PERFORMANCE TESTS AND COOLING EFFECT DISTRIBUTION OF THE V.P.I. FORCED DRAFT COOLING TOWER

by

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and

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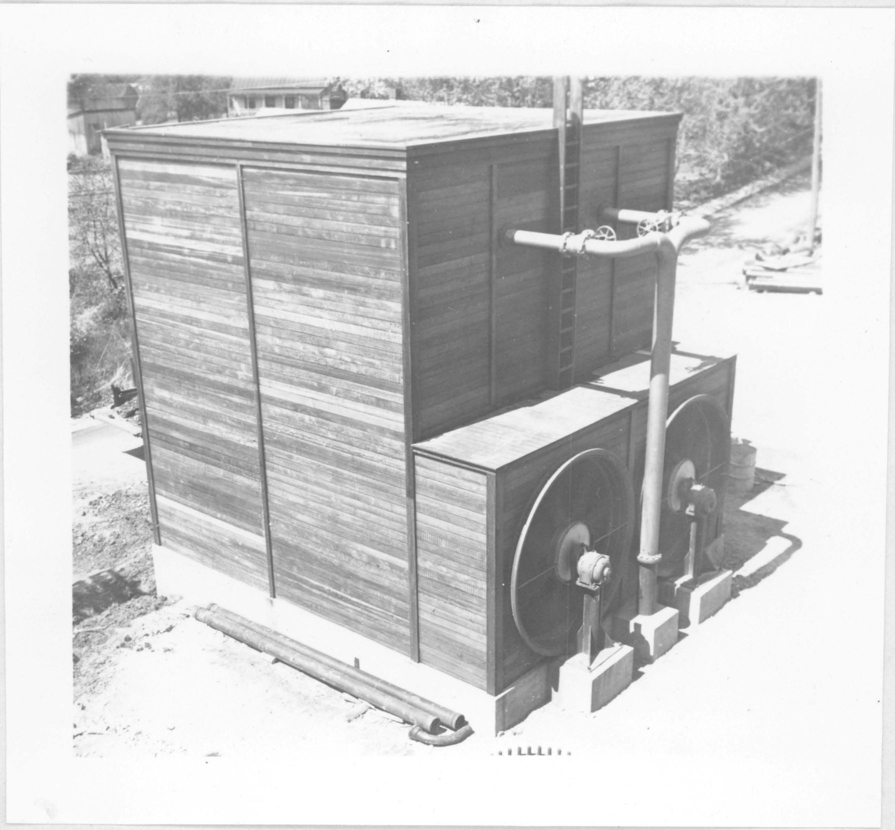

Fig. 1. The Virginia Polytechnic Institute
Forced Draft Cooling Tower

INTRODUCTION

Large quantities of water are being continuously circulated through the condensers of power plants and other heat transfer units of industry. In the early days of industry power plants and industrial establishments were situated near rivers and large streams in order to avail themselves of what was then thought to be an unlimited supply of cooling water. Recently, however, as inland streams and rivers have become overheated,as artesian supplies have become more difficult to obtain, and as other plants have been located where there is not an adequate supply of cooling water, industry has been rapidly turning to atmospheric cooling as a more suitable solution of the water cooling problem.

Condenser pressures are fixed by the temperature of the cooling water circulating through the unit. Lake and stream temperatures vary considerably with geographical location and with seasons of the year, but are much steadier than air temperatures. The limiting factor in atmospheric cooling is the wet-bulb temperature of the air. This is the minimum temperature to which the water can be cooled and is steadier than the air, or dry-bulb, temperature.

Several systems of atmospheric cooling are being used at the present time. Where space permits and where small quantities of water are to be cooled, cooling ponds are adequate. Spray ponds are resorted to as the volume of water necessary increases and the space available decreases. In very restricted areas atmospheric cooling towers of the natural and forced draft types are used. The forced draft tower takes up the least amount of ground space, requiring about one-fifth the space per gallon cooled as that required by a natural draft tower, and about one-tenth the space required for a spray pond of the same capacity.[1*]

The gradual growth and development of the evaporative water cooling process have been such that, in general, "rule-of-thumb" methods prevail in the design of atmospheric water cooling equipment, this being particularly true in the case of forced draft towers. Recent research in psychrometry (given impetus by the growth of Air Conditioning) has given to the engineering profession much scientific data that has been arrived at by rational analysis and verified by research, and these data have displaced many of the "rule-of-thumb" methods that have been so prevalent.

It is the purpose of this study to further the knowledge of the variables of forced draft cooling towers and to develop rational methods of design.

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* Numerical indices refer to correspondingly numbered references in the bibliography at the end of the thesis.

REVIEW OF LITERATURE

Mechanical draft towers are a comparatively recent development. Experiments were begun on them around 1910. At that time they were used in conjunction with natural draft towers, the forced or induced draft fans as then used being put in operation only about five to ten per cent of the time.[2] In 1915 they were still in the experimmntal stage. A rotary form of water cooler of the forced draft type was then being studied.[1]

The high pumping head required to raise the water to the top of the coolinh tower has been a great drawback in the widespread use of this type of equipment, as it has made the pumping rather expensive. Experience and study in the field had brought the pumping head down from 40 feet to 26 feet, which was the standard pumping head for a fotced draft tower, prior to the recent advent of the "Low-Head" tower, which has reduced it to 20 feet. In contrast to these heads the pumping head for a spray type forced draft tower is about 30 feet, and for natural draft towers it averages between 45 and 55 feet.[3]

The low-head tower has several refinements of design which have reduced its cost. The low pumping head has reduced the cost of operation below that of the standard tower. Because of the lower overall height and hence lessened wind resistance a simpler frame design is used, which cuts down the initial cost. A more efficient conversion of air velocity into low pressure has been utilized, and greater freedom from wetting of the fan blades has been worked into the design.[3] The tower tested by the authors, shown in Figs. 1-4, is of the "low head" type.

Atmospheric water cooling is defined as the heat transfer from water to air with the fluids in direct contact.[4] Dean[5] gives the following definitions of terms concerned with atmospheric cooling:

"1. The dry-bulb temperature is the temperature of the atmosphere as registered by an ordinary thermometer.

2. The dew-point is the temperature at which the moisture present is a saturated vapor. If the atmosphere is cooled below this temperature, dew will form.

3. The atmospheric vapor pressure is the absolute pressure of the water vapor in the atmosphere, ordinarily expressed in inches of mercury. It is the pressure corresponding to the dew point as taken from the steam tables.

4. Relative humidity is the percentage ratio of the density of water vapor actually in the atmosphere to the density of water vapor that would exist in the atmosphere in a saturated condition at the observed dry-bulb temperature.

5. The wet-bulb temperature is the temperature a thermometer would register if its bulb were kept moist and if it should be passed through the atmosphere at a velocity sufficient to provide for a rapid heat exchange. It is the temperature at which the rate of heat absorption by the bulb or the moisture upon it thru conduction, as a result of the difference between the dry-bulb atmospheric temperature and the temperature of the wet thermometer bulb, is just equal to the rate of heat dissipation by the moisture of the bulb through evaporation by reason of the difference between the vapor pressure of the moisture on the bulb and the atmospheric vapor pressure."

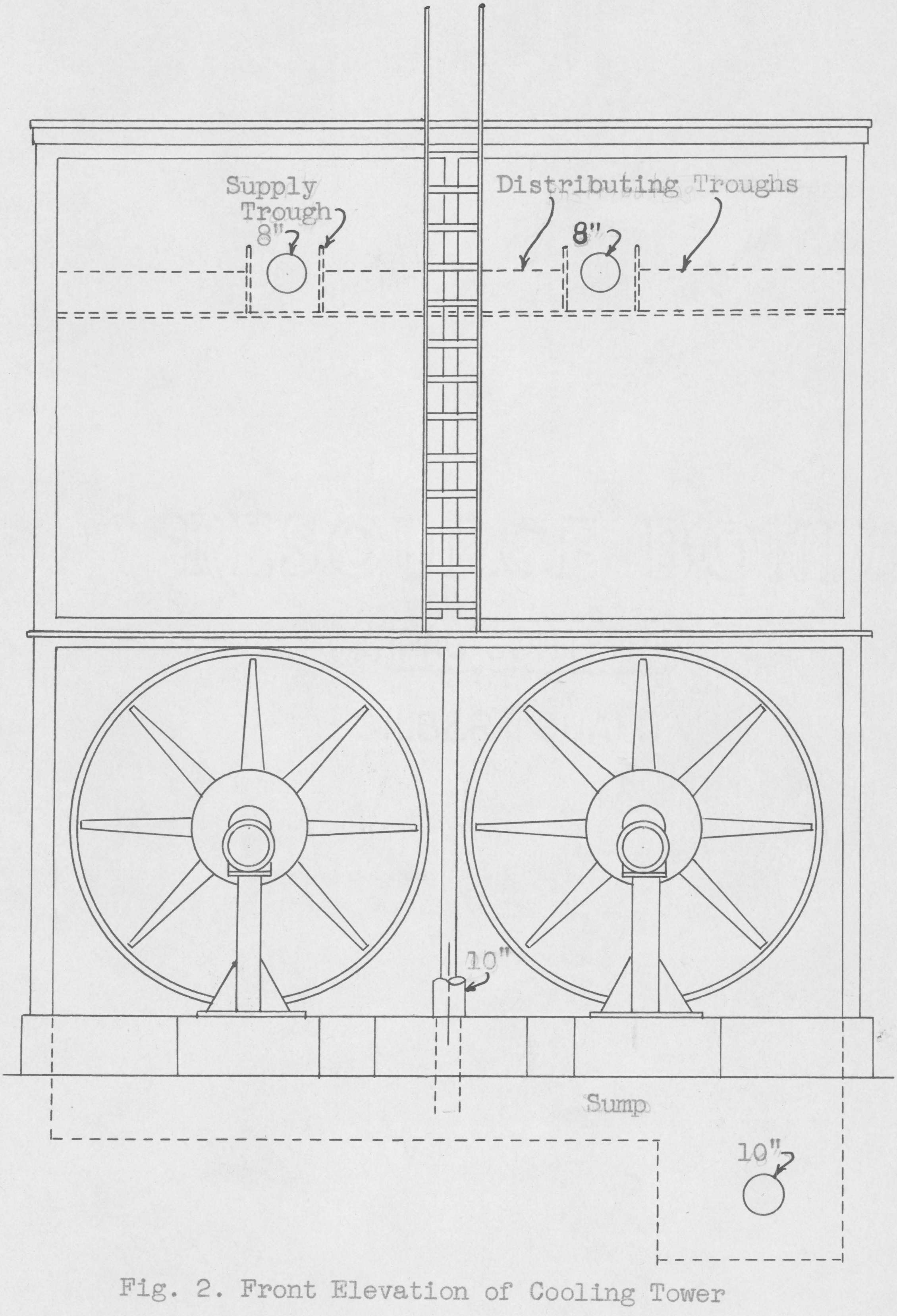


Fig. 2. Front Elevation of Cooling Tower

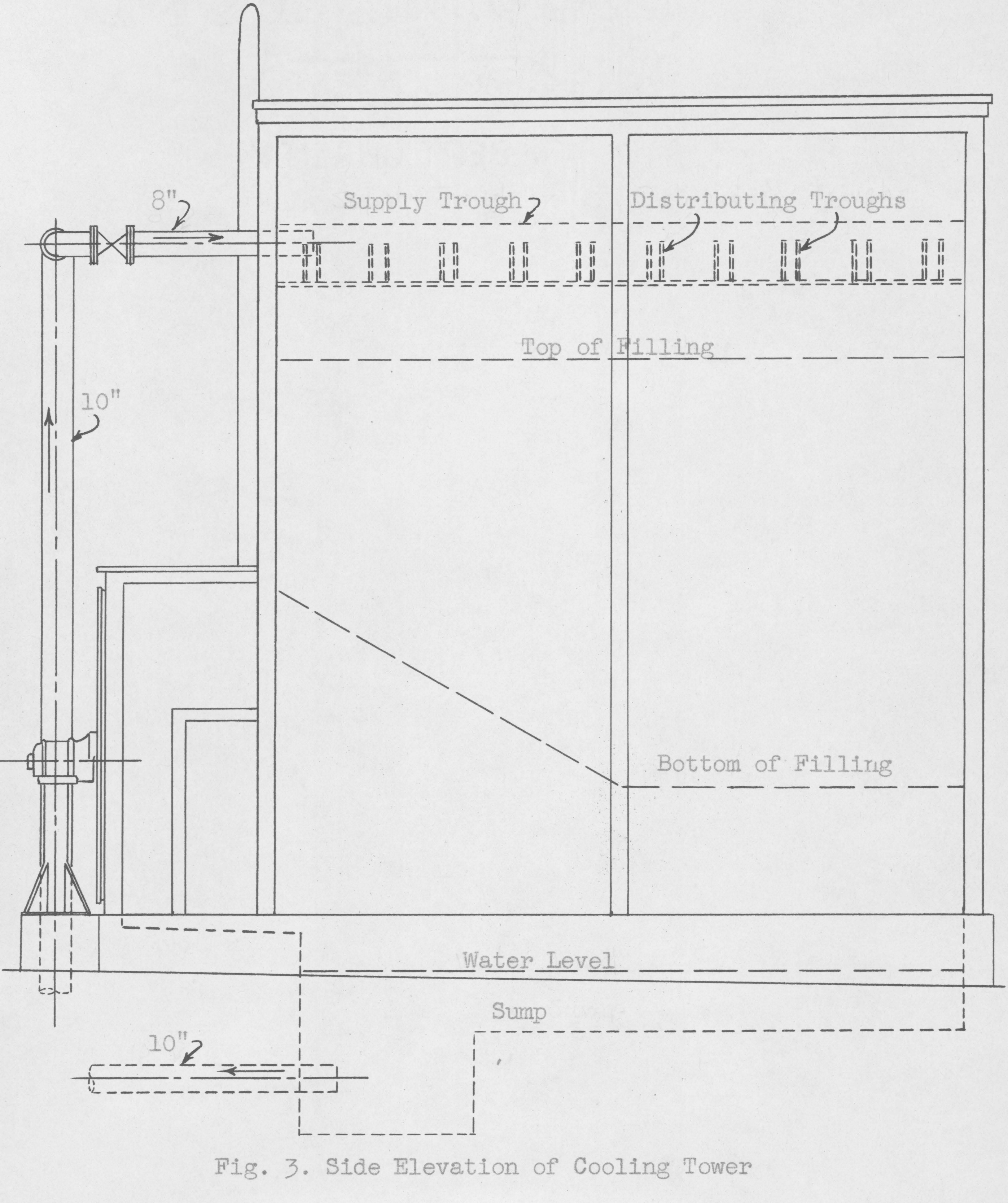


Fig. 3. Side Elevation of Cooling Tower

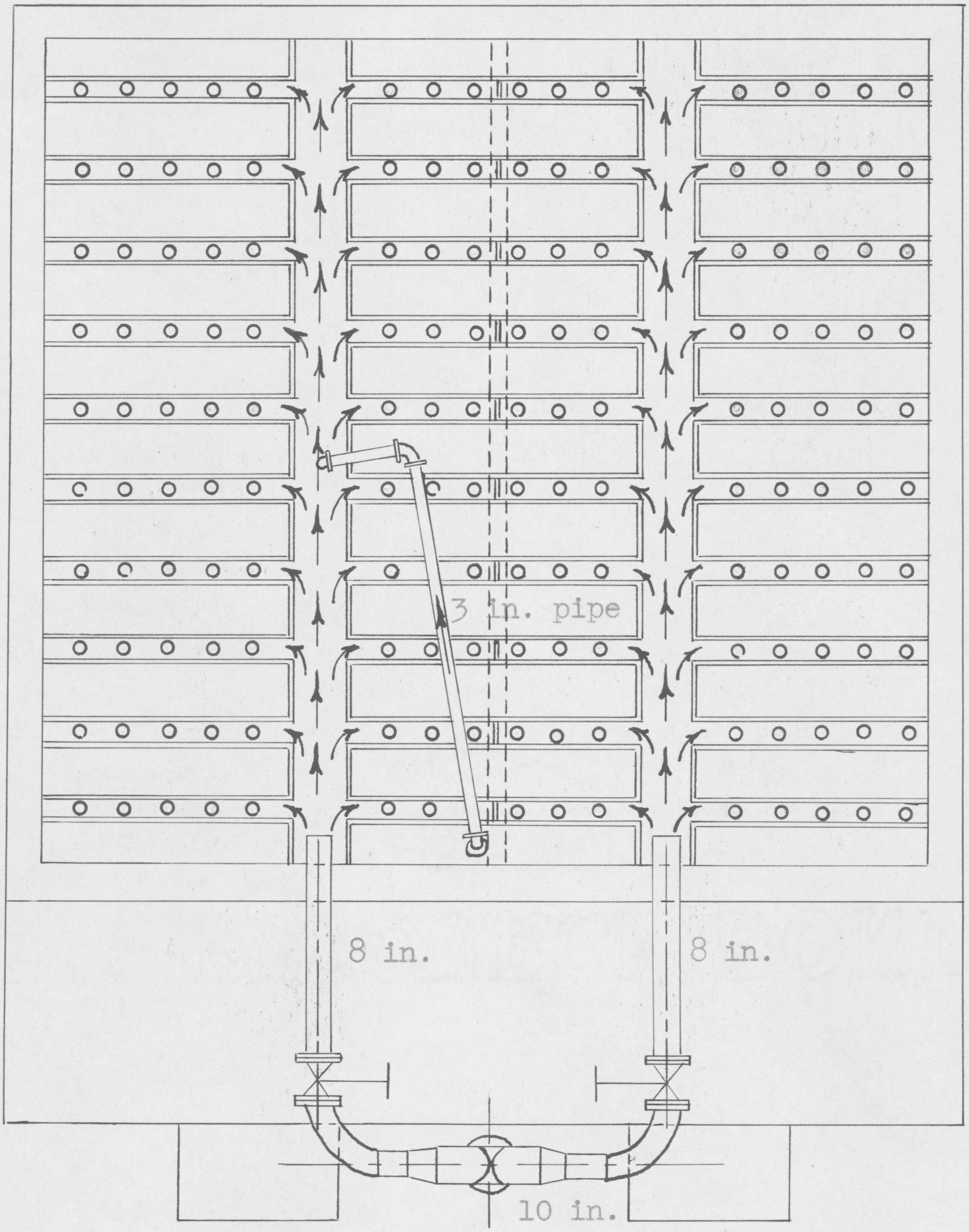


Fig. 3a. Plan View of Cooling Tower

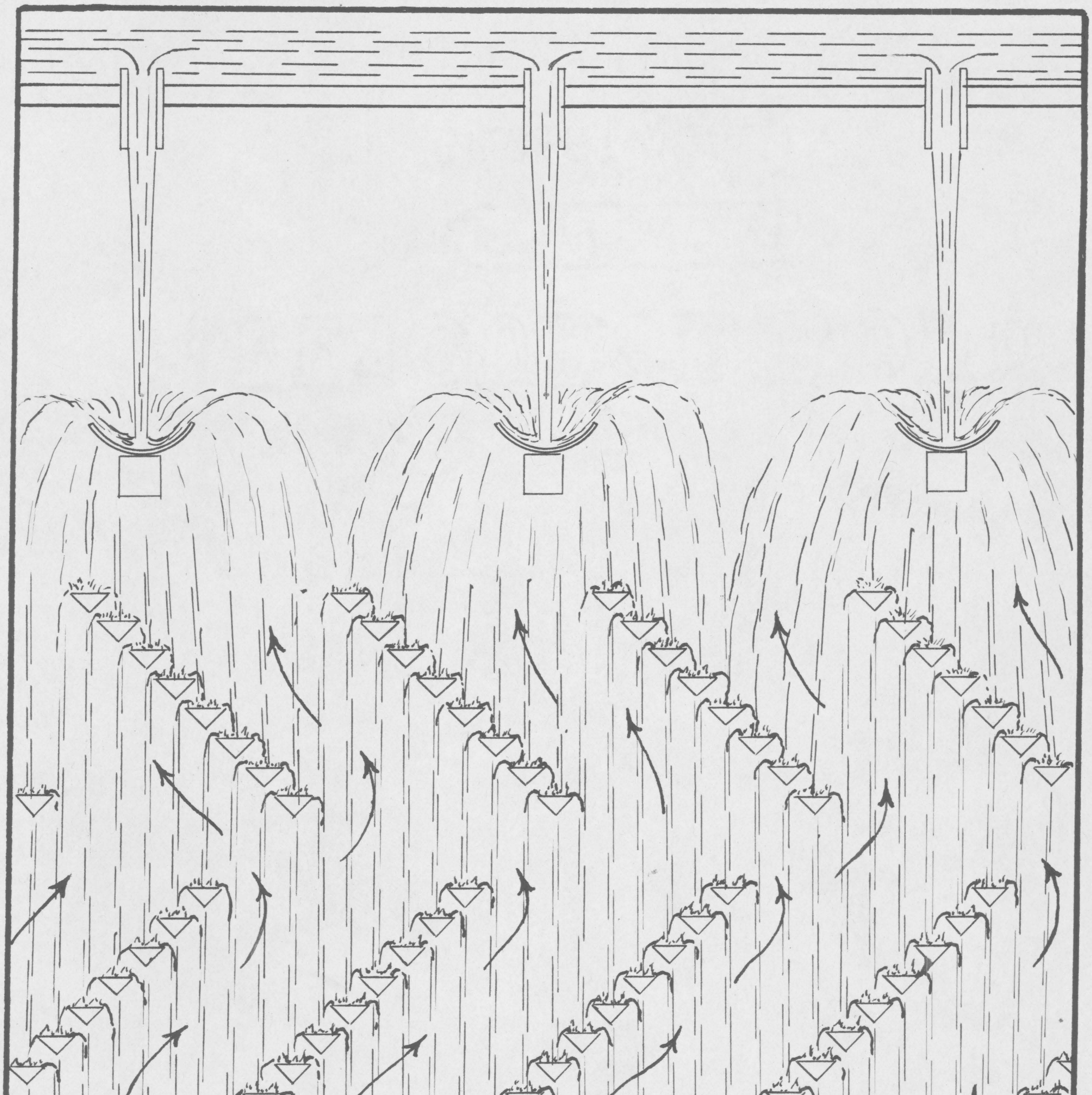


Fig. 4. Detail of Spray System and Tower Filling

Heat transfer from water to air can take place in the following ways: radiation, convection, conduction, and evaporation.Due to the small temperature differences in cooling towers very little of the heat transfer is accounted for by radiation, convection, or conduction. The evaporation of small amounts of the water circulated causes the major cooling effect and may produce as much as 80 to 90 per cent of the total cooling in summer.[4] Evaporation is a cooling process that requires heat potentials to carry it out. The heat of vaporization is absorbed from the water, and therefore cools it, the wet-bulb temperature of the atmosphere being the minimum temperature to which the water can be cooled by evaporation. At some temperature above that of the wet-bulb thermometer the water receives and discharges equal quantities of heat, thermal equilibrium being thus established and the temperature of the water leaving the cooling tower becoming constant.

There are six variables concerned with cooling towers.[4] They are, namely:

1. The wet-bulb temperature of the atmosphere.
2. The initial or entering water temperature.
3. The ratio of cooling surface area to weight of water cooled per unit time.
4. The ratio, by weight, of air supplied to water cooled.
5. The relative velocity of air to cooling surface per unit time.

6. The final or exit water temperature.

The first five variables cooperate to produce the final water temperature, Item 6. If Item 1 or 2 rise, Item 6 also rises. If Item 3, 4, or 5 rise, Item 6 is lowered. Items 3, 4, and 5 are controlled wholly by design in mechanical draft towers.

PURPOSE

The various designers of forced draft cooling towers have their own individual and peculiar methods of calculation and design. Little can be said by way of comparing these various methods, for each designer jealously guards what he considers his "trade secrets". These methods have evolved from many years of experience in the building of cooling towers, but still are almost entirely empirical. It is hoped and believed that intensive research by disinterested parties may provide the engineering profession with the scientific data needed for the rational design of this type of water cooling equipment, and it is for this reason that the present study was undertaken.

A practically unlimited field of research is thus entered into. It is felt that intensive research into any one particular phase would be of little value unless it were correlated with similar work on closely related phases, and that preliminary work should treat with as many of the various phases as is practicable. Hence the primary work should not be confined to any great extent with experimental research, but should rather concern itself with investigation of the general behavior and characteristics of a modern forced draft cooling tower and with study of the various instruments intended for measurements similar to those necessary for detailed testing. It is the purpose of the authors of this thesis to lay a foundation for a series of detailed and related tests extending over a considerable period of time, and to determine those methods of testing which are best suited to the problems involved.

PLAN OF INVESTIGATION

In keeping with the expressed purpose of this report, the major portion of the work is confined to the selection of apparatus and the taking of sufficient test data to indicate the suitability of the apparatus for the application in question. The same data serves to show the general reactions involved in the cooling of water by means of forced draft towers and indicates those phases upon which more detailed investigation should be concentrated.

The general plan of investigation is:

1. Study of all available material upon the constructional and operating features of the tower tested. This material, for the most part, consisted of data and specifications from the manufacturer.
2. Determination of the tower characteristics to be investigated and measured.
3. Selection of apparatus for these measurements.
4. Calibration of apparatus.
5. Experimental test of tower.

From the results obtained from this procedure the recomendations for future study and for test procedure were formulated.

SELECTION OF APPARATUS

The main load on the cooling tower is the cooling of the circulating water for the condenser of the partially-condensing turbine in the power plant. This water is piped to the tower through a standard ten-inch pipe which branches near the top of the tower into two eight-inch pipes, as shown in Fig. 1. Distribution of the water into the left-hand or right-hand unit is controlled by two gate valves installed in the eight-inch lines.

The total flow through the condenser was known to be approximately 1200 gallons per minute, but it was decided that the flow meters selected should be able to handle 1800 gallons per minute, which is the design capacity of the cooling tower. The plan to run detailed tests on one unit as well as an overall efficiency test on the entire tower made it necessary to measure the flow in each of the eight-inch lines rather than just that in the ten-inch line.

The arrangement of the piping system is such that the installation of a venturi meter or weir was not considered practical. The lack of a straight run of pipe of sufficient length preceding an available point of installation eliminated pitot tubes, orifices, and elbow flow meters from consideration. The instruments finally selected for measuring the flow were flow nozzles, installed on the discharge ends of the two eight-inch pipes. Diligent research failed to disclose any standard contour for a nozzle with exact known coefficients or any calibration data on nozzles of the size required. The form selected was the Compressed Air Society standard nozzle[6], as shown in Figs. 5 and 6. Equipment of capacity sufficient for calibration was not available, so the nozzles were installed with the thought in mind that the assumed coefficient of 0.985 was accurate to within one and one-half per cent,[6] and as such was within the expected limits of accuracy for the other apparatus.

The instruments selected to measure the head on the up-stream side of the nozzles were regular "U-tube" manometers, filled with mercury and having a range up to thirty inches. These manometers were installed with the zero of the scale on the same horizontal line as the center lines of the eight-inch pipes.

Water for the air and oil coolers on the back-pressure turbine in the power plant is supplied by a separate piping system to and from the tower, in order that the operation of this turbine will not be affected by any abnormalities occurring in the functioning of the cooling water system of the condensing turbine. The water from the separate and smaller system enters the tower through a three-inch pipe that discharges into the supply trough of the left-hand unit. This pipe is the only portion of the system that is readily available for the installation of a flow meter.

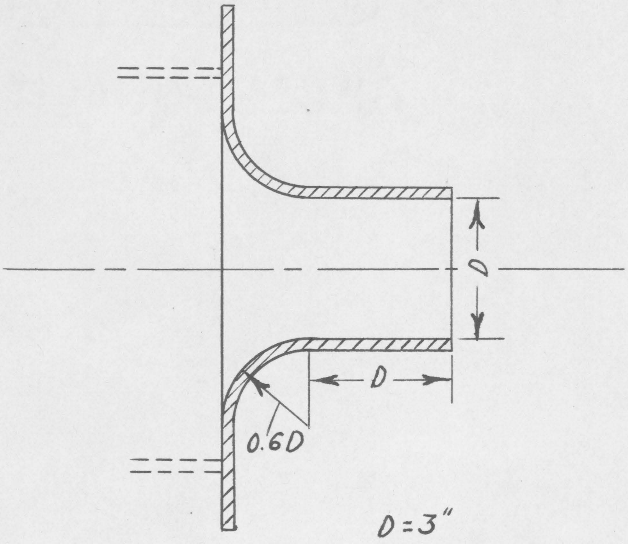


Fig. 5. Compressed Air Society Standard Nozzle

Fig. 6. Flow Nozzles Used on Eight-inch Pipes

When selecting an instrument to measure the flow in this system (known to be approximately 160 gallons per minute) the following factors were considered:

1. Increase in pumping head.
2. Expected accuracy.
3. Simplicity of operation.
4. Simplicity of installation.
5. Cost.
6. Calibration necessary.
7. Adaptability to structural characteristics of the tower.

Failure to meet one or more of these characteristics was responsible for eliminating pitot tubes, flow nozzles, orifices, venturi meters, and weirs from further consideration. An elbow flow meter was finally selected as it complied most satisfactorily with the desired characteristics.

The only increase in pumping head when this meter is installed is the small increment caused by the addition of a standard 90° elbow in the pipe line, and this increase is eliminated when an elbow already in the line may be used. The meter is as simple in operation as could be reasonably desired, for it is an indicating instrument requiring only one manometer reading. The only characteristics in which it does not greatly excell all other similar apparatus are the calibration necessary and the expected accuracy. All instruments considered rated about the same on these points. The two greatest advantages of the elbow flow meter are its very low cost and simplicity of construction, its high ratings here being easily explainable by the fact that it may be constructed of materials normally stocked in a power plant by a person having only average mechanical ability. It consists essentially of a standard 90° pipe elbow with pressure taps in the plane of symmetry of the elbow and located on the inner and outer curves midway of the circular length. A differential manometer connected across the pressure taps completes the apparatus. A view of the instrument constructed by the authors is shown in Fig. 7.

The principle of operation is the inertia effect of a fluid in motion, and is analogous to the centrifugal force developed by a rotating mass. A rational analysis of this action[7] bases its fundamental equations on Bernoulli's theorem and Newton's Second Law of Motion. The final derivation of these two statements yields the equation:

$$h = C_1 \frac{V^2}{2g} \qquad (1)$$

in which:

$$C_1 = \frac{(r_2 - r_1)^2}{(\log_e r_2/r_1)^2} \times \left[\frac{1}{r_1^2} - \frac{1}{r_2^2}\right]$$

(See Fig. 8)

Fig. 7. Three-inch Elbow Flow Meter and Manometer

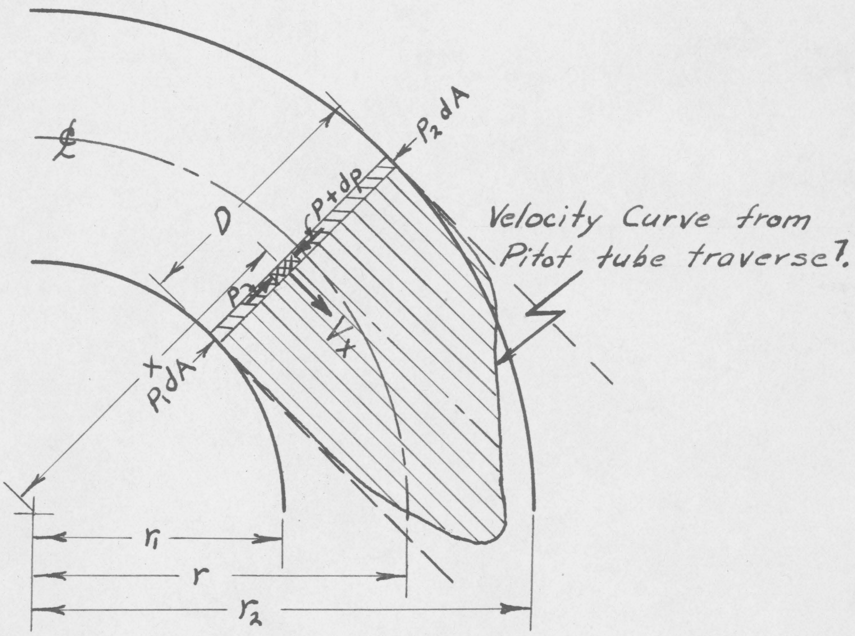


Fig. 8. Diagram of Flow in a Pipe Bend

An approximate analysis of the same action yields:

$$h = C_2 \frac{V^2}{2g} \qquad (2)$$

in which:

$$C_2 = \frac{2D}{r},$$

"r" being the radius of curvature of the centerline of the bend. Combining the constants in each case we arrive at:

$$h = K\, V^2 \qquad (3)$$

in which:

$$K = \frac{C_1}{2g} \text{ or } \frac{C_2}{2g}.$$

Using the real, as opposed to the nominal, dimensions of a standard three-inch screwed pipe elbow, the calculated values of $C_1/2g$ and $C_2/2g$ are 0.0521 and 0.0332 respectively. The curves of the resulting equations, $h = 0.0521\, V^2$ (A) and $h = 0.0332\, V^2$ (B), are shown in Fig. 9, plotted on log-log coordinates.

Calibration data on the instrument constructed uphold the value of two for the exponent of "V" in the general equation $h = K\, V^n$. Curve (C) in Fig. 9 represents the h vs. V curve as determined from the calibration test, the velocity used being that in the straight pipe. Curve (D) in Fig. 9 represents the same data corrected for the decreased velocity through the elbow, the cross section of a screwed elbow being somewhat greater than the cross section of straight pipe of the same nominal size. Discrepancies between the actual and

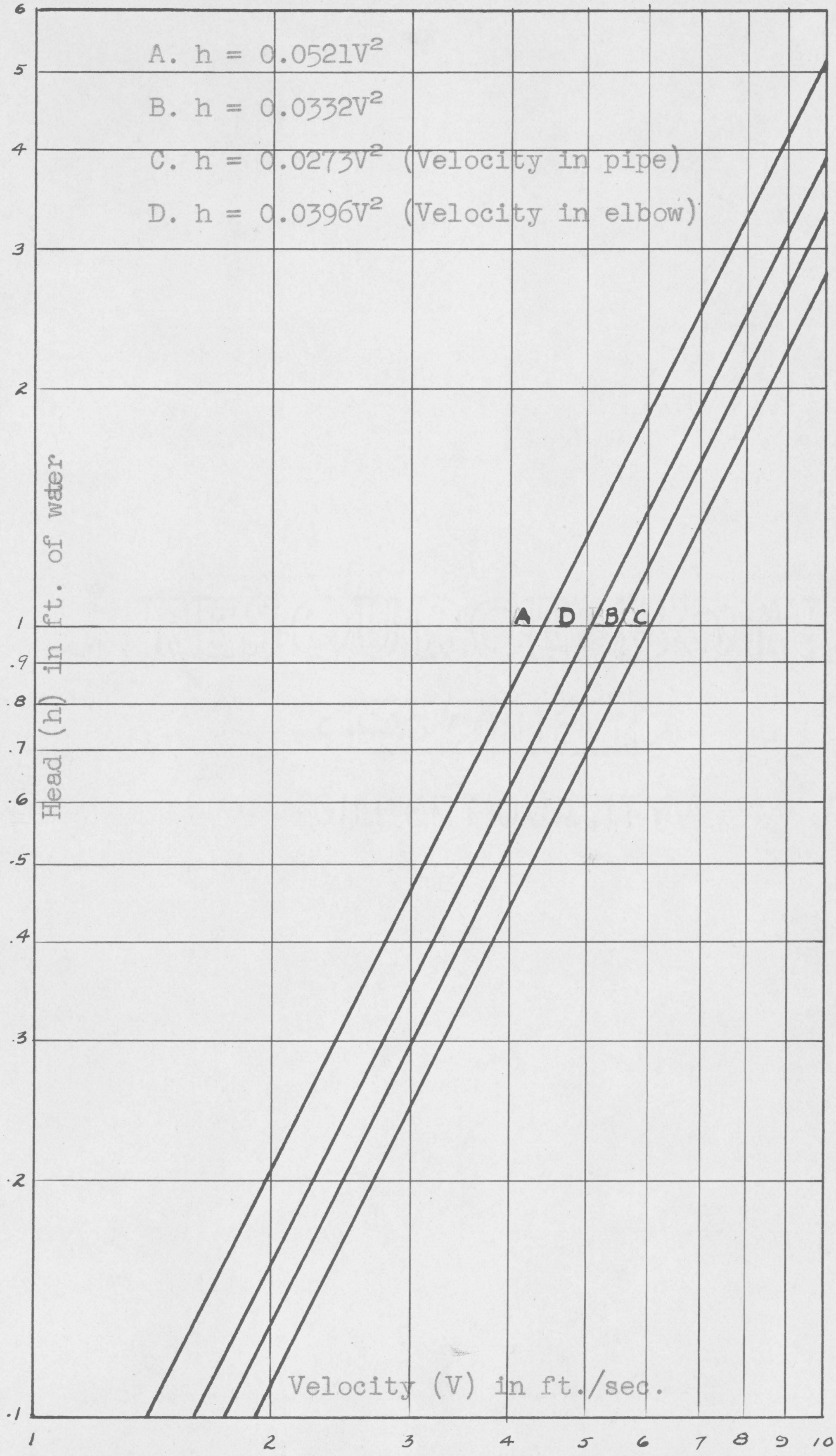


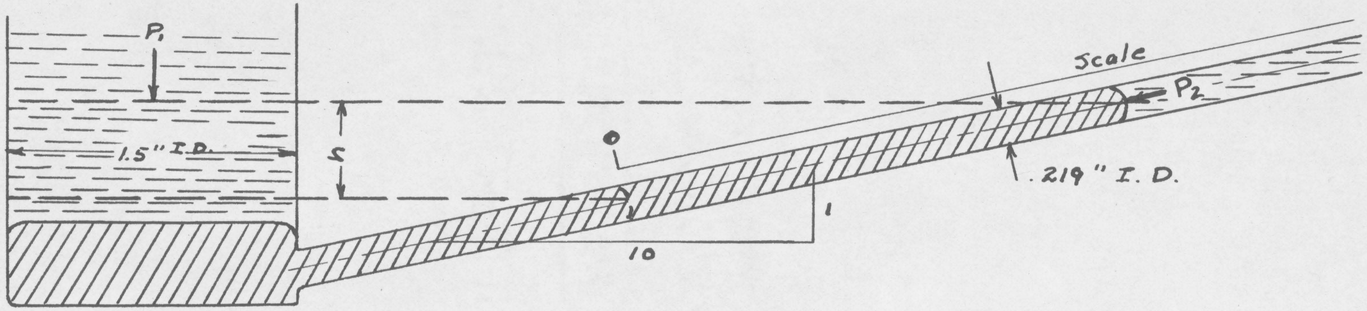
Fig. 9. Head vs. Velocity Curves for Elbow Flow Meter

the theoretical curves are explainable by the fact that the assumptions made in the derivation[7] are not factual in the actual application of the principle.

The probable range necessary for the manometer to be used was calculated from the known rate of flow (about 160 gallons per minute) and from theoretical equation (1). The resultant "h" was found to be 2.000 feet of water, its equivalent being 1.765 inches of mercury. In view of the relatively small head probable it was decided to construct an inclined-tube type of manometer, using mercury as the operating medium. Fig. 10 is a diagram of the instrument constructed. Derivation of the equation for equivalent head yields the equation:

h in ft. of water = 1.273 x scale reading.

For the calibration test the entire apparatus, consisting of elbow flow meter and manometer, was set up in the Mechanical Engineering Laboratory in an arrangement as nearly as possible the same as the arrangement to be used during the test runs on the tower. Eighty inches of straight three-inch pipe preceded the elbow and 72 inches of the same size pipe followed it, both sections being in the same horizontal plane. A standard three-inch elbow at the end of the 72-inch length discharged vertically downward into a weighing tank. The results of the test are shown by curves (C) and (D), Fig. 9.



Note: Scale is graduated to read "h" in inches

Fig. 10. Diagram of Manometer for Elbow Flow Meter

In cooling tower operation a large amount of the cooling of the water is brought about by the evaporation of a small amount of the water circulated. For the tower tested this amounts to slightly over one per cent of the water circulated, including spray loss out of the top of the tower. To take care of these losses water must be continuously added, as the water is recirculated. In normal operation this make-up water is supplied in two ways:

1. Soft water from the drinking fountain and stoker bearings in the Boiler Room is drained into the tower.
2. Periodically any remaining make-up necessary is supplied by a one and one-quarter-inch line which passes zeolite-treated soft water through the oil cooler of the Westinghouse turbine into the ten-inch circulating pipe directly ahead of the circulating pump.

During the test runs the drain line from the Boiler Room was cut out, leaving only the one supply of make-up to be measured. This was accomplished by installing a water meter in the one and one-quarter-inch line. A recording typr of meter was selected so that the total amount of make-up added during any one test could be measured. In as much as the make-up would be added at varying rates an indicating meter was not considered. A meter of the oscillating disk type, which was on hand in the power plant, was used. This meter was calibrated in the Mechanical Engineering Laboratory under conditions under which it would operate. The discharge was measured with the meter operating over a wide range of rates of flow, the water passing through a 35-inch length of pipe with an eight-inch sealing extension and thence into a weigh tank. A calibration constant of 0.99 gives an accuracy within one per cent for the rates of flow used in the tests.

As the make-up water meter was read at the beginning and end of each test it was desirable that the water level in the sump of the cooling tower be the same at the completion of the test as it was at the beginning. The splashing and the wave action on the water surface in the sump made it impractical to read the level inside if any degree of accuracy was desired. A glass U-tube operating on the hydraulic siphon principle was installed on the outside of the tower foundation with a tube extending up through the tower wall and thence downward into the sump. A scale was mounted next to this U-tube, permitting the water level to be read to within one-sixty-fourth of an inch. This increment in elevation of the water surface amounts to four and one-half cubic feet of water, or less than one per cent of the total water added during a four-hour test. To avoid any error from this source the water was brought back to its original level by the end of each test.

Approximately 160,000 cubic feet of air are supplied to the tower each minute by two nine-foot Axial Flow Fans mounted directly on gear-motor sets, as shown in Figs. 2 and 3. Each fan supplies the air for one cell of the tower. The air enters the front of the tower at the bottom, passing through the packing as illustrated in Fig. 4, and is discharged at the top through two layers of spray eliminating louvres. Because of the change in direction more air passes up through the back portion of the tower than through the front. This gives an uneven air distribution at the top. This uneven distribution, the turbulence caused by the louvres, and the large area of the top make it difficult to obtain accurate readings of air flow or velocity at the top of the tower.

The volume of air delivered by the fans could, therefore, be measured on the suction or the exhaust side of the fans. If measured on the exhaust side the air would be very turbulent and difficulty with the spray from the packing would be encountered, as this spray is drawn in toward the hubs of the fans. Thus the suction side of the fans is left as the only feasible place to measure the air velocity or flow. The fan motor and base protrude approximately two feet on the suction side, as illustrated in Fig. 2. Air velocity readings, therefore, should be taken outside of these motors in order to avoid turbulence caused by them. It was considered advisable to construct two ducts approximately three feet long and nine and one-half feet in diameter to confine the entering air and to permit measurement of its velocity when it was in a relatively non-turbulent condition.

The ducts were constructed of galvanized sheet iron. The mouth of each duct was supported by a wooden frame work. The outer edge was supported at about 30 points around its circumference to insure rigidity and to hold it to its circular form. The other end of the duct was strapped snuggly against the venturi ring of the fan housing and supported at about 40 points around its circumference. The ducts are shown in place in Fig. 11.

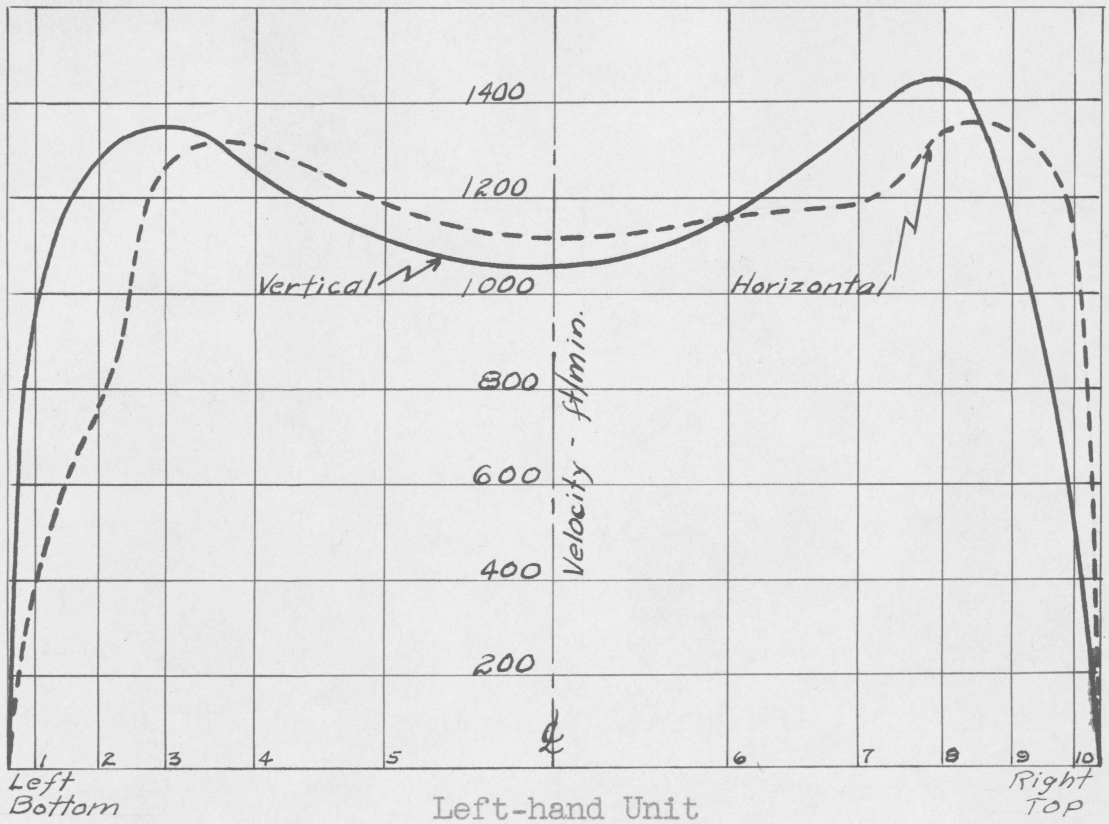
Two different instruments, anemometers and pitot tubes, were considered to be used for measuring the air velocities. Pitot tubes were not used as the slight pressure differences encountered would be insufficient to give an accurate reading. An anemometer was finally selected because of its ease of operation and its positive readings of air velocities, which would eliminate errors due to converting pressure head readings into velocity. As the velocities measured were about 1000 feet per minute they were well within the range of the instrument.

Anemometer readings were taken on two complete traverses for each duct on the horizontal and vertical diameters. The points of reading were taken as described in the Mechanical Engineers' Handbook[10], on page 294, for the ten-point method.

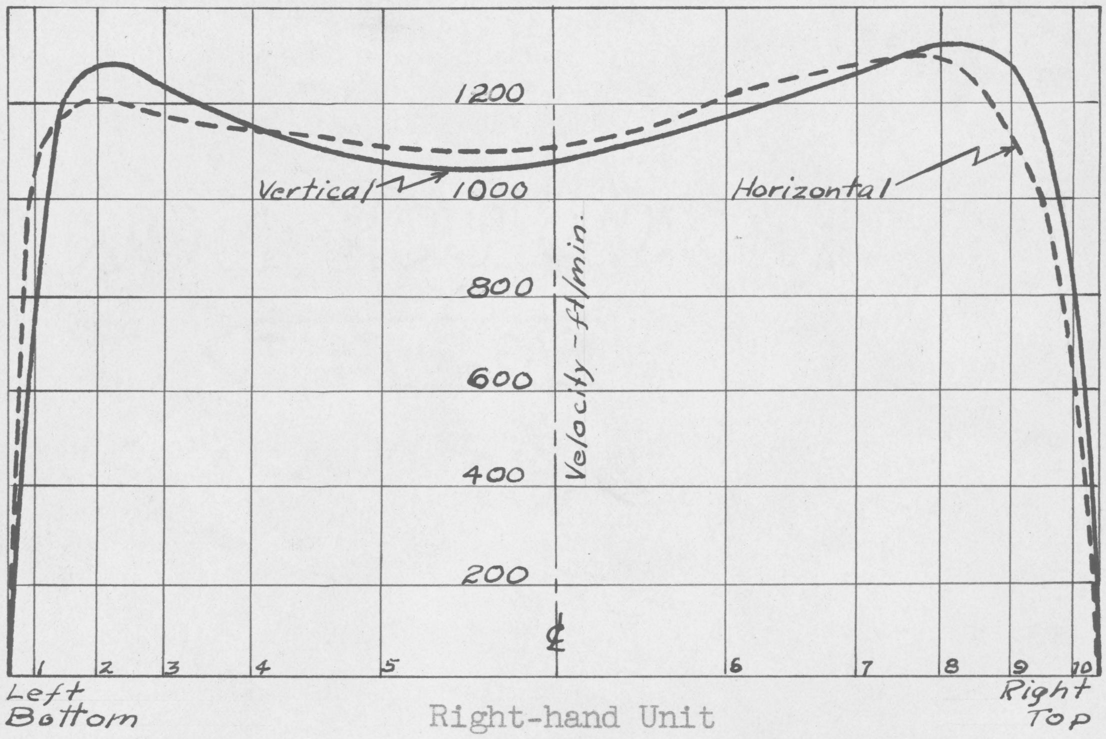
Fig. 11. The V. P. I. Cooling Tower with
Test Equipment Installed

Ten points on each of the two diameters were taken, giving 20 anemometer readings for each duct for each run of each test. Fig. 12 shows typical air velocity distribution in the two ducts.

On each of the vertical posts of the supporting frames of the ducts a one-half-inch pipe was fastened, extending from the top to the bottom. A similar pipe with an oversized pipe tee on each end was placed so that it could slide vertically on the first two members and remain in a horizontal position. Wires attached to the ends of this crossbar ran up through pulleys at the top and then back down to a reel placed on one of the vertical members of the supporting frame. Another oversized tee on the crossbar carried pipe fittings so arranged that when the anemometer was in place on them it was a short distance inside the duct and sufficiently above the crossbar to be clear of eddy currents set up by it. This tee was able to slide horizontally along the crossbar to permit its being located at any point along the horizontal diameter of the duct. Each of the tees was drilled and tapped for a thumb screw used to lock the apparatus in position for a reading. The anemometer was fitted with an off-and-on lever so that the indicating hand operated only when this lever was in the "on" position. A spring attached to the anemometer supporting bracket held this lever in the "off" position. A string attached to the control lever ran through a pulley on the end of the crossbar, permitting



Left-hand Unit



Right-hand Unit

Fig. 12. Velocity Curves of Air Flow in Ducts

the operator to pull the lever into the "on" position and hold it there. Duplicate apparatus on both ducts permitted quick transfer of the anemometer from one duct to the other. The apparatus was so designed that the anemometer could be controlled accurately in any position without disturbance of the entrance air stream.

An Eastman Timer was used to measure the length of time of each run. The second hand of this timer indicated one-quarter second intervals, and could be read accurately at a distance most convenient with the operation of the other apparatus.

The air flow apparatus is shown in use in Fig. 13, measuring the velocity at a point on the horizontal diameter.

The anemometer used was Tycos Anemometer No. 13B-1597, which belongs to the Mechanical Engineering Department. It had been calibrated by the makers some time before, but had been used considerably since then, so was recalibrated against a new Tycos Anemometer belonging to Prof. D. L. McElroy. This new instrument had been calibrated by the manufacturers and had never been used. The two instruments were placed in the wind tunnel of the Aeronautics Laboratory as shown in Fig. 14. They were run simultaneously at velocities in the range expected to be encountered in actual operation. The resulting calibration compared favorably with the original calibration curve for the instrument.

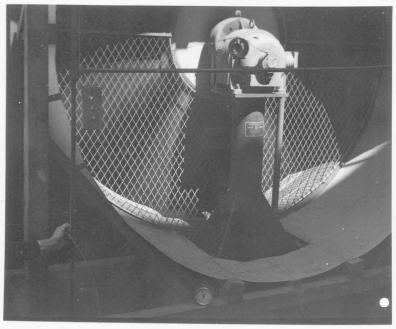


Fig. 13. Air Velocity Apparatus in Use

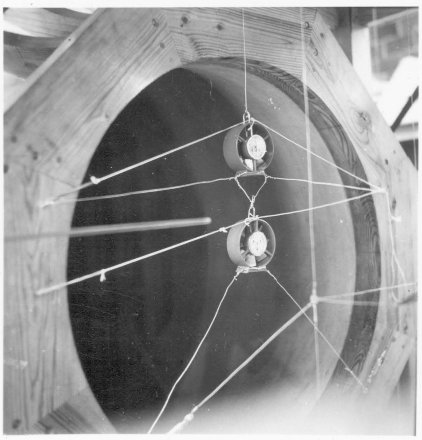


Fig. 14. Anemometer Calibration Set-up

For calculation of performance data knowledge of the following temperatures was necessary:

1. Inlet Air
   a. Wet-Bulb
   b. Dry-Bulb
2. Exit Air
   a. Wet-Bulb
   b. Dry-Bulb
3. Inlet Water
4. Exit Water

Glass stem, mercury filled thermometers 12 inches long were selected to measure these temperatures, identical instruments being used in all cases. These thermometers, with ranges of from 0° fahr. to +120° fahr., were tested at expected test temperatures and were found to agree exactly.

The wet- and dry-bulb temperatures of the inlet air were determined by a sling psychrometer equipped with two of the thermometers, the wet-bulb being covered by a wick that was moistened with distilled water prior to each reading. The psychrometer was used according to the generally accepted methods as outlined in the A.S.M.E. Power Test Codes.[8]

Observations indicated that the humidity and temperature of the exit air were not the same at all points over the top of the tower. Six sets of wet- and dry-bulb thermometers were prepared to measure these temperatures, and these sets were located at various points over the area of the top immediately below the lower spray eliminating louvres. As there was sufficient air velocity past the thermometers at this level to insure adequate evaporation from the wet-bulb they were mounted in a stationary position. The sets constructed consisted of two thermometers mounted parallel on a board and held away from it by small wooden blocks. A metal case over the dry-bulb protected it from spray. Moisture was supplied to the wet-bulb by a wick from a water reservoir mounted between the two thermometers. A photograph of one of these sets is shown in Fig. 15.

The temperatures of the inlet and outlet water were determined by thermometers hung in the distributing troughs at the top and in the sump at the bottom of the tower.

Fig. 15. Wet- and Dry-Bulb Thermometer Set

Calculation of humidity, specific volume, and other properties of the air necessitated the determination of the values of atmospheric pressure during the tests. These were read from a mercury column type of barometer which is one of the regular instruments in the power plant. It is located on the wall of the engine room at approximately the same elevation as the fan house roof of the cooling tower. Corrections for temperature and gravity were made in accordance with the A.S.M.E. Power Test Codes[9].

Each forced draft fan in the cooling tower is driven by a ten horsepower, three-phase, electric motor. The power to each motor is supplied by a 220-volt, 60-cycle, three-wire, system, and the manufacturer's data specifies 26.4 amperes for each leg of the three-phase circuit. As the motors are of the induction type it would not have been possible to measure the amount of power supplied by measuring only the voltage and current without knowledge of the power factor. The actual power input, therefore, was measured by the two watt meter method[10], using two indicating watt meters. Since the power input data was taken only in the tests run on the right-hand unit of the tower the watt meters were connected to the line supplying that fan. The current coils of the instruments were connected to split-core transformers clamped around two separate legs of the circuit.

TEST PROCEDURE

The tests performed were divided into two groups, one group being run on the entire tower and one group being run on the right-hand unit only.

The tests of the first group were of four hours duration each and were divided into one-hour runs, as it required half an hour to make a complete traverse of each air duct with the anemometer. Water level gage and make-up water meter readings were made at the beginning and end of each test. Wet- and dry-bulb temperatures of inlet and exit air, temperatures of inlet and exit water, flow nozzle and elbow flow meter heads, and barometric pressure were read each half-hour and recorded.

The tests of the second group were divided into half-hour runs and were of three hours duration. Readings of temperatures, barometric pressure, flow nozzle and elbow meter heads, and power input to the fan were made on the half-hours and recorded. The water level gage and make-up water meter readings were made as in the first group of tests.

In both groups of tests the anemometer data were being read and recorded continuously. A check on the water level was made at frequent intervals during each run and a valve immediately ahead of the make-up water meter regulated to maintain a constant water level.

A supplementary test on the right-hand unit was also run. This test was performed in the same manner as those in the second group with the exception that the wet- and dry-bulb thermometer sets were placed just below the bottom of the tower filling instead of in their usual place just below the bottom of the spray eliminating louvres at the top of the tower. The purpose of this supplementary test was to determine, if possible, the variation in wet- and dry-bulb temperatures from front to rear of the tower at this lower level.

A check on the effect of the air ducts on power requirements of the fans was made at the conclusion of all test runs. After the air ducts were removed a series of watt meter readings was made, the tower then being in its usual operating condition with no abnormal obstructions to air flow.

EXPERIMENTAL DATA AND RESULTS

Pertinent experimental data and results are shown in Table 1. Data appearing in this table and used in the calculations were obtained from Marks[10], A.S.H. and V.E. Guide[11], Keenan and Keyes[12], and a General Electric Psychrometric Chart (1934). Barometer corrections were made according to the A.S.M.E. Power Test Codes[9]. The values of cooling efficiency and effectiveness of cooling were calculated from equations given by Azbe[13] and Barnard, Ellenwood, and Hirshfield[14].

No attempt was made to plot curves of the different variables, for reasons that will be explained later.

1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16
Test No.	Water Flow #/min.			Air Velocity ft/min.				Entrance Air Temp. °fahr.		Corrected Barometer Reading Inches of mercury	Rate of Flow of Air-Vapor Mixture		Rate of Flow of Dry Air #/min.	Exit Air Temp. °fahr.	
	L	R	Total	L	R	Avg.	Cor. Avg.	Wet-Bulb	Dry-Bulb		cfm	#/min.		Wet-Bulb	Dry-Bulb
1	5485	4230	9715	1199	1213	1206	1102	61.0	72.3	27.61	156,500	11,020	10,930	74.6	74.8
2	6180	4530	10,710	1142	1181	1162	1062	45.4	59.0	27.74	151,000	10,930	10,900	64.8	65.2
3	6060	4230	10,290	----	1262	1262	1152	42.7	52.7	27.96	81,750	6000	5985	63.9	63.9
4	6075	4240	10,315	----	1171	1171	1070	50.0	66.7	27.92	76,000	5440	5410	67.5	67.9
5	6050	4260	10,310	----	1199	1199	1097	49.3	64.5	27.97	72,250	5580	5550	68.2	68.6
6	6060	4260	10,320	----	1187	1187	1085	55.0	75.0	27.90	77,000	5420	5400	----	----
Design			15,000					72.0							

Table I. Experimental Data and Results

1	17	18	19	20	21	22	23	24	25	26	27	28
Test No.	Vapor Entering Tower #/# Dry Air	Vapor Leaving Tower #/# Dry Air	Enthalpy of Exit Air Btu/# Dry Air	Enthalpy of Entrance Air Btu/# Dry Air	Water Temp. °fahr. Sump	Water Temp. °fahr. Trgh	Enthalpy of Water Entering Btu/# Water	Enthalpy of Water Leaving Btu/# Water	# Water Circulated per # Dry Air Circulated	Heat Absorbed by Air Btu/# Dry Air	Heat Given up by Water Btu/# Water	Heat Given up by Water Btu/# Dry Air
1	0.00886	0.01843	38.10	27.10	71.9	84.1	52.11	39.94	0.888	11.00	12.17	10.80
2	0.00343	0.01300	29.90	17.90	62.9	77.4	45.72	30.95	0.983	12.00	14.77	14.50
3	0.00314	0.01270	29.20	16.50	62.3	77.5	45.52	30.35	0.707	12.70	15.17	10.70
4	0.00500	0.01430	32.00	20.30	65.2	79.8	47.82	33.25	0.785	11.70	14.57	11.42
5	0.00400	0.01470	32.50	19.90	65.8	80.0	48.02	33.85	0.768	12.60	14.17	10.90
6	0.00471	-------	-----	23.30	70.0	85.0	53.00	38.04	0.789	-----	14.96	11.80
Design					85.0	98.0						

Table I. Experimental Data and Results (Con't.)

1	29*	30*	31	32	33	34	35	36
Test No.	Make-up Water cfm	Theoretical Evaporation cfm	Theoretical Air Required # Dry Air/min.	% Evaporation and Spray Loss	Efficiency %	Effectiveness of cooling	Power Required by one Fan i.h.p.	Remarks
1	1.62	1.82	10,720	1.04	52.9	0.588	-----	
2	1.69	2.43	13,200	0.92	45.3	0.524	-----	Very windy
3	1.58	2.40	5050	0.96	43.5	0.488	10.85	Tests Nos. 3 to 6 inclusive
4	1.74	2.32	5280	1.05	49.0	0.566	10.80	run on right-hand unit only.
5	1.68	2.25	4790	1.01	46.2	0.540	10.45	
6	1.96	----	----	1.18	50.0	0.578	10.40	
Design				1.20	50.0	0.570	9.40 b.h.p.	

*Items nos. 29 and 30 apply to entire tower, and not to right-hand unit only.

Table I. Experimental Data and Results (Con't.)

SAMPLE CALCULATIONS

Item 1.

From test data sheets

Item 2.

$Q = CA\sqrt{2gH}$ (for flow nozzles)

Q = Rate of flow in cu. ft./sec.

C = Velocity coefficient of nozzle = 0.985

A = Area of nozzle discharge tube in sq. ft.

g = Acceleration of gravity = 32.17 ft./sec.^{2}

H = Head on nozzle in ft. of water

$$Q = 0.985 \times \frac{\pi}{4} \times (3/12)^2 \times \sqrt{2 \times 32.17 \times H} = 0.388\sqrt{H}$$

$$H = \frac{(h' + h'') \times 13.55 - h''}{12}$$

h' = Reading of left leg of manometer

h'' = Reading of right leg of manometer

$$H = \frac{(3.9 + 3.9) \times 13.55 - 3.9}{12} = 8.5 \text{ ft. of water}$$

$Q = 0.388 \times 8.5 = 1.13$ ft.^{3}/sec.

Rate of flow = $1.13 \times 60 = 67.9$ ft.^{3}/min.

$= 67.9 \times 62.3$ #/ft.^{3} = 4230 #/min.

$Q = 212\sqrt{h}$ (for elbow meter)

Q = Rate of flow in ft.^{3}/min.

212 = Calibration constant

h = Manometer reading in inches of mercury

$Q = 212\sqrt{0.906} = 20.15$ ft.^{3}/min.

Rate of flow = 20.15×62.3 #/ft.^{3} = 1255 #/min.

Item 2 = 4230 + 1255 = 5485 #/min.

Item 3.

Calculations identical with those for flow nozzle Item 2.

Rate of flow = 4230 #/min.

Item 4.

Total flow = Item 2. + Item 3. = 5485 + 4230 = 9715 #/min.

Item 5.- 6.

Averages from test data sheets.

Item 7.

Average of Items 5. and 6.

Item 8.

Item 7. corrected according to calibration data for anemometer.

Item 9.- 10.

From test data sheets.

Item 11.

Barometer reading = 27.78 in. Hg. Temperature correction from A.S.M.E. Power Test Codes = -0.14 inches of mercury. Gravity correction from A.S.M.E. Power Test Codes = -0.03 inches of mercury. Corrected barometer reading = 27.78 - 0.14 - 0.03 = 27.61 inches of mercury.

Item 12.

$Q = VA$

Q = Rate of flow of air in ft.^{3}/min.

V = Average velocity of air in ft./min. (Item 8.).

A = Area of duct in square feet. (Area of both ducts used in tests applying to entire tower.)

$$Q = Vx2A = Vx2x\frac{\pi}{4}x(114/12)^2 = 142\ V$$

$$= 1102x142 = 156{,}500\ ft.^3/min.$$

Item 13.

Rate of flow in #/min. = Item 12 x density of air-vapor mixture in $\#/ft.^3$ (From A.S.H. and V.E. Guide).

Rate of flow = 156,500x0.07047 = 11,020 #/min.

Item 14.

Rate of flow of dry air = Item 13 x # dry air per # of mixture.

$$\#\ dry\ air/\#\ mixture = \frac{1}{1 + \#\ vapor/\#\ dry\ air}$$

$$Vapor/\#\ dry\ air = \frac{grains\ vapor/\#\ dry\ air}{7000\ grains/\#} = \#$$

Grains vapor/# dry air from G. E. Psychrometric Chart.

$$Vapor/\#\ dry\ air = \frac{62}{7000} = 0.00886\ \#$$

$$Dry\ air/\#\ mixture = \frac{1.00000}{1.00886} = 0.991\ \#$$

Rate of flow of dry air = 11,020x0.991 = 10,930 #/min.

Items 15 -16.

From test data sheets.

Item 17.

From G. E. Psychrometric Chart. (See Item 14.)

Item 18.

Calculations similar to those for Item 17.

Vapor/# dry air = 129/7000 = 0.01843 #

Items 19 - 20.

From G. E. Psychrometric Chart.

Items 21 - 22.

From test data sheets.

Items 23 - 24.

From Keenan and Keyes steam tables.

Item 25.

$$\text{Water/\# dry air} = \frac{\text{Item 4}}{\text{Item 14}} = \frac{9715}{10{,}930} = 0.888\ \#$$

$$\text{Water/\# dry air} = \frac{\text{Item 3}}{\text{Item 14}} \quad \text{for tests 3 - 6.}$$

Item 26.

$$\text{Btu/\# dry air} = \text{Item 19} - \text{Item 20}$$

$$= 38.10 - 27.10 = 11.00$$

Item 27.

$$\text{Btu/\# water} = \text{Item 23} - \text{Item 24}$$

$$= 52.11 - 39.94 = 12.17$$

Item 28.

$$\text{Btu/\# dry air} = \text{Item 27} \times \text{Item 25}$$

$$= 12.17 \times 0.888 = 10.80$$

Item 29.

$$\text{Make-up in cfm} = \frac{\text{Total make-up during test} \times 0.99}{\text{Duration of test in minutes}}$$

Total make-up and duration of test from test data sheets.

0.99 = Correction factor from calibration test.

$$\text{Make-up in cfm} = \frac{392.9 \times 0.99}{240} = 1.62$$

Item 30.

$$\text{Theoretical evaporation} = \frac{\text{Item 27 x Item 4}}{(H_v - \text{Item 23}) \text{x density of water}}$$

H_v = Enthalpy of water vapor at pertial pressure corresponding to exit air wet-bulb temperature. (From Keenan and Keyes steam tables.)

$$\text{Theoretical evaporation} = \frac{12.17 \text{x} 9715}{(1094.3 - 52.11) \text{x} 62.3}$$

$$= 1.82 \text{ cfm.}$$

Item 31.

$$\text{Theoretical air required} = \frac{\text{Item 27 x Item 4}}{\text{Item 26}}$$

$$= \frac{12.17 \text{x} 9715}{11.00}$$

$$= 10{,}720 \text{ \# dry air/min.}$$

$\text{Theoretical air required} = \dfrac{\text{Item 27 x Item 3}}{\text{Item 26}}$ for tests 3 - 6.

Item 32.

$$\text{Losses} = \frac{\text{Item 29 x density of water x 100}}{\text{Item 4}}$$

$$= \frac{1.62 \text{x} 62.3 \text{x} 100}{9715} = 1.04 \text{ \%.}$$

Item 33.

$$\text{Efficiency} = \frac{\text{Item 22 - Item 21}}{\text{Item 22 - Item 9}} = \frac{84.1 - 71.9}{84.1 - 61.0}$$

$$= 0.529 = 52.9 \text{ \%.}$$

Item 34.

$$\text{Effectiveness} = \frac{E' - E''}{E'}$$

$E' = (p' - p) + 0.011(t' - t)$

$E'' = (p'' - p) + 0.011(t'' - t)$

E = Cooling head in inches of mercury.

t' = Entrance water temperature.

t'' = Exit water temperature.

t = Entrance air wet-bulb temperature.

p' = Saturation pressure of water at t', inches of Hg.

p'' = Saturation pressure of water at t'', inches of Hg.

p = Saturation pressure of water at t , inches of Hg.

$E' = (1.1788 - 0.5407) + 0.011(84.1 - 61.0)$

$= 0.8921$ inches of mercury

$E'' = (0.7886 - 0.5407) + 0.011(71.9 - 61.0)$

$= 0.3679$ inches of mercury

$$\text{Effectiveness} = \frac{0.8921 - 0.3679}{0.8921} = 0.588$$

Item 35.

$$\text{Indicated h.p.} = \frac{\text{sum of watt meter readings x 20}}{\text{746 watts/h.p.}}$$

Watt meter readings from test data sheets.

20 = Current ratio of split-core transformers.

$$\text{i.h.p.} = \frac{405 \times 20}{746} = 10.85 \qquad \text{(For Test No. 3.)}$$

Item 36.

From test data sheets.

DISCUSSION AND INTERPRETATION OF RESULTS

In keeping with the expressed purpose of this report, data taken was primarily concerned with determining the adaptability of the test apparatus for the application and the variables of which further detailed tests should be run. The various factors later calculated were two-fold in purpose:

1. To correlate the data taken.
2. To compare the test with the design and guarantee performance.

It is apparent from the test data that while instantaneous values of air velocity were probably inaccurate, because of wind gusts, the average values over the period of a single test were accurate within reasonable limits. This fact is borne out by the steadiness of the values of average air velocities (Item 8*) and of the values of power input (Item 35) and by the symmetry of the air velocity curves in Fig. 12.

Values of Item 16 are the average values of the readings of five sets of wet- and dry-bulb thermometers. Several different arrangements of these sets over the top of the tower were used in order to determine the distribution of exit air wet- and dry-bulb temperatures across the width and length of the tower. There was found to be no appreciable variation

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*Items 1 through 36 in this discussion refer to Table I.

across the width, there being only a slight decrease in both temperatures at the outer edges. A representative distribution of the temperatures lengthwise of the tower from front to rear is shown in Fig. 16. Some of the factors to which these differences may be attributed are:

1. The division wall between the two units does not continue above the top of the filling.
2. Air flow upward through the rear portion is much greater than that through the front portion.

The results of the supplementary test with the wet- and dry-bulb thermometer sets placed below the filling are of doubtful value as far as humidity readings are concerned. It was impossible to protect the sets from the water spray. It is therefore quite probable that the temperatures indicated were a combination of those of the falling water and of the entrance air, and hence are of no value in calculation of efficiency or heat balance.

It will be noticed in all tests that the heat absorbed by the air (Item 26) is greater than that given up by the water (Item 28), with the exception of Test No. 2, which was run on a very windy day. This apparent inconsistency may be charged to the fact that the heat equivalent of the fan power input was not considered. This factor was not taken into account because the velocity of the exit air could not be determined, as already stated on page 27. It is the opinion of

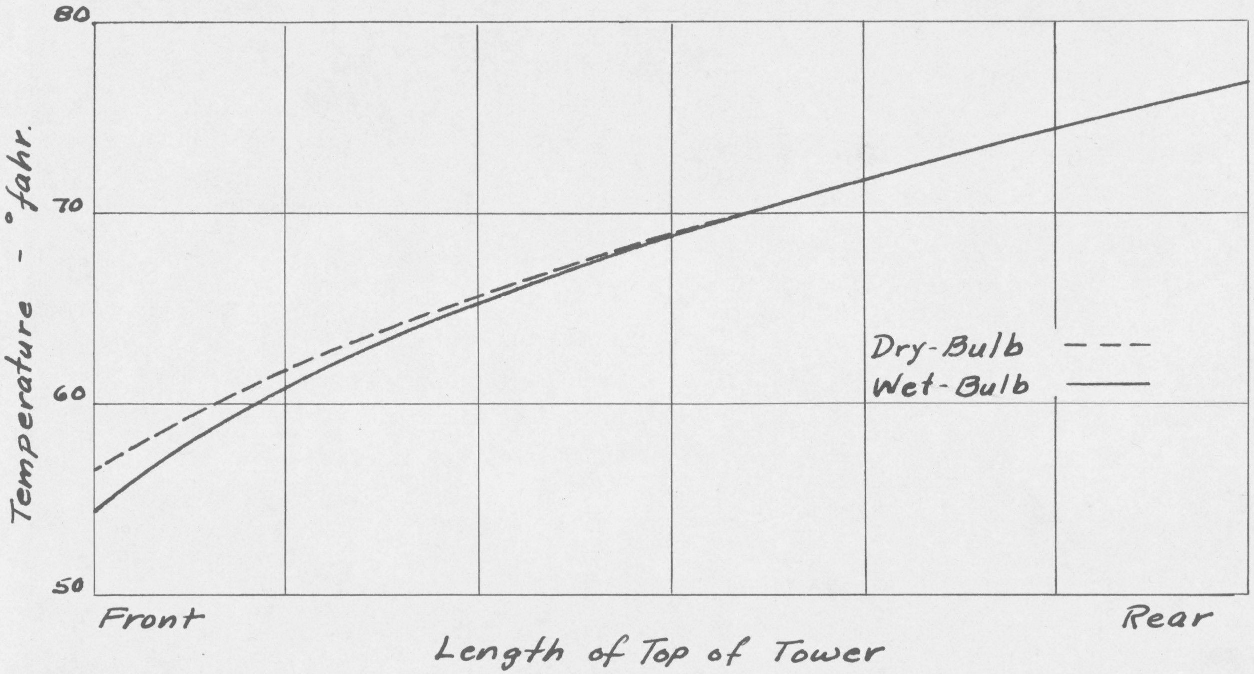

Fig. 16. Temperature Distribution in Exit Air

the authors that the difference between the heat equivalents of the exit air velocity and the power input to the fans would account for the variation between Items 26 and 28.

Items 29 and 30 apply to the entire tower in all tests, as both units drain into the same sump. It is impossible, therefore, to determine the proportion of the make-up going to each individual unit. The theoretical evaporation was calculated on the same basis to allow comparison of the two items. It is felt that variation between these items can be accounted for by the same reasons suggested for the variation between heat absorbed by the air and heat given up by the water.

Study of Items 32 through 36 indicates that design and test performance agree favorably, considering the fact that design conditions are quite different from test conditions in so far as water flow and temperature and entrance air wet-bulb temperature are concerned. There is no known method of interpolating between different sets of operating conditions.

In the check on the effect of the air ducts on fan power consumption the watt meter readings with the ducts removed were exactly the same as the readings taken during the regular test runs.

CONCLUSIONS AND RECOMMENDATIONS

As this thesis is primarily concerned with the selection of instruments for use in the testing of forced draft cooling towers it seems fitting to discuss the performance of the instruments selected and used in the tests run.

The anemometer showed up very well in the measurement of the quantity of air supplied to the tower. The velocities measured were well within the range of the instrument, and it showed itself to be an accurate and easy instrument to manipulate. It was found, however, that the anemometer did not give reliable readings when there was a high wind, due to the turbulent air currents set up. The anemometer is considered a very reliable instrument under normal conditions.

It is apparent from the tests run that ducts ahead of the fans are desirable and reliable in the measurement of air flow. Those constructed by the authors produced no apparent increase in the fan power consumption, and it is concluded that such apparatus has no material effect on the performance of a cooling tower.

The two watt-meter method is the generally accepted way of determining the power in a balanced three-phase circuit. Where the current can not be interrupted for connection of the current coils of a watt-meter, split core transformers can be used for the current connection, the potential connections being made as usual. This method can be used very easily and accurately in this case to determine the power input and the power factor of the fan motors.

Orifices and flow nozzles are considered accurate instruments for measuring large quantities of water flow. There are a number of nozzle forms that have been studied sufficently that reasonably accurate coefficients may be assumed. Small variations in machining occur, however, and, where possible, nozzles and orifices should be calibrated prior to use. The authors would suggest that when cooling towers are being constructed and are to be subjected to testing at any future time, provisions should be made for mounting flow nozzles or orifices in the water supply pipes. Flanges on the discharge end of such pipes provide a means of mounting such instruments in a minimum of time. This point is of particular importance in case the tower is operated continuously.

In smaller pipes carrying steady flows elbow flow meters provide an accurate and low-cost means of water flow measurement. They should be calibrated under conditions paralleling as nearly as possible the conditions under which they are to be used.

While hook and float gages installed in stilling wells are extremely accurate and are applicable for measuring the water level in cooling tower sumps, an instrument of the type used in this test is the simplest to read and is considered sufficiently accurate for such measurement. It is recommended that during the construction of towers a small plugged connection be provided below the minimum water level to facilitate mounting a glass tube as a water level gage. Such an arrangement is desirable not only for testing but also for maintainence of proper water level during normal operation.

Where relative humidities are close to 100 per cent and the air is traveling with considerable velocity with large amounts of entrained water droplets, as is the case in cooling towers, it is very difficult to measure the wet- and dry-bulb temperatures accurately. The chief difficulty lies in the proper shielding of the dry-bulb from the entrained moisture. Where cost is not a deciding factor and tests are to be run over a considerable period of time, resistance thermometers are recommended. The thermometer sets built by the authors gave satisfactory results when measuring the temperatures of the air leaving the tower, but were of no value when used for a similar purpose under more severe conditions existing in and beneath the tower filling.

Sling psychrometers are generally accepted for measuring the wet- and dry-bulb temperatures of still or slowly moving air, and gave what were considered very reliable results in the tests performed by the authors.

In measuring the temperatures of water entering or leaving the tower, resistance thermometers with the bulbs installed in the pipe lines are recommended; but if, as mentioned above, cost is a deciding factor, glass stem thermometers can be placed in wells in the pipes at such points as are readily available. If this is not practical they may be hung in the distributing troughs at the top and in the sump at the bottom. If this later application is used, delayed reading thermometers are preferable, as instruments of the usual construction willchange in readings while they are being raised to a location where they may be read.

Since the make-up water is added periodically a recording meter should be used for measurement of this item. The oscillating disk type is satisfactory if calibrated in the range of flows to be used, and should be used only in this range.

Even though all instruments functioned satisfactorily and the results obtained are considered reliable, nothing of material value, so far as rational design is concerned,could be determined. With a full size tower in actual operation no control over the different variables is possible, and under such conditions the effect of changes of any one variable, with others held constant, is indeterminate. As this was the case in the tests herein discussed, no attempt was made to calculate any such effects or to plot curves of them. It is the recommendation of the authors that any research relating to rational design be conducted on a model cooling tower so constructed that absolute control over the following variables is possible:

1. Entrance water,quantity and temperature.
2. Entrance air, quantity, velocity, temperature and humidity.
3. Tower filling.

It is only through such research that a reasonable approach to rational design can be made.

SUMMARY

Test data of forced draft cooling towers is all too meager, and that available is, in many instances, incomplete. It is the opinion of the authors that putting cooling tower design on a rational basis can be brought about only by two methods. Either complete and intensive study of all existing towers or a thorough study of models, similar to the research conducted on airplanes in wind tunnels and on boat hulls in towing tanks, is necessary. Models used should be so constructed that quantity, condition, and velocity of air; quantity and condition of water; and type and arrangement of filling may be controlled. It is by the second method that the authors believe rational tower design may best be brought about.

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