

Research Article

Combustion Air Pre-heating from Ash Sensible Heat in Municipal Waste Incineration Systems

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Abstract: Heat recovery from bottom ash is more important in municipal waste combustion systems than in any other solid fuel combustion since almost 50% of it comprises of non-combustibles. In this study, an ash cooling system using air as the cooling medium has been modeled for pre-heating the combustion air. Air cooling has several advantages over water cooling methods. The study involves modeling using Gambit tool and is solved with the fluent solver. Municipal solid waste incineration systems have the advantage of being located near the waste collection area apart from the high volume reduction ratio. Improvements in the emission control systems and combustion technology can make incineration a highly feasible disposal method. Low furnace temperature due to heat losses through fuel moisture loss and ash sensible heat loss has been a disadvantage with these systems. In this study, a small percentage of the combustion air is pre-heated in a non-contact type heat exchanger and its effect on the available energy of combustion gases at the evaporator outlet is studied. The study is performed for two different waste samples. Results indicate significant increase in available energy at the evaporator outlet and better relative performance for the lower grade fuel. A comparison is made with similar methods reported in the literature along with a brief discussion on the methodologies adopted. The results confirm the importance of installing ash sensible heat recovery mechanism for waste incineration systems as well as the feasibility of the air based method.

Keywords: Air pre-heater, ash cooler, heat recovery, incineration, solid waste, waste combustion

INTRODUCTION

Municipal Solid Waste (MSW) treatment through incineration converts the organic substances in the waste to produce useful heat along with flue gas and ash. The advantage of incineration is that the process plant can be located near the waste generation area. The volume reduction that can be achieved by incineration is 85 to 90% (World Bank Report, 1999). Wastes generated in the developing world are much lower in their heating value and have high moisture contents compared to that of the developed world. Singh *et al.* (2011) have explored the possibilities of energy generation from wastes generated in the Indian cities and made a critical review of the Indian scenario. Low calorific value reduces heat output during combustion. High moisture content in the waste results in energy drain in the combustion system due to dehydration of the fuel and it is endothermic in nature and reduces bed temperature as reported by Alliston *et al.* (2001). Apart from the low calorific values and high moisture content, another problem with MSW which has received lesser attention in the literature is the high ash contents or non-combustibles. High ash contents in biomass results in a huge loss of heat through ash sensible heat which reduced the temperature of the combustion chamber considerably. The physical heat lost due to this can be

as high as 3% of total heat produced when low calorific value fuel is used in the boiler (Helemer and Stokke, 1998). Ash at high temperatures can also damage the ash handling equipment and hence ash cooling is very important. In this study the performance analysis of an air preheating/bottom ash cooling system used in a municipal waste fired boiler is made. The boiler uses a dumping type travelling grate furnace. The analysis is made by simulating the thermal performance using the two dimensional Fluent solver. The air pre-heating system, the combustion air supply system, fuel combustion and boiler tubes are modeled with Gambit software. The k- ϵ model is used along with the energy model. The flow conditions of the combustion air, pre-heater air and the fuel are given as inputs to the system using a mixing model as described below. The effect of air pre-heating on the evaporator outlet condition is studied. The performance is analyzed for two different fuel qualities available in an urban locality of South India. Finally, a comparison of the system performance with the water cooled method is made.

This study proves the acceptability of air based ash cooling system as a viable alternative to water based systems. It also gives a comparison of performance with the water based method along with the relative performance at different capacities.

Results of performance of different types of ash cooling systems for Grate Firing (GF) and Circulating Fluidized Bed (CFB) firing of conventional fuels are reported in the literature. These include the cooled ash screw system (Sami *et al.*, 2001), the rolling cylinder ash cooler (Zeng *et al.*, 2011), fluidized bed ash cooler (Lu, 2006) and the high strength steel belt ash cooler (Liu *et al.*, 2006). Similar studies were reported also by Shu *et al.* (2007) and Cheng *et al.* (2005). The first three types use water as the cooling medium and the fourth type uses air. Water cooling systems use the boiler feed water for this purpose. Consequently the water gets preheated and is fed into the boiler at higher temperatures. Air cooling systems are used to pre-heat the combustion air used for combustion in the furnace. Air cooled systems as reported by Vincenzo *et al.* (2007) have been installed in a large number of GF and CFB systems. Some of the advantages of air cooled systems over water cooled systems as given below:

- No water contamination as in quenching type coolers
- No handling problems of wet waste
- Unburned combustibles in ash is burned off by air

In this study, initially, the waste characteristics as obtained from the field study are given. This is followed by an explanation of system used for modeling followed by the model description. The ash cooling system is then described followed by the details of the modeling procedure and the boundary conditions. Finally, the result of the fluent solver is discussed along with the graphical representation of the available energy for the two cases of fuel qualities.

METHODOLOGY

Composition of MSW: The assessment of the waste generated from a town in South India was used in this study. A detailed statistical methodology was adopted for sample collection. The sample size and number of samples required for the analysis was done as per the method suggested by Dennis and Albert (1969) (not elaborated in this study). Accordingly two samples of 500 kg each were collected and their proximate and ultimate analysis was done. The sample taken during the dry month of April was named Case A and during the rainy month of October was named Case B. The result of the analysis is given in Table 1 and 2.

The boiler: The different parts of the combustion system considered in the study are shown in Fig. 1. In the travelling grate dumping type furnace, the fuel is fed by hoppers into one side of the furnace and the non-combusted material or ash is removed from the opposite side. The boiler consists of a rectangular furnace and it is a travelling dumping grate type. Water is evaporated

Table 1: Proximate analysis of waste for two cases A and B

Constituent	% by weight case A	% by weight case B
Moisture	15.31	19.60
Volatile matter	24.33	26.21
Ash	43.65	36.50
Fixed carbon	16.71	17.69

Table 2: Ultimate analysis of waste for two cases A and B

Element	% by weight case A	% by weight case B
Carbon	25.81	26.24
Hydrogen	4.51	5.32
Sulphur	0.00	0.00
Nitrogen	0.48	0.46
Oxygen	25.54	26.96
Ash	43.65	41.02
CV (kJ/kg)	10617.14	11930.47

in the evaporator tube walls arranged along the walls of the furnace as shown in Fig. 2. These are erected vertically around the rectangular cross sectioned furnace. Water is evaporated partially in these tubes after which steam separation takes place in the boiler drum. Further heating of the steam takes place in the super heater region further downstream. In the present model, only the region of the boiler up to evaporator is considered.

Sizing of the furnace: The furnace sizing is made from the volumetric and area wise heat loading used for travelling grate furnaces. The furnace is rectangular in cross section and the cross sectional area and height are calculated from the volumetric and area wise heat loading data (Ganapathy, 2002).

The Higher Heating Value of the fuel (HHV) is given by the following relation (Ganapathy, 2002):

$$HHV = \frac{4.18}{100} [(8080 * C) + \{(34500 * (H - O/8))\} + (2240 * S)] \quad (1)$$

where,

- C = The percentage of carbon
- H = The percentage of hydrogen
- O = The percentage of oxygen
- S = The percentage of sulphur

The Lower Heating Value (LHV) of the fuel is given by the following relation (Ganapathy, 2002):

$$LHV = [HHV - (22604 * H) - (\%M * h_{fg})] \quad (2)$$

where,

%M : The moisture content in the fuel in decimal form

Volumetric heat generation rate (q_v) determines the volume of the furnace and in the case of biomass boilers have typical values in the range of 0.176 MW/m³ (Ganapathy, 2002).

The Volume of the furnace (V) is given by:

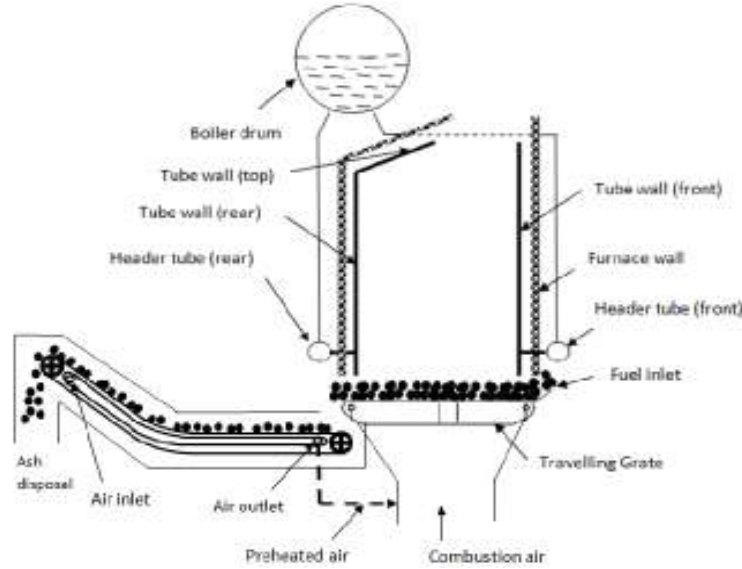


Fig. 1: Parts of the MSW fired boiler with the air pre-heating system

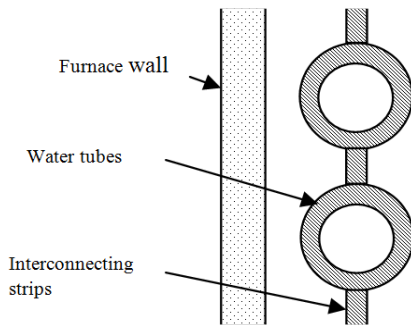


Fig. 2: Construction of tube walls (top view)

Table 3: Recommended values of area heat loading

Boiler capacity (tons per h)	Upper limit of area heat loading MW/m ²		
	ST ≤ 1300°C	ST = 1300°C	ST ≥ 1300°C
130	2.13	2.56	2.59
220	2.79	3.37	3.91
420	3.65	4.49	5.12
500	3.71	4.65	5.44
1000	4.42	5.12	6.16
1500	4.77	5.45	6.33

ST: The ash softening temperature

$$V = \frac{\dot{m}_f \cdot LHV}{q_V} \quad (3)$$

\dot{m}_f = The mass flow rate of the fuel in kg/s

q_V = The volumetric heat load of the furnace mentioned above, kJ/m³

Area heat loading of a furnace (q_A) is the quantity of heat released per m² of grate area which is given for different boiler capacities (Ganapathy, 2002) in Table 3. Lara *et al.* (2013) conducted experimental studies on biomass fired boilers with travelling grate furnaces and obtained efficiency in the range of 75-80%.

The fuel feeding rate is taken as the generation rate of MSW. The operating temperature of the evaporator is taken as 400°C and the corresponding enthalpy of vaporization (h_{fg} value) is taken from steam tables. Ash fusion temperature of 1200°C is anticipated as per results of most biomass samples as reported by Angelo (2012).

The grate area, A_g is calculated using the following relation:

$$A_g = \frac{\dot{m}_f \cdot LHV}{q_A} \quad (4)$$

q_A is the area heat loading of the furnace, kJ/m².

The height of the furnace is calculated by the following equation:

$$H = \frac{V}{A_g} \quad (5)$$

The dimensions of the furnace obtained using the above procedure was 3 m length, 3 m width and 9 m height. Heat from the combustion reactions are transferred to the water flowing through the tubes which are basically classified into three regions. They are the economizer, evaporator and super heater. Out of these three regions, only the evaporator tubes are exposed directly to the flames and hence both convection and radiation heat transfer is pre-dominant in this region. Super heaters and economizers are located away from the flame. In this study, only the evaporator region is considered for analysis. The height of the evaporator region is taken as 5 m.

THE ASH COOLING SYSTEM

The travelling grate discharges the ash into the inlet of the ash cooling system which consists of an

endless metallic belt specially designed to withstand the high temperature and wear. The ash travels along the enclosure which is insulated to prevent heat loss to the surroundings. The system has the provision for air cooling with air flowing in between the onward and the return conveyor belts. The width and speed of the belt determines the height of the ash bed which is usually maintained at 10 to 20 cm in height (Vincenzo *et al.*, 2007). The air passing below the ash conveyor plates get heated by convection heat transfer from these plates. After getting heated up, the hot air is mixed with the combustion air below the grate.

Heat available in the ash, Q_A , which constitutes the hot region of the heat exchanger, is given by the following relation:

$$Q_A = m_f * \frac{\%Ash}{100} * C_A * (T_{Ai} - T_{Ao}) \quad (6)$$

Heat transfer area, A, required is obtained from the heat exchanger relationship:

$$A = \frac{Q_A}{U * LMTD} \quad (7)$$

U is the overall heat transfer coefficient and LMTD is the logarithmic mean temperature difference for counter flow. The hot material being solid state, the heat transfer takes place by conduction and also the speed of movement is very less and hence the ash temperature is taken as the surface temperature of the hot surface. Hence the overall heat transfer coefficient depends on only the heat transfer coefficient on the air flow side. That is $U = h_a$.

The heat transfer relation for flow over inclined flat plate is used. Air side heat transfer coefficient is found using the following relations (Necati, 1985):

$$St_x * Pr^{2/3} = 0.0296 * Re_x^{-0.2} \quad (8)$$

$$St_x = \frac{Nu_x}{Pr * Re_x}, Nu_x = \frac{h_a D}{K}, Re_x = \frac{\rho v D}{\mu}$$

The pre-heating or ash cooling air flow rate is taken as 1.5% of the combustion air supply requirement as per the procedure given by Vincenzo *et al.* (2007). Again the combustion air requirement is calculated from the stoichiometric air calculations after considering 10% excess air. The combustion reactions assuming complete combustion and the stoichiometric air required, A_{Sreq} is given as:

$$A_{Sreq} = \frac{[C * \frac{8}{3} + H * 8 + S - O]}{21} \quad (9)$$

Velocity of air flowing through the heat transfer area is given as follows:

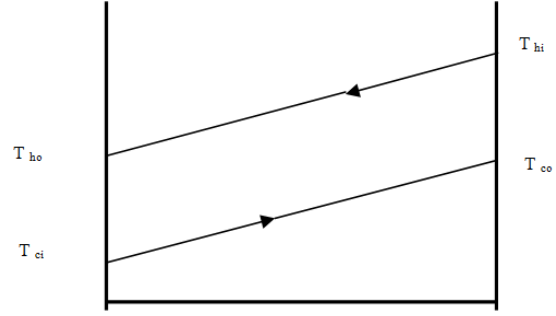


Fig. 3: Temperature profile of heat exchanger

$$v = \frac{A_{req} * m_f * (1+e)}{\rho * W * h} \quad (10)$$

'e' is the excess air percentage in decimal form which is usually 0.15 for solid fuel fired furnaces (Ganapathy, 2002).

The LMTD is taken for the case of counter flow heat exchanger since the direction of ash flow is opposite to that of the cooling air as shown in Fig. 3. The temperature profile is assumed to be linear across the ash and air flow paths inside the heat exchanger:

$$LMTD = \frac{(T_{ho} - T_{ci}) - (T_{hi} - T_{co})}{\ln \frac{(T_{ho} - T_{ci})}{(T_{hi} - T_{co})}} \quad (11)$$

The width of the heat transfer surface is taken as equal to the width of the grate and the length of the heat exchange surface is fixed accordingly from the result of the above calculation.

Heat recovered from unburnt carbon in ash: Unburnts are present in the ash coming out after combustion due to insufficient contact with air. The unburnts are burned during the air supply to the air heater which leak through the metallic parts of the heat exchanger and result in an increase in the temperature of the pre-heated air. Sathyanathan (2011) has given the percentage of unburnts (Y) in bottom ash by the following empirical equation based on experiments on grate fired boilers:

$$Y = 0.0233X^4 - 0.3925X^4 + 1.9277X^4 - 0.1593X^4 + 0.2357 \quad (12)$$

with the residual value of 0.9412.

X is the percentage of particles with size over 50mesh in the ash. This value should be below 2%. The heat received by the pre-heat air due to combustion of the unburnts in air is given as follows:

$$Q_u = m_f * \frac{Ash\%}{100} * \frac{U\%}{100} * HHV \quad (13)$$

Temperature rise of the pre-heat air is given by:

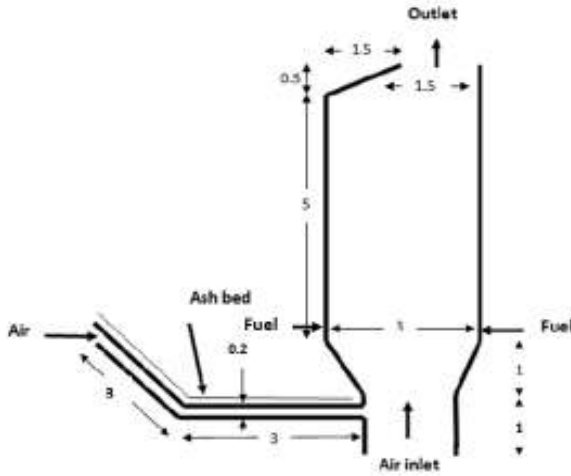


Fig. 4: The model of the MSW fired boiler with the air pre-heating system

$$\Delta T = \frac{Q_u}{m_{pa}c_{pa}} \quad (14)$$

Modeling the system: The dimensions of boiler, furnace and air pre-heating system are shown in Fig. 4. The dimensions are calculated as per the above Eq. (1) to (14). The model is constructed using the Gambit software. The combustion process is modeled by assuming that the process is a mixing of cold combustion air and the products of combustion which possess the heat of combustion in it. The grate is in the form of uniformly distributed rectangular strips which are assumed to be at the combustion temperature. The rectangular strips help in uniformly distributing the gases entering the evaporator region. The uniformity of the mixing is visible in the temperature contours and the velocity contours given in the results. Equation (15) to (19) gives the equivalent values of temperatures and mass flow rates that are used to model the combustion process in the furnace. Triangular meshing is done and the boundary conditions are given as given in Table 4. The combustion air, pre-heating air and the fuel inlet are given as velocity inlets and the corresponding velocities and temperature are obtained from combustion equations. The evaporator outlet is marked as pressure outlet. The boiler walls are considered to be at the temperature of boiling water corresponding to the operating pressure of the boiler:

$$T_{gm} = \frac{m_{pa}T_{pa} + m_{ca}T_{ca} + m_{fa}T_{fa}}{m_{pa} + m_{ca} + m_{fa}} \quad (15)$$

$$T_{ga} = \left(\frac{m_f * HHV}{[(m_{pa} + m_{ca} + m_f * (1 - Ash)) * C_{pg}]} \right) + 300 \quad (16)$$

$$m_{gm} = m_{pa} + m_{ca} + m_{fa} \quad (17)$$

Table 4: Equivalent boundary values of model and corresponding actual conditions

Boundary values for the model	Boundary condition	Actual conditions
Combustion air inlet	Velocity inlet	Eq. (9)
Pre-heating air	Velocity inlet	Eq. (10)
Fuel inlet (2 sides)	Velocity inlet	Eq. (15)-(18)
Evaporator outlet	Pressure outlet	0 Pa (gauge)
Ash cooler/air pre-heater	Wall	Eq. (6)
Evaporator tube walls	Wall	Wall at boiling temperature
Other walls	Wall	Wall with zero heat flux
Grate	Wall	Wall at fuel inlet temperature

$$m_{ga} = m_{pa} + m_{ca} + m_f * (1 - Ash) \quad (18)$$

Equation (15) and (17) represent the temperature and the mass flow rate of the gas after combustion in the case of the model. The actual values of the temperature and mass flow rates of the products of combustion are given by Eq. (16) and (18). The equivalent values of the temperature and the mass flow rate used for the model is obtained by equating the Eq. (15) with (17) and (16) with (18), respectively.

RESULTS AND DISCUSSION

The temperature contours indicate the effect of pre-heating on the temperature above the grate. The velocity contours show the effect of the grate on the velocity of the gases above the grate. The contours indicate uniform velocities above the grate and hence it models the actual combustion bed reasonably well. The temperature and velocity contours of the model are given in Fig. 5 to 7. The maximum temperature in the furnace zone at a height of 0.5 m above the grate is found to be 1070 K (Fig. 5) for the preheating case and 1020 K for the no preheating case (Fig. 6). The velocity contours indicate an increase in the velocity above the grate for the preheating case. Figure 7a and b show the temperature contours for the Case B for no pre-heating and pre-heating, respectively.

The mean temperature of the gas at the outlet of the evaporator region is plotted in Fig. 8 for the MSW Case 1 and Case 2. The plotting is made for different inlet velocities in the pre-heater. It is observed that for increasing values of inlet velocities or mass flow rates, the corresponding mean temperature decreases. The combined effect of the mass flow rate and the mean temperature is understood with the help of the Available Energy (AE) in the gas which is given by Eq. (19):

$$AE = mC_{pg} \left[(T - T_a) - \left(T_a \ln \frac{T}{T_a} \right) \right] \quad (19)$$

m = The mass flow rate of the gases

T = The temperature of the gas obtained from the results of the model

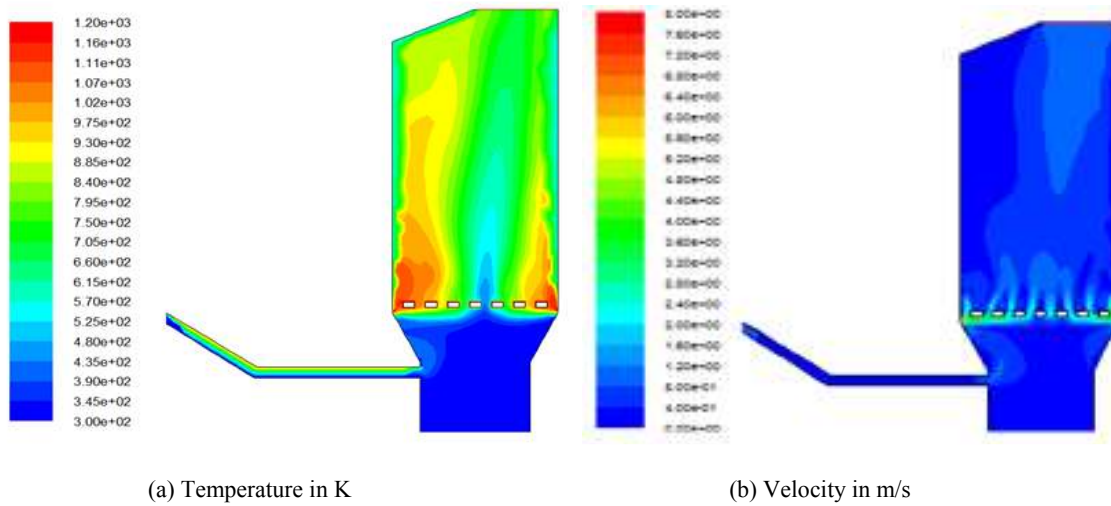


Fig. 5: Temperature (a) and velocity (b), contours for pre-heater inlet mass flow rate of 0.5 kg/s for case A

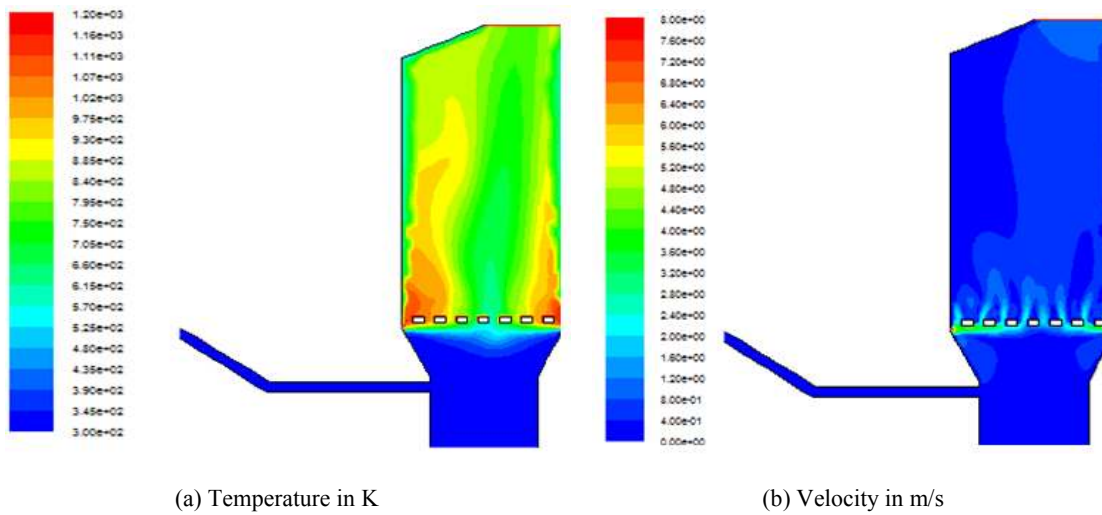


Fig. 6: Temperature (a) and velocity, (b) contours for pre-heater inlet mass flow rate 0 kg/s for case A

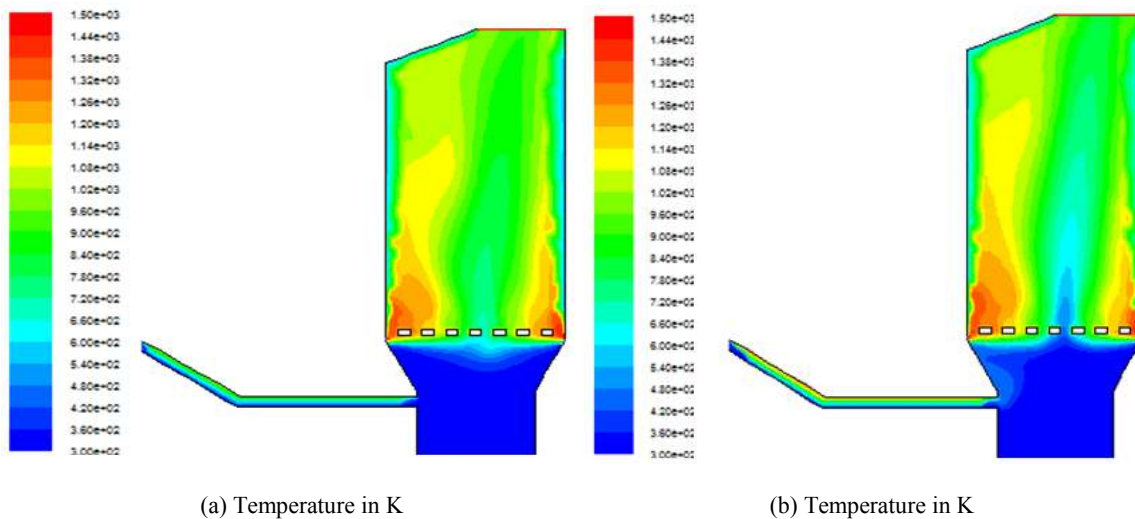


Fig. 7: Temperature contours for pre-heater inlet mass flow rate of (a) 0 kg/s and (b) 0.5 kg/s for case B

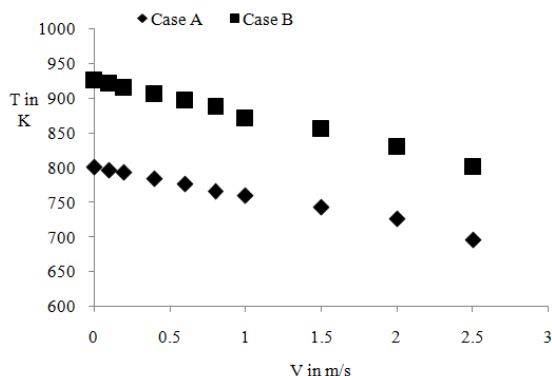
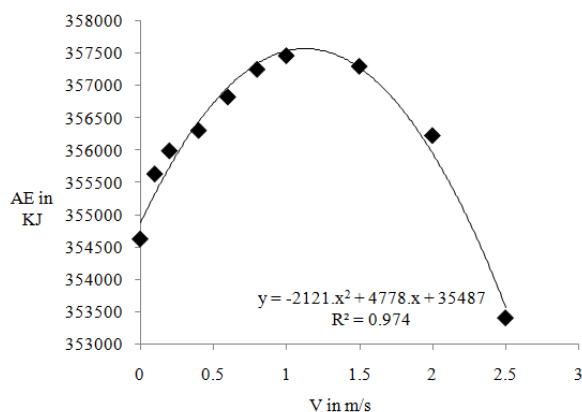
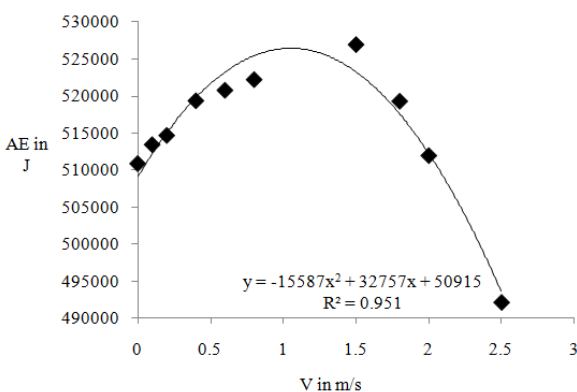


Fig. 8: Average temperature at the evaporator outlet region for case A and B



(a)



(b)

Fig. 9: Variation of available energy for (a) case A and (b) case B

T_a = The ambient temperature
 C_{pg} = The specific heat of the gases

Figure 9 gives the variation in the available energy at the evaporator outlet region for the two cases. In case

A, the maximum available energy occurs for an input velocity of about 1.12 m/s and the corresponding value is 357325 W. In case B, this maximum occurs at 1.33 m/s and the corresponding available energy is 519844 J/s. This indicates that the enhancement effect of pre-heating is more in the case of smaller capacity.

CONCLUSION

An ash cooling system for a MSW based dumping travelling grate type furnace for steam generation is modeled and its performance is analyzed for two different cases of fuel properties as well as at different pre-heating air flow rates. It is seen from the results that pre-heating helps to increase the temperature of the gases above the grate. Available energy was calculated for the different cases and it is seen that pre-heating is more effective for the case with poorer quality fuel.

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