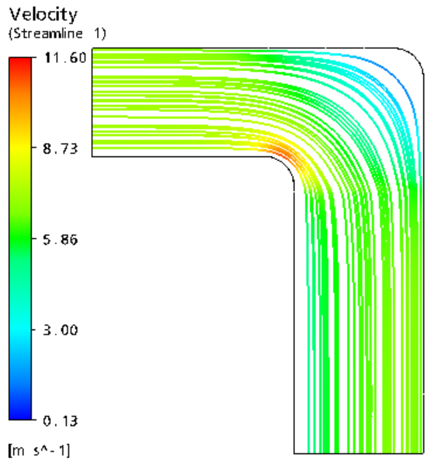

Fig. 10 : Velocity streamlines across flue gas flow path with perforated plate

TABLE V: CFD RESULTS FOR VARIOUS FLUE GAS FLOW RATES

Sl. No.	Flue Gas flow, kg/hr	Flue gas temperature (°C)		Air temperature (°C)		Gas side pressure drop (mmwc)	Heat Transfer coefficient (Kcal/m² hr °C)
		in	*out*	*in*	*out*		
1	95850	171.0	157.5	35.0	147.8	22.9	20.783
2	100000	171.0	158.0	35.0	148.7	25.4	21.178
3	110000	171.0	158.9	35.0	150.6	30.5	22.175
4	120000	171.0	159.8	35.0	152.2	35.6	22.941
5	130000	171.0	160.6	35.0	153.6	40.6	23.715

TABLE VI : CFD RESULTS FOR DIFFERENT TUBE GEOMETRY

Sl. No.	Parameter	Unit	Case-1	Case-2
1	Flue gas flow	Kg/ hr	95850	95850
2	Flue gas inlet temperature	° C	171	171
3	Flue gas outlet temperature	° C	158	158
4	Flue gas side pressure drop	mmwc	22	22
5	Air flow	Kg/ hr	12297	12297
6	Air inlet temperature	° C	35	35
7	Air outlet temperature	° C	148	148
8	Air side pressure drop	mmwc	22.9	22.9
9	Tube OD	mm	63.5	38.1
10	Tube thickness	mm	2.04	2.04
	Tube length	mm	1920	960
11	Number of tubes wide	Nos.	15	40
12	Perpendicular pitch (perpendicular to the direction of air flow)	mm	85	55.0
13	Number of tubes deep	Nos.	50	58
14	Parallel pitch (along the direction of air flow)	mm	85	63.5
15	Total number of tubes	Nos.	750	2320
16	Heat transfer coefficient	(Kcal/m² hr °C)	20.783	22.279
17	Weight of all tubes in APH module	Ton	4.60	4.27

TABLE VII : CFD RESULTS FOR DIFFERENT TUBE PITCH ON AIR SIDE

Sl. No.	Air side tube Pitch, mm	Flue gas temperature (°C)		Air temperature (°C)		Air side pressure drop (mmwc)	Heat Transfer coefficient (Kcal/m² hr °C)
		in	*out*	*In*	*out*		
1	85	171.0	157.5	35.0	147.8	33.0	20.783
2	90	171.0	157.9	35.0	145.0	17.8	19.269
3	95	171.0	158.2	35.0	142.0	10.2	17.885
4	100	171.0	158.6	35.0	139.2	7.6	16.698

V. RESULTS AND DISCUSSIONS

A. Influence of perforated plate at inlet of Air preheater

Tubular air preheaters consist of straight tube bundles which pass through the outlet ducting of the boiler and open at each end outside of the ducting. When flue gas passes through bend portion of flue gas duct, there is unbalance in flow patterns. Fig. 2 represents air preheater module with inlet flue gas duct. Fig. 3 represents meshed geometry of air preheater tube. Simulation is done on existing air preheater inlet duct without perforated plate and results of velocity contours, vector and streamlines are presented in Fig. 4, 5 & 6. This unbalance is carried forward to the air preheater module resulting in uneven distribution of gas through the tubes. In order to have uniform distribution of gas perforated plate is provided at the inlet of APH module as shown in Fig. 7. From Fig. 8, 9 & 10, it is clear that velocity profile at the inlet of air preheater is uniform

B. Flue Gas mass flow influence

The heat transfer and fluid-dynamic performance of an air preheater is strongly dependent on hot fluids mass flow. This study shows the behaviour of the selected air preheater over a wide flow range, while maintaining the geometry and the temperature levels at the baseline situation. Simulation was carried out of for different cases in order to check impact of flue gas mass flow rate on heat transfer keeping other parameters same.

From table V it is clear that as flue gas flow rate increases, velocity and Reynolds number also increase and hence convective heat transfer increases.

C. Tube geometry influence

Tube geometry is important design parameters in design of air preheater modules, since proper selection of tube diameter, thickness, number of tubes in parallel & perpendicular direction etc. results in optimum design of air preheater module. It can be clearly seen from table VI that as we change tube outside diameter from 63.5 mm to 38.1 mm & adjust number of tubes wide & deep in order to keep same pressure drop, there is saving of 7 to 8 % of tube material. Overall size of air preheater is reduced & results in compact design.

D. Tube pitch influence

Tube pitch is one of the most important design parameters in this kind of heat exchangers, because its great influence on the global heat transfer rate of the equipment and its easy industrial implementation. Simulation was carried out of for different cases in order to check impact of variation in tube pitch on heat transfer coefficient keeping other parameters same. The influence of the tube pitch on the heat transfer coefficient and pressure drop can be clearly seen in the table VII. As expected, smaller tube spacing imply higher heat transfer capacity and air pressure drop at fixed air flow rate.

VI. CONCLUSION

From the simulation results, uniform flow of flue gas through air preheater module can be seen as a result of adding perforated plate at the inlet of Air Preheater module. Analysis of air preheater was carried out using K- ε model for prediction of air temperature at air preheater outlet. Reduction in tube pitch by 5 mm increases heat transfer coefficient by 6 to 7%. Increase in flue gas flow rate & temperature results in enhanced heat transfer coefficient. Enhanced heat transfer coefficient of air preheater module results in reduced flue gas temperature at the outlet of air preheater. Approximate 20 °C drop in flue gas temperature results in 1 % of efficiency improvement & hence fuel saving. Modified design of air preheater with change in tube diameter from 63.5 mm to 38.1 mm resulted in saving of 7 to 8 % of tube material. Reduced length of tube resulted in compact design of air preheater, which further reduces structural cost & can be erected in less space. CFD can be used as a tool for substantial reduction of lead times and costs of new designs.

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