

AIR POLLUTION ENGINEERING MANUAL

SECOND EDITION

Compiled and Edited

by

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Table 28. CONVERSION VALUES

Gaseous components	W_i , lb/min	h_i at 300° F, Btu/lb	$Q_i = h_i W_i$, at 300° F Btu/min
CO ₂	3.85	51.3	197.7
H ₂ O	2.89	108.2	312.5
N ₂	21.32	59.8	1,279.0
O ₂	1.08	53.4	57.5
Totals	29.14		1,846.7

Total heat above 60° F in PC at 300° F = 1,847 Btu/min

Volume of dilution air at 100° F

$$\rho \text{ at } 100^\circ\text{F} = 0.0708 \text{ lb/ft}^3 \text{ (From Table D1)}$$

$$\text{Volume} = \frac{408}{0.0708} = 5,760 \text{ cfm}$$

The exhaust system must be designed to handle a sufficient volume of gases at 300° F to provide an indraft of dilution air of 5,760 cfm in addition to the products of combustion.

Quenching With Water

When a large volume of hot gas is to be cooled and only a small quantity of dilution air is needed to capture the air contaminants, some methods of cooling other than dilution with ambient air should be used. Since the evaporation of water requires a large amount of heat, the gas can be cooled simply by spraying water into the hot gas.

For efficient evaporation of water in a gas stream, it has been determined that the gas velocity should be from 500 to 700 fpm and the entire cross section of the stream should be covered with a fine spray of water. If, however, water carryover is undesirable, as in a baghouse, satisfactory settling of the water droplets must be attained; hence, lower velocities are employed. Eliminator plates are seldom used in installations where excessive maintenance due to corrosion or fouling is expected. To reduce further the likelihood of water droplet carryover, place the water spray chamber as far from the baghouse as practical.

Water spray pressures generally range from 50 to 150 psig; however, to reduce the amount of moisture collected, some installations have employed pressures as high as 400 psig. Since the moisture collected in spray chambers readily corrodes steel, the chambers are frequently lined with materials resistant to corrosion.

If the gases discharged from the basic equipment are exceptionally hot, as are those from the cupola furnace, the first portion of the duct should be

refractory lined or made from stainless steel. In some cases, stainless steel ducts with water sprays have been used between the furnace and the quench chamber.

For controlling the gas temperature leaving the quench chamber, a temperature controller is generally used to regulate the amount of water sprayed into the quench chamber. For emergency conditions, a second temperature controller can be used to divert excessively hot gases away from the air pollution control device.

Cooling hot gases with a water quench is relatively simple and requires very little space. Figure 38 shows a quench chamber used to cool the gas-

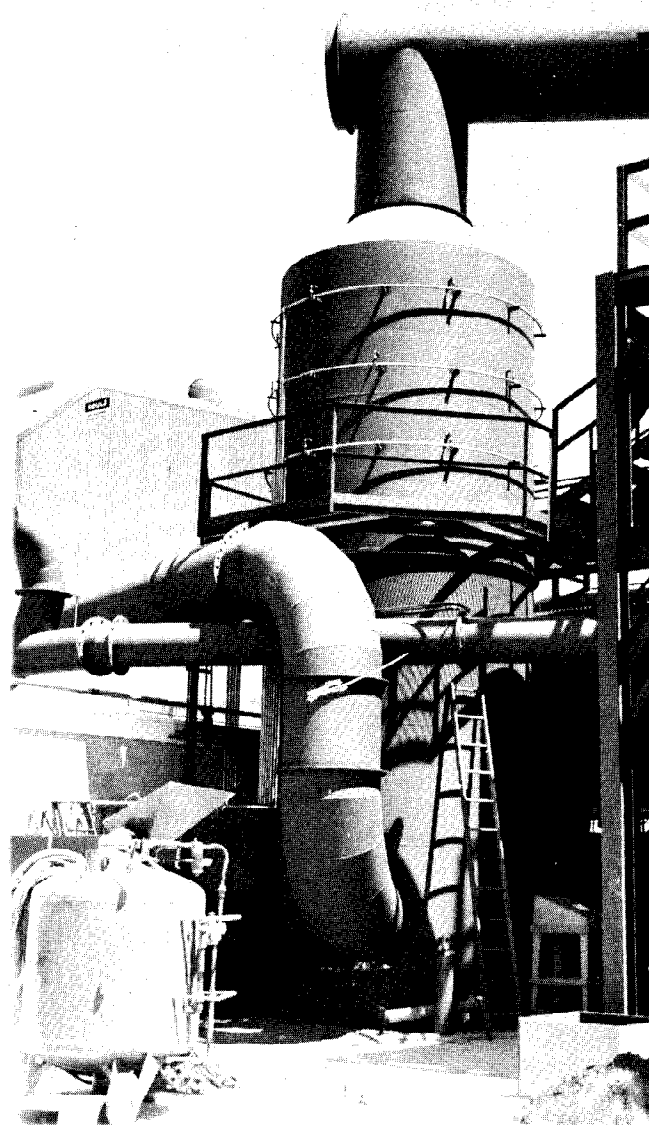


Figure 38. A quench chamber in a baghouse control system serving a cupola furnace (Harsell Engineering Company, Inglewood, California).

eous effluent from a cupola furnace. Quench chambers are little more than enlarged portions of the ductwork equipped with water sprays. They are easy to operate and, with automatic temperature controls, only that amount of water is used that is needed to maintain the desired temperature of the gases at the discharge. Their installation and operating costs are generally considered to be less than for other cooling methods. Quench chambers should not be used when the gases to be cooled contain a large amount of gases or fumes that become highly corrosive when wet. This creates additional maintenance problems, not only in the quench chamber, but in the remainder of the ductwork, the control device, and the blower.

The following example will illustrate some of the factors that must be considered when designing a quench chamber to cool the gaseous effluent from a gray-iron-melting cupola.

Example 16

Given:

32-in.-I. D. cupola. Maximum temperature of gaseous effluent at cupola outlet = 2,000°F.

Weight of gaseous effluent at cupola outlet = 216 lb/min.

Volume of gaseous effluent at cupola outlet = 13,280 cfm at 2,000°F. This volume of effluent includes indraft air at the charging door of the cupola. The temperature of 2,000°F is a maximum temperature.

Assume the effluent gases have the same properties as air. Consideration of the enthalpies of the gaseous constituents in the effluent gas stream will show that this is a valid assumption. Any corrections would introduce an insignificant refinement to the calculations when considered with respect to the accuracy of other design factors.

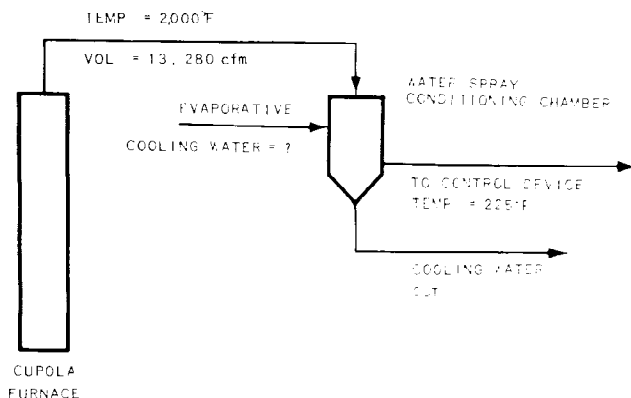


Figure 39. Problem: Determine the water needed to cool the gaseous effluent to 225°F and the total volume of gases discharged from the quench chamber.

Solution:

1. Cooling required:

$$\text{Enthalpy of gas at } 2,000^{\circ}\text{F} = 509.5 \text{ Btu/lb}$$

$$\text{Enthalpy of gas at } 225^{\circ}\text{F} = 39.6 \text{ Btu/lb}$$

$$\Delta h = 469.9 \text{ Btu/lb}$$

$$(216)(469.9) = 101,300 \text{ Btu/min}$$

2. Water to be evaporated:

Heat absorbed per lb of water:

$$Q = h_g (225^{\circ}\text{F}, 14.7 \text{ psig}) - h_f (60^{\circ}\text{F})$$

$$= 1,156.8 - 28.06 = 1,128.7 \text{ Btu/lb}$$

$$\text{Water required} = \frac{101,300}{1,128.7} = 90 \text{ lb/min}$$

3. Volume of water evaporated at 225°F:

$$\frac{379}{18} \left(\frac{460 + 225}{460 + 60} \right) (90) = 2,510 \text{ cfm}$$

4. Total volume vented from spray chamber at 225°F:

$$\text{Cupola} = (13,280) \left(\frac{225 + 460}{2,000 + 460} \right) = 3,700 \text{ cfm}$$

$$\text{Water} = 2,510 \text{ cfm}$$

$$\text{Total} = 6,210 \text{ cfm}$$

Problem Note: In this example, the calculated amount of water required to cool the gases, 90 lb/min or 10.8 gal/min, is only the water that must be evaporated. Since all the water sprayed into a quench chamber does not evaporate, the pump and spray system should be sized to supply more water than that calculated. The amount of excess water needed will depend on factors such as the inlet temperature of the gases, the temperature drop required, the fineness of the water spray, and the arrangement of spray heads. It is not uncommon to size the pump to give 200 percent of the water needed for evaporation. The actual amount of water used should be controlled by the temperature of the gases discharged from the quench chamber.

The loss of heat by radiation and convection from the ducts was neglected. With long duct runs, however, a considerable temperature drop in the gas-

eous effluent could occur, especially if the quench chamber was installed near the downstream end of the ductwork. If the quench chamber is placed near the control device, adequate water entrainment separators must be employed.

Natural Convection and Radiation

When a hot gas flows through a duct, the duct becomes hot and heats the surrounding air. As the air becomes heated, natural drafts are formed carrying the heat away from the duct. This phenomenon is called natural convection. Heat is also discharged from the hot duct to its surroundings by radiant energy.

The rate of heat transfer is a function of the resistances to heat flow, the mean temperature difference between the hot gas and the air surrounding the duct, and the surface area of the duct. It may be expressed as:

$$Q = UA \Delta t_m \quad (33)$$

where

- Q = rate of heat transfer, Btu/hr
- U = overall heat transfer coefficient, Btu/hr-°F-ft²
- A = heat transfer area, ft²
- Δt_m = log-mean temperature difference, °F.

The rate of heat transfer is determined by the amount of heat to be removed from the hot gaseous effluent entering the exhaust system. For any particular basic process, the weight of gaseous effluent and its maximum temperature are fixed. The cooling system must, therefore, be designed to dissipate sufficient heat to lower the effluent temperature to the operating temperature of the air pollution control device to be used.

The rate of heat transfer can be determined by the enthalpy difference of the gas at the inlet and outlet of the cooling system.

$$Q = W \Delta h \quad (34)$$

where

- W = weight of gas flowing, lb/hr
- Δh = enthalpy change between inlet and outlet, Btu/lb.

The log-mean temperature difference is the difference in temperature between the air surrounding the duct, and the inlet and outlet temperature

of the gas. This term, too, is fixed for a particular process. It is calculated as follows:

$$\Delta t_m = \frac{(t_1 - t_a) - (t_2 - t_a)}{\log_e \frac{(t_1 - t_a)}{(t_2 - t_a)}} \quad (35)$$

where

- t_1 = gas temperature of inlet, °F
- t_2 = gas temperature at outlet, °F
- t_a = air temperature, °F.

In many processes the temperature of the gaseous effluent is not constant but varies during different operational phases. The atmospheric temperatures also vary a great deal. In such cases, the cooling system must be designed for the worst conditions that prevail to ensure adequate cooling at all times. The inlet temperature (t_1) chosen must be the maximum temperature of the gas entering the system; t_2 must be the maximum allowable temperature of the gas discharged from the cooling system; and t_a must be the maximum expected atmospheric temperature.

The overall coefficient of heat transfer, U, is the reciprocal of the overall resistance to heat flow. It is a function of the individual heat transfer coefficients, which can be estimated by empirical equations. U must be based on either the inside or outside surface of the duct. For radiation-convection cooling, it is generally based on the outside surface and is denoted by U_o . U_o is defined by the following equation (Kern, 1950):

$$U_o = \frac{h_{i_o} h_o}{h_{i_o} + h_o} \quad (36)$$

where

- h_{i_o} = inside film coefficient based on the outside surface area, Btu/hr-°F-ft²
- h_o = outside film coefficient, Btu/hr-°F-ft².

The inside film coefficient can be solved by the formula (Kern, 1950):

$$h_i = j_H \frac{k}{D} \left(\frac{C\mu}{k} \right)^{1/3} \quad (37)$$

where

$$j_H = \frac{h_i D}{k} \left(\frac{C\mu}{k} \right)^{-1/3} \quad \text{and is plotted against}$$

MISCELLANEOUS DATA

Table D4. ENTHALPIES OF GASES EXPRESSED IN Btu/sct OF GAS, REFERENCE 60°F

°F	N ₂	O ₂	Air	H ₂	CO	CO ₂	H ₂ O ^a
60	0.31	0.31	0.32	0.31	0.32	0.39	-
77	0.31	0.74	0.74	0.73	0.74	0.94	0.36
100	0.74	2.58	2.58	2.55	2.58	3.39	0.85
200	2.58	4.42	4.42	4.40	4.43	5.98	2.98
300	4.42	6.27	6.29	6.24	6.29	8.69	5.14
400	6.27	8.14	8.17	8.09	8.18	11.52	7.33
500	8.14	10.40	10.07	9.89	10.08	14.44	9.52
600	10.02	12.43	12.00	11.77	12.01	17.45	11.81
700	11.93	14.49	13.95	13.61	13.96	20.54	14.11
800	13.85	16.59	15.92	15.47	15.94	23.70	16.45
900	15.80	18.71	17.92	17.36	17.94	26.92	18.84
1,000	17.77	20.85	19.94	19.20	19.97	30.21	21.27
1,100	19.78	23.02	21.98	21.08	22.02	33.55	23.74
1,200	21.79	25.20	24.05	22.95	24.10	36.93	26.26
1,300	23.84	27.40	26.13	24.87	26.19	40.36	28.82
1,400	25.90	29.62	28.24	26.80	28.31	43.85	31.42
1,500	27.98	31.85	30.38	28.70	30.44	47.35	34.08
1,600	30.10	34.10	32.50	30.62	32.58	50.89	36.77
1,700	32.21	36.34	34.66	32.52	34.74	54.48	39.49
1,800	34.34	38.61	36.82	34.45	36.93	58.07	42.26
1,900	36.48	40.90	38.99	36.43	39.12	61.71	45.06
2,000	38.65	43.17	41.18	38.49	41.31	65.35	47.91
2,100	40.84	45.47	43.39	40.57	43.53	69.02	50.78
2,200	43.00	47.79	45.61	42.66	45.74	72.71	53.68
2,300	45.24	50.11	47.83	44.71	47.99	76.43	56.64
2,400	47.46	52.43	50.07	46.82	50.23	80.15	59.59
2,500	49.67	54.73	52.31	48.91	52.47	83.88	62.60
3,000	60.91	64.18	61.39	57.22	61.55	98.96	77.98
3,500	72.31	76.13	72.87	68.14	73.00	118.15	93.92
4,000	83.79	88.29	84.42	79.38	84.56	137.62	110.28
4,500	95.37	100.64	96.11	90.68	96.21	157.20	126.96
5,000	107.04	113.20	107.91	102.42	107.93	176.93	143.92
5,500	118.78	125.89	119.78	114.21	119.70	196.77	161.07
6,000	132.54	139.74	131.73	126.16	131.52	216.77	178.41
6,500	142.37	151.72	143.76	138.35	143.37	236.88	195.82

^aEnthalpies are for a gaseous system, and do not include latent heat of vaporization. L_v = 1,059.9 Btu/lb or 50.34 Btu/sct of H₂O vapor at 60°F and 14.696 psia.

Table D3. ENTHALPIES OF VARIOUS GASES
EXPRESSED IN Btu/lb OF GAS

Temp, °F	CO ₂	N ₂	H ₂ O ^a	O ₂	Air
100	5.8	6.4	17.8	8.8	9.6
150	17.6	20.6	40.3	19.8	21.6
200	29.3	34.8	62.7	30.9	33.6
250	40.3	47.7	85.5	42.1	45.7
300	51.3	59.8	108.2	53.4	57.8
350	63.1	73.3	131.3	64.8	70.0
400	74.9	84.9	154.3	76.2	82.1
450	87.0	97.5	177.7	87.8	94.4
500	99.1	110.1	201.0	99.5	106.7
550	111.8	122.9	224.8	111.3	119.2
600	124.5	135.6	248.7	123.2	131.6
700	150.2	161.4	297.1	147.2	156.7
800	176.8	187.4	346.4	171.7	182.2
900	204.1	213.8	396.7	196.5	211.4
1,000	231.9	240.5	447.7	221.6	234.1
1,100	260.2	267.5	499.7	247.0	260.5
1,200	289.0	294.9	552.9	272.7	287.2
1,300	318.0	326.1	606.8	298.5	314.2
1,400	347.6	350.5	661.3	324.6	341.5
1,500	377.6	378.7	717.6	350.8	369.0
1,600	407.8	407.3	774.2	377.3	396.8
1,700	438.2	435.9	831.4	403.7	424.6
1,800	469.1	464.8	889.8	430.4	452.9
1,900	500.1	493.7	948.7	457.3	481.2
2,000	531.4	523.0	1,003.1	484.5	509.5
2,100	562.8	552.7	1,069.2	511.4	538.1
2,200	594.3	582.0	1,130.3	538.6	567.1
2,300	626.2	612.3	1,192.6	566.1	596.1
2,400	658.2	642.3	1,256.8	593.5	625.0
2,500	690.2	672.3	1,318.1	621.0	654.3
3,000	852.3	823.8	1,640.2	760.1	802.3
3,500	1,017.4	978.0	1,975.4	901.7	950.3

^aThe enthalpies tabulated for H₂O represent a gaseous system, and the enthalpies do not include the latent heat of vaporization. It is recommended that the latent heat of vaporization at 60°F (1,059.9 Btu/lb) be used where necessary.