Hot Air Recirculation by Air Coolers

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ABSTRACT

This paper covers model, plant and field smoke testing on dry air coolers. A technique and equipment for smoke testing is described. From this qualitative and quantitative data, preliminary equations are outlined for determining if an air cooler design will recirculate hot air. Numerous examples of correct and incorrect air cooler layouts are shown for the engineer's consideration.

HOT AIR RECIRCULATION BY AIR COOLERS

I. HISTORY

Commercial dry air cooling dates back to 1930. The first designs were single face vertical tubes, with fan in the vertical plane forcing air through the bundle (Figure 1a). Later, square or rectangular units with vertical tube bundles were made (Figure 1b). These designs were soon in disfavor due to wind direction effects. About 1935, units with horizontal fin tubes of induced draft (Figure 1c) and forced draft (Figure 1d) types were in operation. Before 1940, to save plot plan area, "V" type units were designed. These are shown in Figure 1e (forced draft) and Figure 1f (induced draft). They have the inherent problem of wind direction, requiring wind screens and close attention to hot air recirculation. By 1950, 90 to 100 percent commercial dry air-cooled process plants were built. All during this period the problem of hot exit air recirculation to inlets was noted and various solutions used.

There is a large amount of published information available covering wind effects on buildings of various shapes, and dilution of vent air from vehicular tunnels. Only a small amount of published material is available on air coolers. The bibliography attached contains only a small proportion of the literature extant. Briefly, the most prevalent hot air recirculation problems encountered in the last 35 years were:

1. High air approach velocities to air coolers.

2. Air coolers placed too close to each other in the downwind direction.

3. Air coolers placed in front of downwind obstructions.

4. Air coolers placed at different elevations near each other.

5. Low exit hot air velocities.

6. Indiscriminate plot plan mixing of forced and induced draft air coolers.

7. Inadequate analysis of plot plan layout in view of prevailing summer wind direction. Also, placing air coolers with close temperature approaches on the leeward side.

The majority of these problems were due to lack of knowledge and attention by project engineers. No comprehensive design criteria is available in usable form. There is scattered published data such as: 1) "Hudson River Holland Tunnel Vent Stack Dilution Tests" by McElroy et al, 1943, 2) "Climatic Considerations in the Design of Air Cooled Condensers" by Collins and Mathews, 1958, and 3) Cooling Tower Institute "Bulletin PFM-110" on Recirculation, 1958.

Over the last 20 years, the author's company has accumulated field test data and in the last 5 years developed smoke testing techniques for model and field tests on dry air coolers.

II. SMOKE TEST METHOD

Initially, Hudson Products Corporation utilized various types of smoke bombs and flares to provide the smoke for testing the recirculation of air in various configurations of air coolers. Use of these bombs and flares was discontinued due to: 1) the duration of the smoke generation being too short to adequately illustrate the recirculation, and 2) the smoke generated by the bombs and flares left an undesirable residue on the surfaces of the air cooler.

A resonant pulse generator was acquired in 1966 and has become the principal test unit. The most current unit in use is the Dyna-Fog Chemical Applicator, Model 150-B, manufactured by Curtis Dyna Products Corporation, Westfield, Indiana. The unit operates on regular gasoline using a mineral seal oil formulated for thermal fogging. Primarily, the formulation used should have a flash point greater than 200°F. The unit delivers 25 CFM of gas which vaporizes and expels the oil as fog.

Testing requires some additional equipment including an anemometer, calibrated measuring rod, thermometers, and for permanent visual records, a camera.

The test procedure followed by the company's personnel is as follows:

1. Wind velocity and ambient temperature data are taken at points unaffected by the smoke generator.

2. Positioning the generator so that the full quantity of smoke generated is drawn into the air inlet, exit air temperatures are taken and an exit air traverse is made.

3. Inlet air temperatures around the perimeter of the air cooler are taken with the generator in operation.

4. Inlet area, for approach velocity, measurements are made with the calibrated rod.

5. Photographs are taken for visual record of plume heights and test conditions.

III. THEORY

Figure 2 shows scale model tests at 5 and 15 miles per hour wind velocity for forced and induced draft units. These test data agree very closely with McElroy et al (1) their Figure 16, based on Ve/Vw ratios. Figures 3A and 3B show field tests on both induced and forced draft air coolers in a southwestern plant at 5 miles per hour wind velocity. The Ve/Vw velocity ratios were 3.1 and 1.1, respectively. The approach velocities were 8.3 and 13.3 feet per second. Thus, it is seen that Ve/Vw and Ve/Va ratios are primary factors in air flow around air coolers.

The type of wind is important. With a steady wind, stable conditions usually prevail around the air coolers. Gusty, rolling turbulent winds cause unstable conditions and are not amenable to accurate prediction.

For a single air cooler, such as shown in Figure 4 Unit A, under no wind (Vw = 0), hot air recirculation is determined by:

$$\left(\frac{Ve^2 \rho e}{2g}\right) - \left(\frac{Va^2 \rho a}{2g}\right) = \frac{Positive - no\ recirculation}{Negative - recirculation} \tag{1}$$

With steady wind the equation becomes:

$$\left(\frac{Ve^2 \rho e}{2g}\right) - \left[\left(\frac{Va^2 \rho a}{2g}\right) + \phi\right] = \frac{Positive - no\ recirculation}{Negative - recirculation} \tag{2}$$

Note: $\varnothing = C \left(\frac{Vw^2 \rho w}{2g}\right)$ Negative pressure on downwind side of air cooler.

$C = 0.2$ to 0.6

For an air cooler as shown in Figure 4, Unit B, downwind from Unit A, under steady wind, hot air recirculation is determined by three factors; namely,

1. Forces around Unit A
2. Forces around Unit B
3. Dilution of hot exit gases from Unit A

For the forces around Units A and B, Figure 4, the equations are:

$$\left(\frac{Ve^2 \rho e}{2g}\right)_{(A)} - \left[\left(\frac{Va^2 \rho a}{2g}\right) + \left(\frac{Vd^2 \rho d}{2g}\right) + \phi\right]_{(A)} = \frac{Positive - no\ recirculation}{Negative - recirculation} \tag{3}$$

$$\left(\frac{Ve^2 \rho e}{2g}\right)_{(B)} - \left[\left(\frac{Va^2 \rho a}{2g}\right) + \left(\frac{Vd^2 \rho d}{2g}\right) + \phi\right]_{(B)} = \frac{Positive - no\ recirculation}{Negative - recirculation} \tag{4}$$

Note: If either equation 3 or 4 is negative there will be hot air recirculation.

For diluation effects on Unit B from Unit A, Figures 5 [from McElroy et al (1)] gives approximate intensity. Since, in this case, the units are equal heights the plot cannot be read. Assume Unit B is 10 feet higher, Ve = 20, Va = 10, Vw = 22, diameter of fans = 12, L = 60. Then Ve/Vw = 0.91, L/fan diameter = 5.0, vertical distance/fan diameter = 0.83. From Figure 5 dilution is between 10 to 15 percent. If ambient air is 90°F, and air rise is 40°F, then mixed air to Unit B would be 94 to 95°F. Figure 6 from the Cooling Tower Institute "Bulletin PFM-110", 1958, plots percent recirculation versus tower length in feet. This varies from 1 to 11 percent and is not comparable to dry air coolers except where in banks and free area under banks is 50 percent or more obstructed. The data reported is on thirty different cooling towers and was also plotted against frame height, exit velocity, stack height, wind direction and velocity. No correlation was better than that shown in Figure 6.

One final point should be made. Theoretically, the hot exit gases should have a thermal draft effect. Field and model tests do not show any appreciable difference in tests up to 70°F air temperature rise. This is probably due to the fact that the exit air velocity is always much greater than the velocity which would occur due to thermal draft alone. The authors, therefore, have ignored this effect.

IV. DISCUSSION OF RESULTS

This paper presents various qualitative plus some quantitative field test data from which preliminary equations have been proposed. Additional field data, under rigorous test conditions, is required for complete quantitative analysis. Discussion of results covers testing the present data by the proposed equations.

This test, Figure 3A, was run in a recycling plant in the southwest. Several forced draft units, 13 x 24, each having two 10-ft diameter fans, were being used to cool jacket water. The test conditions were:

$Ve = 7.2 \qquad \rho e = .067$

$Va = 8 \qquad \rho a = .071$

$Vw = 8 \qquad \rho w = .071$

$$\left(\frac{Ve^2 \rho e}{2g}\right) - \left[\left(\frac{Va^2 \rho a}{2g}\right) + \left(\frac{0.2Vw^2 \rho w}{2g}\right)\right]$$

$$= \frac{(7.2)^2 .067}{64} - \frac{(8)^2 .071}{64} - \frac{0.2\,(8)^2 .071}{64}$$

$$= .054 - .071 - .014 = -0.031$$

The test equation indicates self-recirculation, and this is borne out by Figure 3A. Reduction of cooling capacity up to 11 percent was observed.

This test, Figure 3B, was run in the same recycling plant as the previous case; however, it was run on an induced draft unit, 14 x 24, with two 10-ft diameter fans.

The test conditions were:

$Ve = 28 \qquad \rho e = .067$

$Va = 7 \qquad \rho a = .071$

$Vw = 8 \qquad \rho w = .071$

$$\left(\frac{Ve^2 \rho e}{2g}\right) - \left[\left(\frac{Va^2 \rho a}{2g}\right) + \left(\frac{0.2Vw^2 \rho w}{2g}\right)\right]$$

$$= \frac{(28)^2 .067}{64} - \frac{(7)^2 .071}{64} - \frac{0.2\,(8)^2 .071}{64}$$

$$= .821 - .054 - .014 = 0.753$$

The test equation indicates no possibility of self-recirculation, which is borne out by Figure 3B.

This test, Figure 7, was run in Hudson's plant on a forced draft unit (20-32), having two 14-ft diameter fans. By suspending the unit (while running) at four different heights, and smoke testing at each height, it was possible to gauge the effect of varying only the approach velocity. The test conditions were: Ve = 8.1, Vw = 7.0, and Va = 6.1, 6.4, 7.5 and 8.8, respectively in figures A, B, C and D. There was no heat load and the ambient temperature was 71°. Therefore, $\rho_e = \rho_a = \rho_w = .075$.

Referring to figure A:

$$\frac{\rho}{2g}\left[Ve^2 - Va^2 - 0.2\,Vw^2\right] = \frac{.075}{64}\left[(8.1)^2 - (6.1)^2 - 0.2(7)^2\right] = 0.022$$

figure B:

$$\frac{.075}{64}\left[(8.1)^2 - (6.4)^2 - 0.2(7)^2\right] = 0.017$$

figure C:

$$\frac{.075}{64}\left[(8.1)^2 - (7.5)^2 - 0.2(7)^2\right] = -.0002$$

figure D:

$$\frac{.075}{64}\left[(8.1)^2 - (8.8)^2 - -0.2(7)^2\right] = -0.025$$

These results indicate that figures A and B are not self-recirculating; that figure C is right on the line; and that figure D is self-recirculating. These results agree with the corresponding pictures in all four cases.

This test, Figure 8, was run in a Gulf Coast refinery. Two long banks of forced draft units were involved. These banks were about 80 feet apart, as shown in Figure 10. The test conditions were:

$Ve = 7.5$ $\rho e = .066$

$Va = 6.0$ $\rho a = .070$

$Vd = 3.0$ $\rho d = .070$

$Vw = 12.5$ $\rho w = .072$

$$\frac{Ve^2 \rho e}{2g} - \left[\frac{Va^2 \rho a}{2g} + \frac{Vd^2 \rho d}{2g} + \frac{0.2\,Vw^2 \rho w}{2g}\right]$$

$$= \frac{(7.5)^2 .066}{64} - \frac{(6)^2 .070}{64} - \frac{(3)^2 .070}{64} - \frac{0.2\,(12.5)^2 .072}{64}$$

$$= .0580 - .0394 - .0100 - .0351 = -.0264$$

The test equation indicates that the downwind bank will receive hot air from the exit of the upwind bank. The test data indicated that this did happen, and that the percentage of air so recirculating was more than 25 percent of the air to the downwind bank. This reduced the capacity of the downwind unit 26 percent and the overall capacity 13 percent.

Figure 9 covers field tests at a southwestern recycling plant. It illustrates plot plan layout where effects of prevailing hot winds were not considered. The plot plan could have been revolved 180° at no extra cost and all hot air recirculation problems eliminated.

Data on a test with southwest winds gave 13.6 percent hot exit air recirculation from southwest air coolers to central induced draft inlet air. This raised air inlet temperature 5 to 7°F and reduced capacity approximately 10 percent. Essential test data: Ambient air = 95°F; elevation = 2600 feet; Vw - SW = 16. Southwest forced draft unit - Va = 7.8; Ve = 12.2; ρa = 0.064; ρe = 0.0595. Central induced draft unit - Va = 10.4; Ve = 29.4; ρa = 0.064; ρe = 0.0595, Vd = 5.0.

Now, testing southwest air cooler by equation (3):

$$\frac{(12.2)^2 \times .0595}{64} - \left[\frac{7.8^2 \times .064}{64} + \frac{5.0^2 \times .064}{64} + .2\frac{16^2 \times .064}{64}\right] = +.0014 \text{ lbs/ft}^2$$

This indicates the southwest bank of air coolers is near hot air recirculation on itself on downwind side. During the test, occasional puffs of hot air were noted at ground level by the author. Note that if a C value of 0.3 had been used, equation (3) would have given a negative answer.

Now, equation (4) for central induced draft unit gives:

$$\frac{29.4^2 \times .0595}{64} - \left[\frac{10.4^2 \times .064}{64} + \frac{5.0^2 \times .064}{64} + 0.2\frac{16^2 \times .064}{64}\right] = +.619 \text{ lbs/ft}^2$$

This is well above the recirculation point and C could be the maximum (0.6) without causing recirculation. Here, it is appropriate to point out that induced draft units, with Ve = 20 - 28 fps versus forced draft Ve = 9 - 12 fps, will always have less hot air recirculation than forced draft.

In this test, hot air recirculation was from the southwest coolers exit into the inlet of the central induced draft units, which were one foot higher elevation than exit of the southwest units. Refer to Figure 5, which is correlation of stack dilution taken from McElroy et al (1). From layout and test data Vw/Ve = 1.3; X/W = 1.5; Y/W = 0.11 then dilution is between 10 and 15 percent. Now, there are 880,000 scfm of 45°F above ambient air temperature from the southwest unit. 1,280,000 scfm of air is drawn through the west inlet of the central induced draft unit which is raised 5°F. This calculates 13.6 percent recirculation, based on Figure 5.

Some ten other tests were run with wind from the south and southeast, all confirming recirculation on the induced draft unit. They are not reported here due to effects of obstructions.

Figure 10 is an end elevation of a bank of air coolers in a natural gasoline plant in the southwest. This illustrates the problem of adding a unit to an existing bank of coolers. The new induced draft unit was installed as shown.

Field test data in 3 foot space between old and new units gave: ambient air = 83.5°F; mixed air = 100°F; elevation = 100 ft; Ve old unit = 9.5 (ρair = .065); Vd between units = 13.5; Ve new unit = 23.4 (ρair = .065); Va - old = 8.4; Va - new = 8.4. This gives 24 percent hot air recirculation and approximately 35 percent reduction in cooling capacity of new unit. If new unit had been placed joining old units, and air inlets at the same level, this problem would have been avoided.

Figure 11 is the end elevation of an overhead air-cooled condenser and a bottoms cooler installed at different elevations in a Canadian refinery. Prevailing hot winds recirculated nearly all of the bottoms cooler exit gases (Ve) to the overhead condenser.

Figure 12 refers to two banks of forced draft units in an Alaskan plant. These units were 18 x 30, each having two 12-ft diameter fans. These units have a relatively high air approach velocity due to the air access area being limited to approximately 50 percent of the bundle face area. Test conditions were:

$Ve = 9.3 \qquad \rho\mathrm{e} = .070$

$Va = 17.5 \qquad \rho\mathrm{a} = .076$

$Vw = 12.0 \qquad \rho\mathrm{w} = .076$

$$\frac{(9.3)^2 .070}{64} - \frac{(17.5)^2 .076}{64} - \frac{0.2(12)^2 .076}{64} = .094 - .364 - .034 = -0.305$$

The test equation indicates considerable self-recirculation, which was borne out by test data.

Figure 13 illustrates some installations of jacket water coolers on gas pipeline stations. When wind direction is across coolers to compressor building the hot air recirculates back from the building to the air coolers. Assume following data: Induced draft - 12-ft diameter fans; Ve = 25; Vd = 4.2; Va = 8; Vw = 14.7; ρe -.066; ρd and ρa = .071.

Thus, equation (3);

$$\frac{(25)^2 \times .066}{64} - \frac{(8)^2 \times .071}{64} - \frac{(4.2)^2 \times .071}{64} - \frac{.2(14.7)^2 \times .071}{64} = +.506$$

There is no recirculation.

Now referring to Figure 5, X/W = 3; Y/W = 3.7; Vw/Ve = 0.59. This shows 50 percent of hot exit air is trapped by compressor building and a substantial percent is returned to the air cooler. This has been checked in the field and substantiated. Obviously, a forced draft unit would have a lower hot exit gas trajectory and a larger percent recirculation.

There is another problem with this type of layout. When wind is from over the building, a low pressure turbulent zone is built up between the air cooler and the building which will cause hot air recirculation, especially on forced draft units.

V. CONCLUSIONS

1. The four equations presented here are reasonable tests for design engineers to determine possibility of hot air recirculation.

2. Additional field test data under rigorous conditions are needed for refining equations (1) to (4).

3. The smoke test described here is a useful visual tool for analysis of hot air recirculation problems.

4. Forced draft air coolers are more susceptible to recirculation than induced draft due to low exist velocities (Ve).

5. Detailed study of the effects of obstructions and turbulent wind conditions are needed.

6. Figure 14 presents known correct layout criteria.

7. Using reasonable care and the equations herein, air-cooled installations may be economically designed to substantially eliminate air recirculation, as is true of the majority of the installations operating today.

8. Data showing effect on $\varnothing$ when wind velocity, Vw, approaches and exeeds Ve is needed to establish proper value of the constant C at various Ve/Vw ratios.

NOMENCLATURE

C	=	Constant in equation for ∅ (Value varies from 0.2 to 0.6)	
Cd	=	Concentration of Contaminant along axis of Discharge Stream – See Figure 5	
Cs	=	Concentration Ratio – See Figure 5	
Vw	=	Wind Velocity	ft/sec
Ve	=	Exit Velocity Based on fan throat free area for induced draft and total fin tube bundle face area for forced draft air coolers – See Figure 4	ft/sec
Va	=	Approach Velocity Based on periphery of air cooler and unobstructed height above ground level – See Figure 4	ft/sec
Vd	=	Downward Velocity Between Two Air Coolers Based on horizontal plane area between two air coolers and amount of air to the affected peripheries of the two air cooler – See Figure	ft/sec
X	=	Horizontal Distance for Air Cooler	ft
Y	=	Vertical Distance Between Air Cooler Exit and Next Inlet	ft
∅	=	$\left(\frac{Vw^2 \rho w}{2g}\right)$ Low pressure area downwind	lbs/ft^2
ρw, ρe, ρa	=	Air density, lbs/ft^3 referred to wind, exit air, and approaching air	

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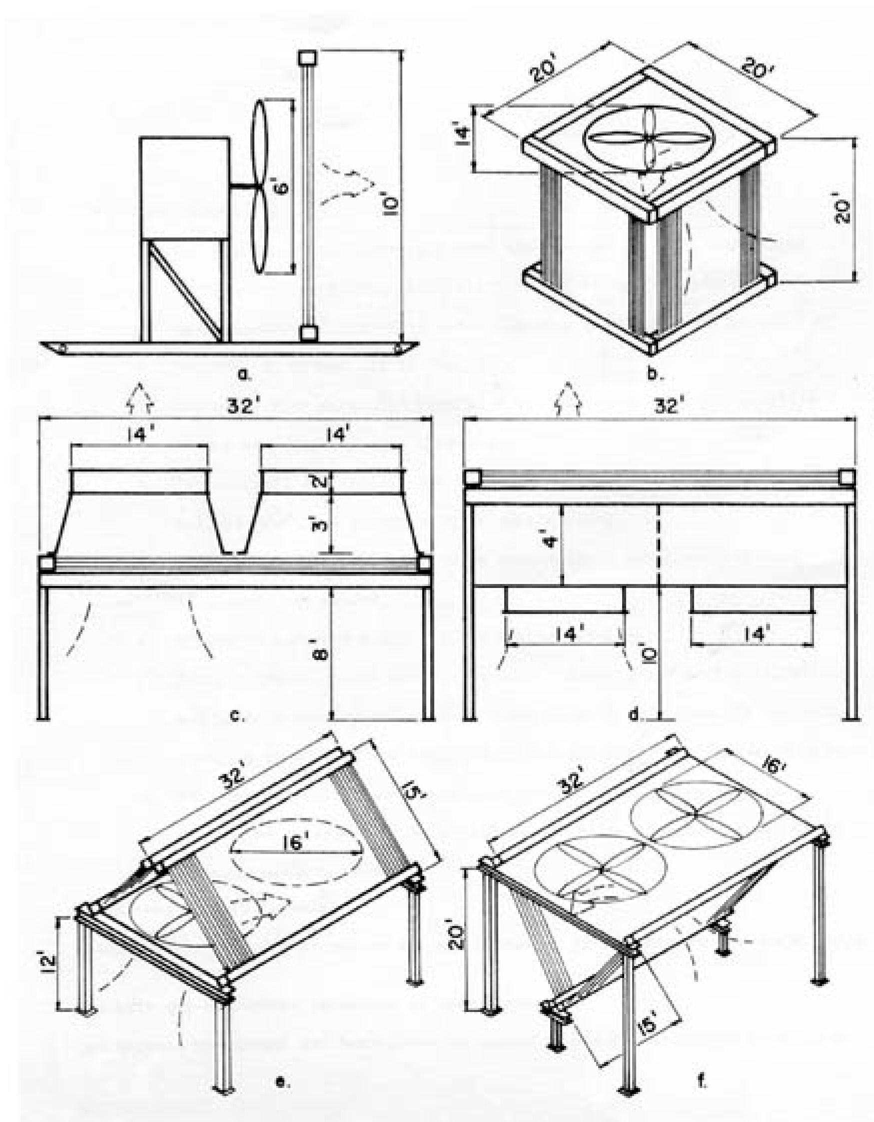

Figure 1 These figures illustrate various common types of dry air coolers. (a) Single face – vertical tube forced draft; (b) Multi-faced vertical tube induced draft; (c) Horizontal tube induced draft; (d) Same as (c) forced draft; (e) Inclined tube "V" - forced draft; (f) Same as (e) induced draft.

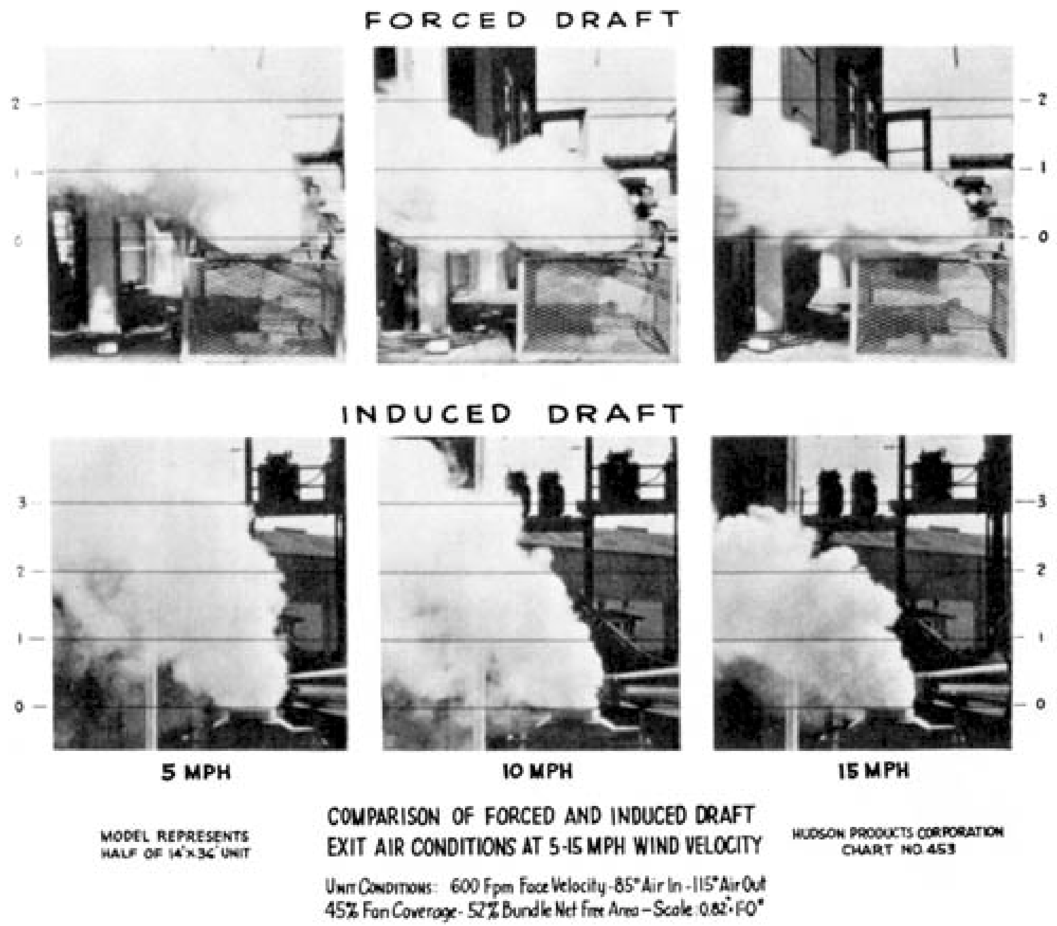

Figure 2 Comparison of forced and induced draft exit air conditions at 5-15 mph wind velocity; unit conditions: 600 fpm face velocity - 85° air in - 115° air out, 45% fan coverage, 52% bundle net free area - scale: 0.82" = 1' - 0"; Model represents half of 14' x 36' unit.

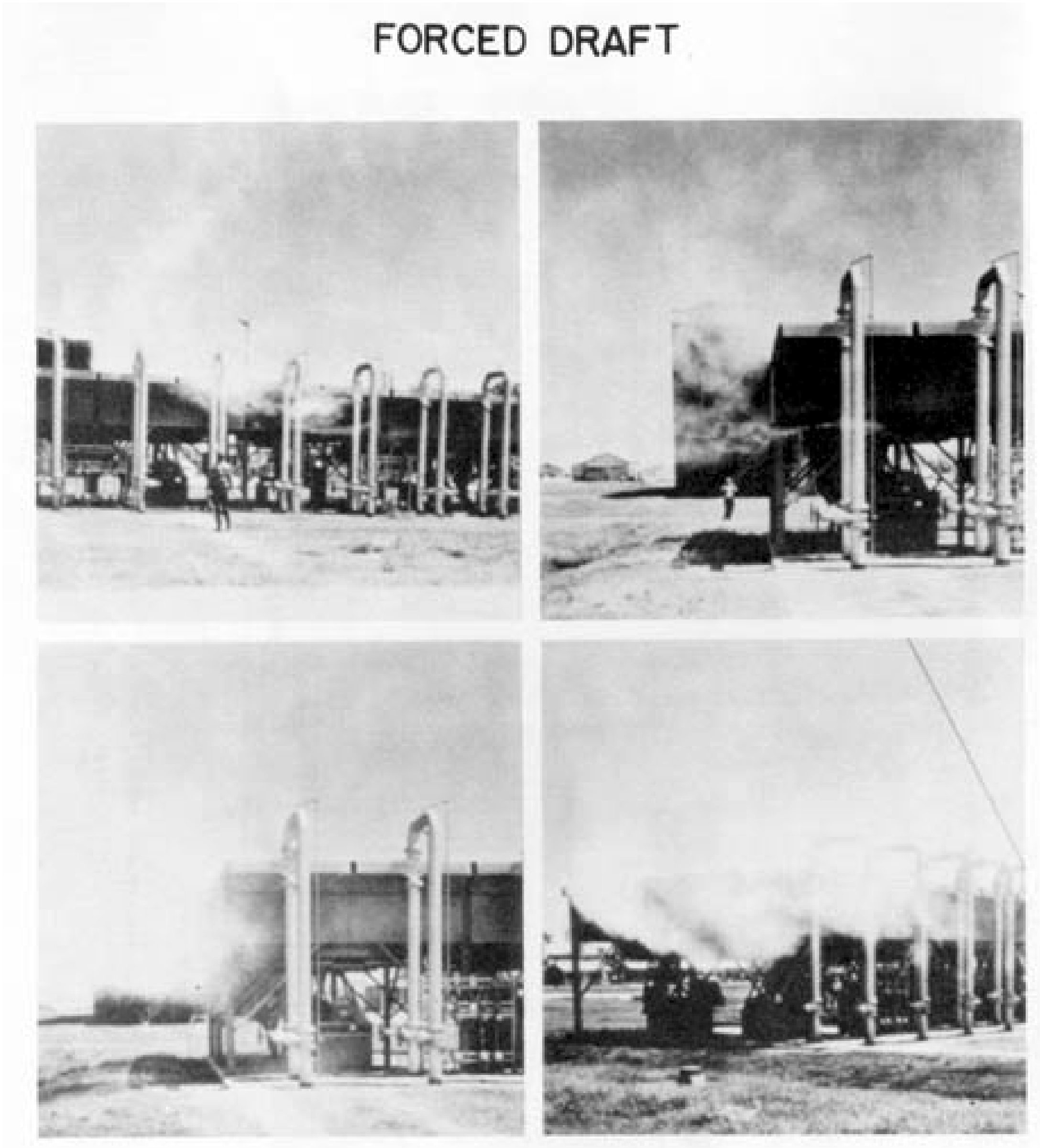

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Figure 3A Field test smoke pictures of forced draft air cooler at a southwestern recycling plant

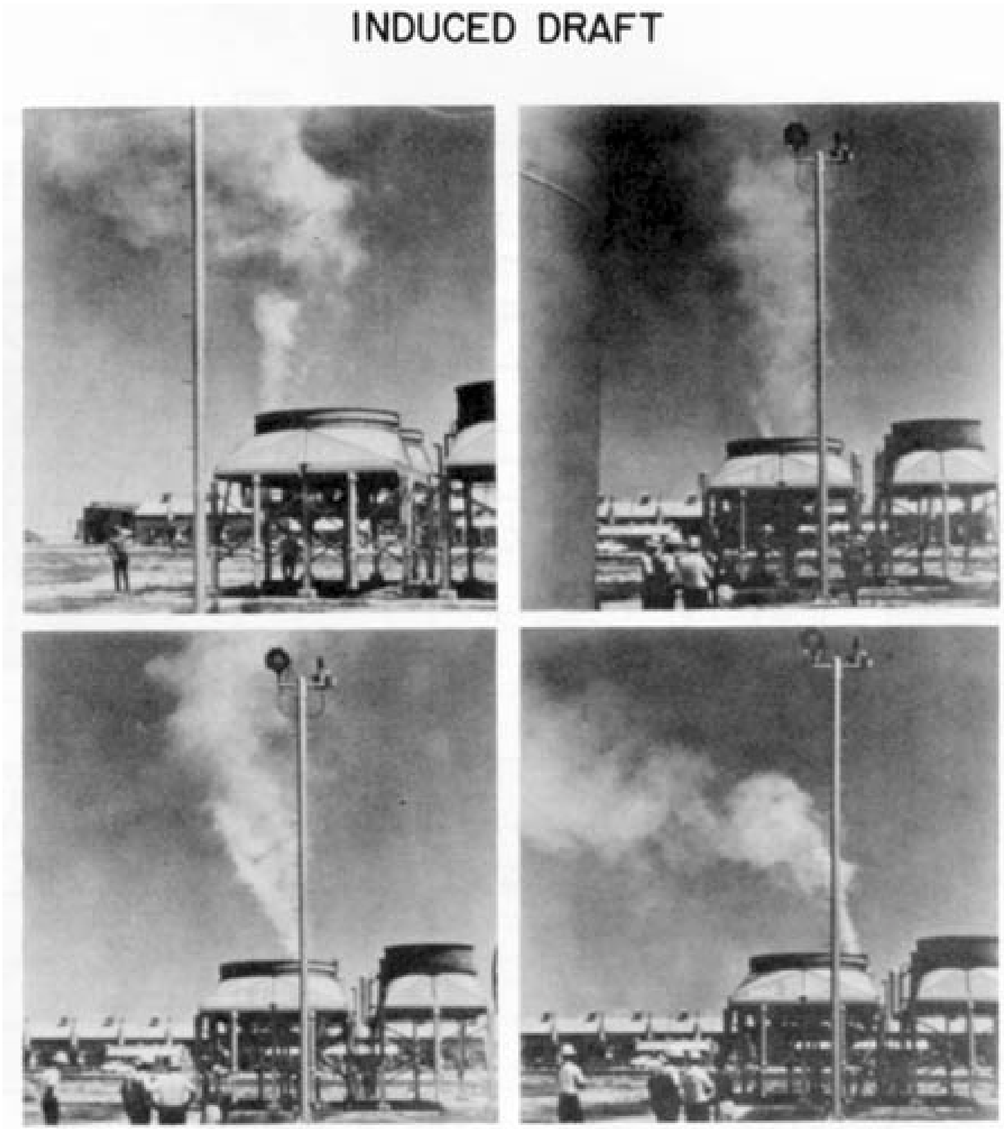

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Figure 3B Field test smoke pictures at same plant as 3A on induced draft air coolers.

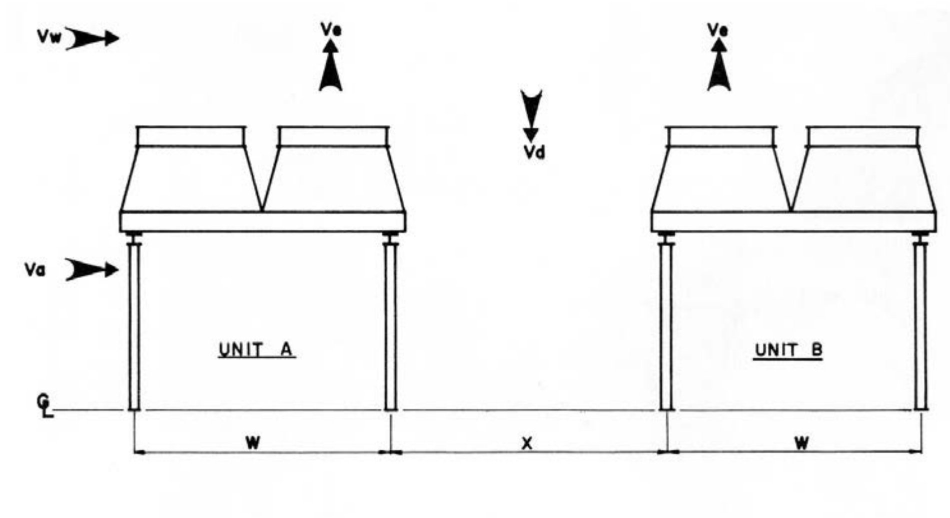

Figure 4   Force diagram and nomenclature for induced or forced draft air coolers.

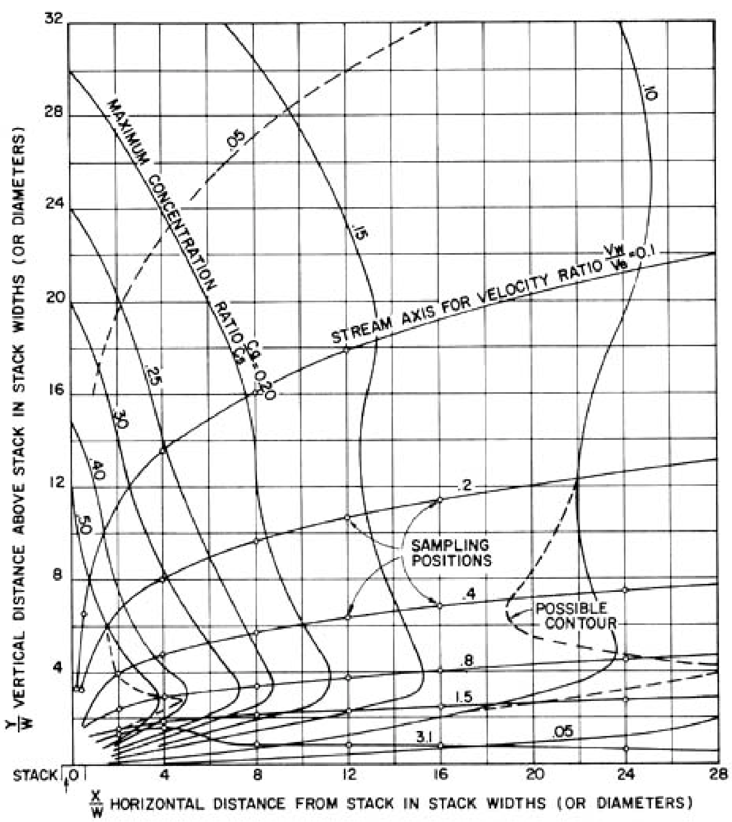

Figure 5   Maximum relative gas concentrations in undisturbed streams in the vicinity of a stack.

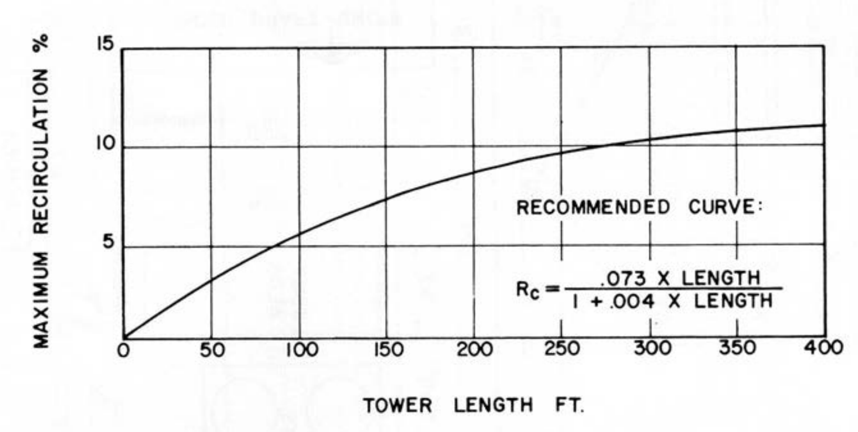

Figure 6 Maximum percent recirculation versus cooling tower length. Induced draft cooling towers only.

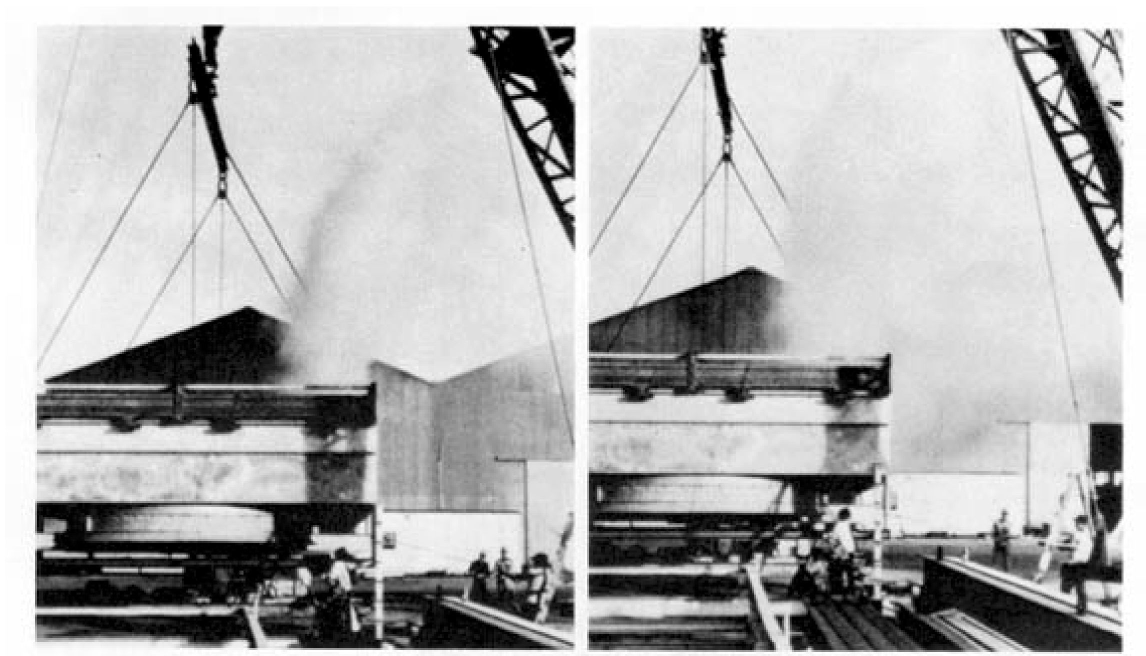

Figure 7A Smoke test on suspended air cooler at Va = 6.1 and Ve = 8.1, no recirculation.

Figure 7B Smoke test on suspended air cooler at Va = 6.4 and Ve = 8.1, no recirculation.

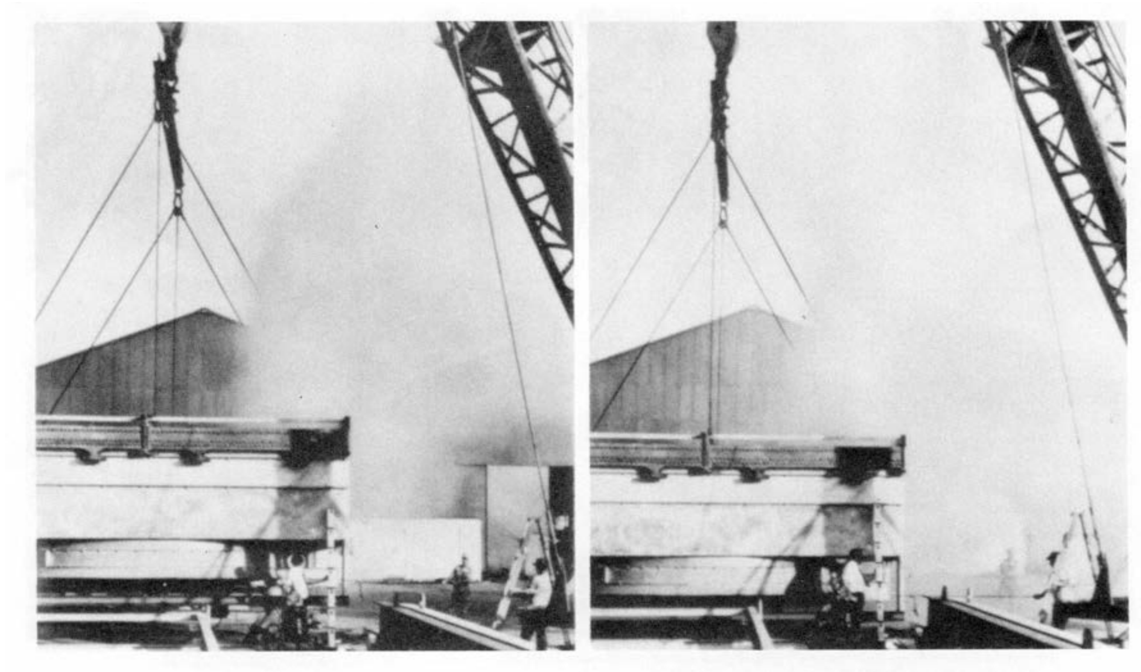

Figure 7C Smoke test on suspended air cooler at Va = 7.5 and Ve = 8.1, note some recirculation.

Figure 7D Smoke test on suspended air cooler at Va = 8.8 and Ve = 8.1, note the recirculation.

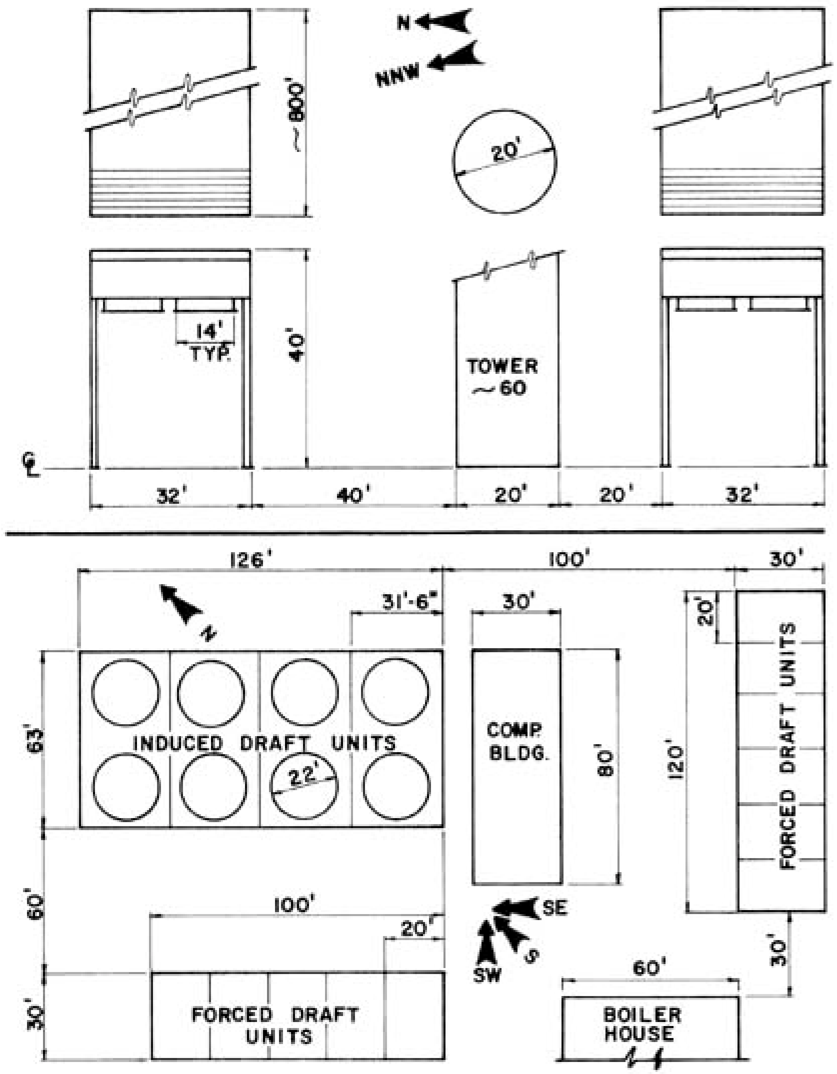

Figure 8 End elevation of two banks of overhead force draft air cooled condensers at refinery in southwest. Severe hot air recirculation on downwind unit.

Figure 9 Plot plan layout of air coolers in a southwestern recycling plant. Hot air recirculation on center unit with prevailing hot winds from southwest. Revolving layout 180 degrees would have corrected problem.

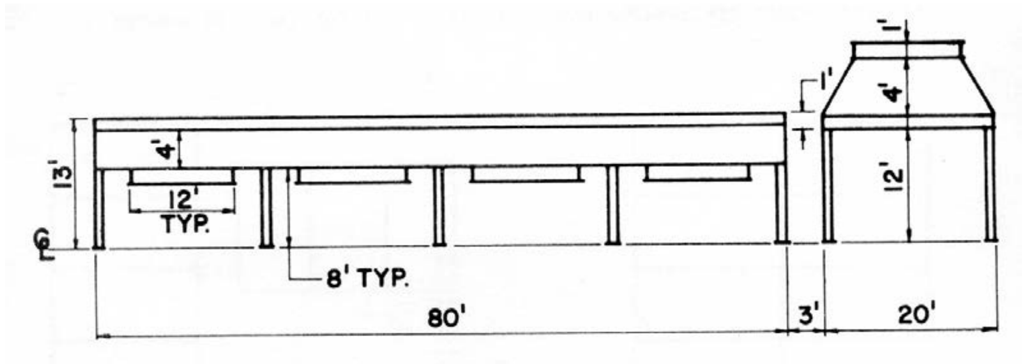

Figure 10 End elevation of air cooler layout in a southwestern natural gasoline plant. Hot air recirculation occurred where new induced draft air cooler was added to existing forced draft air cooler bank as shown.

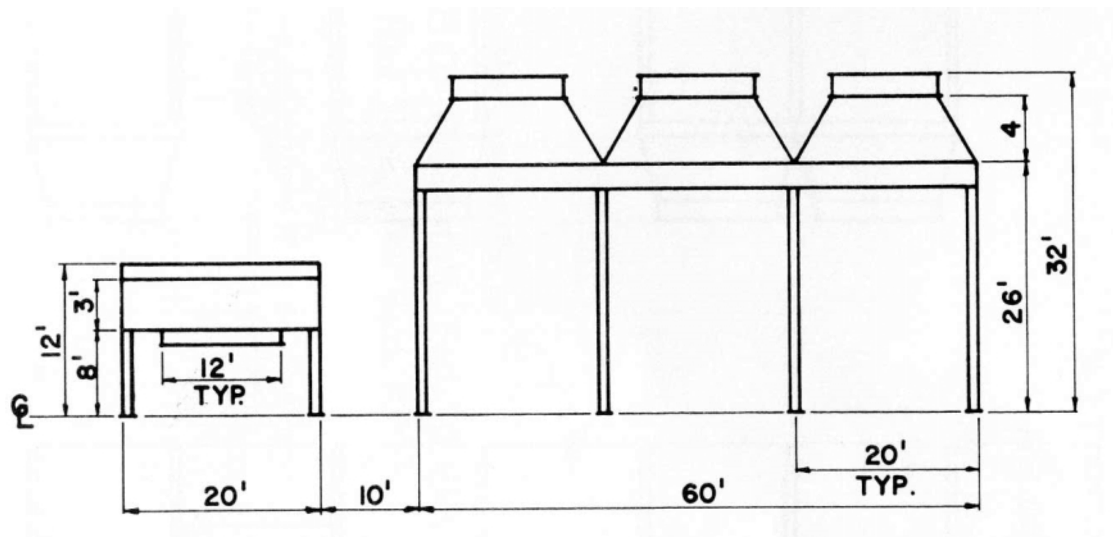

Figure 11 End elevation of air coolers installed in Canadian refinery. Hot air recirculated from air cooler at lower elevation.

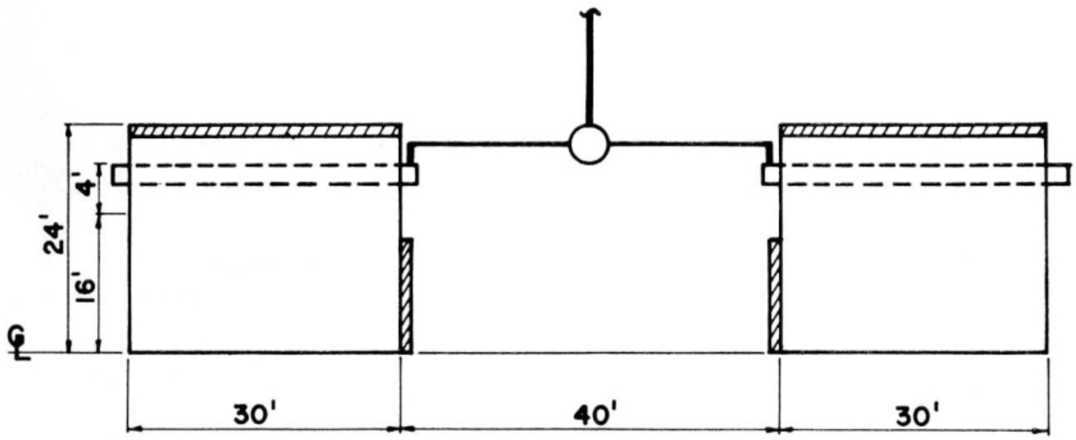

Figure 12 End elevation of two banks of forced draft air coolers installed in Alaskan plant. Hot air recirculated on both banks.

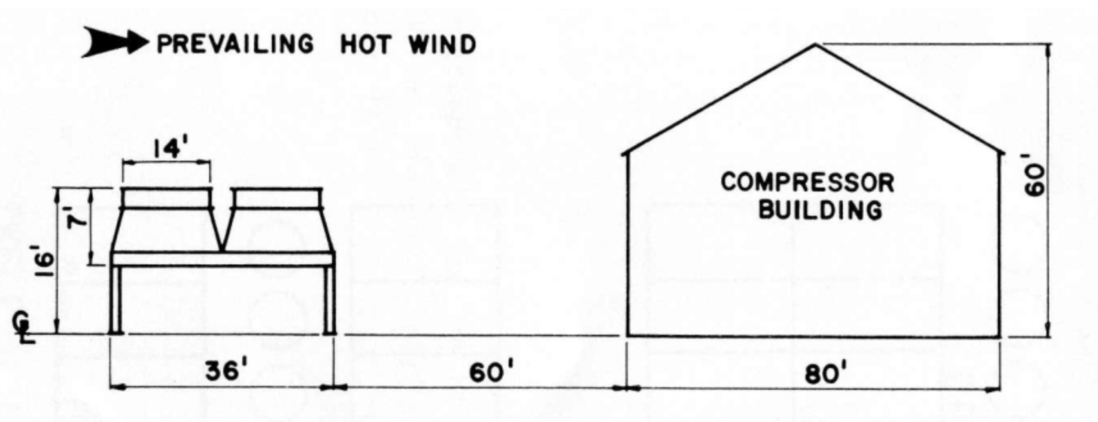

Figure 13 Jacket water coolers installed on gas pipelines near compressor building. Hot air recirculation due to building obstruction.

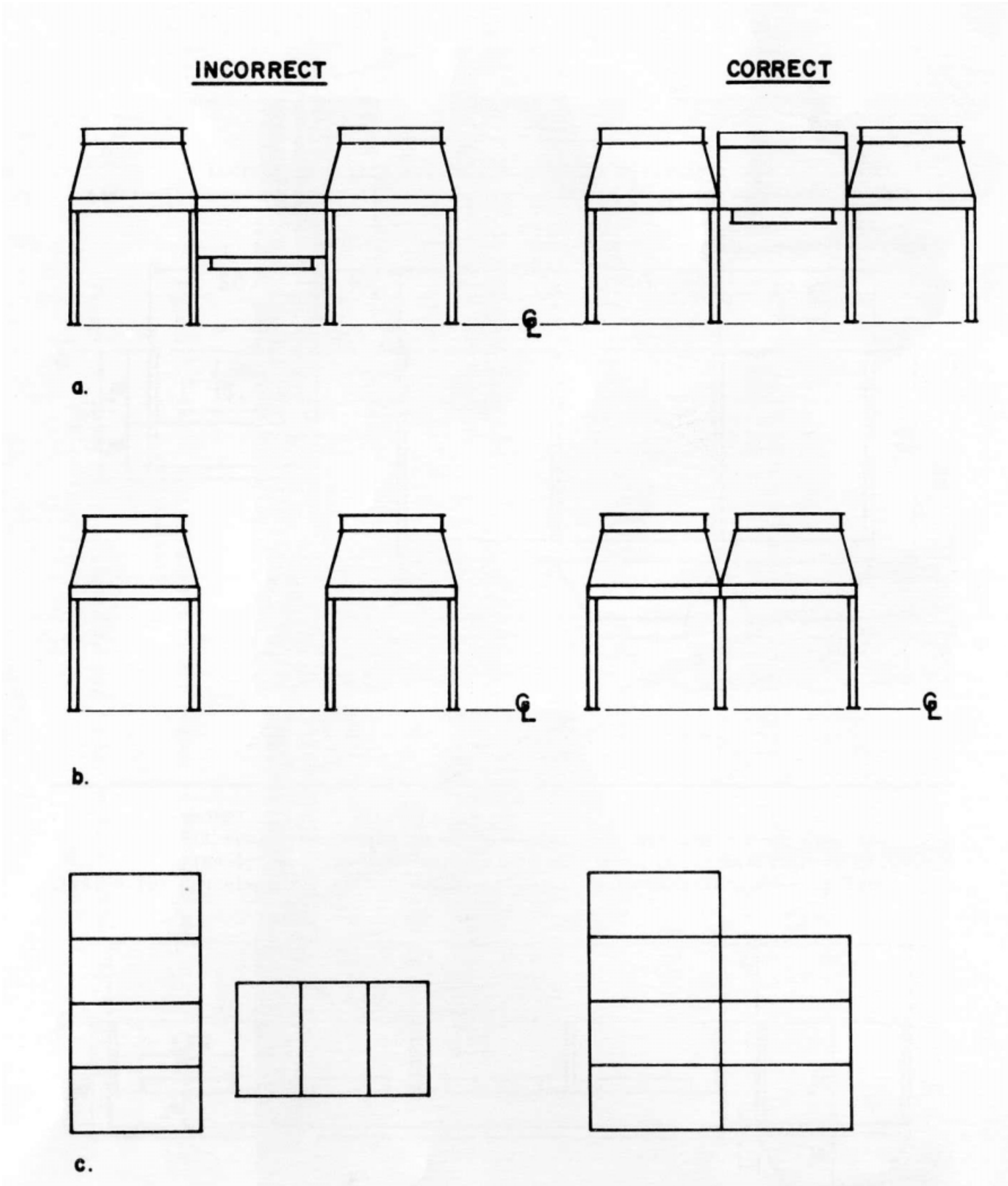

Figure 14(a)(b)(c) Incorrect and correct air cooler layouts.

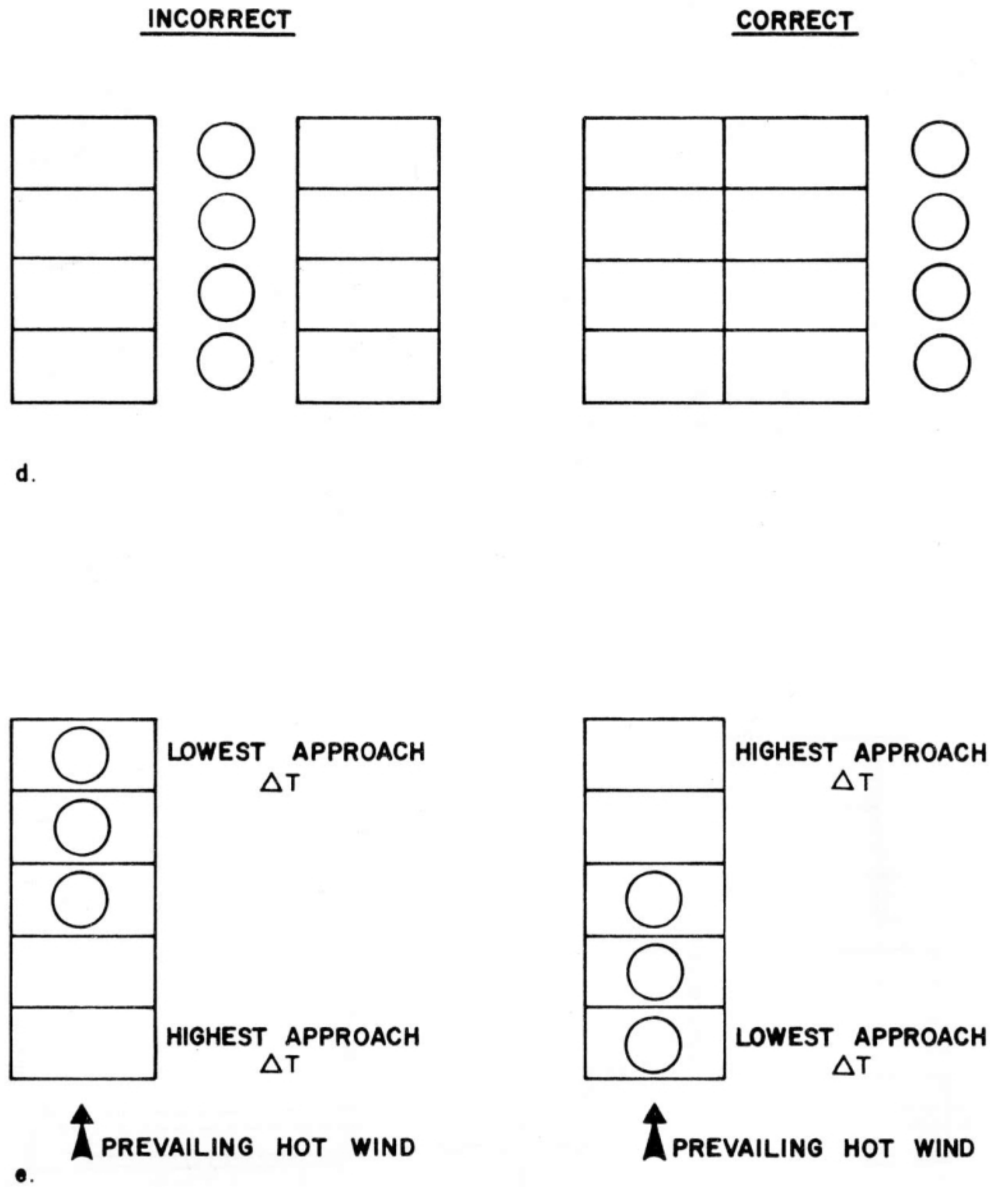

Figure 14(d)(e) Incorrect and correct air cooler layouts.