

Municipal Solid Waste Incinerator Design: Basic Principles

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Received October 20, 2018; Revised November 23, 2018; Accepted December 09, 2018

Abstract The paper presents some basics and the steps required when the design of an incinerator for heat recovery or waste treatment is being thought of. It is mostly important for designers in developing countries and students where the advanced design tools and computer modelling are not easily accessible. Waste management has become a major concern world - wide and amidst various waste treatment methods like recycling, composting; incineration is the method that treats the non-reusable and non-organic portion of wastes. Incineration is a complex process due to the heterogenous nature of wastes. Incinerators cannot be designed properly without the knowledge of the combustion science involved and the characteristics of the wastes. Aspects of prime importance in design to be considered are: the incineration mechanisms and their selection, the grate firing systems, furnace geometries, secondary air injection, the 3Ts, the heating value or calorific value of the waste, theoretical Air to Fuel ratio and the excess air requirement. The incinerator internal sizing requirements, chamber sizing, incinerator residence time and retention time, the air injection, as well as the estimation of fuel requirements and the flame temperatures need to be assessed. No one method can be used alone to handle all waste streams effectively, thus an integrated waste management system which not only deals with different methods of treating wastes but also issues such as waste streams, collection, environmental benefits, economic optimization and social acceptability.

Keywords: *municipal solid waste, incineration, design, heating value, air fuel ratio, waste management*

Cite This Article: Mohammed Ben Oumarou, Alhaji Bukar Abubakar, and Sahabo Abubakar, "Municipal Solid Waste Incinerator Design: Basic Principles." *Sustainable Energy*, vol. 6, no. 1 (2018): 11-19. doi: 10.12691/rse-6-1-2.

1. Introduction

Amidst various waste treatment methods like recycling, composting; incineration is the method that treats the non-reusable and non-organic portion of wastes. Modern incinerators are generally of two types: the mass-burn type, in which waste is fed directly into the furnace as it is received, and the refuse-derived fuel type (RDF), where the waste must be sorted, sized and separated into fuel fractions and non-combustible fractions. Both combustion methods pose a number of challenges, such as heavy metals contaminants and their potential leachability. Systems available are: (i) Moving grate furnaces (i.e. Martin System, Düsseldorf System, and Von Roll System) [1]; (ii) Fluidized Bed Furnaces [2]; (iii) Rotary furnaces [3]. Other systems are also available as: Shelf furnaces, Thermo select process, Gasification, Pyrolysis, and the Plasma process [4].

The most preferable method of waste management is, of course, through waste minimization, waste separation at the source at best. A successful waste management system will require: a strong political and effective education; access to facilities and infrastructures for the collection of

source separated recyclables; intensive efforts to implement centralized collection and treatment; the introduction of variable rate charging, or "pay as you throw" and "The Polluter Pays" schemes for household wastes; landfills bans for untreated wastes and the integrated management of residual waste, including energy recovery facilities.

Waste management has become a major concern world-wide and incineration is now being increasingly used to treat waste which cannot be economically recycled. The combustion of conventional well specified fossil fuels is a very complex process since it involves two-phase turbulent reacting flow including radiant heat transfer. Incineration is even more complex since the waste is poorly specified and its composition varies from moment to moment. In the past, the design of incinerators has not been based on fundamental understanding and modelling of the process, and empirical rules have had to be used. Incinerator design thus requires a judicious combination of fundamental combustion science, ingenious engineering guided by an understanding of the mixing process, and last but not least, practical experience of previous failures and successes [5].

According to Gohlke *et al.* [6], through incineration, problems of solid wastes management and pollution can be addressed as: a reduction in the volume and mass of

wastes; detoxification; the destruction of organic components of biodegradable waste that may generate landfill gas (LFG); waste heat recovery potential and no long-term liability.

However waste incinerators may produce unwanted toxic combustion by products or pollutants of environmental concern if they are not well designed and operated [7].

Landfill is still the preferred option in developing nations. While land filling handles the waste in the short term, what happens when all available space is fully utilized? When burying waste, a country also faces the possibility of contaminating ground water [8].

This paper presents some of the fundamental basic formulas and steps required when designing an incinerator for heat recovery or waste treatment. It is mostly important for designers in developing countries and students where the advanced computer modelling and design tools are not available.

2. Incineration Mechanisms and Their Selection

2.1. Grate Firing Systems for Municipal Waste Incineration

Incinerators can be subdivided into grate firing systems, fluidized bed systems (stationary and circulating FB's) and fixed bed systems.

In MSW incineration, grate firing are the common systems with a wide distribution and a long-term experience.

Typical grate firing systems for refuse incineration are shown in Figure 1:

- (i). Reciprocating grates,
- (ii). Roller grates and,
- (iii). Reversed feed grates.

Depending on the grate type, the furnace geometry and the secondary air injection concept have to be optimized. Typically, the primary combustion air cools the grate bars. Besides the air cooling effect is limited for very high

calorific values of the refuse and the bars can be damaged or destroyed by intense heating.

For this reason water cooled bars have been developed and successfully applied in various plants. The primary stoichiometry can be optimized with respect to the burnout behaviour or gaseous emissions such as CO and NO. However, greater complexity and a higher susceptibility to malfunctions are the main disadvantages [9].

2.2. Furnace Geometries

Furnace geometry is one of the main features to influence and to optimize the primary combustion. The direction of the flue gas flow in comparison to the waste transport on the grate is one of the most important factors.

Three (3) standard geometries for municipal waste incineration plants have been established and used worldwide as shown in Figure 2:

- (a) Parallel flow,
- (b) Counter flow and,
- (c) Centre flow.

The main advantages and disadvantages of the different systems are tabulated in Table 1.

A lot of variants in geometries deviating from the base concepts as discussed above are on the market. This fact does not so much depend on real process reasons but is due to manufacture specific developments or patent situation.

2.3. Secondary Air Injection

By dividing the total combustion air (having an over-stoichiometry of 1.3 to 1.8) into primary air and secondary air; the combustion conditions in the furnace can be controlled to give near-stoichiometric conditions.

A typical position for secondary air injection is to the transition area between the furnace and the first pass of the boiler (radiative pass, Figure 3a). The parameters are the number of nozzles, the nozzle diameter, the nozzle spacing and the horizontal and/or vertical inclination.

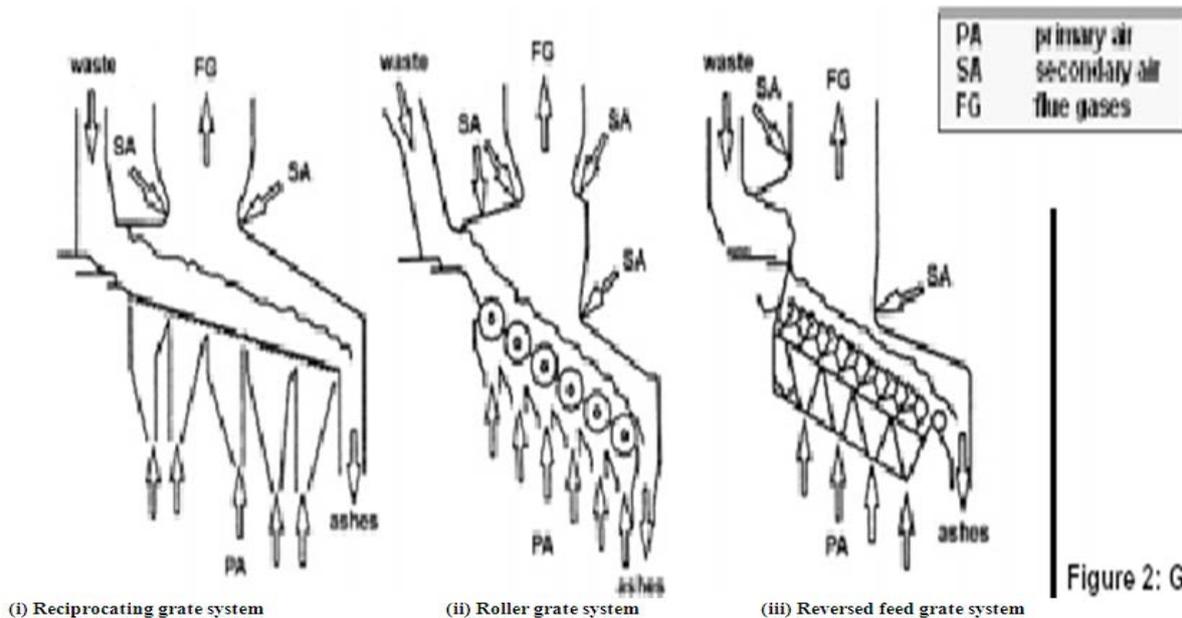


Figure 1. Grate types (schematic) for municipal waste incineration plants [9]

Figure 2: G

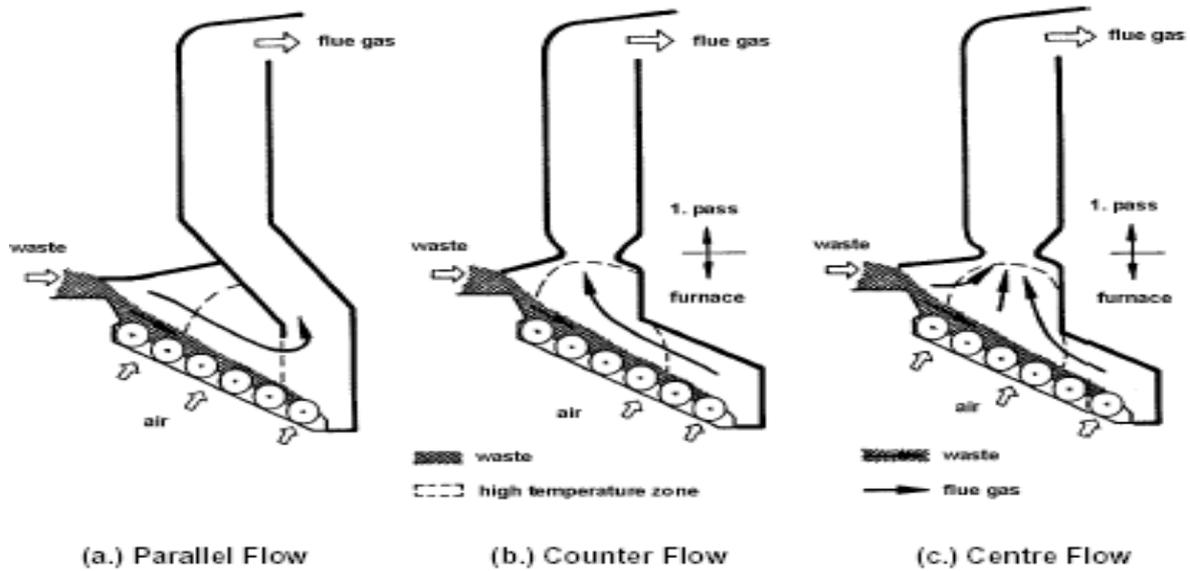


Figure 2. Furnace geometries (schematic) for municipal waste incineration plants [9]

Table 1. Advantages and disadvantages of the different furnace geometries [9]

	Parallel flow	Counter flow	Centre flow
Advantages	Pyrolysis gases pass through the hottest area and therefore are burnt out satisfactorily. Suitable for lower LCV.	Energy transfer from the main combustion area to the drying and gasification area. Suitable for higher LCV.	Very flexible for different heat release distributions on the grate. Suitable for wider range of LCV.
Disadvantages	Energy transfer from main combustion area to ignition area is just by radiation.	Pyrolysis can bypass the hottest area and may cause burnout problems	Flow and mixing pattern after the passage to the one (1) pass sensible to disturbances.

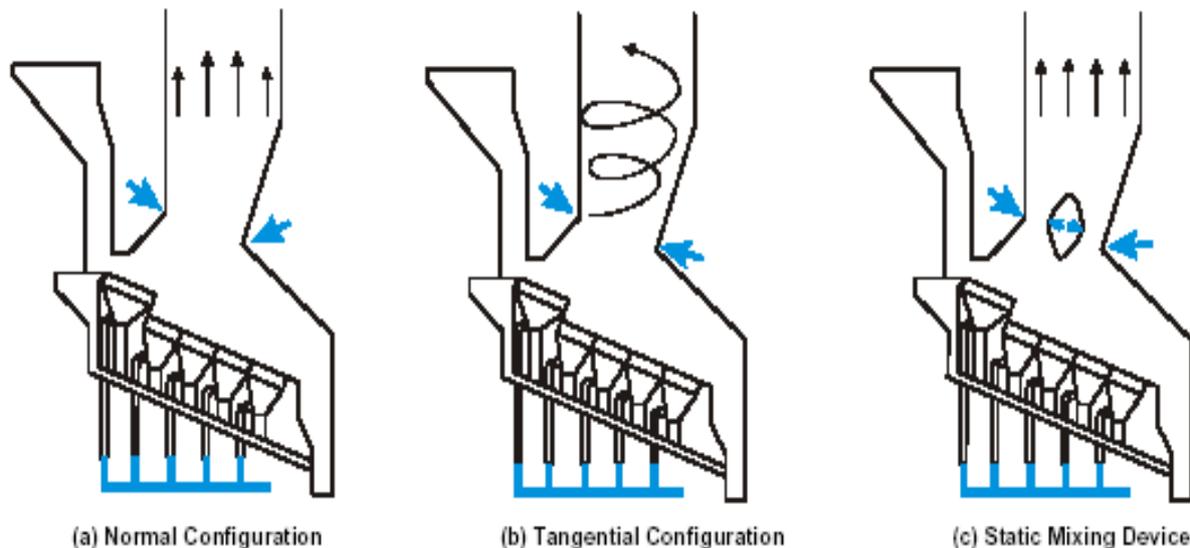


Figure 3. Different secondary air injection concepts in municipal wastes incineration [9]

New developments are tangential nozzle arrangements—similar to the firing concepts of tangentially fired boilers (Figure 3b) or static mixing devices in the flue gas pass (Figure 3c).

3. Factors Affecting Incineration

A good combustion process depends on the waste repartition on the grate, the effectiveness of the waste mixing to allow for a good contact between the combustible and air, a sufficient temperature in the drying-pyrolysis zone (primary air pre-heating and intense radiation) as

well as a sufficient post combustion chamber temperature. The temperature, time, turbulence ("3T") rule's summarizes the fundamental parameters to be respected:

- (a) The temperature of combustion must be regulated and adequate (between 850 and 1200°C) in the post combustion chamber to destroy the gaseous pollutants. The combustion temperature should be enough to allow for the drying and ignition of the MSW as it reaches the grate and should generally not exceed 1200°C due to the mechanical properties of the grate material, the combustion chamber refracting walls and the various other devices (boiler, economizer, super heater, etc...). The thermal power being fixed,

temperature excesses in the post combustion chamber affect the flue gases treatment devices, which work around 200°C.

- (b) The residence time in the furnace should be sufficient enough (45 minutes to 1 hour for wastes and 2 to 4 seconds for gases in the post combustion chamber). The rule of 850°C /2s imposes a residence time of 2 seconds at 850°C with a minimum oxygen concentration of 6% in a controlled and homogeneous manner in critical working conditions after the last injection of air [9].
- (c) Gas flow in the post combustion chamber must be turbulent. This turbulence is ensured by the high velocity secondary air injection (50 to 80 ms⁻¹) in the burning zone using small diameter nozzles. This injection brings in an intimate mixing of combustion gases and fresh air, which theoretically oxidizes the un-burnt fraction and the gaseous pollutants.

4. Mass and Energy Balances

Heat balance calculations can be made by using the mean specific heat of each component of the products of combustion over the range of temperatures; alternatively, the enthalpy (kJ/kg) of the gases at the temperatures involved may be used [10].

4.1. Heating Value or Calorific Value

The calorific value of refuse (fuel) is a complex function of the elemental composition of the refuse. It is a property of fundamental importance. Test values are obtained by burning a 1g or 2g samples in oxygen at 25 MPa pressure in a bomb calorimeter (ASTM standard D 2015-73). Two different heating values are cited. The value obtained directly from the calorimeter is called the gross heating value or high heating value (HHV) since all water formed remains a liquid. A second heating value, the net heating value or low heating value (LHV) is formed if water is allowed to evaporate.

The LHV is of the more practical of the two as it is the heating value of the fuel minus the latent heat of the vaporization of the water vapour present in the exhaust gases. The LHV can be found by subtracting 2.395 kJ/ (grams of water in the combustion product) from the calorific value given in kJ/gram [11].

$$Q_{low} = Q'_{high} - 2.395w_m \text{ kJ} / \text{g}, \quad (1)$$

where w_m -(H₂O in combustion product) kg original refuse.

It should be noted that; kg H₂O-total amount of water, initial moisture in sample plus water formed in combustion.

Also, the ash free, dry heating value (LCV) can be calculated to within 2% accuracy by using the Dulong-Berthelot formula [11]:

$$Q'_d = 81.37C' + 345 \left[H' - \frac{(O' + N' - 1)}{8} \right] + 22.2S, \quad (2)$$

Q'_d -cal/g while elements are in weight %.

The “as-burned” (i.e.: moisture and ash free) heating value:

$$Q'_{AB} = Q'_d \left[1 - \frac{(M'_0 - A'_s)}{100} \right]. \quad (3)$$

However, the Bioe and Vondracek formula is more often used in the municipal wastes field as it gives slightly different and more accurate results [10].

$$Q'_d = \left(\begin{array}{l} 15,122[C] + 50,000[H] \\ +4,500[S] + 2700[N] - 4771[O] \end{array} \right), \quad (4)$$

where: M'_0 and A'_s are the mass percentage of moisture and ash respectively at the point of combustion.

The total heat release

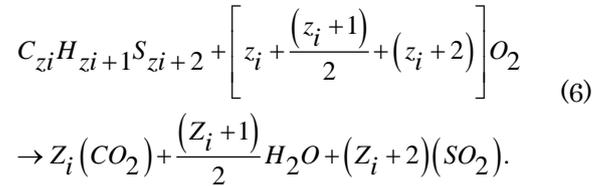
$$Q = 81.0C' + 341.5(H' - \frac{O'}{8}) + 21.8S'. \quad (5)$$

In cal/g, where $H' - \frac{O'}{8}$ is the available hydrogen.

4.2. Theoretical Air to Fuel Ratio

The oxygen for the combustion process comes from the air supplied to the burners. In design, it is in customary practice to supply sufficient air for complete combustion plus excess air to allow for incomplete mixing. For any fuel, the moles of air theoretically required for complete combustion are determined from the moles of oxygen required [11].

For fuel containing carbon, hydrogen and sulphur, we may write a balanced equation of the form:



Since air consist of 79% N₂ and 21% O₂ the ratio of N₂ moles to O₂ moles is 79/21 = 3.76.

Fuel is mostly burned in the oxygen from air.



Where:

$$\begin{aligned} A = \left\{ \begin{array}{l} C_{z_i} H_{z_i+1} S_{z_i+2} \\ + \left[z_i + \frac{(z_i+1)}{2} + (z_i+2) \right] (O_2 + 3.76N_2) \end{array} \right\}, \\ B = \left\{ \begin{array}{l} Z_i (CO_2) + \frac{(Z_i+1)}{2} H_2O + (Z_i+2)(SO_2) \\ + \left[z_i + \frac{(z_i+1)}{2} + (z_i+2) \right] \times 3.76N_2 \end{array} \right\}. \end{aligned}$$

For each mole of carbon and sulphur in fuel, 4.76 mole of air is required [11] for each kilogram atom of available hydrogen in the fuel, 4.76/2 = 2.38 mol of air is needed. Since the molecular weight of air is 28.9g/g mol the mass of air required per gram of carbon $m'_{a/c}$ is:

$$m'_{a/C} = \left(\frac{1g}{12gC / g \cdot mol \cdot C} \times \frac{4.76g \cdot mol \cdot air}{g \cdot mol \cdot C} \times 28.9g / g \cdot mol \cdot air} \right) = 11.47 \frac{g \cdot air}{g \cdot carbon}, \quad (8)$$

$$m'_{a/S} = 43g \cdot Air / g \cdot sulphur,$$

$$m'_{a/H} = 34.4g \cdot Air / g \cdot hydrogen.$$

Hence

$$m'_a = \left[11.47C' + 34.4 \left(H' - \frac{O'}{8} \right) + 4.3S' \right] \frac{1}{100}, \quad (9)$$

where m'_a -grams of air required for complete combustion of 1g of fuel (mass air/mass fuel).

4.3. Excess Air Requirement

When incinerating refuse or fuel is being oxidized, some minimum air is stoichiometrically required for combustion. For incineration processes however, far more air is supplied than is actually required on a purely chemical reaction basis. The percentage excess air is defined as:

$$P = 100 \left[\frac{\left(\frac{\text{Amount of air supplied}}{\text{Amount of refuse}} \right)}{\left(\frac{\text{Amount of air needed}}{\text{Amount of refuse}} \right)} - 1 \right], \quad (10)$$

P-excess air.

The 50 percent excess air correction method [12] suggests 50% excess air for more diluted flue gases. Consequently the contaminant concentration is a function of the percentage excess air. This factor may vary to 4.0 and depends on waste characteristics such as volatility, heating value, bulk density, etc. Generally low heating value-low volatile waste has an excess air factor of 2.0. As heating value and volatility increase so will this factor [12]. Furthermore, the excess air, together with moisture content, waste combustible content and waste combustible heating value greatly affect the adiabatic (theoretical) flame temperature.

As a first order approximation, of any combustible material, the as-received (without any removal of moisture) waste heating value can be used [10].

Table 2. Typical excess air values for various incinerator types [10]

Incinerator type	Excess air range (%)
Fluid bed	40-60
Gaseous waste	10-15
Liquid waste	15-30
Multiple chamber (retort, in line)	100-200
Multiple hearth	100-125
Rotary kiln	100
Water wall	65-100

4.4. Incinerator Internal Sizing Requirements

The minimum internal volume of the incinerator is a basic consideration of critical importance. This volume is a function of the total heat released per hour from the burning refuse. The internal volume requirement excludes the ash pit and can be evaluated from [12]:

$$Q_{thr} = \frac{M_b}{Q_{av}}, \quad (11)$$

$$V_{i \min} = \frac{Q_{thr}}{258,750W / m^3}. \quad (12)$$

Where:

$V_{i \min}$ -minimum theoretical internal volume (m^3)

Q_{thr} -total heat released (W)

M_b -mass of refuse burned per hour (kg/hr)

Q_{av} -average heating value (kJ/kg)

It should be noted that if auxiliary fuel (gas or oil) will be burned, its heating value should be added to the total heat release determined by the equation (11).

Depending on the nature of the predominant refuse to be treated (i.e.: wet, dry, municipal, medical, etc) two types of incinerators are considered: Type I-dry garbage and rubbish incinerator and type II-wet garbage incinerators. For these types, the basic design requirements are as follows [13]:

Table 3. Preliminary design factors of MSW incinerators [13]

	Type of incinerator	
	I	II
Effective grate area per kilogram of refuse per hour(square meter)	0.022	0.04
Ratio of hearth area to grate area	1: 1	1: 1
Effectiveness of hearth area in terms of grate area (percent):		
Firebrick hearths	60	60
C.I.(Cast Iron) grate bars	80	80
Horizontal cross-sectional area of mixing chamber in terms of effective grate area(percent)	25	20
Horizontal cross-sectional area of combustion chamber in terms of effective grate area(percent)	60	30
Cross-sectional area of flue in terms of effective grate area(percent)	25	10
Cross-sectional area of stack in terms of effective grate area(percent)	22	10
Ratio of height of arch above grate to width of furnace not to exceed	1: 1	1: 1

The overall specific heat over the range of temperatures is somewhat difficult to calculate [10] because the variation with temperature is not linear. The average specific heat is given for ranges of temperatures, assuming combustion air enters at 21°C [10].

4.5. Chamber Sizing

Chamber sizing is based on heat release. There is a limit to the quantity of heat that can be released in a particular furnace chamber. Heat release is the amount of heat generated when combustible material burns. When combustible gases burn or when liquids burn in suspension, the volume of the furnace will limit the total amount of heat released within that chamber. The furnace volume

must be large enough to allow release of the heat generated by the anticipated waste and the supplemental fuel [12].

When a solid or sludge waste is fired, the heat release (heat generated per chamber volume or hearth area per hour) of that waste is characterized by the area of the surface on which it is placed, i.e., the hearth or grate. In general, an incinerator volume can be calculated by two approaches: by incinerator dimensions (Equations 11 and 12) or by heat release rate. Table 4 and Table 5 list typical heat release values from supplemental fuels and for some common incinerator systems respectively.

$$V = \pi (r^2)(L). \quad (13)$$

r=radius of the kiln, m.

L= length of the kiln, m.

Table 4. Typical Heat release Rates of various incinerator systems [10]

Incinerator type	Heat release (kJ/m ³ -h)
Fluid bed*	350,000-500,000 (bed area)
Multiple chamber	300,000-400,000
Multiple hearth	300,000-400,000
Multiple hearth heat release*	250,000-350,000 (hearth area)
Gaseous waste incinerator	3,000,000-10,000,000
Liquid waste incinerator	1,000,000-3,000,000
Rotary kiln	500,000-1,500,000
Solid waste grate*	150,000-300,000 (grate area)

*Heat release values based on surface area not volume.

Table 5. Supplementary Fuel Combustion characteristics [10]

No. 2 fuel oil * heat content 39,100 kJ/L, density 0.91 kg/L			
Percent excess air (%)	10	20	30
kg air/L	13.78	15.03	16.29
kg dry gas/L**	13.84	15.09	16.34
kg water/L**	1.04	1.05	1.07
Natural gas, heat content 37,256 kJ/m ³ , density 0.80 kg/m ³			
Percent excess air (%)	5	10	15
kg air/m ³	12.09	12.66	13.24
kg dry air/m ³ **	11.4	11.97	12.55
kg water/m ³ **	1.65	1.65	1.66

*residual fuel oil;

**products of combustion.

4.6. Incinerator Residence Time

Residence time means the length of time that the combustion gas is exposed to the combustion temperature in an incinerator. This is an important criterion in the design of waste incinerators, and it is calculated at the typically mandated 990° or 1100°C. This time is typically measured from the location where the last over fire air ports are placed [14]. Residence time is determined by the velocity of the gases and the distance they travel through the combustion chamber as shown below:

$$t = \frac{V}{q}. \quad (14)$$

t=residence time, sec.

V=combustion chamber volume, m³.

q=combustion gas flow rate, m³/s.

$$t = \frac{\left(\text{chamber volume, m}^3 \right)}{\left(\text{gas volume flow rate, m}^3 / \text{s} \right)}, \quad (15)$$

or

$$t = (\text{chamber length, m}) / (\text{gas velocity, m/s}). \quad (16)$$

4.7. Incinerator Retention time

Retention time means the length of time that the solid materials remain in the primary combustion chamber during incineration. It can be expressed as follows:

$$t = 0.19 \frac{L}{DSN}. \quad (17)$$

t= retention time in minutes.

L= kiln length in metres.

D= kiln diameter in metres.

S= slope of kiln m/m.

N= rotational velocity in rpm.

It should be noted that residence time and retention time have a great influence on the flue gas temperature as shown in Table 6.

Table 6. Residence times in the after-burning zone as a function of the temperature [9]

Flue gas temperature (°C)	Residence time	
	After burning zone (sec)	Furnace + after burning zone (sec)
950	0.3	1.4
900	1.25	2.35
850	2.15	3.25
800	3.3	4.4

4.8. Air Injection

The total combustion air (having an over-stoichiometry of 1.3 to 1.8) is mainly divided into primary and secondary air as to control the combustion conditions to give near-stoichiometric conditions [15]. The partitioning ratio of primary to secondary air is between 80/20 (old plants) to 40/60 (for new plants) [9].

The task of secondary air is to complete the burnout of the hydrocarbons and carbon monoxide [16] say to bring the air-to-fuel to the required level. Furthermore, secondary air can be used as a mixing device for flue gases. Basically, secondary air injection concepts are Normal, Tangential configurations; Static mixing device and rotating cylinder with additional secondary air nozzles respectively. A typical position for secondary air injection is to the transition area between the furnace and the first pass of the boiler. Primary air or under-bed is mainly supplied from under the bed.

4.9. Grate Cooling

The grate bars are cooled by the primary combustion air [17]. Thus the degree of freedom for the stoichiometry in the primary combustion area (furnace) is reduced [18]. However, the cooling effect is limited for every high calorific values of the refuse and the bars can be damaged or

destroyed by intense heating. To this effect water cooled bars have been developed and successfully applied in various plants [19]. The major advantage here is that the primary stoichiometry can be optimized with respect to the burnout behaviour or gaseous emissions such as CO and NO.

4.10. Turbulence and Mixing

In order to achieve high combustion efficiency in incinerators, it is particularly important to achieve good mixing between the primary combustion products (primarily CO and organics) and a stoichiometric excess secondary combustion air [20]. This mixing can be promoted by a range of physical parameters as well as by promoting highly turbulent flow of gases [6, 21]. Physical parameters which are used to promote mixing include:

- Location and direction of secondary air jets,
- Volume and pressure of secondary air addition,
- Changes in flow direction (s),
- Other baffling techniques (dual-mode firing, tangential firing, inclined firing, etc.).

To the provision of overall mixing using the above methods, promote local mixing of the combustible gases and air. The degree of turbulence is typically assessed by use of the Reynolds number Re .

$$Re = \frac{VD}{K}. \quad (18)$$

V = average velocity, m/s.

D = diameter (or equivalent diameter) of flow stream, m.

K =kinematic viscosity, m^2/s .

For a rectangular or other shape of flue, the equivalent diameter (D_e) should be used.

$$D_e = \frac{2ab}{(a+b)}, \quad (19)$$

where a and b are the dimensions of the sides of the rectangle, and this value of D_e would be used in lieu of D when applied to noncircular cross section.

4.11. Air Injection Nozzles Arrangement

Air injection nozzles arrangement and number play an important role in MSW incineration. Basically, four arrays are used for an optimum operation with the following injection angles: Nozzle array 1: (-30°), nozzle array 2: (-10°), nozzle array 3: (-5°) and nozzle array 4: (0°) with respect to the post combustion chamber wall. Such arrangement leads to uniform temperature, velocity and species distributions resulting from the improvement to the mixing process. Furthermore, it could reduce the concentration of the incomplete product of combustion CO by 150% [22].

5. Basic Flame Kinetics and Heat Transfer

5.1. Flames Structures/Burners

In an incinerator combustion chamber, waste is thermally decomposed through an oxygen-deficient,

medium-temperature combustion process ($800\sim 900^\circ C$) producing solid ashes and gases. This chamber includes fuel burners usually used to start the process. These burners often use oil or gas as combustible. To direct the energy (heat) deposition, the burner direction or altitude can be changed. To suit a particular firing diagram, burners can be pointing downward in the drier while directed up in the secondary chamber [11]. The most frequently used burners are circular in design or cell type in design [23]. The burners usually fire horizontally or at slight inclinations and are often mounted in the corners of the walls so that the flames can be directed in a circular pattern referred to as tangential firing [11]. Another popular arrangement is for the burners to be inserted in the walls at opposite sides of the furnace: This arrangement is called opposed firing.

Gases produced in the lower chamber are burned at high temperature ($900\sim 1200^\circ C$) by another fuel burner in the post combustion chamber using an excess air to minimize smoke and odours [24].

5.2. Estimation of Fuel Requirements

The fuel requirements are known from knowledge of the total steam requirements and the heat released in the furnace per unit mass of fuel. It is achieved by solving the heat balance around the entire furnace [25]:

$$\begin{aligned} & \text{Enthalpy of entering streams} + \text{heat of reaction} \\ & = \text{enthalpy of exit streams}, \end{aligned} \quad (20)$$

$$D = E + F, \quad (21)$$

where:

$$\begin{cases} D = \dot{m}_a h_a + \dot{m}_F Q_0'' + \dot{m}_1 h_1 \\ E = \dot{m}_1 h_{ss} + \dot{m}_F \left[\sum_i \frac{\dot{m}'_i}{MW_i} (\bar{c}_p)_{m,i} (T_{exit} - T_0) \right] \\ F = \dot{m}'_w h_{fg} \dot{m}_F + \dot{m}'_A (\bar{c}_p)_A \dot{m}_F (T_{ash} - T_0) \end{cases}$$

In most instances, preheating of the air is done within the furnace [26], the air enters the furnace at room temperature and its enthalpy is negligible, thereby eliminating the $\dot{m}_a h_a$ term.

$$\begin{aligned} & \dot{m}_1 (h_{ss} - h_1) \\ & = \dot{m}_F \left\{ \begin{aligned} & Q_0'' - \sum_i \frac{\dot{m}'_i}{MW_i} (\bar{c}_p)_{m,i} (T_{exit} - T_0) \\ & - \dot{m}'_w h_{fg} - \dot{m}'_A (\bar{c}_p)_A (T_{ash} - T_0) \end{aligned} \right\}, \end{aligned} \quad (22)$$

or

$$\dot{m}_F = \frac{\dot{m}_1 (h_{ss} - h_1)}{\left[Q_0'' - \sum_i \frac{\dot{m}'_i}{MW_i} (\bar{c}_p)_{m,i} (T_{exit} - T_0) - \dot{m}'_w h_{fg} - \dot{m}'_A (\bar{c}_p)_A (T_{ash} - T_0) \right]}, \quad (23)$$

where

\dot{m}_a = mass of air required per hour,
 h_a = specific enthalpy of entering air,
 \dot{m}_F = mass of fuel required per hour,
 Q_0'' = higher heating value (standard heat of reaction with product water as liquid per unit mass of fuel).
 \dot{m}_1 = mass of feed water required per hour,
 h_1 = specific enthalpy of entering feed water,
 h_{ss} = specific enthalpy of superheated steam,
 m'_i = mass of product i in flue gas per unit mass of fuel,
 m'_A = mass of ash per unit mass of fuel,
 $(\bar{c}_p)_{m,i}$, MW_i = mean molar specific heat of product i and molecular weight of product in flue gas, respectively.
 $(\bar{c}_p)_A$ = mean specific heat of ash, kJ/kg.°C,
 m'_w = mass of water vapour discharged in the flue gas per unit mass of fuel,
 h_{fg} = heat of vaporization of water at 1 atm, kcal/kg,
 T_0 = base temperature (temperature at which Q_0'' is evaluated).

5.3. Theoretical Flame Temperature

The heat released by combustion (Q) raises the temperature of the flow stream (W) of the products of combustion. The temperature rise (ΔT) of the products is dependent on the specific heat (c_p) of the mixture of gases.

The calculation of the theoretical flame temperature is based on the assumption that the heat released by the combustion process is completely absorbed by reaction products and excess air. The temperature of the product gases is the flame temperature of interest [10], [11], [27] often termed T_{exit} , T_2 or T_f .

$$Q = W \times c_p \times \Delta T = W \times c_p \times (T_2 - T_1), \quad (24)$$

$$T_2 = T_1 + \frac{Q}{(W \times c_p)}. \quad (25)$$

The typical value of c_p [10] can be used to make fairly accurate estimate of the theoretical flame temperature and the final temperature of the gases after adding excess air.

5.4. Actual Flame Temperature

Actual flame temperatures are always lower than the adiabatic (theoretical) flame temperature since there is always a substantial quantity of heat released to the environment. Since the heat release per unit mass of fuel burned Q_1 , is known as a function of temperature, the heat balance becomes.

$$\begin{aligned} & \sum_j^n (Q'_o)_j \\ & = \left[\sum_i \frac{m_i}{MW_i} (\bar{c}_p)_{m,i} (T_{\text{exit}} - T_0) + m_w h_{fg} \right] \\ & = \left[-(\bar{c}_p)_{m,a} (T_{\text{air}} - T_0) \frac{m_a}{MW_a} + Q_1 m_F \right]. \end{aligned} \quad (26)$$

Note that the heat released from combustion is the lower heating value (LHV), because the moisture content is not condensed in order to absorb the latent heat of condensation. At this stage, (during design) there is need to simultaneously determine the flame temperature and the value of Q_1 . The evaluation of these quantities require the knowledge of the interchange of energy between an isothermal black enclosure with area and a gray gas flame filling the enclosure. The rate of energy radiation by the flame q_{rad} is given by:

$$q_{\text{rad}} = A_T \sigma T_f^4 (\varepsilon_G)_f. \quad (27)$$

However, the wall, at absolute temperature T_c , will also radiate energy and a portion of this will be absorbed by the gas. The rate of energy absorption by the gas is obtained from:

$$q_a = A_T \sigma T_c^4 (\alpha_G)_c \approx A_T \sigma T_c^4 (\varepsilon_G)_c. \quad (28)$$

The net rate of radiation interchange between the gas and the black enclosure is then

$$(q_{\text{rad}})_{\text{net}} = A_T \sigma \left[(\varepsilon_G)_f T_f^4 - (\varepsilon_G)_c T_c^4 \right]. \quad (29)$$

Now considering the net rate of interchange between the entire volume of gas and a portion of the enclosure (e.g. area subtended by boiler tube) having area A_c , we may write

$$q_{TC} = \sigma \left(A_T \mathfrak{I}_{TC} T_f^4 - A_c \mathfrak{I}_{CT} T_c^4 \right). \quad (30)$$

In this case,

$$\begin{aligned} \mathfrak{I}_{TC} &= (\varepsilon_G)_f \frac{A_c}{A_T}, \\ \mathfrak{I}_{CT} &= (\varepsilon_G)_c. \end{aligned} \quad (31)$$

To extend this concept to more realistic case of the combustion chamber lined with non-black, cold tubes which absorb radiation and a refractory that reradiates all the radiation which fall on it. Equation (30) still gives the interchange between the flame and area A_c , which now is the effective area but \mathfrak{I}_{TC} and \mathfrak{I}_{CT} must be re-evaluated to include effect of the refractory and the emissivity of area A_c .

$$\mathfrak{I}_{TC} = \frac{1}{\frac{A_T}{A_c \varepsilon_t} + \frac{1}{(\varepsilon_G)_f} - 1}, \quad (32)$$

and

$$\mathfrak{I}_{CT} = \frac{A_T / A_c}{\frac{A_T}{A_c \varepsilon_t} + \frac{1}{(\varepsilon_G)_c} - 1}. \quad (33)$$

Since the refractory reradiates all the radiation it receives, its emissivity does not enter into the calculation.

The total rate of heat transfer from the flame in the boiler region, q , is the sum of the radiative heat transfer plus convective heat transfer.

$$q = q_{TC} + hA_C(T_f - T_c), \quad (34)$$

where h is the convective heat-transfer coefficient.

In most cases the convective heat transfer is only a small fraction of the radiation heat transfer in the combustion chamber and is often neglected. Assuming sufficient turbulence so that exit gas temperature and average flame temperature are equal and the absolute temperature of the absorber is less than one-half the flame temperature, the radiation from the cold surface is not a major factor and use of a single emissivity is allowable.

Equation (30) thus becomes;

$$q_{TC} = \sigma A_T \epsilon_{TC} (T_f^4 - T_c^4). \quad (35)$$

Where

A_c =effective tube area.

ϵ_c =emissivity of boiler tubes.

A_T =total surface area of enclosure (tube area+refractory).

6. Conclusion

Incinerator design cannot be effected without the knowledge of the basic combustion science, engineering and the characteristics of the waste.

No one method can be used alone to handle all waste streams effectively, thus an integrated waste management system which not only deals with different methods of treating wastes but also issues such as waste streams, collection, environmental benefits, economic optimization and social acceptability.

Note

In order not to alter the authenticity of some tables from the literature; they are reported as they are with their original units.

References

- [1] Babcock & Wilcox Vølund. Computational Fluid Dynamics in Waste-to-Energy Technology. Pamphlet, June 2003.
- [2] Harvey Alter. The History of Refuse Derived Fuels. Resources and Conservation. Elsevier Science Publishers, Amsterdam, Netherlands, 1987.
- [3] Yongxiang Yang, Marc J. A. Pijnenborg and Marcus A. Reuter. Modelling of the fuel stream and Combustion in a Rotary-Kiln Hazardous Waste Incinerator. 3rd International Conference on CFD in the Minerals and Process Industries CSIRO, Melbourne Australia; December 2003: 10-12.
- [4] Gluszynski Pawel. Types of Incineration based on Technological Features. Waste Prevention Association, Greenpeace International publication, 1995.
- [5] Swithenbank J., V. Nassezadeh, R. Goh and R.G. Siddall. Fundamental Principles of Incinerator Design; Asia-Pacific Journal of Chemical Engineering; 7 (5-6): 623-640.
- [6] Gohlke Oliver, Sri M. K. and Martins E. A.. New grate-based waste-to-energy system produces inert ash granulates. Waste management World, pp: 37-46, May-June 2003.
- [7] Connett Paul. Municipal Waste incineration; A Poor Solution for the Twenty First Century. 4th Annual International Management Conference on Waste-to-Energy, Amsterdam, 24 November 1998
- [8] Kerdsuwan Somrat. Design of 50 kg/hr controlled Air Hospital Waste Incinerator. The Waste incineration Research Centre (WIRC) publication, Bangkok Thailand, 2005.
- [9] Goerner Klaus. Waste Incineration State of the Art and New Development. IFRF Combustion Journal, Article number 200303, July 2003.
- [10] Lee C. C. and Huffman G. L.. Solid Waste Calculations: Thermodynamics used in Environmental Engineering. Waste Incineration Handbook; U.S. Environmental Protection Agency, Cincinnati, Ohio, 2001: 2-47, Part 2.
- [11] Weisman Joel and Roy Eckart. Modern Power Plant Engineering. 2nd edition, Prentice Hall International, New Delhi, 1985.
- [12] Cheremisinoff N. Paul and Richard A. Young (1976) *Pollution Engineering practice Handbook*. Ann Arbor Science publishers Inc., Michigan.
- [13] Kavanaugh Paul F.. *Engineering Design: Incinerators Mobilization Construction*. US Department of the Army Corps of Engineers, Washington, D.C., April 1984.
- [14] Hasselri Floyd, C. C. Lee; Huffman G. L.; Thomas C. Ho. *Basic Combustion and Incineration*. Mc Graw-Hill, 2001.
- [15] Jorgensen K. and Madsen O. H.. (2002) Modern control system for MSW plants. *Transactions of the institution of Chemical Engineering*, 78 part B, 2002: 15-20.
- [16] Berkowitz N.. *An Introduction to Coal Technology*. Academic Press New York, 1979: 155-159.
- [17] El Asri R. and Baxter D.. Process control in municipal solid waste incinerators. *Waste Management and Research*, 22 (3): 177-185, 2004.
- [18] Brunner Calvin R.. *Handbook of incinerator Systems*. McGraw-Hill, New York, 1991.
- [19] Gierend Chr. and Born M.. *Fuzzy Control in waste incineration plants; experiences in several types of plants*.ACHEMA Fair, Frankfurt/M, Germany, 22/27. June 2000.
- [20] Skrotzki B. and Vopat W.. *Power Station Engineering and Economy*. Mc Graw-Hill; New York, 1960.
- [21] Randall T., Marcel Dekker and Simeon N. Oka. *Waste-to-Energy in the United States*. Westport, Connecticut: Quorum Books, ISBN 0-89930-844-9, 1994.
- [22] Klasen Thomas and Goerner Klaus. Numerical Calculation and Optimization of a large Municipal Waste incinerator plant. *2nd International Symposium on Incineration and Flue gas treatment technologies*, Sheffield University, UK, 1999.
- [23] Krenz J. H.. *Energy.: Conversion and Utilization*. Allyn and Bacon, Boston, 1981.
- [24] Pruss A., Giroult E. and Rushbrook P.. *Safe Management of wastes from health-care activities*. World Health Organization-Geneva, 1999.
- [25] Ashner F.. *Planning Fundamentals of Thermal Power Plants*. John Wiley & Sons, New York, 1978.
- [26] Lyons J.. *Optimizing Designs of Fossil Fired Generating Units*. Power Engineering, pp: 50-56, February 1979.
- [27] Perry J. H.. *Chemical Engineers Handbook*. 4th edition, Mc Graw-Hill, New York, 1976.