TEMPERATURES AND AIR DISTRIBUTIONS IN LARGE RECTANGULAR INCINERATOR FURNACES

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ABSTRACT

Proper design and operation of municipal incinerators with combustion and excess air delivered in correct quantities, at the right place and at the right time will result in controlled optimum operating furnace temperatures, reduced refractory maintenance cost by virtue of limited slagging on refractory walls, reduced stoker maintenance cost, considerable reduction in potential particulate emissions.

This paper includes some basic theories of furnace design and their application in modern refuse burning furnaces, some preliminary and partial test data, observations based on long experience in the operation of various types of municipal incinerators, and finally, some recommendations for the design of combustion-air systems. The paper is intended to be an incentive for more detailed research and testing programs aimed specifically at the effects of overfire air distribution on temperature gradient and combustion efficiency of a furnace.

INTRODUCTION

Twenty-five years ago when municipal refuse contained 65 percent garbage and 35 percent rubbish (average moisture content of 55 percent auxiliary burners and/or underfire air preheaters of silicon carbide or cast iron tubes were usually included in the incinerator design as the only means of maintaining adequate combustion temperatures and guarantying a residue with minimum percentage of combustibles. Blower-powered overfire air jets were not required or incorporated in refuse burning incinerator design.

With the end of World War II, salvaging of newspaper became uneconomical and in some instances a nuisance. The newspapers found their way to the incinerator furnaces and caused not only increased but often uncontrollable furnace temperatures. Emergency overfire air doors seemed to provide the answer. (In 1937, two 90T/24 hours mutually assisting cell-type units were built for the City of Rochester, N. Y. with manually operated overfire air doors above the stoking doors of each cell. The purpose of these doors was dilution, i.e., admission of cooling air, if temperature exceeded safe limits.) The doors, manually or automatically operated, were located at the top of ignition and/or combustion chambers. Some plants, incorporating this design on circular mechanically stoked furnaces are operating successfully today. (Jersey City, N. J. — 600T/24 hrs. — 4 units — completed 1957, Scarsdale, N. Y. — 150T/24 hrs. — 2 units — completed 1959.)

The increased quantity and changing quality of the municipal refuse over the past fifteen years led to a small revolution in incinerator design. When refuse reached the heat value of a lower grade fuel, the designer was compelled to use design parameters approaching criteria (such as furnace volume and overfire air application) used in the design of coal furnaces. To meet the demands of changing refuse quality, New York City advertised in 1948 for bids to build a 800 T/24 hrs plant with “travel-
ling grate” stokers and continuous feed system. This plant, as well as a number of plants of the same general type with improvements and variations built since, shows that although some similarity exists between a modern incinerator and a coal-fired boiler, the gap between the two furnace designs will remain wide because of the variable characteristics of typical municipal refuse.

FURNACE TEMPERATURES

Furnace temperature and its control has been discussed in a number of publications, mostly from the point of view of the combustion process but these publications have partially or completely disregarded furnace configuration and combustion and excess air distribution. The ultimate goal of any furnace designer is to obtain a furnace with as nearly uniform temperature distribution throughout the combustion chamber as possible. Although it is impossible to obtain such equilibrium conditions in the primary chamber of an incinerator, every effort should be made to at least approach the ideal solution. Some recent installations seem to disregard this important requirement.

Nearly all furnaces designed before the nineteen forties were characterized by the absence of overfire air and by 100 percent underfire air. Provided that the refuse distribution over the grate area (either manually or automatically stoked) was uniform, the temperature throughout the furnace volume above the refuse bed was nearly constant. In the absence of overfire air and turbulence in the primary chamber, the flame was usually very long and a large portion of the combustion process was completed in a secondary combustion chamber. The products of combustion had been subjected to turbulence and mixing in passing through high velocity flues or deflection baffles between the primary and secondary chambers. The secondary chamber temperature was sometimes higher than the primary chamber temperature. This not unusual effect in this type of incinerator was largely due to the delayed combustion of CO to CO₂ and to the combustion of free carbon in the form of smoke. The delayed combustion occurred only when the temperature was maintained at or above a 1500-1600 F level and sufficient oxygen was present.

As mentioned before, the trend to dryer refuse brought into the picture the use of overfire air jets. Unfortunately, the results of research in the field of the coal burning furnaces were not always consulted in incinerator design. Research data, dating back to 1951 (Bituminous Coal Research, Inc. BCR Aid to Industry 500-300 Revision of Technical Report No. 7, Copyright 1951), show that maximum combustion efficiency is obtained with air jets located in side or end walls of various furnaces only a few inches above the fuel bed, supplying the combustion and excess air to the portion of the flame characterized by maximum luminosity and therefore by maximum radiation heat transfer. The luminosity of the flame is due to the formation of incandescent soot particles and to a lesser degree to CO₂ and water vapor. Side wall air jets, located directly above the fuel bed, reach the richest part of the flame and create turbulence and mixing at the earliest stage before hydrocarbons crack to soot, providing maximum time for the completion of the combustion process.

It has also been observed that air jets located high above the fuel bed have a chilling effect on furnace gases after the flames have lost their luminosity. This is especially evident when the refuse moisture is high and the furnace temperature is below 1400 F. In this case the air jets mix with the furnace gases and cause further cooling below the ignition temperature of soot. The heat transfer occurs mainly by the entrainment of the hot gas by the cold air jet and their progressive mixing without an appreciable transfer by radiation.

Practically all municipal incinerators that include an overfire air system are equipped, with perhaps a few exceptions, with roof nozzles to provide high and/or low pressure air. The roof air nozzles had been introduced into the incinerator design some years ago in an attempt to provide proper turbulence and mixing of air with the flame. Some justification of this design could be based on the waterwall boiler concept. The high efficiency of a waterwall boiler depends on the radiation heat transfer from a long luminous flame to the water tubes. The furnaces are usually relatively high and a constant quality fuel (coal) with adequate underfire air will supply a uniform long flame. In such a furnace, the roof air nozzles will probably provide satisfactory operation, as long as the air jets reach the luminous portion of the flame.

It is difficult, however, to apply the same design parameters to a refractory-lined refuse furnace. The variable characteristics of refuse, which can change not only from one day to the next, but actually from hour to hour, result in different flame length. The effect of the roof air jet, which might be beneficial at one instant, can produce a cooling effect and smoky operation in the next.

It has been observed at several recent plants that unless the refuse contains a high percentage of cardboard and dry paper, it is practically impossible to operate a furnace with roof nozzles without visible smoke emissions.
from the chimney. As a matter of fact, some of these furnaces have to be operated with considerable reduction in capacity when the moisture content in the refuse rises appreciably.

Numerous visual tests and some dust loading and gas analysis tests (by Wisconsin Chemical & Testing Company for Morse Boulger, Div. of Hagan Industries, Inc.) performed on a number of industrial incinerator installations showed that a proper side air distribution can and will provide a smokeless operation. The elimination of smoke is inherently accompanied in all instances by almost complete combustion within the furnace itself (primary chamber), shortening of flame, practically uniform temperature in all parts of the furnace and reduction or elimination of slag.

The shortening of the flame results in a “clean” furnace where the refractory enclosure is exposed to more uniform temperatures transmitted to a great extent by convection from clean, uniform stream of gases. It was found that relatively low furnace temperatures can provide a smokeless operation. The optimum furnace temperatures measured in the furnace itself were found to be 1600-1800°F.

The experience with some high refractory furnaces undoubtedly prompted a new design trend, characterized by a reduced furnace height in an attempt to bring the over-fire air closer to the luminous flame. Unfortunately, the roof nozzle air systems were maintained as part of the new design with some undesirable results such as extreme temperatures in some parts of the furnaces, impingement of flame on furnace walls due to air jets adjacent to side walls, penetration of air jets into the refuse bed, resulting in localized high burning rate, and in some instances penetration of the air jets and entrained flame below the grates. The effect of high temperatures and flame impingement on the refractory enclosures, as well as the effect of the flame and heat penetration below the grates on the mechanical equipment, is highly undesirable and can be eliminated by the use of side wall air jets.

In view of unusually rapid deterioration of refractory enclosures on three recent municipal incinerator plants, due apparently to extreme temperatures and thermal shock, we decided to conduct temperature tests within the furnaces themselves in an attempt to determine the efficiency and accuracy of present automatic furnace temperature control systems. Two series of tests were performed on two different plants.

TESTS

The use of conventional thermocouples requires a location remote from the flame that will minimize their rapid deterioration from to flame temperatures. Most of the plant furnace thermocouples are therefore located at the furnace exit or in the flue, yet they are used as controlling media to maintain constant furnace temperature. We inserted temporary thermocouples, connected to a

FIG. 1 PLANT NO. 1. SCHEMATIC FURNACE LAYOUT
portable potentiometer, into a zone where maximum furnace temperatures were anticipated.

Fig. 1 shows the furnace of Plant No. 1, which is a low set furnace with two rows of four roof air nozzles each and a series of side wall air nozzles at the end portion of the horizontal travelling grate.

The deterioration of the refractory walls seemed to be more pronounced in the vicinity of the roof air jets, and a visual observation indicated that the air jets adjacent to the side walls created a "torch" effect when the refuse was burning with a long flame. The "air torch", distinguishable by a bright yellow to white color, reached almost to the surface of the refuse bed, causing the adjacent flame to impinge on the furnace walls. The thermocouple was therefore inserted through the side wall of the furnace below the roof nozzle; it protruded approximately 24 in. from the hot face of the wall into the furnace.

A spare (plant) chromel-alumel thermocouple with stainless steel protective well was used, and the opening around the well was packed with ceramic fibre to minimize cold air infiltration. The calibration was done at ambient temperature with a mercury thermometer. The stationary plant thermocouple was of identical type, inserted in an airtight capped stainless steel sleeve and was calibrated sometime before the test by the instrumentation manufacturer. The series of readings were started approximately 10 minutes after the furnace was put into operation.

The graphic representation for Tests 1 and 2 of the variation of the furnace and the furnace exit temperatures (Fig. 2), taken simultaneously, show a definite time delay between the rise of the temperature in the furnace and the reaction of the termocouple at the furnace exit. As a matter of fact, in some instances the controlling plant...
thermocouple does not respond at all to the rising temperature in the furnace, apparently because of dilution and stratification of the gas stream due to overhead air from roof manifold No. 2 and side wall air nozzles which are located at the end of the furnace.

The log shows also that the "furnace" temperature decreased somewhat after the roof nozzle above the temporary thermocouple was closed, indicating that the localized "air torch" effect was minimized.

The values of the information obtained at Plant No. 1 was of limited importance. We decided therefore to perform a similar test on Plant No. 2.

Plant No. 2 furnace (Fig. 3) is of a medium height with 4 rows of 6 roof air nozzles in each row and a series of side wall nozzles above the rocking-type stoker. The tests were made after the furnace was in operation for several hours so that temperatures at a saturation condition could be obtained. The temporary thermocouple was inserted through the roof of the furnace between the roof air manifolds No. 1 and 2, approximately 2 ft from the side wall. Again a spare plant chromel-alumel thermocouple with stainless steel protective well was used and the opening around the well was packed with ceramic fibre. The instrument was calibrated at ambient temperature.

The stationary plant thermocouple was of identical type, installed in an airtight stainless steel sleeve. It was calibrated a few days before our test by the instrumentation manufacturer, using a spare thermocouple at the same location in conjunction with a portable potentiometer.

It shall be noted that the roof air manifold No. 3 was kept closed during the test in view of its location and unusually large diameter nozzles.

Since it was suspected that the roof air nozzles, arranged in rows perpendicular to the centerline of the furnace were the cause of localized temperatures exceeding considerably the temperatures registered by the stationary up-pass thermocouple, it was decided to take four consecutive test runs, closing the roof manifolds in successive order.

Again the readings, taken simultaneously at the test thermocouple and the plant thermocouple, suggest that the response of the thermocouple at the furnace exit is very poor in respect to the furnace temperatures. In some instances the differential values reached alarming proportions. It is to be noted that subsequent closing of each roof nozzle manifold in sequence produced a reduced differential reading. The differential was reduced considerably during the fourth run after all roof nozzle mani-

FIG. 3 PLANT NO. 2. SCHEMATIC FURNACE LAYOUT
folds were closed, with the exception of manifold No. 4 which was left open approximately 50 percent. The results suggest that the use of the roof air nozzles results in an effect similar to an air curtain effect, i.e., the separation of the furnace into a number of zones with different temperatures isolated to a great extent from the controlling thermocouple. Consequently, a localized overheating of the furnace can occur without being detected by the controlling media. As a matter of fact, overheating of certain parts of the furnace occurred while the furnace was operated at 1800 F. This was indicated by readings from thermocouple at the furnace exit previous to the described temperature test. The test readings show an average furnace exit temperature of 1600 F.

It has to be stated that the overfire air quantity on both plants is controlled automatically by the thermocouple at the furnace exit. While the various roof nozzles and manifold have been closed manually, the air flow through the remaining roof and side wall nozzles was increased automatically to maintain a relatively constant flue gas temperature at the furnace exit.

The formation of an “air curtain” by the roof air nozzles has been observed in both plants on several instances while air distribution adjustments were made. The open-

<p>| TABLE 1 |
| SUMMARY OF SURVEY: COMBUSTION AIR |</p>
<table>
<thead>
<tr>
<th>Plant No.</th>
<th>Description</th>
<th>Capacity T/24 Hrs</th>
<th>Overfire</th>
<th>ACTUAL AIR CFM</th>
<th>Underfire</th>
<th>Wall Cooling</th>
<th>Total</th>
<th>*Stoichiometric + Excess Air for 1800 F - CFM</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Low set furnace with travelling grate</td>
<td>200</td>
<td>12,000</td>
<td>13,000</td>
<td>--</td>
<td>25,000</td>
<td>27,800</td>
<td></td>
</tr>
<tr>
<td>2</td>
<td>Medium height furnace with rocking grate</td>
<td>200</td>
<td>20,700</td>
<td>20,700</td>
<td>3,000</td>
<td>44,400</td>
<td>27,800</td>
<td></td>
</tr>
<tr>
<td>3</td>
<td>Low set furnace with travelling grate</td>
<td>200</td>
<td>12,000</td>
<td>13,000</td>
<td>--</td>
<td>25,000</td>
<td>27,800</td>
<td></td>
</tr>
<tr>
<td>4</td>
<td>High furnace with travelling grate</td>
<td>200</td>
<td>65,500 Low Press. 2,000 High Press.</td>
<td>Duct from overfire air fan</td>
<td>7,500</td>
<td>75,000</td>
<td>27,800</td>
<td></td>
</tr>
<tr>
<td>5</td>
<td>Low set furnace with reciprocating grate</td>
<td>75</td>
<td>Duct from underfire air fan</td>
<td>15,400</td>
<td>--</td>
<td>15,400</td>
<td>10,200</td>
<td></td>
</tr>
<tr>
<td>6</td>
<td>High furnace with travelling grate</td>
<td>350</td>
<td>53,000</td>
<td>Duct from overfire air fan</td>
<td>8,000</td>
<td>61,000</td>
<td>49,100</td>
<td></td>
</tr>
<tr>
<td>7</td>
<td>Medium height furnace with rocking grate</td>
<td>125</td>
<td>5,000 Low Press. 600 High Press. (22,000)</td>
<td>16,200</td>
<td>1,600</td>
<td>23,400</td>
<td>17,000</td>
<td></td>
</tr>
<tr>
<td>8</td>
<td>Medium height furnace</td>
<td>100</td>
<td>16,000</td>
<td>6,100</td>
<td>3,180</td>
<td>25,280</td>
<td>13,680</td>
<td></td>
</tr>
<tr>
<td>9</td>
<td>High furnace with rocking grate</td>
<td>150</td>
<td>Duct from underfire air fan</td>
<td>25,000</td>
<td>--</td>
<td>25,000</td>
<td>20,750</td>
<td></td>
</tr>
</tbody>
</table>

*Quantity of air based on 5000 Btu/lb refuse and heat losses through 9 in. superduty refractory enclosures.
ing of the roof manifolds adjacent to the charging chutes resulted sometimes in a positive pressure zone between the roof manifold and the rear furnace wall, causing the gases and the smoke to escape through the refuse in the charging chute, around stoking doors, stokers and other openings. In most instances, the furnace draft settings had to be increased momentarily to overcome the formation of a positive pressure zone.

The effect of localized excessive temperatures on the furnace enclosure can be considerable: differential expansion of the refractories, spalling, overheating of the supporting structure, considerable slagging at temperatures in excess of 1800 F, possible overheating of stokers due to the penetration of the roof jets above or into refuse bed, etc.

**DISCUSSION OF AIR DISTRIBUTION**

In preparing general recommendations for design criteria combustion-air systems, a survey of 9 different plants was made in an attempt to establish a trend. It was surprising to find that there is actually no trend at all. The total air required for a 1800 F heat balance was compared with various design specifications. The actual total air supply provided for by the specifications varied from approximately 93 percent to 270 percent of the stoichiometric plus excess air required. Table 1 lists the survey findings and shows the variations of design air quantities. It is obvious that some variations had to be expected, taking into account different characteristics of refuse for which the plants were designed; nevertheless, all the plants investigated are of a recent design, and refuse with an average heat content of 5,000 Btu lb was probably considered for most, if not all of them.

It has been established by experience in actual operation that optimum safe performance of a given furnace is obtained usually by providing ± 70 percent of total air as overfire air and up to ± 60 percent of total air as underfire air. The total of 130 percent of total air for a given heat balance is used only for the selection of combustion air fans and is never used at one time. Automatic “cascade” controls have to be provided to change the ratio of overfire air to underfire air according to furnace temperature variation. It is possible, with properly adjusted controls, to obtain virtually constant furnace temperature within ± 50 F with a wide range of refuse quality.

The quantities of underfire air mentioned will probably be subject to discussion by various stoker manufacturers. Some manufacturers might be satisfied with lower percentage of underfire air while others might request as much as 100 percent of total air to be available as underfire air. We believe that it is essential to caution the furnace designers at this point that the type of stoker used with its inherent underfire air requirements must be considered carefully in the configuration and size of the furnace. The furnace and the stoker form a system which must be designed for specific requirements and substitution of stokers of different types might result in design changes in the furnace enclosure itself.

The distribution of the overfire air is, of course, of prime importance. While some further research on this subject is needed, it appears that sidewall air nozzles, located a short distance above the burning refuse bed, provide the best performance. The selection of the number of nozzles and their size is given by the flow needed and maximum penetration of the air jet, which should be less than the width of the furnace, in order to minimize or eliminate the impingement on the opposite refractory walls.

**CONCLUSION**

It appears that most of the recent incinerator furnaces have been designed with partial disregard for temperature limitations of the refractory enclosures considering the actual furnace temperatures encountered. While many designers assume an average operating temperature range of 1600 to 1800 F, some specifications stipulate a temperature range of 2200 to 2300 F as a “safe upper limit” of the refractory lining. The characteristics of the current refractory enclosures and the intimate supporting structures indicate that temperatures in this range, over a prolonged period of time, will cause some deterioration and shorten considerably the life of the refractories. Nevertheless, we can see from the data given in this paper that these high temperatures can and do exist over relatively long periods of time and that they are due primarily to inadequate overfire air supply or improper overfire air distribution.

While temperatures of 2200 to 2300 F are not uncommon in various industrial furnaces designed for special purposes, they will result in serious problems in incinerator applications. As mentioned before, temperatures in excess of 1800 F will cause excessive slagging due to the presence of low fusion point fly ash, diversity of chemical composition of various waste materials, such as plastics, glass, etc. Although the formation of slag on the refractory walls was attributed to various conditions, it appears that high temperatures are the main contributing
factors. The temperature control is, therefore, of utmost importance and the proper amount of overfire air properly distributed, is the most logical controlling media at the present time.

Although the recommendations made in this paper are based on long experience in incinerator design and actual operation, the authors sincerely hope that more research and discussion on the subject of furnace temperatures and air distribution will bring further improvements in the art of incineration.

REFERENCES