Flue Gas Cooling

Abstract

When waste heat is not utilized, cooling of flue gases in special cooling chambers becomes necessary, if the dust-collecting and associated equipment is constructed of metal. The two most common methods of cooling are by dilution with ambient air, and by evaporation of water in the gas stream. Usually, they are combined. Load sharing depends largely on acceptable visibility of stack effluvia and upon economy. Automatic control of the cooling system appears necessary. Explanation of computations with tabulations and graphs plus a description of an existing installation are included in the text.

Introduction

The last decade has seen a change in the appearance of many incinerator plants: the addition of chambers, ducts, piping, and mechanical equipment in the area between the combustion chamber and the chimney.

The settling chambers, with baffles and ports, though still common, are giving way to water spray, cyclone or other mechanical fly ash collectors with cooling chambers and induced draft fans. For safe and efficient operation, these plants include control systems of considerable complexity compared with the earlier ultra-simple instrumentation.

This development is due to a sudden awareness of the extensive pollution of the atmosphere by gaseous and particulate substances. Drastic reduction of solid, fine-particle matter in the atmosphere is now the concern of officials in most communities and increasingly the subject of governmental regulation and enforcement. Fly ash control is fast becoming the most difficult problem facing the incinerator designer. Not only is the refinement of the fly ash separation process a difficult task — to it is added the control of pollution of streams receiving waste water from spray-collecting chambers or cooling chambers.

The problem of designing equipment to withstand the highly corrosive nature of water leaving the process chamber, as well as the abrasive action of the fly ash (the latter being no small matter), places an additional demand upon the ingenuity of the designer.

In order to achieve the degree of gas cleansing demanded today by many of our communities, the designer must turn more and more to the mechanical types of fly ash collectors: small diameter multi-cyclones, water scrubbers, and — coming in for serious consideration — electrostatic precipitators. The latter, though not yet in use in this country, are in successful operation in many municipal incinerators in Europe.

If the designer's choice is the multi-cyclone collector, or the electrostatic precipitator, or a different type constructed of metals or other low temperature materials, it becomes necessary to add a gas-cooling device. This is done to assure that the entering flue gases are within the temperature range of 600 to 700 F, with no possibility of even flashes of higher temperatures.

The exit temperature depends upon the materials used downstream from the cooling chamber. Since, however, experience and economical evaluations justifying
the use of the high temperature alloys are not generally available this paper is based upon the exit temperatures stated above.

The two agents commonly used as cooling materials are air and water. Both have advantages and disadvantages. Generally, a combination of the two is found most acceptable. However, they must first be considered separately, so that the characteristics of each may be properly evaluated.

Cooling by Air Admixture

In many ways, this appears the simplest method. Air is available at no cost. Its introduction into the gas stream is simple. Instrument-controlled louvered openings with or without compressed air injectors may admit the cooling medium to the gas stream. Mixing cold air with the hot gases is a continuous operation, thus the temperatures of the surrounding refractories remain relatively stable, and thermal shock damage is minor.

It is only when the designer considers the quantities of combined combustion gas and cooling air that the high cost of this method becomes evident. Though the gas volume contracts as the cooling progresses, the cooling air adds another stream of gas of such a quantity that the total volume of the combined gas increases (see Table I). Cooling by this method from, for example, 1500°F to 700°F results in a combined quantity 142 per cent of the volume of the combustion gases before cooling. Since the enlarged quantity of the combined gas stream demands increased sizes of cooling chamber, ducts, fly ash separation equipment, induced draft fan and chimney, the increase in plant construction costs may be considerable.

Cooling by Water Admixture

As contrasted to gases cooled by air dilution, the volume of the combined cooled gases and superheated vapors — when water is the cooling medium — is considerably smaller than the volume of the gases before cooling (see Table I). In this case, when cooling from 1500°F to 700°F, there is a reduction in volume of 27.3 per cent. This is economically reflected in smaller equipment. Against this saving, there must be charged the increased maintenance costs of refractories and equipment, due to corrosion and abrasion, plus the cost of pumps, piping, nozzles and waste water treatment, if such is required.

TABLE I

VOLUME CHANGES OF 1000 CU FT OF GASES FROM 1500°F AS COOLED BY VARIOUS METHODS

<table>
<thead>
<tr>
<th>To Temp. F</th>
<th>Vol Entering</th>
<th>Lb Required</th>
<th>Total Lb Gases</th>
<th>Total Volume</th>
<th>Lb 70°F Water</th>
<th>Total Cu Ft Gases</th>
<th>Total Cu Ft Vapor</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Cu Ft</td>
<td>Dilution Air</td>
<td>Leaving</td>
<td>Cu Ft</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td>at 80°F</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>1500</td>
<td>1000</td>
<td>20.20</td>
<td>0.00</td>
<td>20.20</td>
<td>1000</td>
<td>0.00</td>
<td>1000</td>
</tr>
<tr>
<td>1300</td>
<td>898</td>
<td>20.20</td>
<td>3.57</td>
<td>23.77</td>
<td>1054</td>
<td>0.59</td>
<td>898</td>
</tr>
<tr>
<td>1100</td>
<td>796</td>
<td>20.20</td>
<td>8.57</td>
<td>28.77</td>
<td>1130</td>
<td>1.25</td>
<td>796</td>
</tr>
<tr>
<td>900</td>
<td>694</td>
<td>20.20</td>
<td>16.01</td>
<td>36.21</td>
<td>1233</td>
<td>1.97</td>
<td>694</td>
</tr>
<tr>
<td>700</td>
<td>592</td>
<td>20.20</td>
<td>28.20</td>
<td>48.40</td>
<td>1415</td>
<td>2.88</td>
<td>592</td>
</tr>
</tbody>
</table>

* Cooling by supplying heat to a waste heat boiler or similar equipment.

** By dry gases are meant the combustion products containing the moisture of the refuse but not including moisture added by the cooling process.
Most incinerator plants are charged regular water rates. If not the cost to the community to provide the water is a fair charge and should be part of the cost comparison estimate. Table I shows that with 100 per cent efficiency, 2.88 pounds of water is required to cool 1000 cubic feet (1500 F) from 1500 to 700 F or approximately 274 gallons of water is needed per ton of refuse burned. Assuming 65 per cent efficiency, this becomes 420 gallons per ton of refuse. At a cost of $0.25 per 100 cubic feet, the water cost per ton burned would be $0.14 or between 2 and 4 per cent of the total burning cost for an efficient and well-run plant. This cost may be reduced by settling the slurry and treating and recirculating the waste water. Recirculation often becomes attractive if untreated fluid waste cannot be disposed of. This may be the case where a sewer is too far away or too high to make connection practical, or where the nearest stream cannot safely absorb the untreated waste water. Further, the local water pollution control plant may not be able to handle the additional load or the sewers may be too flat to assure that the solid matter will not settle and seriously reduce the flow in an already-inadequate system.

Quantities and Dimensions

The two decisions which must be made before the design may proceed are as follows:
1) re-circulation or 100 per cent make-up of spray water;
2) proportional division of cooling load between air and water.

These go hand in hand since an increase in air cooling load reduces the quantity of water available for re-circulation — possibly to a point where re-circulation becomes unnecessary. It is generally advisable to make several cost estimates to determine the least expensive and least troublesome proportion of the cooling agents.

Estimates may be tabulated as shown in the next column. Items given provide an illustration. Items may be added and subtracted, and the list should not be considered definitive.

After the first estimate is made, it is less time-consuming to follow up with additional estimates for different conditions. The investigation of three different proportions will probably make it possible to settle on the answers to 1) and 2) above.

Certain basic data must be developed in order to make the estimates discussed. These are:

a) burning rate
b) refuse composition
c) theoretical air for combustion
d) excess air before cooling chamber
e) quantity of combustion products
f) temperature at entrance to cooling chamber
g) temperature at exit from cooling chamber
h) volume of flue gases entering cooling chamber
i) volume of flue gases leaving cooling chamber
j) size of fly ash collector
k) size of ducts and chimney
l) static pressure of induced draft fan
m) size of induced draft fan.

Items a) to e) inclusive are familiar to the incinerator designer. Consideration of these does not belong under this subject.

Items f) and g), the gas temperatures immediately before and after the cooling process, must be established. The cooled gas temperature depends upon the tolerance of the materials containing the gas stream. As this normally will be a low alloy steel plate, a top temperature of 700 F is safe. The temperature at the cooling chamber inlet depends upon furnace temperature and excess air, but it may be taken to average 1700 F. An average temperature must be chosen since it will determine the annual quantities of water consumption and the cost of electric power to operate the

Tabulation of Estimate

<table>
<thead>
<tr>
<th>Air Cooling</th>
<th>Water Cooling</th>
<th>Est. Cost</th>
</tr>
</thead>
<tbody>
<tr>
<td>Per cent of each Quantities</td>
<td>X</td>
<td>X</td>
</tr>
<tr>
<td>Combined Quantity</td>
<td>X</td>
<td></td>
</tr>
<tr>
<td>Dimensions Cool. Chamber</td>
<td>X</td>
<td>X</td>
</tr>
<tr>
<td>Dimensions Duct to fly ash coll.</td>
<td>X</td>
<td></td>
</tr>
<tr>
<td>Dimensions Duct to induced draft fan</td>
<td>X</td>
<td>X</td>
</tr>
<tr>
<td>Dimensions Duct to chimney</td>
<td>X</td>
<td>X</td>
</tr>
<tr>
<td>Fly ash coll. and assoc. equip.</td>
<td>X</td>
<td>X</td>
</tr>
<tr>
<td>Induced draft fan</td>
<td>X</td>
<td>X</td>
</tr>
<tr>
<td>Chimney</td>
<td>X</td>
<td></td>
</tr>
<tr>
<td>Louvered air inlets and inlet nozzles</td>
<td>X</td>
<td>X</td>
</tr>
<tr>
<td>Injector piping</td>
<td>X</td>
<td>X</td>
</tr>
<tr>
<td>Air compressor or shore in plant compressor</td>
<td>X</td>
<td>X</td>
</tr>
<tr>
<td>Water pumps</td>
<td>X</td>
<td>X</td>
</tr>
<tr>
<td>Piping, nozzles and valving</td>
<td>X</td>
<td>X</td>
</tr>
<tr>
<td>Metering</td>
<td>X</td>
<td>X</td>
</tr>
<tr>
<td>Cool. Chamber drain</td>
<td>X</td>
<td>X</td>
</tr>
<tr>
<td>Water Treatment and settling</td>
<td>X</td>
<td>X</td>
</tr>
<tr>
<td>Re-circulation pumps and associated equipment</td>
<td>X</td>
<td>X</td>
</tr>
<tr>
<td>Fly ash disposal</td>
<td>X</td>
<td>X</td>
</tr>
<tr>
<td>Automatic control and instrumentation</td>
<td>X</td>
<td>X</td>
</tr>
<tr>
<td>Water cost per year</td>
<td>X</td>
<td>X</td>
</tr>
<tr>
<td>Power cost per year</td>
<td>X</td>
<td>X</td>
</tr>
<tr>
<td>Maintenance per year</td>
<td>X</td>
<td>X</td>
</tr>
<tr>
<td>Operation cost per year</td>
<td>X</td>
<td>X</td>
</tr>
</tbody>
</table>

* Add B to the amortized annual cost of A.
auxiliary equipment and the induced draft fan. A maximum temperature must, however, also be selected to determine equipment and duct sizes and the capacity of a system to supply sufficient quantities of cooling agents to assure that the selected cooling chamber outlet temperature will not be exceeded.

Usually 2000°F should be satisfactory, especially when an alarm is incorporated in the system. An additional hand-operated bank of spray water nozzles will be provided by a conservative designer. The remaining items listed may be determined by well-known methods.

There remains to establish the quantities of cooling air and water. These may be calculated by the formulas:

\[
Q_a = \frac{Q_g (T_i - T_m) C_{pg}}{(T_m - T_i) C_{pa}} \]
\[
Q_w = \frac{Q_g (T_m - T_o) C_{pg}}{H x f + (T_3 - T_2) (1 - f)}
\]

where

- \(Q_g\) = quantity flue gas - pounds per hour
- \(Q_a\) = quantity cooling air - pounds per hour
- \(Q_w\) = quantity spray water - pounds per hour
- \(T_i\) = temperature 0°F - flue gas at inlet to cooling chamber
- \(T_m\) = temperature 0°F - flue gas after cooling by air
- \(T_o\) = temperature 0°F - flue gas at outlet from cooling chamber
- \(T_1\) = temperature 0°F - ambient at air inlet louver
- \(T_2\) = temperature 0°F - inlet water
- \(T_3\) = temperature 0°F - waste water at cooling chamber drain
- \(H\) = heating value in Btu per pound of steam from 70°F water, atmospheric pressure and superheated to \(T_o\).
- \(C_{pg}\) = specific heat, constant pressure, flue gas
- \(C_{pa}\) = specific heat, constant pressure, cooling air
- \(C_{pw}\) = specific heat, constant pressure, spray water
- \(f\) = proportion of water evaporated.

These formulas are based on an arrangement where air cooling precedes water cooling. \(T_m\) is a function of the selected proportion of air to water cooling discussed above.

\(T_2\) should be the highest temperature of the water supply which may be taken as 70°F, unless unusual conditions prevail. This is based on a design that does not call for re-circulation of waste water. Where re-circulation is used, \(T_2\) may be assumed to be 100°F and \(T_3\) at 150°F. These values have been established at the installation discussed below.

Air cooling efficiency depends on proper mixing of air and flue gas. In a good design, it is believed that the mixing will be thorough and \(f\) may be taken as 100. Water cooling efficiency depends on the location, type and maintenance of the spray nozzles. For a well-designed layout, it may be taken at 65 per cent. However, more spray water capacity should be added to assure safety in operation under unusual conditions until more operating data is available. It is recommended that the total nozzle capacity, including a hand-operated bank, be based on 33 per cent efficiency and a total cooling gap from 2000 to 700°F.

Fig. 2 and Table II may be used for the trial estimates suggested above, assuming that \(T_i\) is 1800°F and \(T_m\) is 1100°F. Fig. 2 will provide the answer that 741 pounds of air and 76 pounds of water are required, and that the final volume will be 54,470 cubic feet at 700°F for 1000 lb of gases. Fig. 2 is based on Table II in which additional details are given. By subtraction and addition, answers may be found to any choice of temperatures.

**Description of an Installation**

The East 73rd Street incinerator in the City of New York has been in operation since September, 1957. It was constructed in the expanded structure of an existing plant. In three furnaces of 220 tons per day it burns 660 tons per 24 hours. The general arrangement of a furnace unit is shown in Fig. 3.

**Furnace and Cooling Chamber.** Each furnace is equipped with two travelling grates set in tandem, the feeding grate above and at an angle of 25 degrees to the

*Fig. 2. Combination cooling with air and water. 1000 lb gases at 1800°F cooled to 700°F. Computations based on inlet air temperature at 80°F, inlet water temperature at 70°F. Slight inaccuracy using constant specific heat value (0.24) ignored.*
burning grate. The drop between the two grates is four feet.1

From the furnace, an average of 92,000 pounds per hour of combustion gases enter the cooling chamber through a throat 14 ft high by 8 ft wide. Each side wall of the throat has 10 air ports located in a vertical line, for introduction of cooling air. This air is aspirated through 9 in. x 10½ in. rectangular air ports by a jet of compressed air at 30 in. water column. The jet has a 1/2 in. nozzle centered in the air inlet casting. The air jets also create turbulence and therefore a better mixing of the cooling air with the combustion gases. The air inlet ports on each side wall are connected to a common plenum chamber. The air quantity admitted through the ports to the throat is controlled by louver dampers on the front walls of the plenum chambers. The dampers are positioned automatically at any required opening from fully closed to fully opened.

The mixture of flue gases and cooling air then enters the cooling chamber structure in a sudden expansion of width of structure from 8 ft to 20 ft-2 in. The change in dimensions aids in mixing the combustion gases and the cooling air so that hot zones due to stratification in the gas flow may be avoided. Three horizontal banks of water spray nozzles are located in vertical succession in the cooling chamber wall facing the furnace. A fourth horizontal bank of water spray nozzles is located in the far wall.

1 For a more detailed description of the double traveling grate incinerator, see V. Westergaard, "Traveling Grates Find a New Field of Usefulness" by Combustion, Vol. 26, No. 12, June 1955, p. 40.

When air admixture is insufficient, the highest bank of water spray nozzles is automatically activated, followed by the second and third banks, as needed. The fourth bank is manually operated, and is for emergency use in case the automatic cooling system fails to keep the temperature at the desired level. The cooling chamber is constructed of high heat duty refractories. The floor slopes to a center drain which terminates under the water surface of the residue conveyors below.

**Fly Ash Collectors.** From the top of the second pass of the cooling chamber, the gases — reduced to temperatures between 600 and 650 degrees — pass through a horizontal metal flue into the fly ash collector. Each collector consists of 336 cyclones with 8 in. diameter inlet tubes and 6 in. diameter outlet tubes, and is divided into four separate sections of equal size, with a common discharge hood connected to the induced draft fan.

The efficiency of the multi-cyclone type of collector is related to the velocity of the fly ash-laden gases passing through it. In order to keep this velocity from dropping below the proper value, each of the four sections is preceded by a vertical louver damper in the inlet duct. These dampers are operated by air cylinders, controlled by push-pull buttons on the operating instrument board. They are either fully open or fully closed. This permits varying the capacity of the fly ash collector by 25, 50, 75 and 100 per cent of full utilization.

Each of the four collector sections has its own fly ash hopper, sealed by a rotary star feeder. The four rotary feeders for each collector are operated by a continuous shaft and a single electric motor drive mechanism. All fly ash hoppers discharge into a single scraper

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**TABLE II**

**COMBINATION AIR AND WATER COOLING**

(1000 LB GASES AT 1800 F. COOLED TO 700 F.)

100 PER CENT EFFICIENCY

<table>
<thead>
<tr>
<th>Air Cooling to (Temp. °F)</th>
<th>Lb Air Required at 80 F</th>
<th>Total Lb Combined Gas and Air</th>
<th>Additional Water Cooling to 700 F Required at 70°F</th>
<th>Total Cu Ft Dry Gases (700 F)</th>
<th>Total Cu Ft Vapor (700 F)</th>
<th>Total Gases and Vapor Cu Ft (700 F)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1800</td>
<td>0</td>
<td>1000</td>
<td>196</td>
<td>23.4</td>
<td>29,200</td>
<td>9200</td>
</tr>
<tr>
<td>1500</td>
<td>211</td>
<td>1211</td>
<td>173</td>
<td>20.1</td>
<td>35,300</td>
<td>8100</td>
</tr>
<tr>
<td>1300</td>
<td>410</td>
<td>1410</td>
<td>151</td>
<td>18.1</td>
<td>41,100</td>
<td>7100</td>
</tr>
<tr>
<td>1100</td>
<td>686</td>
<td>1686</td>
<td>120</td>
<td>14.4</td>
<td>49,100</td>
<td>5600</td>
</tr>
<tr>
<td>900</td>
<td>1096</td>
<td>2096</td>
<td>75</td>
<td>9.0</td>
<td>61,100</td>
<td>3500</td>
</tr>
<tr>
<td>700</td>
<td>1775</td>
<td>2775</td>
<td>0</td>
<td>0.0</td>
<td>80,800</td>
<td>0</td>
</tr>
</tbody>
</table>
conveyor system. This delivers the collected fly ash to a bifurcated chute, each leg of which empties into a residue conveyor. A flap gate in the chute is pneumatically operated and connected electrically to the residue conveying equipment so that the conveyor system empties into the operating residue conveyor.

The rotary feeders and the scraper conveyors may be operated manually or by an automatic device which starts the conveyors at set intervals and operates them for a length of time which is adjusted to suit the need. The conveyors and feeders are electrically interlocked and start sequentially, beginning with the one which discharges to the residue conveyor.

**Induced Draft Fan.** The combustion gases are drawn through the cooling chambers and fly ash collectors by induced draft fans located above the fly ash collectors (Fig. 3). The fans discharge into a common breeching above the roof which terminates in the chimney.

Each fan is capable of handling 190,000 cubic feet per minute of gases at 700°F, against a static pressure of 5.5 in. water column. It is driven by a 300 hp motor through a variable speed fluid drive and a speed reducer. The variable speed unit is controlled by the negative pressure in the furnace in such a manner that a pre-determined draft is maintained in the furnace under the varying operating conditions.

**Cooling Agent Quantities.** Because of the variability in quantity and temperature of the combustion gases, uncertainties in regard to the efficiency of the cooling system (and no experience to lean upon), liberal figures were used in determining the maximum quantities of air and water required. The quantities selected for a furnace, and proven more than sufficient, were as follows:

- **cooling air — 40,000 cfm at 100°F.**
- **cooling water — 145 gpm.**

**Automatic Control**². The automatic control system performs two distinct functions:

- a) it maintains a predetermined pressure in the furnace;
- b) it adjusts the equipment to provide a predetermined outlet temperature from the Cooling Chamber.

A) Summing up the operation of the furnace draft control system:

1) The negative pressure in the furnace is translated into an air signal in a “Master Draft” controller.

2) The electric current demand of the induced draft fan motor is translated into an air signal in a current converter/pneumatic transmitter combination.

3) The air signal from the “Master Draft” Controller is modified by the air signal from the pneumatic transmitter. Thus the fan speed will not demand an input current above a pre-set maximum. This function is performed by a “Slave Draft” Controller.

4) The air signal from the “Slave Draft” Controller adjusts the position of a piston in the control drive.

5) The position of the piston in the control drive, through a linkage system, adjusts the output speed of the fluid drive unit in the drive of the induced draft fan.

6) The fan speed adjusts the negative pressure in the furnace, and the cycle is completed.

B) Summing up the operation of the temperature control system:

1) The temperature of the combustion gases is measured in four locations, in front of the inlet of the sections of each fly ash collector. Temperatures are translated into air signals in the pneumatic transmitter.

2) The four air signals are weighed in the high pressure selector relays which suppress the three lower air signals.

3) The highest air signal is transmitted to the recording controller which records the signal in terms of temperature and supplies an output air signal.

4) This latter air signal successively activates (through mercoid switches)

- a) cooling air aspirator fan;
- b) cooling air inlet dampers;
- c) water spray solenoid valves.

(This system described was furnished by The Bristol Company and is one of several different types of control methods available in the trade.)

**Suggestions**

1) Determine air and water requirements by assuming 100 per cent utilization of cooling air and a minimum of 33 per cent of the cooling water, ending as superheated steam at the temperature of the cooled gases. A temperature rise in waste water of 50°F may safely be assumed.

2) Slope wall opposite spray nozzles so that the spray water (by then quite acid) may not easily seep through cracks to the metal back-up structure where it can be very destructive.

3) Slope cooling chamber bottom steeply for effective draining of the slurry.

4) Place nozzles well inside the wall to protect them from the strongly eroding effect of moist combustion gases. Conical openings in the wall will permit this.

5) Experiment with nozzle metals to find the best for prolonged use. A metal such as Carpenter 20 may be found better than many of the stainless varieties.

6) Tilt nozzle supply pipes downward towards the nozzle end to reduce water spillage outside chamber.

7) Arrange instruments so that the state of erosion of the nozzle orifice may be determined by observing changes in water pressure and consumption.

8) It was found that the outlet vanes in the cyclone units easily clogged. They were removed without observable change in the efficiency of the fly ash collector.

²For details of the operation of the control system, see Appendix.
9) A study to determine the most suitable refractory for the cooling chamber wall opposite the spray nozzles is recommended. Considerable spalling has been experienced. This may be reduced by a more suitable design and material.

Appendix

Control System

A. Control of Furnace Draft. Furnace draft is maintained by altering the speed of the induced draft fan to meet changing load conditions. The speed change is accomplished through a fluid drive inserted in the fan drive between the constant speed motor and the speed reducer. The speed reduction in the fluid drive is governed through a series of control instruments by:

a) the draft in the furnace;
b) the magnitude of the current drawn by the fan motor.

Two draft connections are made through the refractory wall of the furnace located 8 to 9 ft above the burning stoker. Means are provided for rodding the furnace connections in order to remove slag in front of the tube openings. The negative pressure in the furnace is transmitted by one combining tube to a "Master Draft" Controller (MDG), Fig. 4) located on the furnace instrument board (Fig. 5). This instrument operates as follows (Fig. 6).

The tube from the draft connections in the furnace is connected in the Master Draft Controller to an inverted oil-filled bell differential pressure measuring element. This, through a lever system, moves a vane in and out between air jet nozzles. The back pressure from the nozzles varies as the leading edge of the vane passes between the jets. The back pressure reacts on a needle valve inserted in the power air supply line, throttling the pressure of the 20 psi air supplied by an air compressor,
FIG. 6. SCHEMATIC DIAGRAM OF PROPORTIONAL CONTROL UNIT.

The output pressure varies with, and is directly proportional to, the magnitude of the draft conditions in the furnace and is called the air signal. A circular chart on the controller records the negative pressure in the furnace.

The air signal from the “Master Draft” controller is transmitted to a “Slave Draft” Controller (SDC-1, Fig. 4), located on the furnace instrument board (Fig. 5). This instrument is similar to the above-mentioned instrument except that it receives air signals from two sources, integrating them into one output air signal. It functions in a manner similar to that described above except that it contains two bellows, one receiving its input air signal from the “Master Draft” Controller, the other receiving its input air signal from the motor current sensing instrument described below.

The two bellows activate, through an interconnected leverage system, a vane between air jets (as described above) and an output air signal is produced varying from 3 to 15 psi, reflecting both furnace draft conditions and fan motor current consumption. The lever system is so designed that the resulting maximum output air signal pressure cannot call for a fan speed that will produce a running load current above a pre-set value. The settings for both the desired negative furnace pressure and the maximum fan motor current are adjustable. The air signal is transmitted from the “Slave Draft” Controller to the pneumatic transmitter. A circular chart on the “Slave Draft” Controller records the running load current consumption of the induced draft fan.

The motor current sensing equipment consists of a current converter. The converter is an ac/dc transducer type, connected to a current transformer with a secondary providing a maximum current of 5 amperes (TCC-1 and CT, Fig. 4).

The output signal of the current converter is calibrated in millivolts. The signal from the converter is transmitted to an indicating potentiometer (IMCT-1, Fig. 4). Both converter and indicating potentiometer are located on the wall in back of the induced draft fan. The operation of this instrument is as follows.

Any change in the signal from the converter creates a dc voltage imbalance in the potentiometer circuit. This dc voltage imbalance is converted to an ac voltage by a “Syncroverter” switch. The Syncroverter output voltage is then increased by a transformer before being amplified electronically to a power level sufficient to operate a balancing motor. The balancing motor adjusts the sliding contact on a slide wire to restore electrical balance in the potentiometer circuit and simultaneously moves the indicator dial to indicate the value of the variable being measured (Fig. 7).

At the same time, the balancing motor moves a cam segment and linkage which changes the position of the valve stem in the pneumatic transmitter. The output pressure from the pneumatic transmitter varies with the position of the valve stem from 3 to 15 psi. The output air signal is piped to the “Slave Draft” Controller.

The “Slave Draft” Controller output air signal is piped to the Air-Operated Control Drive, Piston Type which is located at, and connected by linkage to, the fluid drive. The Control Drive consists of:

a) A power unit which provides the action needed to adjust the fluid drive to the output speed required;

b) A control unit which activates and positions the power unit as needed. The power unit is a double-acting air cylinder. The piston is moved in the cylinder in accordance with the differential in air pressure on each side of the piston or held in any position when the pressures are equalized.

The control piston is positioned in accordance with the demand output air signal from the “Slave Draft” Controller which is piped to the activating bellows of the positioner of the Control Drive. The positioner consists,
essentially, of two opposing forces, maintaining a piston position that depends upon the magnitude of the forces (Fig. 8). The positioner is balanced when the force exerted upward on a balance beam by bellows is equalled by a force exerted downward by the positioning spring. The force exerted by the positioning spring is modified by the location of the piston in the cylinder through a lever, gear and cam arrangement. When the forces exerted by the bellows and by the spring are in balance, the balance beam holds the pilot valve in its "neutral" position; the pressures on either side of the piston are then equal and the piston is held in position. A change in pressure in the "Slave Draft" control output air signal will thus cause the fan speed to change until balance is again restored.

B. Control of Flue Gas Temperature. The control system is arranged to admit cooling air first. As the demand for cooling increases beyond what can be provided by the cooling air, the three banks of water sprays are successively activated. The automatic system controlling the cooling process is shown diagrammatically in Fig. 9 and operates as follows.

Four thermocouples are installed in the inlet flue in front of each of the four louver dampers of each dust collector as described above (Fig. 3). Each of the four thermocouples is wired to an indicating potentiometer transducer (T1T-1A to 1D, Fig. 9) located on the furnace instrument board (Fig. 5). The operation of this instrument has been described in the section "Control of Furnace Draft". The output air signals from these four instruments, each varying proportionally with the changes in temperatures sensed by its thermocouple are piped to three High Pressure Selector Relays (R1a to R1c, Fig. 9). These select the highest pressure air signal from the four transducers and thus signal the highest temperature sensed by any of the four thermocouples in the cooled gas stream.

The High Pressure Selector Relay functions as follows (Fig. 10). The two input pressures apply their respective forces on opposite sides of a neoprene diaphragm. The higher of these pressures causes the diaphragm to cap the output nozzle of the lower pressure line. This leaves the higher pressure directly connected to the output port and seals off the lower pressure input source.

The selected air signal is transmitted to the cooling chamber recording controller. This Proportional Air-Operated Free-Vane Controller has been described in the preceding section. It records the highest of the four sensed temperatures of the cooled flue gas on a circular chart. It also supplies an output air signal varying in pressure proportionally with the gas temperature.
The following equipment is activated by the output signal from the cooling chamber recording controller:

1) One mercoid switch (Fig. 9) which starts the cooling air aspirator fan when the air signal reaches 2 psi pressure. The fan remains in operation as long as the air signal pressure does not go below 2 psi.

2) Two positioners and damper operators (DO1 and DO2, Fig. 9). The louver dampers begin opening at 3 psi and are fully opened at 9 psi. The method of operation of this equipment is discussed below.

3) Three mercoid switches (PS2, PS3 and PS4, Fig. 9) each connected to a solenoid valve (S, Fig. 9) controlling a bank of cooling water nozzles. These are activated successively as the air signal pressure increases above 9 psi pressure.

The damper operating mechanisms each consist of two units, the positioner and the operator. They operate in the following manner (see Fig. 11).

Power air at 15 psi is let through pipe (R) to pilot valve (OO) which controls the flow through pipe (AX) to the diaphragm of the operator. The pilot valve is operated by spring (TT) and the movement of floating lever (H) acting on stem (XX). By means of lever (H) the pilot valve is operated jointly by the pressure of the output signal from cooling chamber controller and the movement of the stem of the operator. The air signal coming to the spring loaded bellows (JJ) through pipe (Y) gives movement to the bellows which is in proportion to the pressure of the signal. This movement is transmitted to the pilot valve through lever (H). As the operator responds to the change in air pressure over the diaphragm, valve stem (AE) moves. This motion is transmitted through levers (AC) (BB) and (Q) to the floating lever (H) through the adjustable pin (J) and lever (Z). This latter movement of lever (H) readjusts the position of the pilot valve until equilibrium is re-established and the control system is stabilized. The operator stem is directly connected to the air inlet louver by a linkage system.

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