A QUANTITATIVE STUDY OF RECIRCULATION IN COOLING TOWERS

by

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Typed by Nancy Kerley

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The author wishes to express his sincere appreciation for the advice and patient assistance received from Associate Professor Wesley W. Smith of the Department of Mechanical Engineering. He also wishes to thank the other members of the Mechanical Engineering Department for their support and much needed help during the preparation of this thesis. The Fluor Products Company has been generous in contributing the needed equipment.

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INTRODUCTION

When the atmosphere is not yet saturated with water vapor, contact with water will cause a spontaneous mass and heat transfer to take place, until the atmosphere eventually becomes saturated. If the system is an isotropic one, then the water will be cooled to a definite temperature which is the so called wet bulb temperature of the atmosphere. This is the kind of heat transfer which occurs inside an industrial cooling tower where water is cooled by direct contact with air to a temperature approaching the wet bulb temperature of the ambient atmosphere. In contrast to the usual indirect contact exchanger there is a combination of heat and mass transfer occurring in the exchanger. The performance of this kind of cooling apparatus is dominated by the enthalpy potential of the tower inlet air, which is the difference between the enthalpy of the inlet air and the enthalpy of the saturated air at the subject point. The enthalpy of the atmosphere depends heavily on how much water vapor is contained in the atmosphere. The weight percentage of water vapor in the atmosphere is usually called the specific humidity of the atmosphere. As a rule the lower the moisture

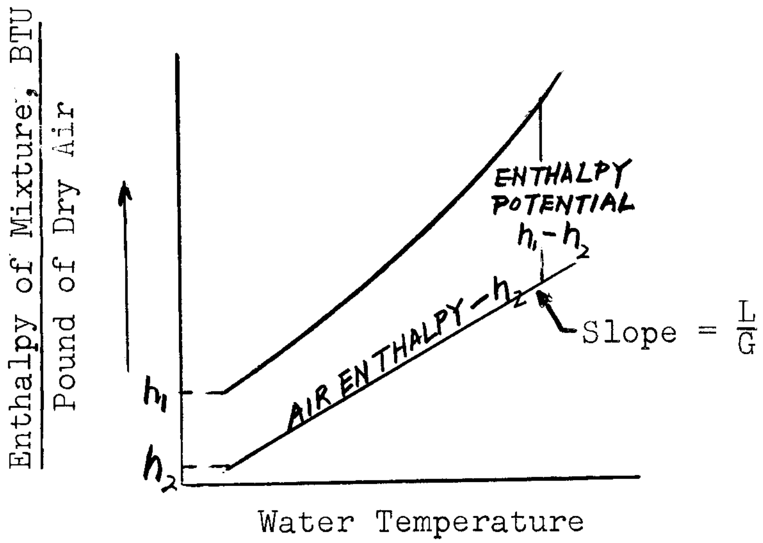

Figure 1A. Graphical Relation of Water Temperature to Enthalpy BTU of Mixture Per Pound of Dry Air.

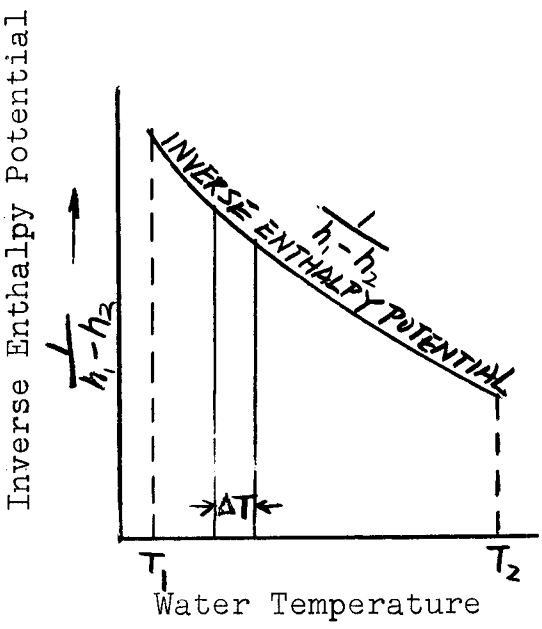

Figure 1B. Graphical Relation of Water Temperature to Inverse Enthalpy Potential of Mixture Per Pound of Dry Air.

content in the atmosphere, the higher its enthalpy potential will be. Therefore it is desirable that the tower inlet air contain no more moisture than the ambient atmosphere. Tower performance is described (6, p. 38) by the following equation where KaV/L is called the tower characteristic:

$$KaV/L = \int_{T_1}^{T_2} \frac{dT}{(h_1 - h_2)} \qquad (1)$$

Tests have indicated that its relation with the L/G ratio is of the following form:

$$KaV/L = C(L/G)^{-N} \qquad (2)$$

C and N are experimental constants that have different values for each individual tower. Therefore the tower performance can be improved either by a decrease in the L/G ratio or an increase in the "a" value in the KaV/L term. Decreasing the L/G ratio results in less water per pound of air and therefore in a larger tower structure and more fan horsepower. The value of "a", the effective contact area per unit volume of tower, can be increased by a water distribution system that breaks the water into fine sheets or particles and by increasing the density and design of the packing.

The value of KaV/L from equation (1) is dependent on the enthalpy potential. The enthalpy potential is decreased by any increasing of the moisture of the ambient atmosphere before it enters the tower. Thus the importance of preventing recirculation becomes evident.

Though recirculation has long been a problem in the design and testing of the water cooling tower, complete understanding of the various factors contributing to recirculation has not yet been achieved. This problem is realized as an extremely variable phenomenon influenced by many elements of weather, tower configuration, and tower orientation. Also many authors have agreed that accurate prediction of the recirculation under all conditions is probably not possible (3, Forward). At present the calculation of recirculation factor is based upon a saturated exit air, although it is known there are entrained water droplets in the exit air stream (16, p. 1680). Such an apparent contradiction may be a factor which leads to the uncertainty of the recirculation factor.

In an effort to obtain consistent information on the recirculation factor with reasonable expenditure of both time and money, the author used scale models of cooling towers in a low speed wind tunnel having a throat measuring four by eight feet. After considering several ways of tracing the recirculated air, the simple idea of measuring the moisture content was chosen since very small quantities can be detected with wet bulb thermometers or thermocouples.

Air having entered the model being tested was withdrawn, heated, and saturated with moisture in a closed system external to the tower and then blown from the tower stack. Details of the construction required are given later. Wet bulb thermocouples were used to detect the moisture in the various areas inside and outside the tower.

The effect on recirculation of wind speed, tower orientation, and tower configuration was investigated.

In comparing this author's experimental data with that in the report on recirculation by the Cooling Tower Institute (3), it appears that the influence of exit air condition may not be a negligible factor in the study of the recirculation of water cooling towers.

THE CONDITION OF EXIT AIR AND ITS BEHAVIOR AFTER LEAVING A COOLING TOWER

The motion of the exit air of the cooling tower is quite like the plume emitted from the chimney of a power plant. Its chance to touch the ground is a function of its density and the crosswind speed. When the plume enters the crosswind it quickly develops a stirring motion caused by the obstacles on the ground or by the thermal convection (14, p. 207). When the air is very stable the plume quickly reaches its equilibrium level horizontally, often with very little dilution because of the negligible stirring motions in the environment. Sometimes it may be carried into sideways meanderings, particularly when the wind is light. The sun's heating effect on the earth's surface varies continuously resulting in variable ground temperature. The air itself is not a good absorber of the sun's radiation, but the ground will transfer some heat energy to it by conduction thus creating an energy potential field immediately above the ground surface and causing a stirring motion. When the exit air enters the natural atmosphere no matter what precautions were taken, this kind of stirring can not be avoided. Also it has been observed that neither the emission velocity nor the buoyancy potential difference of the warmer saturated atmosphere has any appreciable retarding effect on this turbulent motion. Furthermore this turbulent motion will produce a speedy forced diffusion of the exit air with the surrounding atmosphere, but the whole volume of the mixture will tend to move upward if there are no other interfering forces. A similar situation exists when a lighted cigarette is placed over a heated hot plate. The smoke moves upward then turns into a violent motion but generally continues to rise. The conclusion is that thermal convection alone will have negligible effect on recirculation in a cooling tower.

Crosswind causes the air leaving the tower to go into a stirring motion evidenced by downwash and eddies (14, p. 212). The downwash and the eddies will be treated in two separate paragraphs because of their importance.

Downwash is caused by the crosswind deflecting the air leaving the tower a sufficient amount so as to be caught in the leeward eddies. Consequently the air humidity on the ground level of the leeward side of the tower will be higher. It should be noted, however, that the crosswind also reduces the moisture concentration of the air stream through forced diffusion.

The eddy region is produced by the separation of the wind stream around the corners of the tower. Inside the eddy region the air is in a circular motion, and the whole region is a low pressure domain. Therefore once any part of the exit atmosphere of the cooling tower has been carried into this region, it becomes part of the eddy and may be sucked into the cooling tower air intake louvers. Therefore in a crosswind the leeward side of a tower has the possibility of a higher local air humidity.

In addition to configuration, orientation, etc., the downwash of the exit air and water mixture may be influenced by the entrained water droplets. Because of the weight of the water particles, they will quickly drop to the ground in the vicinity of the tower, and part of them will be evaporated during the falling period while the rest will raise the humidity at ground level.

When there is a crosswind blowing, falling droplets and downwash saturated air could appreciably raise the humidity on the leeside of the cooling tower. It is a question whether downwash of the saturated particles or fall out of the fine water particles has the most effect on the recirculation factor of water cooling towers, since both increase the local moisture content.

THE COOLING TOWER MODEL TESTING ANALYSIS

Model studies are made in order to avoid costly mistakes and to obtain design information. A "cut and try" method of design may be undertaken with models when it would be costly if it were undertaken with the full scale system. Model testing has long been employed in aeronautical engineering, dam construction, surface vessel design, and recently in many building structures (4, p. 13). The use of models for the cooling tower investigation is particularly appropriate, not only from the standpoint of cost, but also because atmospheric conditions can be controlled in the model tests.

It is desirable in model testing to have both geometric similarity and dynamic similarity. Geometric similarity between the model and the prototype means that a reasonable scale factor is chosen to shrink the size of the prototype. Sometimes the deviation in the surface roughness may lead to difficulty in interpreting experimental results in fluid flow tests. In that case it is wise to make the testing model as large as possible. The cooling tower test models used were made as large as practical considering the test facilities. However, surface roughness was not considered as being of major importance. Dynamic similarity is often required in fluid mechanics model testing. It is not important in this experiment. Usually this similarity only means that the Reynolds number used in the model testing must be the same as that for the prototype. The local Reynolds number is an indication of the relation between the kinetic force and the viscous force of the fluid at a certain location. Therefore a local geometric characteristic length must be chosen. Since our geometric scale factor reduces the lengths, the other terms such as μ , ρ , and V of the Reynolds number must be enlarged. Actually the Reynolds number is most important when the study involves wall friction and boundry layer of streamlined shapes. For a bluff body, however, the case is somehow different. Experiments had been made in an effort to correlate the relation between drag force to breadth-to-length ratio and the drag force to the Reynolds number (8, p. 198). When the Reynolds number is higher than a specific value, the drag force is almost stable and independent of any further increase of Reynolds number. This can be explained by the fact that the flow separates at the corners of a bluff body, and the shearing force at the surface of the wall is small compared to the over all drag force. The entire drag force is caused by the strong pressure difference between the front side and the back side of a bluff body. When a model of a bluff body is made in a geometrically similar shape, the drag force condition will be nearly the same as that of the prototype. The cooling tower models, being bluff bodies, should be influenced little by change in Reynolds number providing it is sufficiently high (8, p. 198). These are shown in Figures 2 and 3.

New York University has done a series of tests on the fall out of the plume emitted from a power plant chimney. Data obtained from model testing were compared with the field records and show remarkable agreement (14, p. 789-795). Since the saturated air emitted from a cooling tower is similar to the plume from a chimney in many respects, the plume path equations determined experimentally in that experiment will be taken as a guide in this experiment. In the chimney equations it was found that the total rise of the emitted atmosphere was caused by two factors: one is the emitted air velocity, and the other is the thermal buoyancy. Stated in mathematical terms, they are:

Velocity rise:

$$Z_v = \frac{z_v}{D} = z\ max\ (1-0.8\ \frac{z\ max}{D} / \frac{x}{D})\ , \quad x > 2\ z\ max \qquad (3)$$

$$\frac{z\ max}{D} = \frac{4.25}{1 + 0.43\ \frac{V_w}{V_s}} \quad \frac{V_s}{V_w} \sqrt{\frac{\rho_s}{\rho_w}} \qquad (4)$$

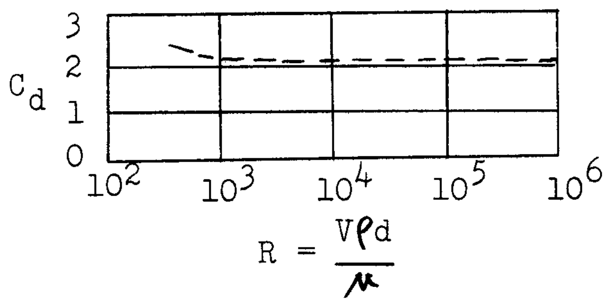

Figure 2. Drag Coefficient C_d Versus Reynolds Number.

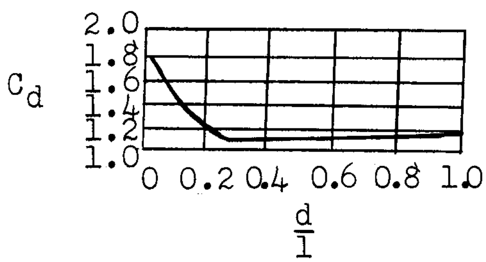

Figure 3. Drag Coefficient C_d Versus Breadth-length d/l.

Buoyancy rise:

$$Z_b = \frac{z_b}{D} = 5.01 \frac{1}{V_s^2/gD} \frac{V_s}{V_w} (1 - \frac{\rho_s}{\rho_w}) Z \qquad (5)$$

The total rise of the air above the top of the chimney at downwind distance X equals Z_v and Z_b (14, p. 791). For geometric similarity the following ratios must be the same (14, p. 792):

$$(X/D)_M = (X/D)_P \qquad (6)$$

$$(V_s/V_w)_M = (V_s/V_w)_P \qquad (7)$$

$$(V_s/GD)_M = (V_s/GD)_P \qquad (8)$$

$$(\rho_s/\rho_w)_M = (\rho_s/\rho_w)_P \qquad (9)$$

The buoyancy rise although appreciable in a chimney is a rather small factor in a cooling tower, and therefore it was neglected in this testing for simplicity. This leaves us a freedom in the choice of crosswind velocity and the model tower exit air velocity. Only the V_s/V_w and ρ_s/ρ_w terms need to be the same as the prototype in this experiment. Also notice that there is no requirement of the same Reynolds number since it does not appear in these equations. In these tests, different V_s/V_w values were chosen and the recirculation was determined for the model towers with the exit air saturated.

The space flow pattern adjoining the air intake of water cooling towers should be considered. Will geometrically similar air intakes guarantee similar space flow patterns? Dalla Velle investigated space flow patterns adjoining flanged and unflanged circular and rectangular exhaust openings (17, p. 77). The air intake of a cooling tower functions similarly to the inlet opening of an exhaust opening. It seems justifiable to apply Dalla Velle's empirical equations to the model cooling tower air intakes. In his equation he explains that in order to induce any velocity V_a in the space in front of a flanged or unflanged and along the hood axis, the relation is that the flow rate equals to the velocity times the contour area. This can be written as below

$$Q = V_a(10X^2 + A_f) \qquad (10)$$

Therefore the contour area A_c expressed as a function of distance X measured outward along the opening of the inlet of exhaust hood in feet was $(10X^2 + A_f)$, where A_f becomes insignificant as the X extended to a large distance. Dalla Velle's findings can also be put in terms of the ratio of the indicated contour area to the face area.

$$\frac{A\ \text{contour}}{A\ \text{face}} \quad \text{(see Table 1)} \qquad (11)$$

These are applicable to all shapes, (i.e. round, square, rectangular up to an aspect ratio of 3:1). In the case of rectangular openings the distance is expressed in diameters of the equivalent round openings having the same area. Since the model is made to be geometrically similar, the space flow pattern adjoining the air intake will be similar.

Table 1.

RELATION BETWEEN AXIAL AND VELOCITIES OF HOODS

Distance measures along hood axis, in fraction of hood face diameter	Area of contour intercepting this point expressed as a ratio with face area: i.e. $\frac{A\ contour}{A\ face}$
1/4	1.8----------------2.2
1/2	3.5----------------4.0
3/4	6.0----------------7.0
1	10.0---------------13.0

Based on the above analysis, and on the thesis (12) the recirculation factor is assumed to be a function of the following form:

$$R = F\left[(V_S/V_W), (P_S/P_W), (H/B), (\theta)\right] \qquad (12)$$

Since the air emitted from the model in this experiment is pure saturated, the experimental data are only valid when the exit air condition of the prototype tower is a stream of pure saturated air. Any deviation from the fixed exit air condition in this experiment, ought to be investigated by further testing.

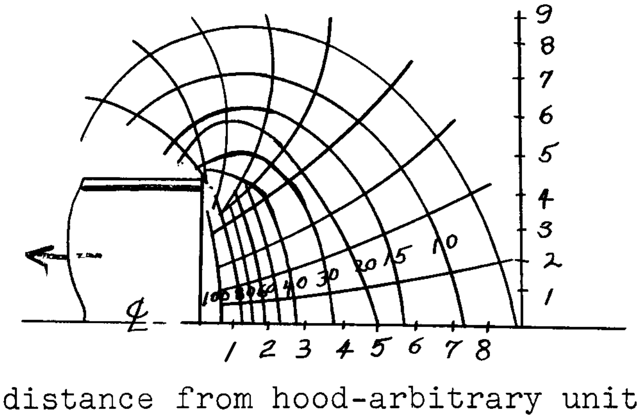

Figure 4. Dalla Velle's Experimentally Determined Velocity in Front of a Circular Opening.

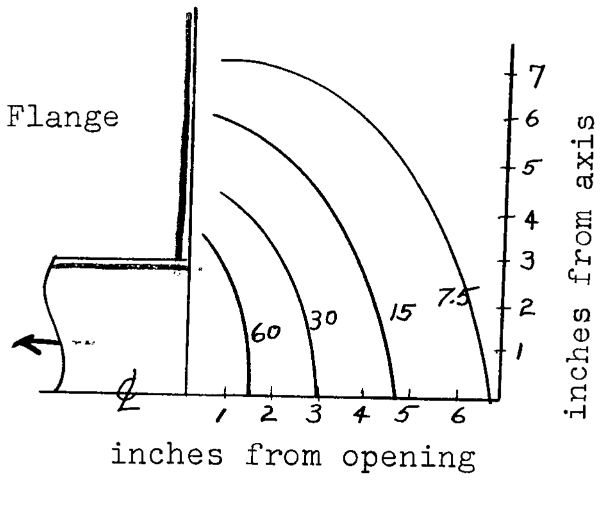

Figure 5. Velocity Contours in Front of a Flanged Circular Opening According to Dalla Velle.

EXPERIMENTAL APPARATUS

The whole apparatus is composed of four parts:

(1) The structure of the cooling tower model.

(2) The air distribution and humidifying system.

(3) The air flow control and measuring system.

(4) The low speed wind tunnel.

Model Tower

The basic model tower is a replica of a tower 35' x 35' x 60' high. As shown in Figure 7, it is made of 1/8 inch thick plywood, and the model has a scale of one hundred to one, resulting in a base size of $4\frac{1}{4}$" x $4\frac{1}{4}$" and a height of $7\frac{1}{2}$" with a round fan stack on the top. A four inch hole is provided in the bottom for the air pipes. The stack floor is removable so that the tower height can be changed by putting in additional stages. The fan stack is 1 1/8 inch in height and 2 inches in diameter, and is made of stainless steel sheet metal. The ratio of the height of the tower to its width is (H/B). Its influence in the recirculation will be checked in this experiment.

The inside volume of the model is divided into two separate compartments. The lower and the upper compartments are separated by a floor made of plywood, the joints are well sealed, and the surface of the floor is coated with a layer of special sealing compound. All these processes are done for the prevention of any interflow of air or moisture between these two compartments. The lower compartment is the fresh air intake region and the upper compartment is the saturated air exit path. Two opposite side walls of the lower compartment have air intakes. Each intake has seven 45 degree upward louvers. The louvers are removable to facilitate inspection and work in the lower compartment.

At the center part of the separating floor, there is a two inch diameter hole through which the exit air pipe protrudes. This pipe is well insulated on the inside with a layer of 1/8 inch thick corkboard insulation, and a layer of thin aluminum foil. This combination of insulation will cut down both the conduction and the radiation.

Air Heating and Humidifying System

The tower air enteres the louvers much in the same manner as in the prototype but insteac of leaving through the fan stack directly, it is drawn downward through the annulus formed by the concentric two inch and four inch pipes. It is heated and humidified before being forced back through the two inch pipe to the model upper compartment.

The apparatus used to heat, humidify, and pump the air consist of a steam heater, a steam spray humidifier, and a blower along with the necessary housings and ducts. The air blower is a Buffalo Forge No. 2EH type, which is driven by a Peerless Electric Company Series No. 519909 two phase 3500 rpm 1 1/8 hp electric motor. The blower is rated at 15 CFM free delivery and four inches water at shut off. It is housed inside an air tight housing. The housing is made of $\frac{1}{4}$ inch thick plywood and is reinforced inside by a wooden frame. In order to be air tight all the edges and the corners of the housing were sealed by two layers of plastic tape and then painted with grey enamel.

The air heater is a fin tube type, crossflow and steam heated. After the air has been heated it enters the humidifying zone. A steam spray is located at the outlet of the air heater so that the air will be humidified in this region. Before the air is sent back into the upper compartment of the model tower it must pass the humidifying box which is a $6\frac{1}{2}$" x 9" x 12" wooden cabinet having baffles arranged to give the air a zig-zag path for the purpose of increasing contact time with the steam. The saturation condition was checked by dry and wet bulb mercury thermometers fixed at the end of the humidifier. The air temperature and humidity were controlled by two manual steam valves. One was used to control the amount of steam sent into the steam spray, the other controlled the steam to the heater.

Air Flow Control and Measurement

Orifices were used in the control and measurement of the air flow. These were installed with flange taps in the pipe leading to the blower housing. The ratio V_S/V_W of the inlet air flow to the wind was used as the independent variable in the investigation of recirculation.

Temperature Measurement

Copper-constantan thermocouples with cloth wicks were placed in the low speed tunnel upstream and downstream from the tower model and also in the upper and lower compartments of the model. Five junctions in the lower compartment of the model were connected to five junctions upstream from the model in a manner to form a thermopile measuring the difference in temperature.

The thermoelectric potential of copper-constantan is approximately 0.02 MV/F for the range of 60 F to 120 F (1, p. 305), therefore the thermopile will generate an EMF of 0.1 MV/F. This was measured by a Leeds Northrup No. 8662 potentiometer. This permits us to readily measure a temperature difference of 0.1 F.

Figure 9 is a photograph of the tower model. The tower can be rotated in order that the influence on recirculation of tower orientation to the wind can be investigated.

The low speed wind tunnel used in this experiment was previously used in a qualitative study of the same subject (12). The velocity profile across the test section is fairly uniform (12, p. 22). The wind speed is readily varied by shutters at the tunnel exit (12, p. 14). The tower model was mounted slightly above the floor of the wind tunnel with the pipes extending to the auxiliary apparatus below. All the other equipment was placed outside and underneath the wind tunnel so that no obstacle interference other than the tower itself was inside the wind tunnel. For a better understanding of the apparatus, please see Figures 6 to 10 which are the photographs and the drawings of the apparatus.

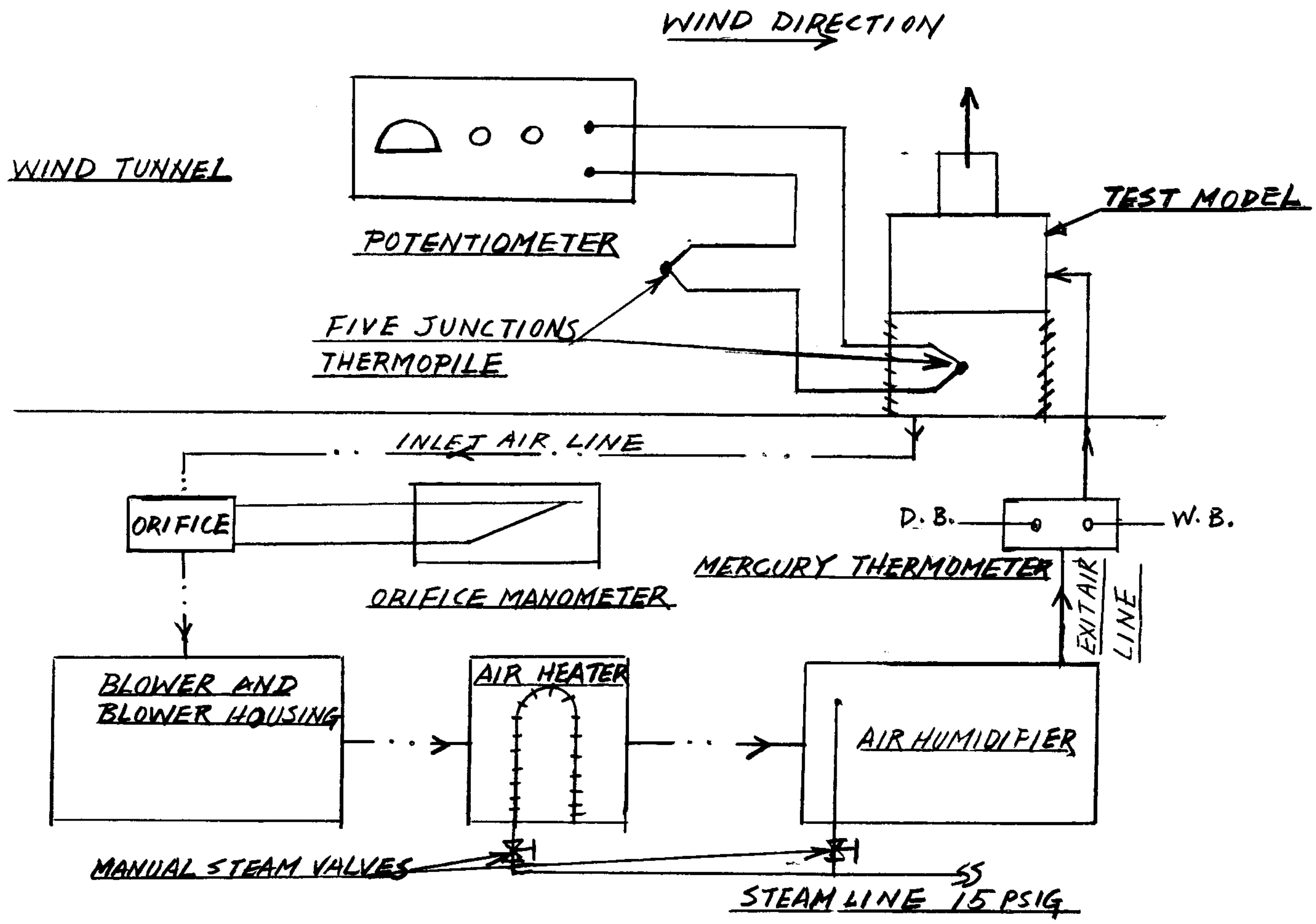

Figure 6. Schematic Flow Diagram of the Experiment

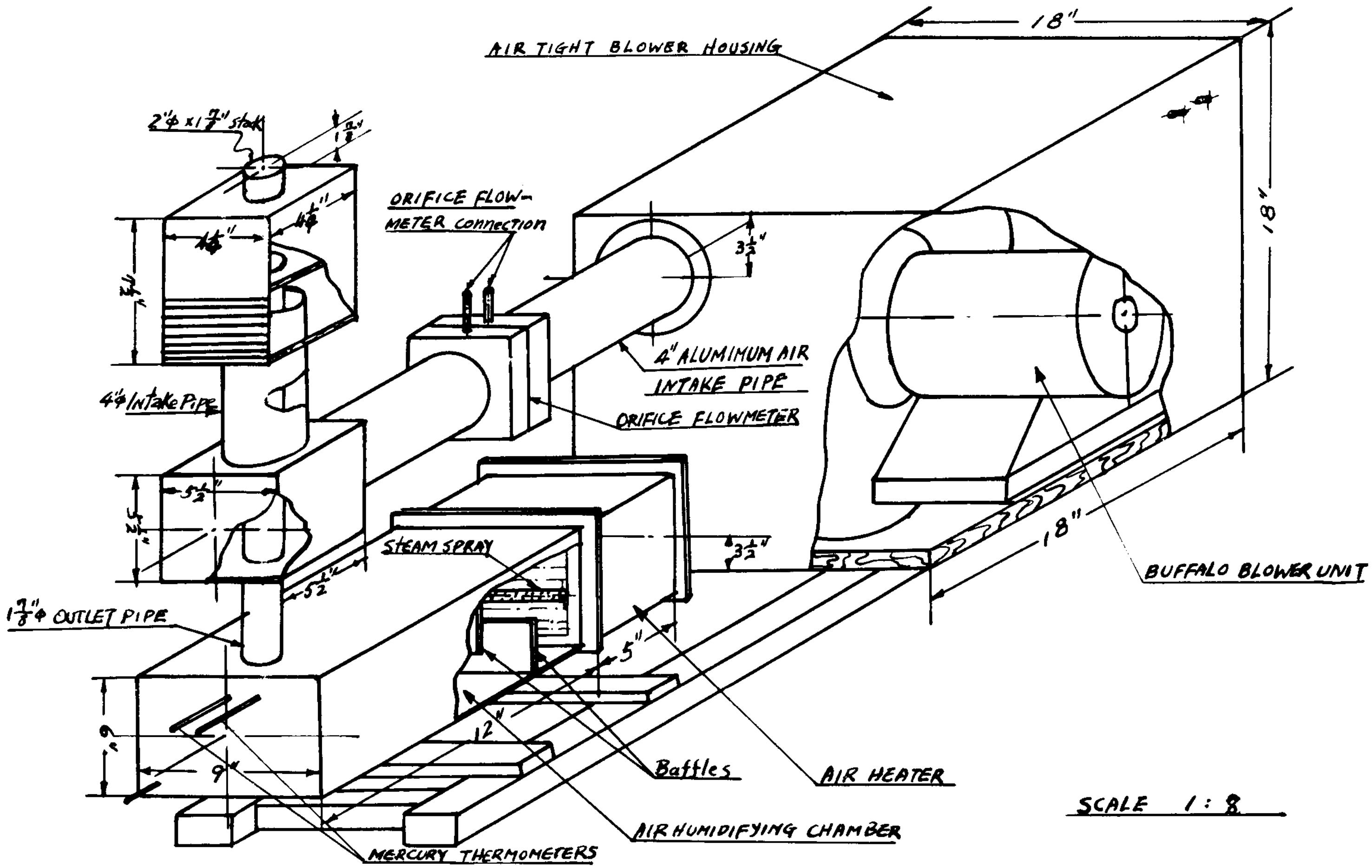

Fig 7 Diagram of the assembled apparatus

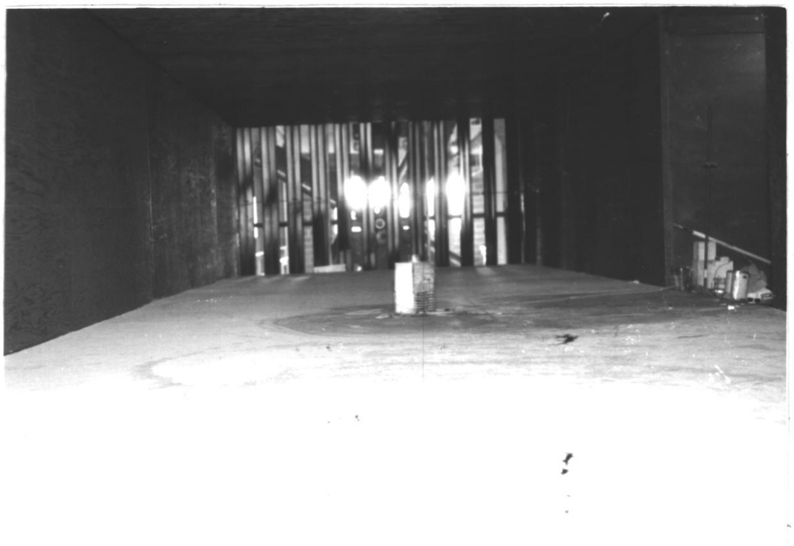

Figure 8. Low Speed Wind Tunnel and the Cooling Tower Model.

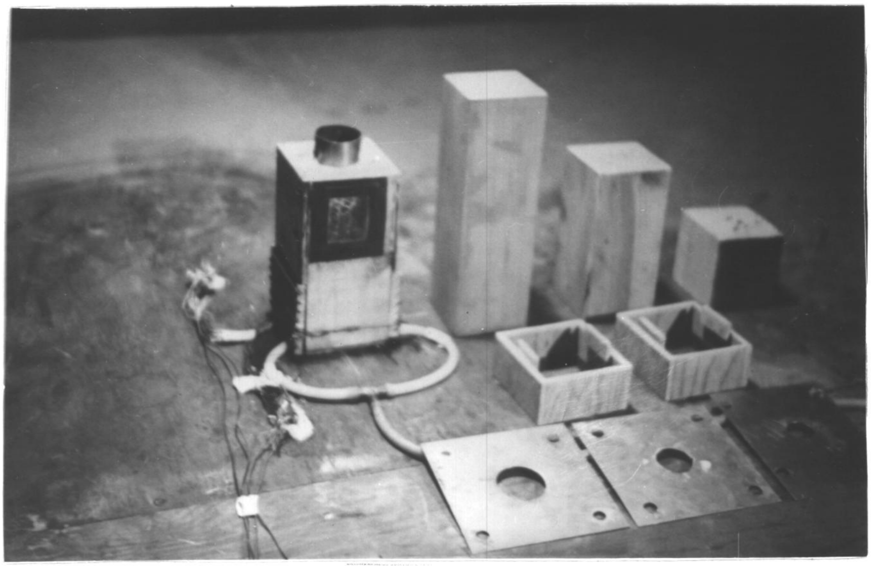

Figure 9. The Interchangeable Parts of the Tower Model.

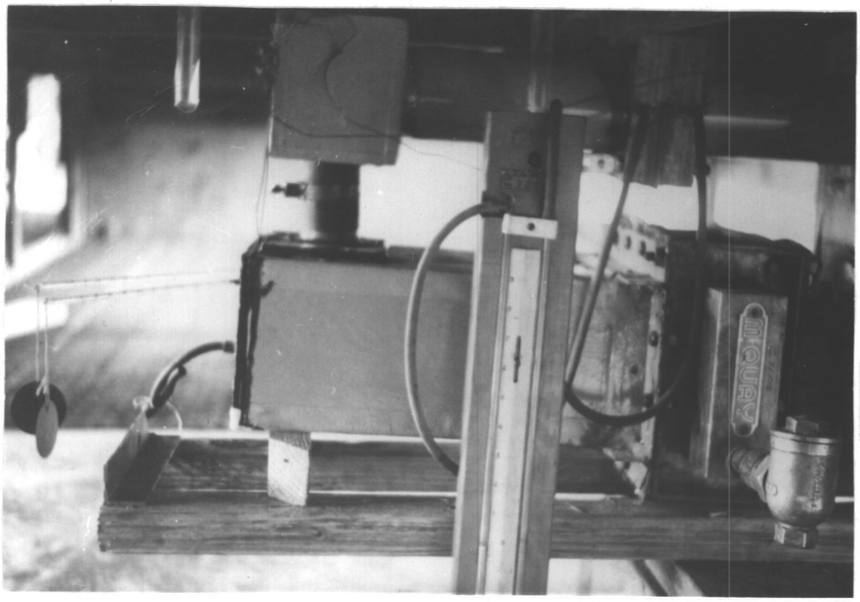

Figure 10. The Arrangement of Air Heating and Humidifying Apparatus.

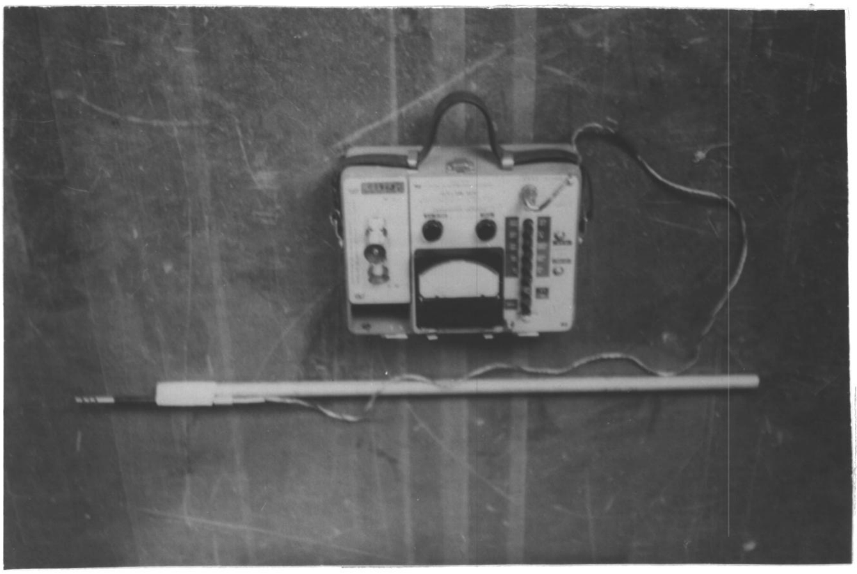

Figure 11. The Anemometer Used in the Determination of the Air Speed in Wind Tunnel.

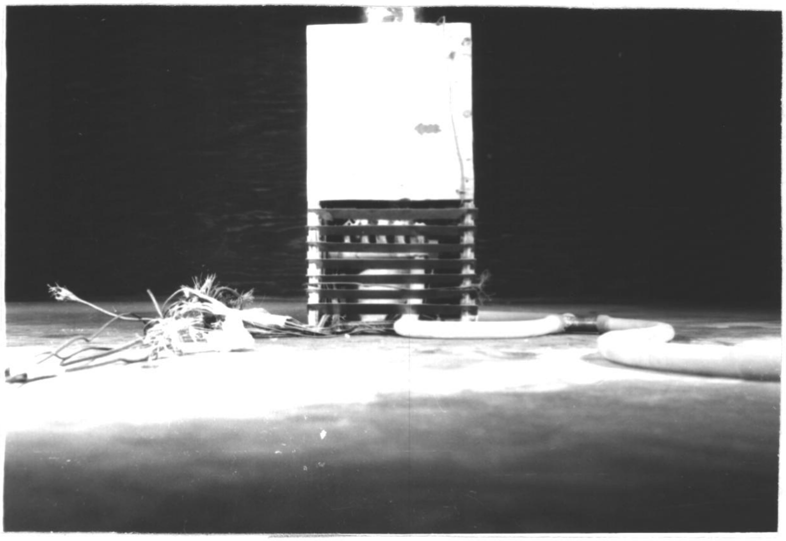

Figure 12. View of the Thermopiles and the Water Distributors Housed Inside the Tower Model.

TESTING PROCEDURE

The testing procedure can be classified into five steps:

I. Preparation for testing.

1. Check the general condition of the apparatus, the thermopiles and the wicks.
2. Fill the gravity water flask and the water pans with clear water (distilled water preferred), and check that the water flows freely to all the wicks of the thermocouples.
3. Start the blower and the wind tunnel fan motors, and check the tower air path for any leaks.
4. After the tunnel and the blower have run five minutes or more record the mercury thermometer readings located at the air outlet of the air humidifier.
5. Check the wiring connections and balance the potentiometer.
6. Measure and adjust the tunnel wind speed to the testing range with a calibrated anemometer (speeds of 5, 10, and 13 MPH were chosen).

II. Adjustment of exit air condition

1. Start the air heater and the air humidifier. Turn on the air heater steam and the steam

spray, watching the reading of the wet and dry bulb mercury thermometers. The exit air condition in this experiment was set at the 120°F saturated air. During test period the exit air condition was checked at 10 minute intervals, making sure that the saturated air temperature did not vary by more than 4°F from 120°F.

III. Data taking process.

1. Take the inlet air wet bulb temperature reading under zero recirculation condition, achieved by connecting a rubber hose to the tower fan stack and discharging well down wind. Since there is no possibility of recirculation in such an arrangement the EMF reading of the thermopiles must be that having to do with zero recirculation.
2. Remove the above hose and read the EMF of the thermopile. Record the tower air flow rate, the tunnel MPH and all temperatures.
3. Repeat steps 1 and 2 before each test made. The following variables were studied:
   (1) The crosswind speed to exit air speed ratio V_S/V_W.

(2) The tower height to its width ratio H/B.

(3) The tower orientation.

IV. Shut down process.

Close the steam valves and drain all the water pans, etc. Leave the blower and the tunnel fan running for 10 more minutes in order to purge the equipment. Make a final safety check.

V. Calculation.

Since the thermopile junctions were spaced around the lower compartment, it was assumed that the wet bulb indicated was the true value of the inlet air, and therefore any deviation from the upstream junction indication was considered to be a valid basis for calculation of recirculation. Figure 13 and the formulae below should be self explanatory.

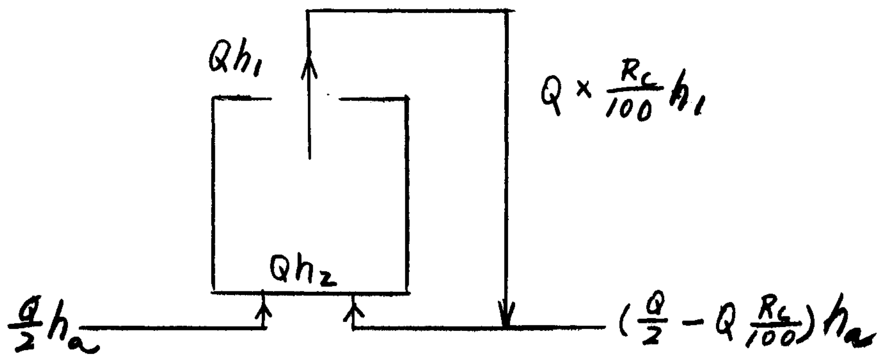

Figure 13. Schematic Diagram of Recirculation in Cooling Tower.

$$Q h_2 = \frac{Q}{2} h_a + \left(\frac{Q}{2} - Q\frac{R_c}{100}\right) h_a + Q \times \frac{R_c}{100} h_1 \qquad (13)$$

Solving for the Recirculation Factor R

$$R = \frac{h_2 - h_a}{h_1 - h_a} \times 100\,\% \qquad (14)$$

For Detailed Calculation see Apprendix

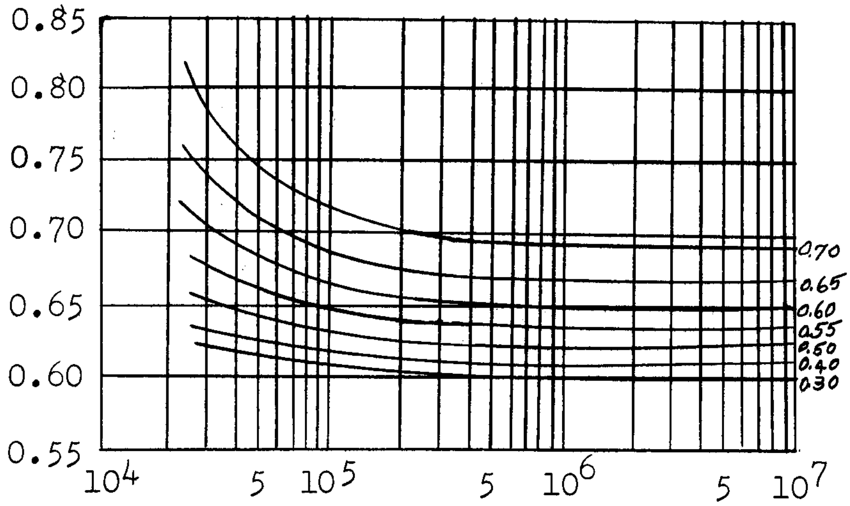

Figure 14. Flow Coefficient K Corrected for Velocity of Approach for Square-edged Orifices Employing Flange Taps

Adapted from "Fluid Meters, Their Theory and Application." A.S.M.E. 1937 4th ed. New York

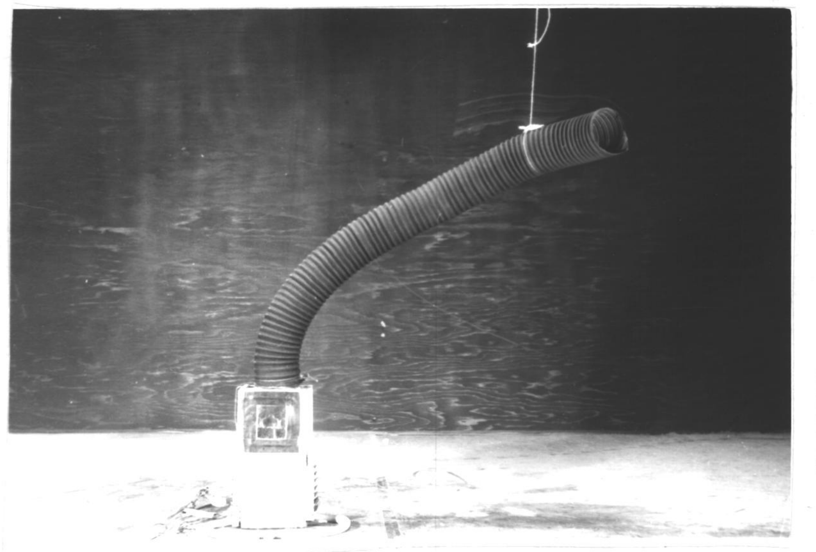

Figure 15. Arrangement of Apparatus in the Determination of Zero Recirculation EMF Reading.

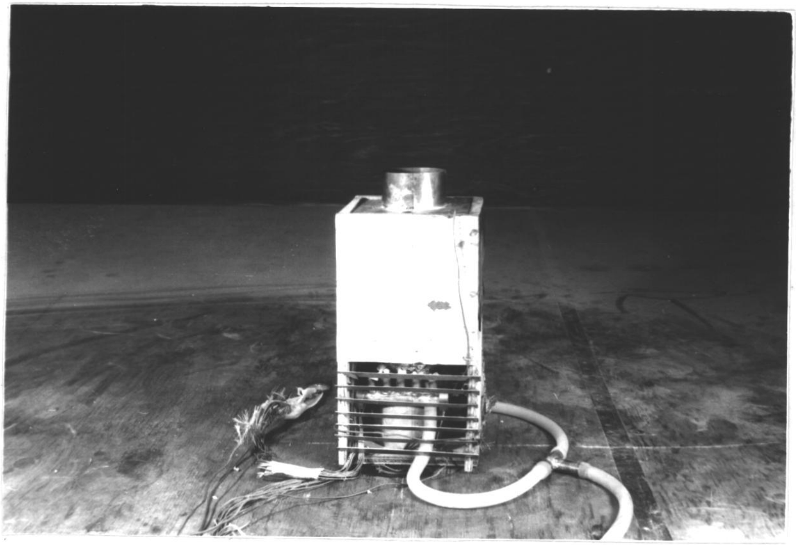

Figure 16. View of the Model Ready for Test.

Figure 17. Appearance of the Cooling Tower Model with an Additional Stage.

Figure 18. View of the Cooling Tower Having $\frac{H}{B} = 3.53$.

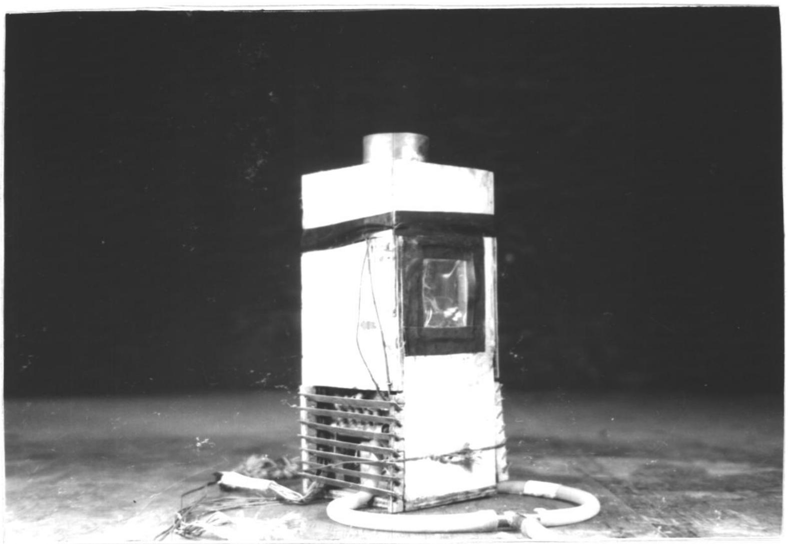

Figure 19. View of the Cooling Tower Model Standing in a 45 Degree Approaching Crosswind.

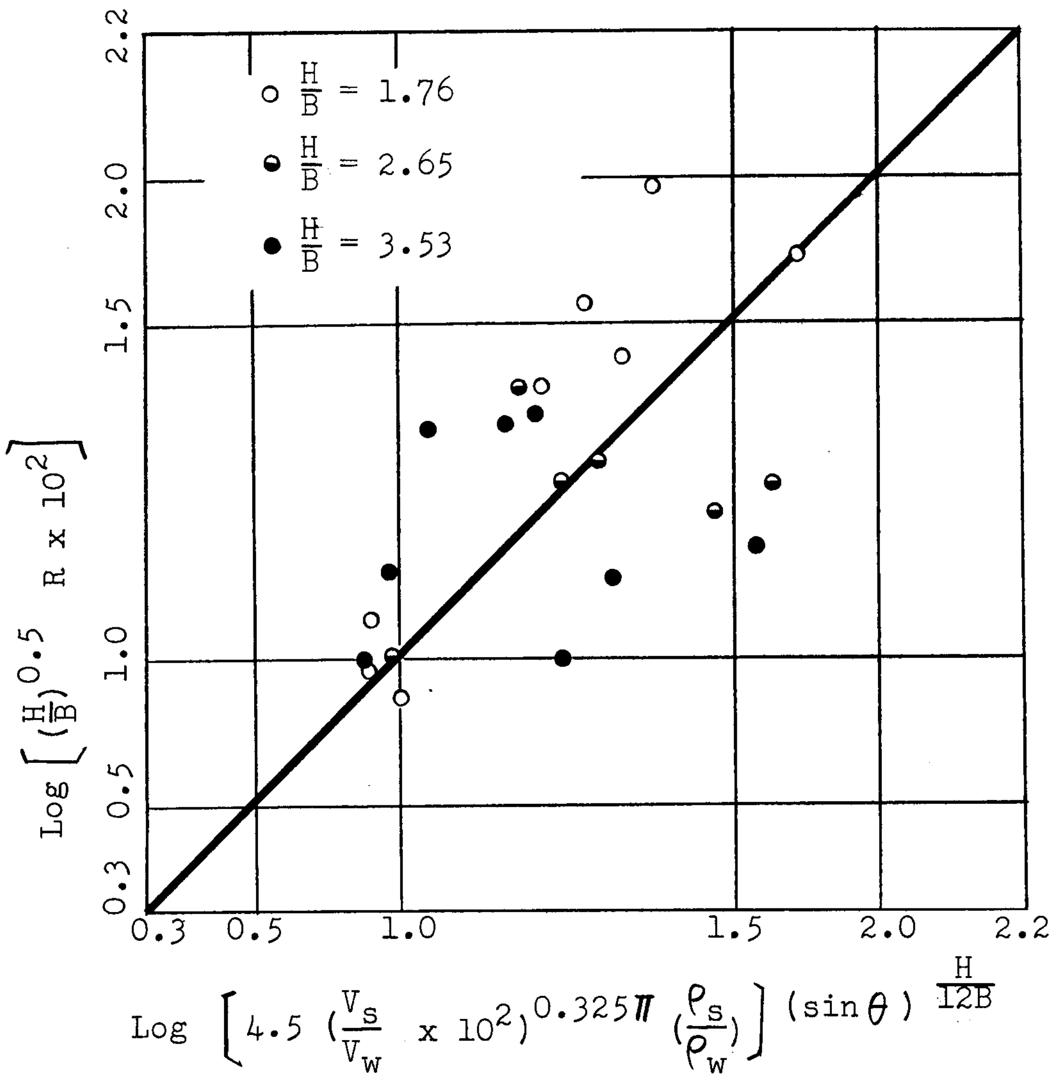

Figure 20. Plotted Experimental Results Using Eq. (15) ($\theta = 90°$)

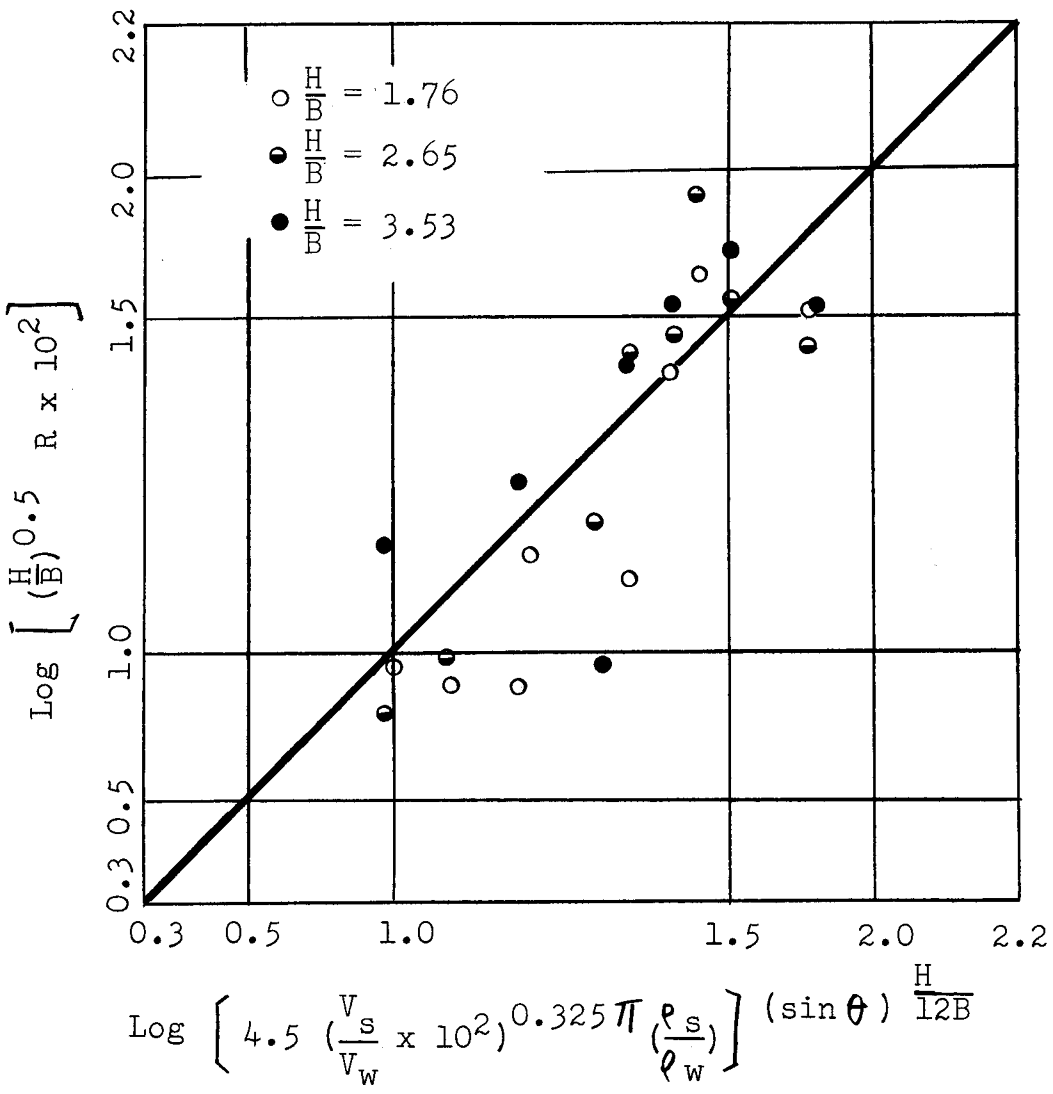

Figure 21. Plotted Experimental Results Using Eq. (15) (θ = 45°)

THE EXPERIMENTAL RESULT AND RECOMMENDATIONS

The exit air was kept at approximately 120°F saturated condition during the testing period, and since the tunnel air varied only slightly, the ratio ρ_s/ρ_w in this experiment had a value very nearly 0.91 in all tests. Using this value in equation (15), the results given in Tables 3A and 3B were obtained. Figures 20 and 21 show data points and the developed empirical equation below:

$$R \times 10^2 \left(\frac{H}{B}\right)^{\frac{1}{2}} = \left[4.5 \left(\frac{V_s}{V_w} \times 10^2\right)^{0.325\pi\left(\frac{\rho_s}{\rho_w}\right)}\right] (\sin\theta)^{\frac{H}{12B}}, \quad (15)$$

Equation (15) tells us the following:

1. The recirculation percentage is decreasing when the H/B ratio is increasing. This agrees with the technique report PFM-110 from the Cooling Tower Institute (3, p. 3). The length of tower in that report corresponds to the "B" variable in this experiment, which is grouped with the variable "H" in a diamentionless ratio of "H/B". The H/B ratio appears in the left side of the equation which shows that the recirculation is a square root relation to the H/B ratio, and in the right side of the equation it appears as an exponent of the $\sin\theta$ exponent relating the tower configuration with the tower orientation. Though this equation was not

derived analytically the experiment implied that the H/B ratio is an important factor in recirculation.

2. The recirculation is decreasing with the decreasing of ρ_S/ρ_W and the V_S/V_W ratio. The V_S/V_W ratio has two meanings. One is the load condition of the cooling tower and the other is the change of cross-wind speed. The V_S/V_W ratio in the equation (15) appears in a nonlinear relationship to the recirculation. It is known that the crosswind produces a wake on the leeward side of the tower structure, and the pressure difference in the region is essentially independent of the wind speed (8, p. 198). Therefore this V_S/V_W ratio is more meaningful when it is considered as a mass transportation factor in the recirculation phenomenon, a factor which determines the amount of downwash exit air in the leeward side of the tower structure.

3. The direction of an approaching wind has an influence on the recirculation, as is indicated in equation (15) as the exponent of the exponent $\sin\theta$. Since the value of a $\sin\theta$ function is between 0 to 1, and the ratio H/12B often has a value less than 1, the combined influence as shown in the exponent of the parenthesis on the right side of the equation can

be very small unless H/12B is large. A high H/B ratio is preferable in order to have a small percentage of recirculation in the cooling tower.

As a matter of fact, the maximum recirculation factors obtained in this experiment have values smaller than the field test values reported by Cooling Tower Institute in Bulletin PFM-110. In the field tests some recirculation percentages were higher than 8 per cent (3, p. 3). The highest measured in the model tests was 0.81 per cent. This is not a surprise when compared to the previous qualitative analysis (12, p. 31-36). This qualitative analysis made by Mr. Riggs shows photographs of arrangements of models quite similar to the author's. These have been interpreted as indicating that only a small amount of recirculation is probable. However it must be borne in mind that only a fraction of a degree rise in the tower entering wet bulb can seriously impair the performance when the tower is operating at a condition involving a close approach to the wet bulb temperature.

It must be understood that in the model experiments performed the exit air was a pure saturated air, and the wind speed and wind direction were stable for each test. The author claims no superiority of equation (15) for conditions other than those used here.

The author also believes that before the recirculation phenomenon can be thoroughly understood, more studies need to be made. The following experiments are suggested:

1. Variable exit air condition, including saturated air with entrained water particles.
2. Unstable wind speed and direction.
3. Various shapes of the cooling tower.
4. Academic study of the natural aerodynamics.

Table 2A.

CALCULATED RECIRCULATION FACTOR DATA

$\frac{H}{B}$	$\frac{V_s}{V_w}$	$\frac{\rho_s}{\rho_w}$	θ (Degree)	ΔT_w °F	R (%)
1.765	0.1265	0.91	90	0.33	0.244
1.765	0.0880	0.91	90	0.00	0.000
1.765	0.0632	0.91	90	0.42	0.311
1.765	0.0558	0.91	90	0.25	0.185
1.765	0.0490	0.91	90	0.011	0.081
1.765	0.0440	0.91	90	0.020	0.015
1.765	0.0340	0.91	90	0.070	0.052
1.765	0.0278	0.91	90	0.080	0.056
1.765	0.0214	0.91	90	0.120	0.084
2.65	0.1265	0.91	90	0.23	0.17
2.65	0.088	0.91	90	0.29	0.215
2.65	0.0632	0.91	90	0.67	0.50
2.65	0.0588	0.91	90	0.25	0.185
2.65	0.0490	0.91	90	0.21	0.162
2.65	0.0440	0.91	90	0.06	0.044
2.65	0.0340	0.91	90	0.12	0.09
2.65	0.0278	0.91	90	0.08	0.056
2.65	0.0214	0.91	90	0.05	0.037
3.53	0.1265	0.91	90	0.23	0.17
3.53	0.0880	0.91	90	0.36	0.267
3.53	0.0632	0.91	90	0.8	0.56
3.53	0.0558	0.91	90	0.25	0.185
3.53	0.0490	0.91	90	0.18	0.133
3.53	0.0440	0.91	90	0.06	0.044
3.53	0.0340	0.91	90	0.12	0.09
3.53	0.0278	0.91	90	0.0	0.00
3.53	0.0214	0.91	90	0.1	0.074

Table 2B.

CALCULATED RECIRCULATION FACTOR DATA

$\frac{H}{B}$	$\frac{V_s}{V_w}$	$\frac{\rho_s}{\rho_w}$	θ (Degree)	ΔT_w °F	R (%)
1.765	0.1265	0.91	45	0.5	0.371
1.765	0.0880	0.91	45	0.015	0.011
1.765	0.0632	0.91	45	0.95	0.705
1.765	0.0558	0.91	45	0.28	0.207
1.765	0.0490	0.91	45	0.36	0.267
1.765	0.0440	0.91	45	0.24	0.178
1.765	0.0340	0.91	45	0.03	0.022
1.765	0.0278	0.91	45	0.07	0.052
1.765	0.0214	0.91	45	0.12	0.084
2.65	0.1265	0.91	45	0.15	0.112
2.65	0.0880	0.91	45	0.09	0.066
2.65	0.0632	0.91	45	1.1	0.81
2.65	0.0558	0.91	45	0.16	0.119
2.65	0.0490	0.91	45	0.16	0.119
2.65	0.0440	0.91	45	0.25	0.185
2.65	0.0340	0.91	45	0.02	0.015
2.65	0.0278	0.91	45	0.1	0.014
2.65	0.0214	0.91	45	0.08	0.056
3.53	0.1265	0.91	45	0.1	0.074
3.53	0.0880	0.91	45	0.0	0.0
3.53	0.0632	0.91	45	0.9	0.664
3.53	0.0558	0.91	45	0.09	0.067
3.53	0.0490	0.91	45	0.15	0.112
3.53	0.0440	0.91	45	0.16	0.112
3.53	0.0340	0.91	45	0.15	0.011
3.53	0.0278	0.91	45	0.10	0.074
3.53	0.0214	0.91	45	0.08	0.056

Table 3A.

DATA INTERPRETED BY EQ. (15) IN LOGARITHM VALUE

$\frac{H}{B}$	$\frac{V}{V} \times 10^2$		$R \times 10^2$	Log A*	Log B#
1.765	12.65	90	24.4	1.5108	1.7294
1.765	8.80	90	0	0.1234	1.5694
1.765	6.32	90	31.1	1.6162	1.4264
1.765	5.58	90	18.5	1.3906	1.3664
1.765	4.90	90	8.1	1.0319	1.3184
1.765	4.40	90	1.5	0.2995	1.2504
1.765	3.40	90	5.2	0.8394	1.1474
1.765	2.78	90	5.6	0.8716	1.0584
1.765	2.14	90	8.4	1.0477	0.9414
2.65	12.65	90	17	1.4420	1.7294
2.65	8.80	90	21.5	1.5440	1.7294
2.65	6.32	90	50.0	1.9106	1.7294
2.65	5.58	90	18.5	1.4788	1.7294
2.65	4.90	90	16.2	1.4211	1.7294
2.65	4.40	90	44	0.8551	1.7294
2.65	3.40	90	9.0	1.1658	1.7294
2.65	2.78	90	5.6	0.9598	1.7294
2.65	2.14	90	3.7	0.7798	1.7294
3.53	12.65	90	1.7	1.5043	1.7294
3.53	8.80	90	26.7	1.7004	1.7294
3.53	6.32	90	56	2.0221	1.7294
3.53	5.58	90	18.5	1.5411	1.7294
3.53	4.90	90	13.3	1.3978	1.7294
3.53	4.40	90	4.4	0.9174	1.7294
3.53	3.40	90	9.0	1.2281	1.7294
3.53	2.78	90	0.0	0	1.7294
3.53	2.14	90	7.4	1.1431	1.7294

*A = $R \times 10^2 \, (H/B)^{\frac{1}{2}}$ (H/12B)

#B = $\left[(V_s/V_w \times 10^2)^{\,0.325\pi} \, (\rho_s/\rho_w) \right]^{\sin\theta}$

Table 3B.

DATA INTERPRETED BY EQ. (15) IN LOGARITHM VALUE

$\frac{H}{B}$	$\frac{V}{V} \times 10^2$	θ	$R \times 10^2$	Log A*	Log B#
1.765	12.65	45	37.1	1.6928	1.64
1.765	8.80	45	1.1	------	1.51
1.765	6.32	45	7.05	1.9710	1.36
1.765	5.58	45	20.7	1.4394	1.29
1.765	4.90	45	26.7	1.5499	1.24
1.765	4.40	45	17.8	1.3738	1.18
1.765	3.40	45	2.2	------	1.12
1.765	2.78	45	5.2	0.8394	1.0
1.765	2.14	45	8.4	1.0477	0.89
2.65	12.65	45	11.2	1.2608	1.60
2.65	8.80	45	6.6	1.0311	1.44
2.65	6.32	45	8.1	2.1201	1.35
2.65	5.58	45	11.4	1.2685	1.26
2.65	4.90	45	12.0	1.2808	1.21
2.65	4.40	45	18.5	1.4788	1.16
2.65	3.40	45	1.5	------	1.07
2.65	2.78	45	7.4	1.0808	0.98
2.65	2.14	45	5.6	0.9598	0.87
3.53	12.65	45	7.4	1.1431	1.55
3.53	8.80	45	0.0	------	1.40
3.53	6.32	45	6.64	1.0961	1.28
3.53	5.58	45	6.7	1.0000	1.22
3.53	4.90	45	11.2	1.3231	1.18
3.53	4.40	45	11.2	1.3231	1.13
3.53	3.40	45	1.1	1.3141	1.03
3.53	2.78	45	7.4	1.1431	0.955
3.53	2.14	45	5.6	1.0221	0.85

$*A = R \times 10^2\ (H/B)^{\frac{1}{2}}$

$\#B = \left[(V_s/V_w \times 10^2)^{0.325}\ (\rho_s/\rho_w)\right]^{\sin\theta\ (H/12B)}$

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APPENDIX

SYMBOLS USED

A_c	Contour area of an exhaust opening	
A_f	Face area of an exhaust opening	
a	Area of transfer surface per unit of tower volume	ft^2/ft^3
B	Tower width	inch
C	Constant in Eq. (2)	
C_d	Total drag force coefficient	
D	Diameter of fan stack	inch
G	Air mass flow	$\frac{\text{lb dry air}}{\text{hr ft}^2\text{ ground area}}$
H	Tower height	inch
h	Enthalpy	BTU/lb
K	Overall transfer coefficient	$\frac{\text{lb/hr ft lb water}}{\text{lb dry air}}$
L	Water mass flow	lb/hr ft^2 ground area
n	Constant in Eq. (2)	
Q	Air flow rate	ft^3/min
T	Temperature	°F
V	Volume of cooling tower	ft^3
V_a	In flow air velocity of exhaust opening	ft/min
V_s	Speed of the exit air	ft/min
V_w	Speed of the wind	ft/min
X	Distance outward exhaust opening	ft.
x	Distance downward of the cooling tower	ft.

Z_v	Exit air main stream height due to the eject velocity	ft.
Z_b	Exit air main stream height due to bouyancy force	ft.
θ	Angle of the approaching wind direction	degree
ρ_s	Density of exit air	mass/ft^3
ρ_w	Density of the ambient air	mass/ft^3
Z	Height of chimney	ft.

SAMPLE CALCULATION

A. Room air condition

Dry bulb temperature 68°F

Wet bulb temperature 58°F

B. Intake air quantity

Orifice manometer reading 0.4 in. H_2O

Orifice diameter ratio $D/D_1 = 0.438$

Velocity correction factor (from figure 14)

$K = 0.65$

$v = \sqrt{2gh} = \sqrt{64.4 \frac{0.4}{12}} = 1.47$ ft/sec

$Q = KAV = 0.65 \times 0.0167 \times 60 \times 1.47 = 0.96$ C.F.M.

@ $T_D = 68°F$, $T_W = 58°F$

C. Recirculation factor calculation

(1) $h_2 = h_a + R_c\,(h_1 - h_a)$

$$R = \frac{h_2 - h_a}{h_1 - h_a}$$

(2) The conditions of testing

$H/B = 1.67$

$V_s/V_w = 12.65 \times 10$

$= 90$ degree

(3) Measured EMF differences

EMF = 0.033 MV

Converted wet bulb temperature

difference

$T_{wb} = 0.33$ F

From Psychrometric chart find the enthalpy of atmosphere

$T_{1_{db}} = 120$ F $\qquad T_{1_{wb}} = 120$ F

$H_1 = 119.4$ BTU/LB.

$T_{a_{db}} = 68$ F $\qquad T_{a_{wb}} = 58$ F

$H_a = 24.5$ BTU/LB.

$T_{2_{db}} = 68$ F $\qquad T_{2_{wb}} = 58.33$ F

$H_2 = 24.73$ BTU/LB.

Then

$$R_c = \frac{H_2 - H_a}{H_1 - H_a}$$

$$= \frac{24.73 - 24.5}{119.4 - 24.5} = 0.224\ \%$$

D. Exit air speed calculation

Cross sectional area of the tower fan stack = $0.0218\ ft^2$

$$\frac{Q(120\ F)}{Q(\ 68\ F)} = \frac{16.51}{13.33} = 1.24$$

$$V_s = \frac{0.976\ CFM}{0.0218\ FT} \times 1.24 = 45\ ft/min.$$