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SINGLE-CELL COOLING TOWER PERFORMANCE STUDIES

Part I: The Effect of Reduced Tip Clearance on Fan Performance

Part II: The Effect of Reduced Tip Clearance on Cooling Tower Performance

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UNION CARBIDE

OAK RIDGE GASEOUS DIFFUSION PLANT
OAK RIDGE, TENNESSEE

prepared for the U.S. DEPARTMENT OF ENERGY *under*
U.S. GOVERNMENT *Contract W-7405 eng 26*

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ABSTRACT

Union Carbide Corporation-Nuclear Division at the Oak Ridge Gaseous Diffusion Plant has been involved in cooling tower testing for some time. Most recent efforts have been directed toward evaluating the effects of reduced blade tip clearance on fan and tower performance. This paper consolidates these efforts which were originally documented in two separate reports.

Data and results are presented herein which show that large improvements in fan efficiency (static and total), significant monetary savings, and substantial increase in tower performance can result from reducing the fan blade tip clearance on cooling towers.

NOMENCLATURE

Symbols

A_s	Area of Fan Stack, sq ft
a	Area of Water Interface, sq ft/cu ft
C_p	Cost of Power, mils/kwh
CFM	Volumetric Air Flow Rate, cfm
G	Mass Flow Rate of Dry Air, lbm/min
g_c	Gravitational Constant, 32.174 lbm-ft/lbf-sq sec
K	Unit Mass Transfer Conductance lbm/hr-sq ft
L	Mass Flow Rate of Water, lbm/min
$\vec{n}$	Unit Vector Normal to Plane of Blade Rotation
P	Power Input to Fan Motors, kw
Δp_s	Static Pressure Drop Across Fans, in. Water
Δp_t	Total Pressure Drop Across Fans, in. Water
S	Power Savings, $/fan-yr
V	Active Tower Volume, cu ft/sq ft
$\bar{V}$	Average Velocity of Exit Air, fpm
$\vec{V}$	Air Velocity Vector, fpm
ε_{GR}	Efficiency of Gear Reducer
ε_m	Electrical Efficiency of Fan Motors
ε_s	Static Fan Efficiency
ε_t	Total Fan Efficiency
ρ_a	Density of Exit Air, lbm/cu ft

Subscripts

RBTC	Reduced Blade Tip Clearance Condition
NBA	New Blade Angle
OBA	Old Blade Angle

SINGLE CELL COOLING TOWER PERFORMANCE STUDIES

Part I: The Effect of Reduced Tip Clearance On Fan Performance (K-892 Tower)

INTRODUCTION

Union Carbide Corporation-Nuclear Division operates the Oak Ridge Gaseous Diffusion Plant (ORGDP) in Oak Ridge, Tennessee for the United States Department of Energy. The forced draft, wet cooling towers at this facility are some of the largest of their kind in the world. Four of these towers are designed to remove a combined total of over 100 million British Thermal Units (BTUs) of heat energy per minute. Approximately ninety percent (90%) of the 2,180 megawatts of electrical energy to be input to the ORGDP uranium enrichment process will be dissipated as heat energy from these towers. With 140°F on-tower temperature at design flow, these towers will sustain one of the higher operating temperatures allowable from the standpoint of continuing tower life. These four towers, with two smaller ones, will be collectively cooling 330,000 gallons per minute of recirculating cooling water (RCW) at full capacity. The evaporative and drift losses at these operating conditions will require twenty million gallons per day of make up water.

These operational requirements make cooling towers the second most important auxiliary operation, after electrical power, which serves the uranium enrichment gaseous diffusion process. The preservation and modification of these structures for continued, improved usage is of prime significance to the ORGDP. All but one of these towers was built prior to 1955, and several later major modifications have been necessary.

Low process power levels in recent years have resulted in spare cooling tower capacity which, in turn, made it possible to improve certain towers by replacing damaged fill, installing new motors and gear reducers, standardizing fan blades, etc. However, in view of the planned increase in the demand for separative work in the 1980's essentially all of the available cooling tower capacity at the ORGDP will be required. In fact, some analyses have shown that the available cooling capacity during this period will only be marginally adequate. Thus, a tower modification test program was begun to determine the most effective methods of obtaining increased cooling from existing towers.

The first parameter to be evaluated was reduced blade tip clearance (RBTC) between the fan blade tips and the fan shroud. One cell of the K-892H Tower (Cell 2) was instrumented and tested first because this tower offered the largest potential savings resulting from improvements. This tower is a Foster-Wheeler counter-flow type built in 1954 of untreated redwood. Fill and structural deterioration made it necessary to rebuild the tower with treated redwood. This eleven-cell tower has two eight-bladed fans per cell driven by 50 horsepower motors. The K-892H tower has a companion eleven-cell tower, K-892G, which shares the same cold water reservoir.

Recording of test data began in 1975 with information being collected from both fans (fans 3H and 4H) of Cell 2H. Additional tower and weather data were also recorded so that cell performance as well as fan efficiencies could be analyzed. The collection of test data was completed in 1976. Part I of this paper will detail the preparation and testing involved in evaluating RBTC with respect to fan efficiency. Part II of the report will show the effect of RBTC on cell performance.

Description of Polyurethane Tip Seal

Cell 2 of the K-892H Tower is serviced by Fan Number 3H on the east half and Fan Number 4H on the west half of the cell. Surrounding each fan is a truncated, conical, sheet metal shroud which funnels the air into the fan and out of the tower. Ideally, no clearance should exist between the tips of the eight blades of each fan and the respective shroud, but the clearance on Cell 2, which resulted from the asymmetry of the shrouds, was a minimum of 1-3/4 inches and a maximum of 4 inches as described in Table I. The space between the blade tips and the inside of the shroud allowed recirculation of air resulting in less air being pulled through the cell.

The efficiency of the fans could be improved by eliminating the tip clearance entirely. Unfortunately, a zero clearance on a 20-foot diameter fan is not practical with most seal rings because of the danger of a blade striking the seal and becoming detached from the hub. Consequently, a material was searched for that would resist the adverse conditions that exist on a cooling tower, permit a near zero tip clearance, and tear away if struck by a blade. Polyurethane was used on the fan shrouds of Cell 2H because it met the conditions specified and was easily molded and applied. A rigid polyurethane foam, Stafoam AA602,† was selected. This material is manufactured by the Expanded Rubber and Plastic Corporation. Sample pieces with densities of 2, 6, and 8 pounds per cubic foot were evaluated for weight, shear strength, and ease of application. The sample with a density of 2 pounds per cubic foot was chosen since it possessed the most desirable properties. A metal mold was then prepared which was 1 foot wide, 1/2 foot thick, and 1/8 of the circumferential length of a fan shroud at the plane of the blades. The mold possessed the same circumferential curvature as the shrouds.

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†In this document, brand names are used for descriptive purposes only. Similar materials from other manufacturers would be equally acceptable.

TABLE I

CELL 2 BASE CONDITION* TIP CLEARANCE ON K-892H COOLING TOWER

Date Measured	Fan Designation	Blade Designation (8 Blades Per Fan)	Blade Angle, Degrees	Distance From Shroud To Fan Tip Measured At Four Circumferential Positions, In. 0°	90°	180°	270°
February 1976	4H	1	30	2-1/4	1-3/4	3-1/2	3
		2		2-1/4	1-3/4	3-1/2	3
		3		2-1/2	2	3-3/4	3-1/2
		4		2-1/4	1-3/4	3-1/2	3-1/4
		5		2-1/2	2	3-3/4	3-1/2
		6		2-1/4	1-3/4	3-1/2	3
		7		2-1/2	2	3-3/4	3-1/4
	↓	8	↓	2-1/4	2	3-1/2	3
February 1976	3H	1	30	2-1/4	1-3/4	3-1/2	3-1/4
		2		2-1/4	1-3/4	3-1/2	3-1/4
		3		2-3/4	2-1/4	4	3-3/4
		4		2-1/2	2	4	3-3/4
		5		2-1/4	2	3-3/4	3-1/2
		6		2	1-3/4	3-3/4	3-1/4
		7		2	1-3/4	3-3/4	3-1/4
	↓	8	↓	2-1/4	2	3-3/4	3-1/2

*Base Condition: No tip seal present to prevent air recirculation around the fan blades.

Note: A complete set of series 67-53-1 blades were present on each fan during all 1976 testing.

An area 1 foot above and below the plane of the blades on each fan was buffed to improve adhesion. After molding, the solid foam was placed, one section at a time, against the buffed area and glued with a mixture of Versamid-140 and Epon-828 resin which was painted onto the foam and shroud surfaces. A 3/8-inch by 1-foot by 7-foot piece of plywood was bolted through the foam to the shroud until the resin cured. The plywood was then removed.

The 6-inch thick foam blocked the path of the blade tips and had to be reduced in thickness. Hand removal was found to be inadequate. Five wire wheel brushes, each 4 inches in diameter, were locked onto the stainless steel shaft of a pneumatic drill. The drill was then clamped onto a blade tip, the shaft being perpendicular to the foam. Careful adjustment of the depth of cut gave a uniform, accurate removal of the excess polyurethane as the cutting blade was rotated. The blade tip-to-foam distance then approached zero clearance. Each of the eight-per-fan foam sections were installed in this manner. Areas which were cut too deeply were repaired by painting the liquid foam directly into the void and allowing it to cure.

To insure stability, washered bolts were added through the foam both above and below the plane of the blades. The bottom edge of the polyurethane tip seal was then beveled to reduce air drag. The tip seal installations on Fans 3H and 4H were completed by the end of February 1976, after beginning in January 1976. The extreme blade tip clearance at any point was no greater than 1/2 inch and the average clearance was less than 1/4 inch. A picture of the polyurethane seal after installation on Fan 4H can be seen in Figure 1.

DESCRIPTION OF TESTS

A total of seven separate tests will be discussed here. Four complete tests at various water loading rates were made under the RBTC conditions, and three complete tests were made under the existing or original condition. No changes were made to the blade angles between these two sets of tests.

During each test numerous measurements were made; however, only those measurements directly related to fan performance will be discussed here.

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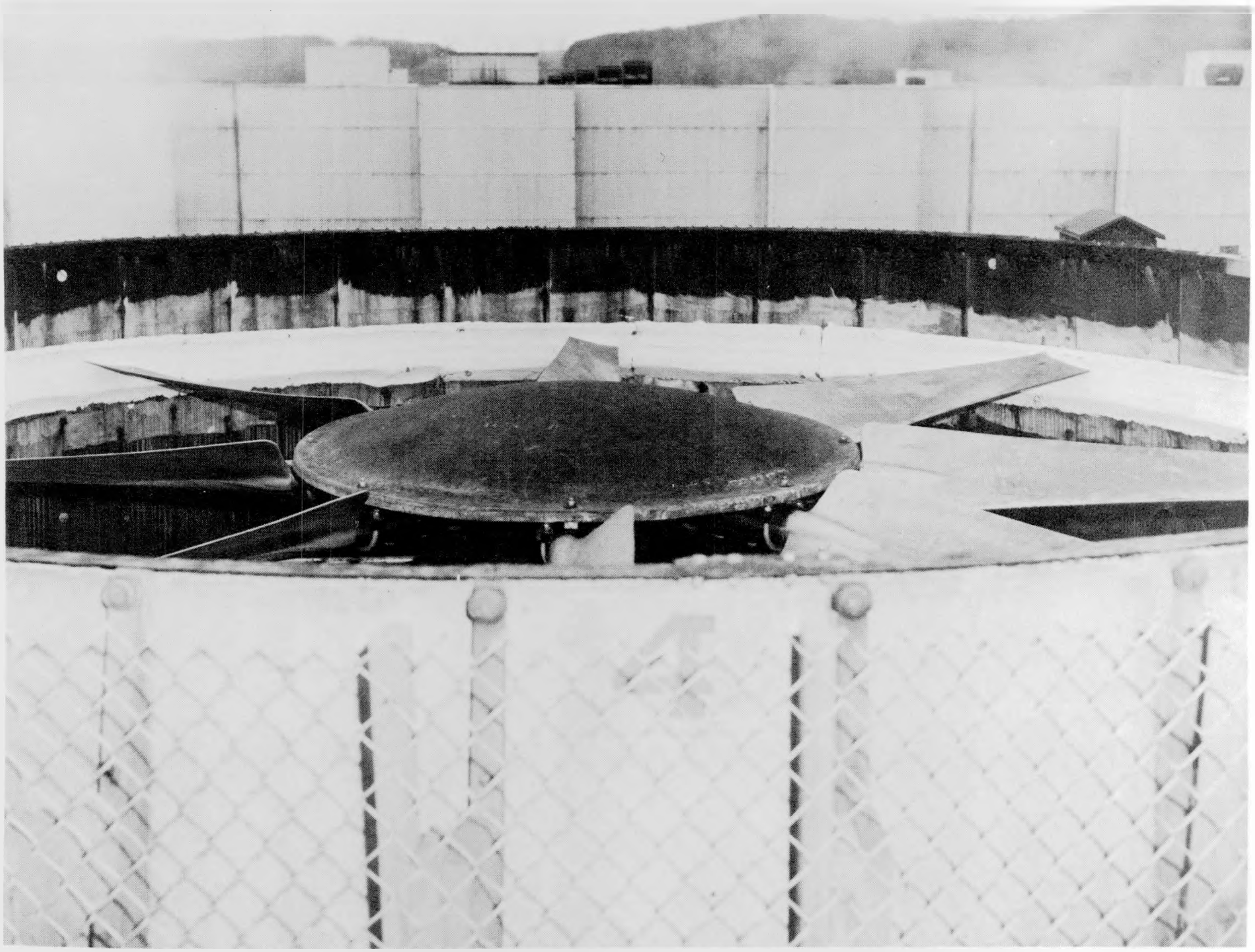

Figure 1

A PHOTOGRAPH OF FAN 4H SHOWING THE POLYURETHANE TIP SEAL

Velocity and Temperature Profiles of the Exit Air

Following the recommendations of CTI[1ψ] and ASME[2] for induced draft cooling towers, air velocity and air temperature measurements were made by traversing the discharge side of each fan. From the velocity measurements the volumetric flow rate of air per fan was determined by evaluating:

$$CFM = \int_{As} \vec{V} \cdot \vec{n} \, dAs \tag{1}$$

where:

CFM = Air volumetric flow rate, cfm

$\vec{V}$ = Air velocity, fpm

A_S = Fan stack area, sq ft

$\vec{n}$ = Unit vector normal to plane of blade rotation

A calibrated, R. M. Young propeller vane anemometer and the associated recording devices were used to obtain the velocity measurements. These instruments were able to sense the air speed as well as the direction of flow. This was accomplished by allowing the vane portion of the anemometer to rotate freely so that the plane of propeller rotation could become perpendicular to the direction of air flow. It was necessary to determine the vertical component of this measured velocity in order to calculate the volumetric flow rate of air as per Equation (1). This procedure was satisfactory until it became necessary to traverse the fan hub. At this point due to the abrupt change in air flow direction, the vane portion of the anemometer would strike the hub jarring the traverse rail. This constituted a slight safety hazard since the vane anemometer assembly could have become detached from the traverse rail and fallen onto the fan. In an attempt to avoid this problem several traverses were made after constraining the anemometer in a position normal to the plane of the fan blades. An estimate of the vertical component of velocity was therefore measured. A comparison of the volumetric air flow rates calculated from these measurements with volume flow rates calculated from measurements made with a freely rotating anemometer showed a difference of less than 2%. Thus, by making the measuring procedure safer, very little was sacrificed in accuracy. In fact the instrument readings obtained with a constrained anemometer did not fluctuate nearly as much as those obtained when the instrument was allowed to rotate freely.

[ψ]Superscripted numerals designate similarly numbered references listed at the end of this report.

The velocity and temperature measurements were recorded at stations determined by dividing the fan stack into five equal areas and calculating the center of each area. The calculations were performed during 1975, and were not checked until after all seven tests reported here had been completed. Unfortunately, the measuring station nearest the center of the fan stack was positioned incorrectly. As Figure 2 shows the proper location of this station was slightly outside the hub rather than on the hub. It would be difficult to estimate the velocity profile existing near the hub from the other available measurements, so no attempt was made to estimate the true average velocity of the inner most area. It can be stated with confidence, however, that the true velocity of the inner most area is greater than the essentially zero value which was measured during the tests. Thus, the calculated volumetric air flow rates presented here are on the conservative side in an absolute sense, perhaps by as much as 5% to 10%. It should be pointed out though, that the data and results presented here should be viewed in a relative rather than absolute sense. Thus, this error should have no impact whatsoever on the information presented.

Exit air wet-bulb temperatures were measured at each station with a calibrated 100-ohm copper resistance temperature detector (RTD) covered with a sponge type wick. This RTD had an accuracy of $\pm$ 0.10°F including instrumentation inaccuracies. The actual temperatures were read in degrees Fahrenheit from a Leeds and Northrup Scanner-Programmer. The incorrectly located measuring station should have only a minimal effect on the average exit temperature.

During each test two diametral traverses were made per fan. A single traverse consisted of ten velocity and temperature measurements. The picture shown in Figure 3 was taken during one of the traverses performed on Fan 4H. The vane anemometer is vaguely visible at the left of the picture.

Static Pressure Drop Across the Fans

The static pressure drop across each fan was measured with an impact tube fabricated from 7/16-inch ID hard-drawn copper. Polyvinyl chloride (PVC) tubing connected the impact tube to the low side of an Ashcroft Digigage, which sensed differential pressure in inches of water. The high side of the Digigage was left open to the atmosphere.

Measurements were taken at four circumferential locations inside the fan stack, beneath the blades. The impact tube had a 90-degree bend about 1 foot from its exposed end so that it could be positioned with respect to the air flow. All measurements were made with the impact tube pointing vertically downward. By virtue of Bernoulli's equation for the flow of nonviscous, incompressible fluids, measurements obtained in this manner were very near the static pressure drop across the fan. Not only did this technique give the most accurate estimate of the static pressure drop but it also greatly reduced the fluctuations in the readings. The true static pressure drop for each test was taken to be the average of the four circumferential readings.

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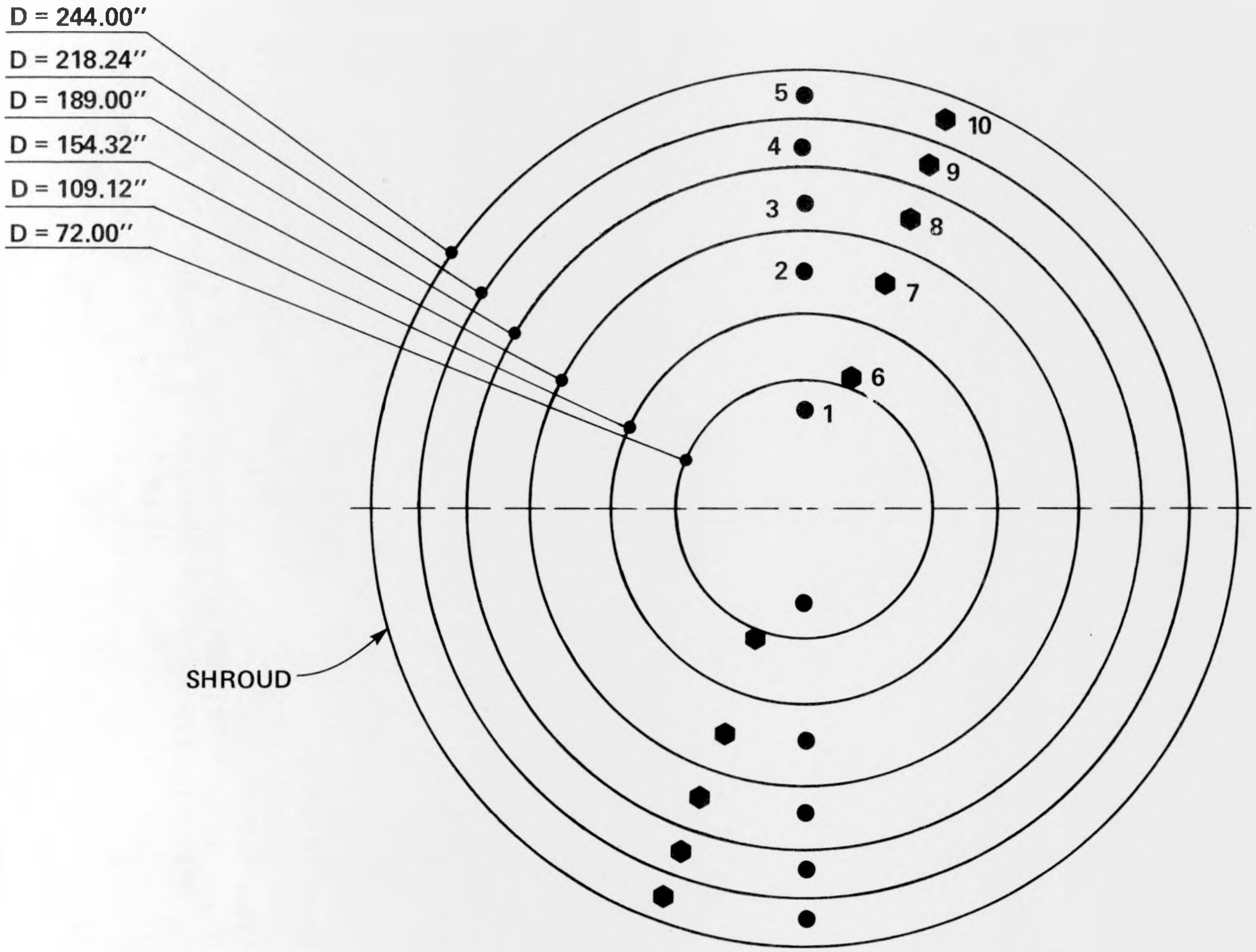
INCHES FROM CENTER TO STATION	
1 – 27.28	6 – 38.58
2 – 65.86	7 – 66.83
3.– 85.83	8 – 86.27
4 – 101.81	9 – 102.07
5 – 115.56	10 – 115.74



● = ACTUAL MEASURING STATIONS

⬢ = CORRECT LOCATION OF MEASURING STATIONS

Figure 2

A SCHEMATIC DRAWING OF A FAN SHOWING THE LOCATIONS OF VELOCITY AND TEMPERATURE MEASURING STATIONS

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Figure 3

A PHOTOGRAPH MADE DURING A TRAVERSE OF FAN 4H

Power supplied to the Fan Motors

The power supplied to the fan motors was measured by a calibrated three-phase wattmeter manufactured by Scientific Columbus. Typically, readings were taken each half-hour. Variations occurring during a given test were insignificant. The power required by the fans themselves was estimated by assuming a combined efficiency of 95% for the motors and gear reducers.

Weather Data

Since meteorological conditions, especially wind speed and direction, have a substantial effect on tests of this nature, the following data were continuously recorded during each test:

(1) Ambient Temperature

(2) Relative Humidity

(3) Wind

(4) Wind Direction

A small weather station, Figure 4, was utilized to obtain these measurements. This station was located a considerable distance (more than 100 feet) from the K-892 Tower in order to eliminate any potential influence from the K-892 Tower itself. The weather data are summarized in Table II.

DISCUSSION OF RESULTS

Table III was constructed in order to summarize some of the key features of each of the seven tests. Considering Fans 3H and 4H separately, it is evident that RBTC increased the average velocity developed by Fan 3H some 9.6%. Similarly, the average velocity developed by Fan 4H was increased some 6.4%. The average power required by the two fans also increased - 9.7% for Fan 3H and 9.3% for Fan 4H.

From this simplified approach, it would seem that implementation of RBTC would not be economically justifiable. The power consumption increased by a larger percentage than the volumetric air flow rate (volumetric air flow rate is very nearly proportional to average air velocity). This, however, does not tell the complete story. The efficiency with which this additional power is utilized must also be considered. In other words the ratio of fan power <u>output</u> to fan power <u>input</u>, both with and without RBTC, must be determined. This implies that an expression for fan efficiency must be considered. If the RBTC increases the fan efficiency significantly, then it should be considered an economically attractive alternative.

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Figure 4

A PHOTOGRAPH OF THE WEATHER STATION

TABLE II

SUMMARY OF WEATHER DATA OBTAINED DURING TESTS

Run No.	Date	Dry Bulb Temperature, °F	Relative Humidity, %	Average Wind Speed, mph	Maximum Wind Speed, mph	General Direction
1	8-10-76	80-83	46-50	5	9	Southeast
2	8-11-76	84-87	53-57	7	9	West
3	8-12-76	88-90	48-52	5	9	Southwest
4	8-13-76	86-90	45-49	5	9	West
5	9-24-76	78-80	38-42	4	6	Southwest
6	9-29-76	68-72	78-82	5	6	Southwest
7	10-1-76	69-73	40-44	7	11	North

TABLE III

SOME FEATURES OF EACH TEST

		(RBTC) Test Number						(Original) Test Number		
		1	2	3	4	Average	Average	5	6	7
Fan 3H	Exit Air Velocity, fps	17.72	17.20	16.98	17.19	17.27	15.76	14.90	15.66	16.72
	Power Input to Motor, kw	46.82	47.22	46.48	46.40	46.73	42.61	42.92	42.30	42.62
Fan 4H	Exit Air Velocity, fps	17.14	16.67	17.08	16.91	16.95	15.93	15.74	15.87	16.17
	Power Input to Motor, kw	49.92	50.43	49.78	49.60	49.93	45.67	45.90	45.48	45.64

Static Efficiency

Two expressions[1] are normally used to calculate the efficiency of cooling tower fans. The first expression utilizes the measured static pressure beneath the fan stack to determine a static efficiency as follows:

$$\varepsilon_s = 1.175 \times 10^{-4} \frac{\overline{V} \cdot A_s \cdot \Delta p_s}{\varepsilon_m \cdot \varepsilon_{GR} \cdot P} \qquad (2)$$

where: ε_s = Static Fan Efficiency

$\overline{V}$ = Average Vertical Air Exit Velocity, fpm

A_s = Area of Fan Stack, sq ft

Δp_s = Measured Static Pressure Drop Across Fan, in. water

ε_m = Motor Electrical Efficiency

ε_{GR} = Gear Reducer Electrical Efficiency

P = Measured Power Input to Motor, kw

The static efficiency is required by the standard CTI Bid and Inquiry Form[3]. Static efficiency gives no credit to the fan for the velocity imparted to the air. It is actually a measure of the fan's ability to move a certain volume of air against an imposed resistance. Field measurements of static efficiency normally vary between 50% and 70%.[3]

Total Efficiency

Total fan efficiency[1] is another important indicator of fan performance. It is calculated from the relationships (3) and (4) below:

$$\varepsilon_t = 1.175 \times 10^{-4} \frac{\overline{V} \cdot A_s \cdot \Delta p_t}{\varepsilon_m \cdot \varepsilon_{GR} \cdot P} \tag{3}$$

$$\Delta p_t = \Delta p_s + 5.342 \times 10^{-5} \frac{\rho_a \overline{V}^2}{2\, g_c} \tag{4}$$

where: ε_t = Total Fan Efficiency

Δp_t = Total Pressure Drop Across Fan, in. water

ρ_a = Density of Exit Air, lbm/cu ft

g_c = Gravitational Constant, 32.174 $\frac{\text{lbm - ft}}{\text{lbf - sq sec}}$

From an examination of Equations (2), (3), and (4) it is evident that total efficiency and static efficiency differ in the sense that the former gives the fan additional credit for the velocity imparted to the air. Total efficiencies are somewhat larger than static efficiencies ranging from about 60% to 80% when measured in the field.

From the experimental data obtained during this investigation, static and total fan efficiencies were calculated. The results both with and without RBTC are shown in Tables IV and V. In order to obtain these efficiencies, the combined efficiency of the electric motors and gear reducers, $\varepsilon_m \cdot \varepsilon_{GR}$, was assumed to be 95%.

Table IV shows that even though Fan 3H was not performing too poorly in its original state (ε_S ranged from 55% to almost 62%) RBTC still improved its performance substantially (from about 7% to just over 36%). The same trend is evident when total fan efficiencies are considered.

In order to visualize the reasons why RBTC increases fan efficiency, Figure 5 was constructed. This figure shows the variation with position of vertical air velocity from Fan 3H both with and without RBTC. Results from Runs 1 and 5 were chosen as being representative cases for

TABLE IV

FAN 3H PERFORMANCE RESULTS (K-892 TOWER)

	RUN NUMBERS						
	1	2	3	4	5	6	7
Air Flow Rate x 10^{-6}, cfm	0.345	0.335	0.331	0.335	0.290	0.305	0.325
Average Vertical Air Velocity, fps	17.72	17.20	16.98	17.19	14.90	15.66	16.72
Δp_s, in. water	0.725	0.850	0.825	0.838	0.658	0.638	0.656
Δp_t, in. water	0.791	0.912	0.885	0.900	0.705	0.689	0.715
P, kw	46.82	47.22	46.48	46.40	42.92	42.30	42.62
ε_s, %	66.08	74.59	72.67	74.84	55.00	56.91	61.87
ε_t, %	71.81	79.73	77.68	80.05	58.60	61.19	67.03

TABLE V

FAN 4H PERFORMANCE RESULTS (K-892 TOWER)

	RUN NUMBERS						
	1	2	3	4	5	6	7
Air Flow Rate x 10^{-6}, cfm	0.334	0.325	0.333	0.329	0.307	0.309	0.315
Average Vertical Air Velocity, fps	17.14	16.67	17.08	16.91	15.74	15.87	16.17
Δp_s, in. water	0.638	0.625	0.663	0.738	0.419	0.400	0.375
Δp_t, in. water	0.695	0.683	0.724	0.798	0.471	0.453	0.430
p, kw	49.92	50.43	49.78	49.60	45.90	45.48	45.64
ε_s, %	52.39	49.82	54.86	60.56	34.66	33.62	32.01
ε_t, %	57.37	54.29	59.75	65.28	38.81	37.89	36.52

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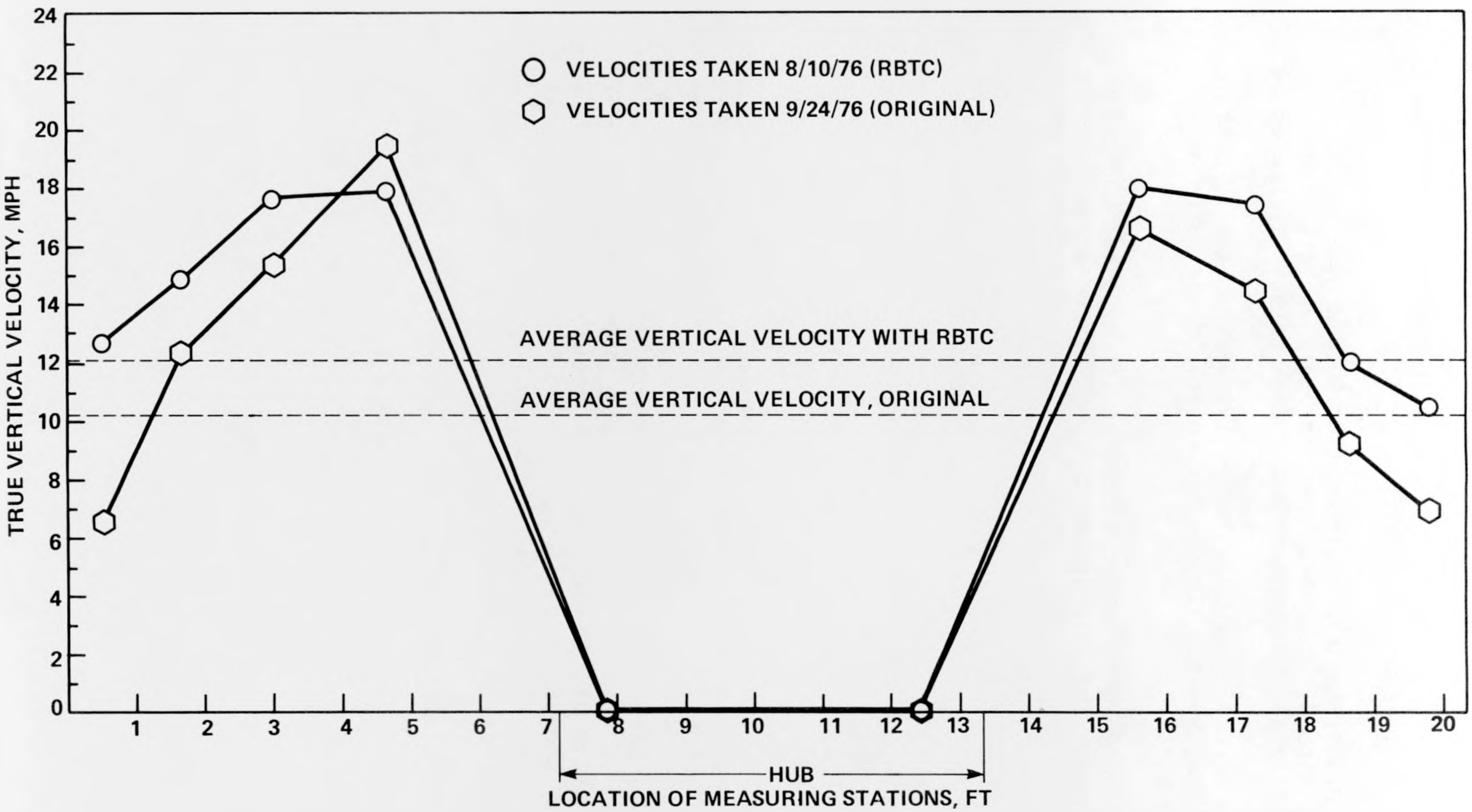

Figure 5

VELOCITY PROFILES OBTAINED FROM MEASUREMENTS ON FAN 3H comparison. As shown by the figure, RBTC does basically two things, both of which improve fan efficiency:

1. By minimizing air recirculation between the blade tips and shroud, the average vertical air veolcity and, therefore, the volumetric air flow rate is increased.

2. The exit air, vertical velocity profile is more uniform so that the air flow is more evenly distributed across the stack.

Fan 4H was performing rather poorly in its original condition as Table V shows. The calculated static efficiencies were in the neighborhood of 33%. Addition of the tip seal improved the static efficiency from about 44% to 89%. Figure 6 was constructed by utilizing the traverse data collected from Fan 4H during Runs 1 and 5. The comments made above concerning Fan 3H are also applicable here.

Since the tip seal reduced the blade tip clearance for both fans to approximately 1/4 inch, it follows that the lower efficiencies of Fan 4H with RBTC are probably due to other factors. For instance, the combined efficiency of the electric motor and gear reducer could be somewhat less than the efficiency of the motor and gear reducer driving Fan 3H. This is possible since both gear reducers were reconditioned prior to testing; however, the motor associated with Fan 3H was reconditioned while the other motor was not. This could help explain the consistently larger power requirement of the motor which drives Fan 4H. There are also strong indications from the tower performance measurements that the water loading is not uniformly distributed within this cell. The east side of the cell which is served by Fan 3H is forced to cool more than one-half of the total water flowing to Cell 2. What effect this might have on the individual fan performances is not complete clear at this time; however, it may help explain the differences.

Resulting Monetary Savings

The above discussion could explain the disparity between the measured efficiencies of Fans 3H and 4H; however, the purpose of this part of the report is to show the improvement in efficiency - especially total efficiency - which can be attributed to RBTC.

The economic benefit derived from the efficiency increase can be determined by calculating the quantity of additional electrical power that would be required to obtain the same increase in air flow. It is obvious from information contained herein that, at the same fan blade angle, RBTC results in increased volumetric air flow rate at the expense of a larger power consumption. An increase in blade angle could also provide an

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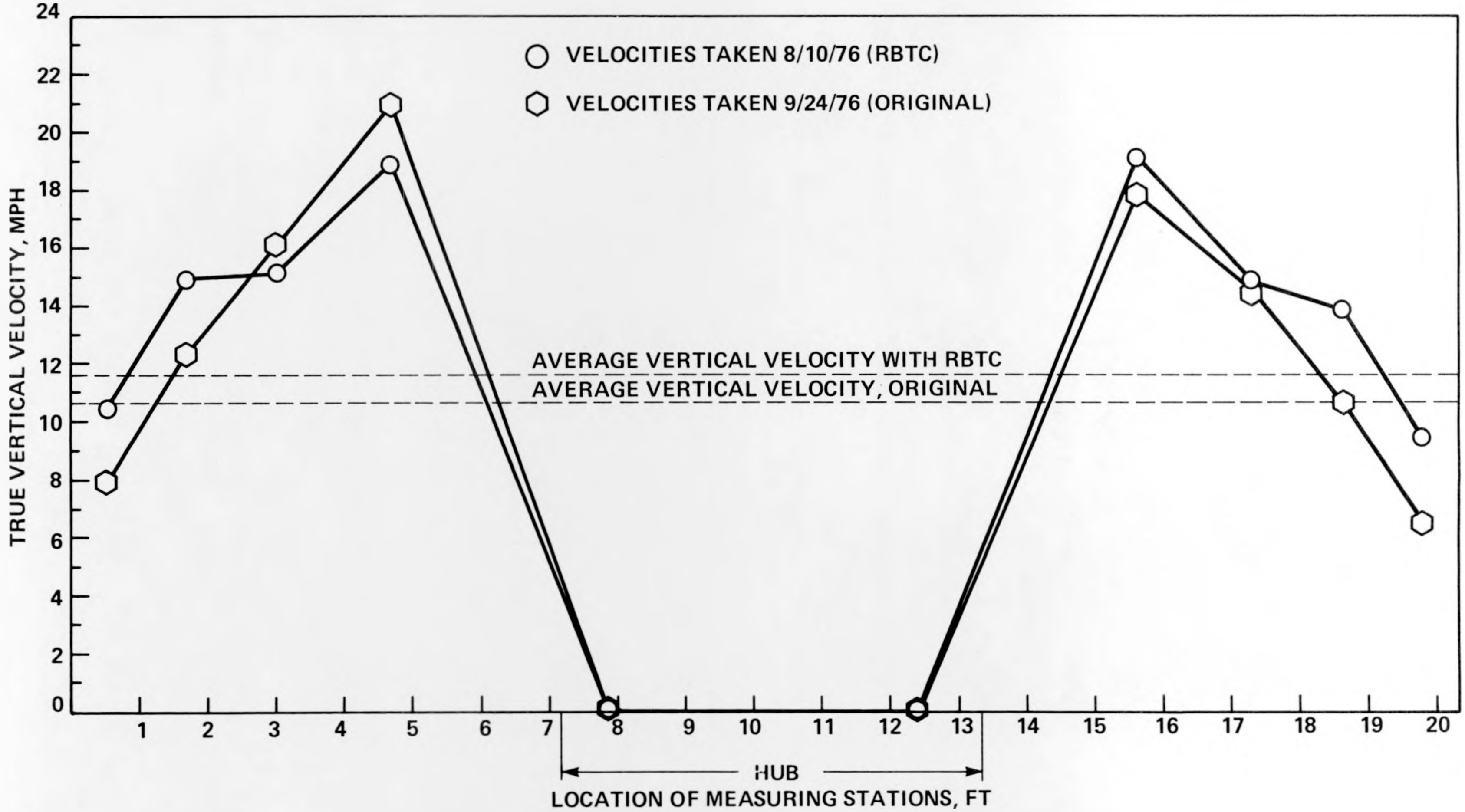

Figure 6

VELOCITY PROFILES OBTAINED FROM MEASUREMENTS ON FAN 4H identical increase in air flow with no need for RBTC[§], but the power required to drive the fan would naturally become larger. In fact it can be shown that, for a change in blade angle, power input and volumetric air flow are approximately related by:

$$\frac{P_{NBA}}{P_{OBA}} = \left(\frac{CFM_{NBA}}{CFM_{OBA}}\right)^3 \tag{5}$$

where:

P_{NBA} = Power Required at New Blade Angle, kw

P_{OBA} = Power Required at Old Blade Angle, kw

CFM_{NBA} = Volumetric Air Flow at New Blade Angle, cfm

CFM_{OBA} = Volumetric Air Flow at Old Blade Angle, cfm

From this expression it is possible to determine the approximate power required to obtain a certain increase in volumetric air flow rate by a change in blade angle provided the power requirement and volumetric air flow rate are known at the original blade angle. By assuming a power cost, it is then possible to calculate a power savings which would result from RBTC, with the following expression:

$$S = 8.76\ C_p\ (P_{NBA} - P_{RBTC}) \tag{6}$$

where:

S = Power Savings, \$/fan-yr

C_p = Cost of Power, mils/kwh

P_{RBTC} = Power Required with RBTC, kw

Table VI shows the power savings calculated by applying Equations 5 and 6 to the results obtained in this investigation. A power cost of 20.0 mils/kwh was assumed. Using a tip seal rather than changing blade

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[§]Such a change would not provide an improvement in air flow distribution as does RBTC.

angles to obtain a given increase in volumetric air flow can result in power savings ranging from \$877 to \$1688 per fan-year. Since the K-892 Tower employs forty-four fans, the total power savings for the entire tower would range from about \$39,000 to about \$74,000 per year based on continuous operation. The current capital investment for polyurethane tip seal installation on the K-892 Tower should range from about \$69,000 to \$86,000 including material and labor charges. Consequently, the total annual savings could be as high as \$1400 dollars per fan-year assuming an interest rate of 10% per year and a ten-year period for recovery of capital.

TABLE VI

SUMMARY OF AVERAGE FAN PERFORMANCE RESULTS

	Fan 3H		Fan 4H	
	RBTC	Original	RBTC	Original
Average Static Fan Efficiency, %	72.05	57.93	54.41	33.43
Average Total Fan Efficiency, %	77.32	62.27	59.17	37.74
Average Air Flow x 10^{-6}, cfm	0.337	0.307	0.330	0.310
Average Power Input to Motor, kw	46.73	42.61	49.93	45.67
Power Savings, \$/fan-yr	1688		877	

Part II: The Effect of Reduced Tip Clearance on Cooling Tower Performance (K-892 Tower)

DESCRIPTION OF MEASURING TECHNIQUES AND TESTING PROCEDURES

Fourteen tests will be discussed here. Six of these tests were conducted with RBTC, while eight were conducted under existing conditions. Data were obtained in both 1975 and 1976. Acquisition of the 1975 data was coordinated by P. A. Jallouk, a former member of Thermal Systems Development, ORGDP. Table VII provides a summary of all tests, along with other pertinent information. A large quantity of measurements were taken during each test. A number of these measurements were discussed in detail in the previous section and will not be repeated here. Certain of the other measurements are needed to determine tower performance. These will be discussed in the following sections.

Air Inlet Wet-Bulb Temperatures at Louvers

The wet-bulb temperature of the air just prior to entering the cooling tower was measured by sixteen 100-ohm copper Resistance Temperature Detectors (RTD's). Eight RTD's were located on the east face of the tower, and eight were located on the west face. Figure 7 is a schematic drawing of the east side of Cell 2H which shows the location of the RTD's. To obtain accurate wet-bulb temperature readings, each RTD was covered with a special type of cylindrical wick constructed from a sponge, Figure 8. Samples of this type of wick were tested by the author under controlled, laboratory conditions and found to give a very accurate indication of wet-bulb temperature. All wicks met the requirements established by the Cooling Tower Institute (CTI).[4]

During testing periods, an air velocity between 950 and 1,050 ft/min was maintained across each wick. This was accomplished by drawing air through each plastic bottle with a variable speed blower, Figure 8. The speed, or rpm, of each blower was controlled by an autotransformer. A distribution system was employed which enabled a single blower to provide air for a total of four plastic bottles. This arrangement is shown schematically in Figure 9. Prior to testing, a hot-wire anemometer was used to detect the air velocity across each wick. The autotransformer setting and, consequently, the blower speeds were adjusted until acceptable air velocities were obtained. The RTD readings were recorded by a Leeds and Northrup scanner-programmer. All RTD's were shielded from direct sunlight.

Table VII

LISTING OF ALL SINGLE-CELL COOLING TOWER TESTS

Test No.	Date	Blade Angle, °	Fan Blade Tip Condition	No. of Traverses Made[a]
1	9/29/75	32	EBTC	4
2	9/30/75	32	EBTC	4
3	10/02/75	32	EBTC	4
4	10/09/75	32	EBTC	4
5	8/10/76	30	RBTC	4
6	8/11/76	30	RBTC	4
7	8/12/76	30	RBTC	4
8	8/13/76	30	RBTC	4
9	8/23/76	30	RBTC	None
10	8/24/76	30	RBTC	None
11	9/17/76	30	EBTC	None
12	9/24/76	30	EBTC	4
13	9/29/76	30	EBTC	4
14	10/01/76	30	EBTC	4

(a) A standard test consisted of two 10-point traverses per fan which resulted in a total of four traverses per cell.

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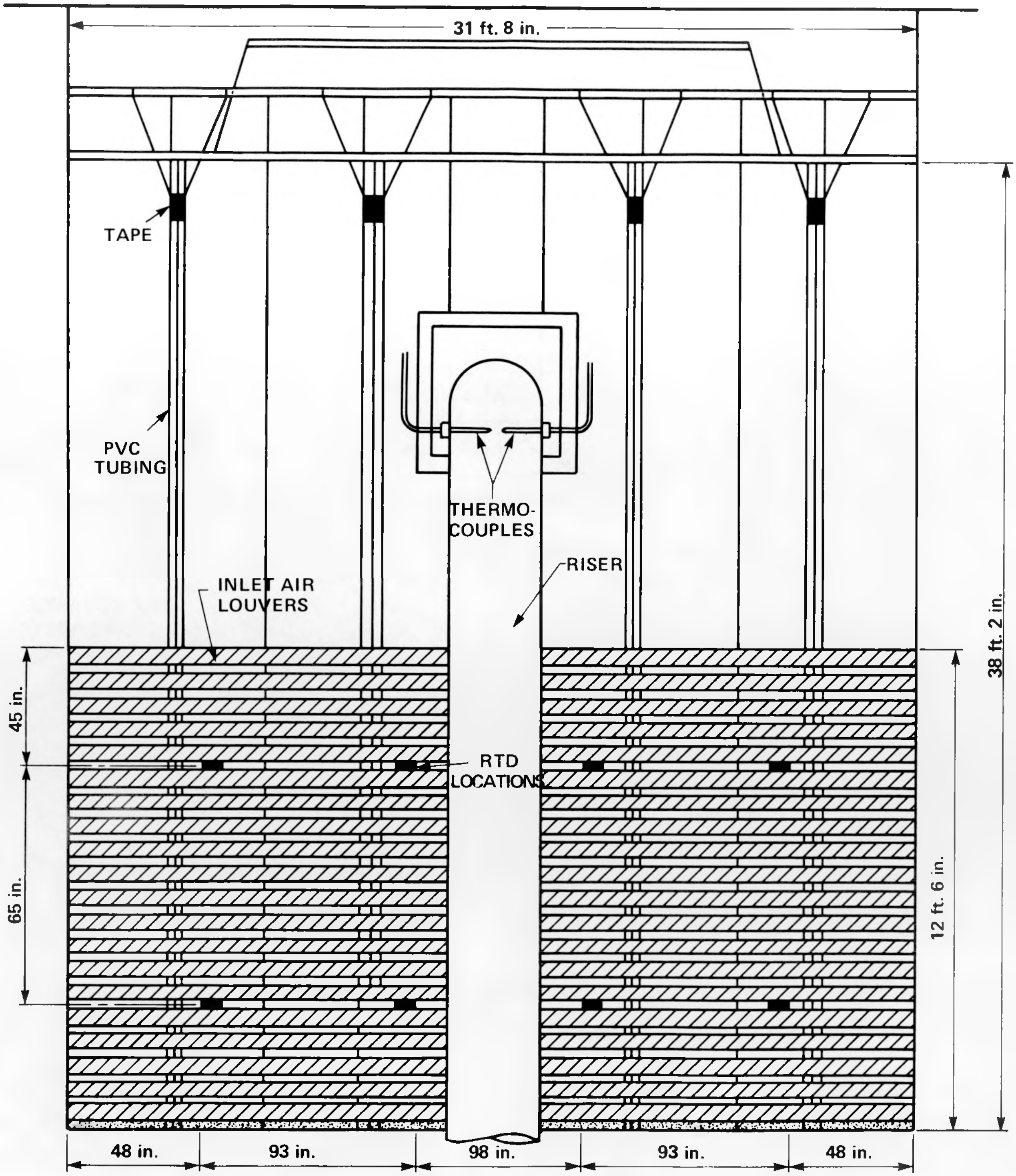

Figure 7

LOCATION OF RESISTANCE TEMPERATURE DETECTORS
AT AIR INLET LOUVERS

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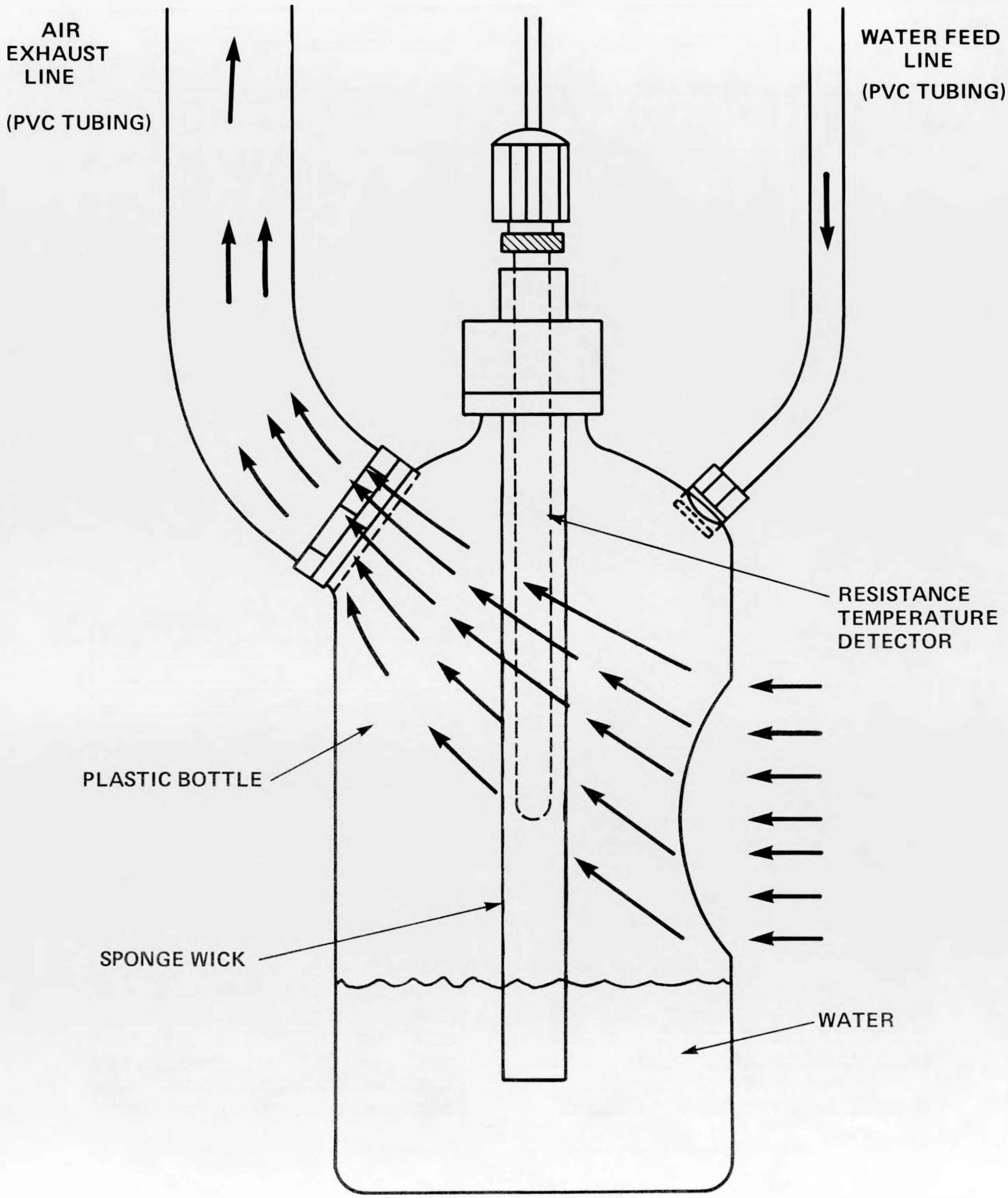

Figure 8
SCHEMATIC OF AIR WET-BULB MEASURING SETUP AT AIR INLET LOUVERS

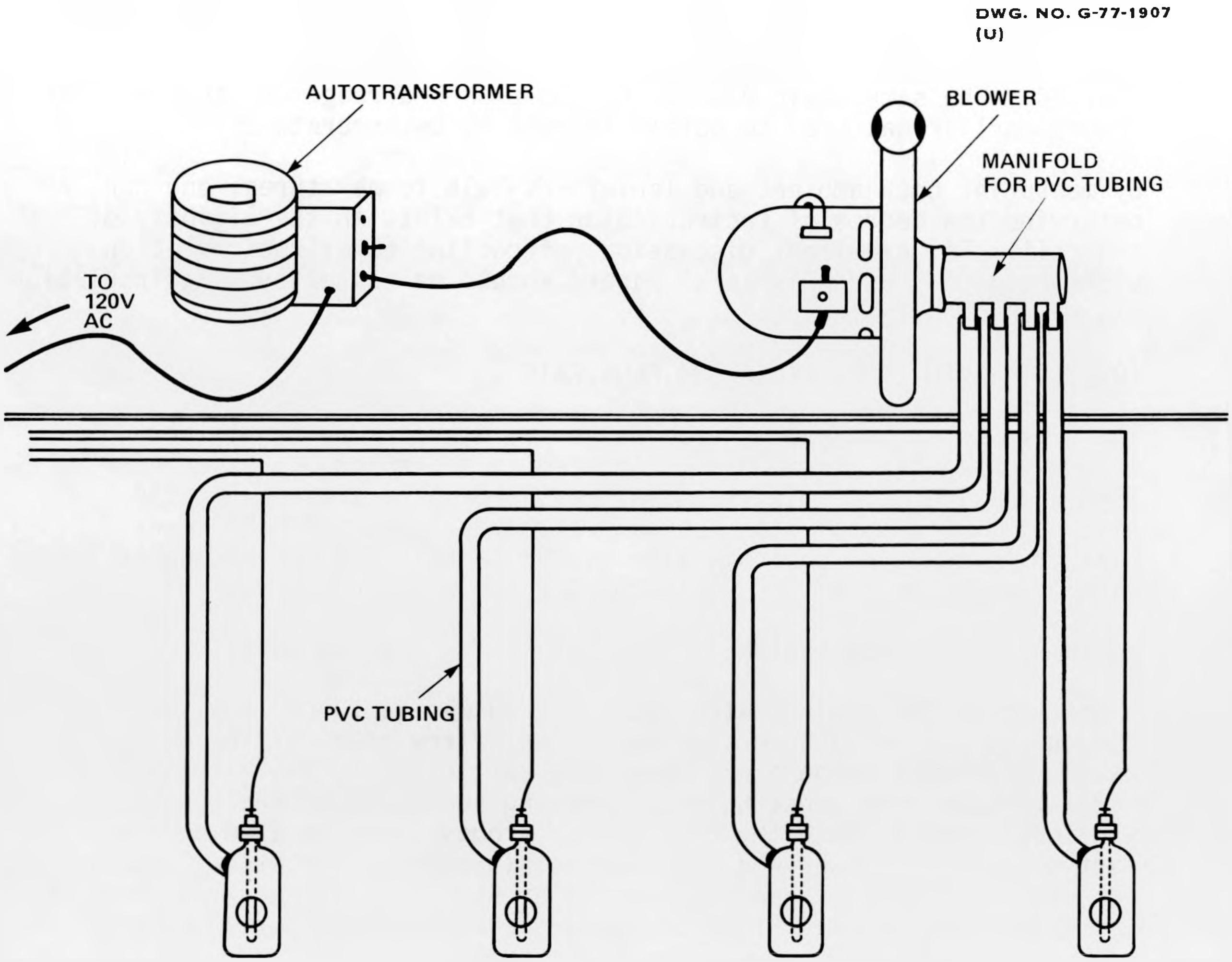

Figure 9
SCHEMATIC OF THE ASPIRATION APPARATUS

AMBIENT AIR INLET WET-BULB AND DRY-BULB TEMPERATURES

Ambient air inlet wet-bulb and dry-bulb temperature measurements were made according to the recommendations of the American Society of Mechanical Engineers (ASME).[2] Measurements were taken at six stations during the tests. The location of each station relative to Cell 2H is shown in Figure 10. The stations were approximately 100 ft from the louvers, and measurements were made some 5 ft above ground level. All temperatures were measured by calibrated mercury thermometers accurate to $\pm$ 0.1°C. The same basic plastic bottle-blower arrangement that was discussed earlier was used to obtain the wet-bulb temperatures.

By measuring both ambient and louver wet-bulb temperatures, one can determine the degree of recirculation that exists in the vicinity of the cell. For excellent discussions of cooling tower recirculation, Lichtenstein's[5] and Reisman's[6] papers should be consulted. Recirculation was minimal on test days.

TOWER-ON WATER TEMPERATURE AND FLOW RATE

The tower-on or hot water temperature was measured by two calibrated copper-constantan thermocouples located in the riser, as shown in Figure 7. The water flow rate was measured by an Ellison hot tap high-pressure Annubar flow element which was located in the riser near the catwalk on the east side of the tower. The thermocouples were accurate to $\pm$ 0.5°F, while the Annubar was accurate to $\pm$5%.

TOWER-OFF WATER TEMPERATURE

A portion of the cooled water from Cell 2H was captured in plywood troughs to obtain an accurate indication of the tower-off or cold water mixed-mean temperature from this particular cell. A total of eight troughs were constructed comprising about 13% of the cross-sectional area beneath the fill. Four troughs were located on the east side of the tower and four were positioned on the west side, as depicted in Figure 11. A 100-ohm copper RTD was positioned in each trough to give an indication of the temperature distribution of the cold water before it entered the tower basin.

This mode of measurement was the most reasonable and the simplest of several alternatives. Unfortunately, there is some uncertainty concerning whether or not the true mixed-mean temperature can result from this measuring technique, since the temperature of a large portion of the water is not detected.

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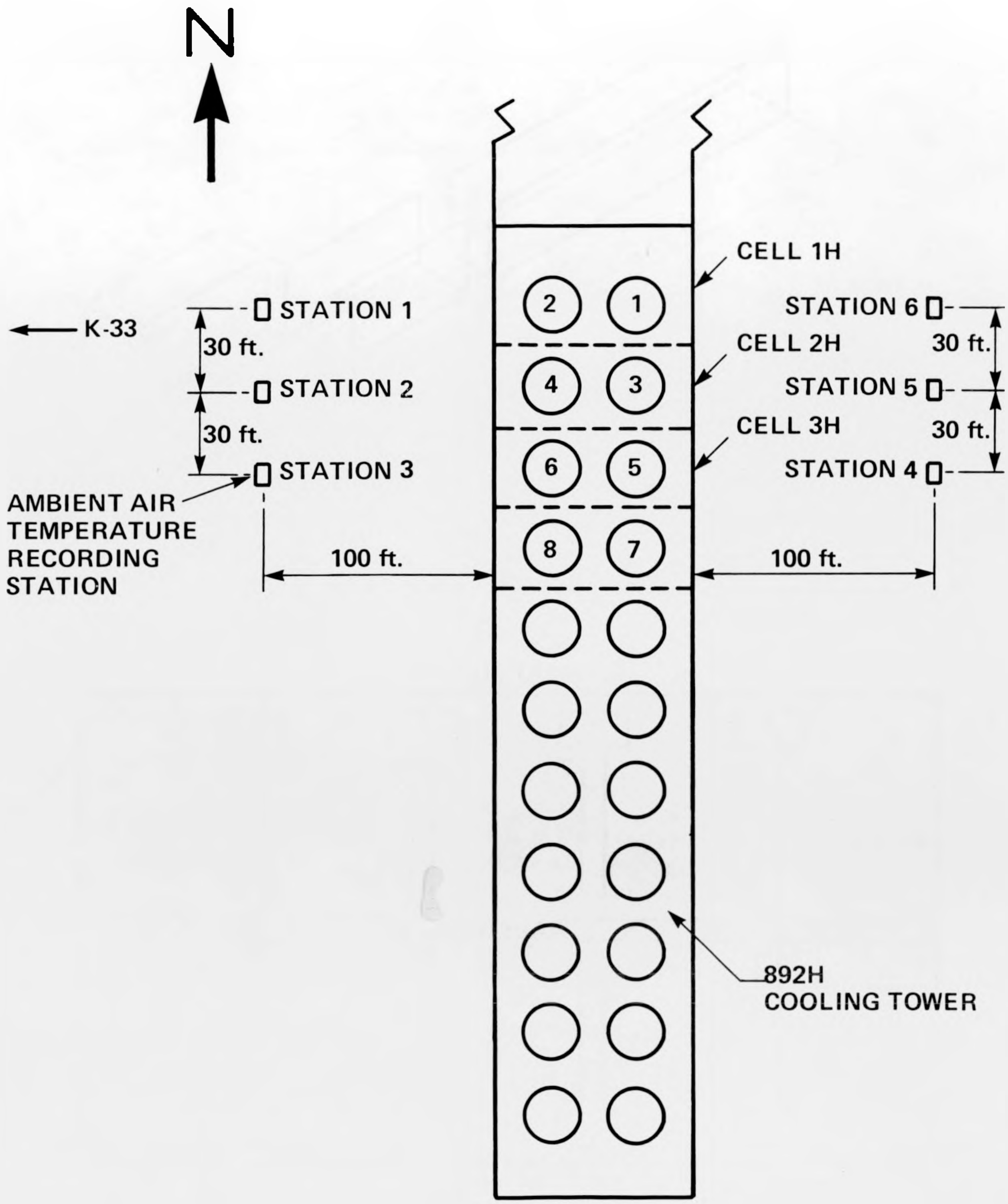

Figure 10

LOCATION OF AMBIENT AIR TEMPERATURE READING
STATIONS RELATIVE TO CELL 2H

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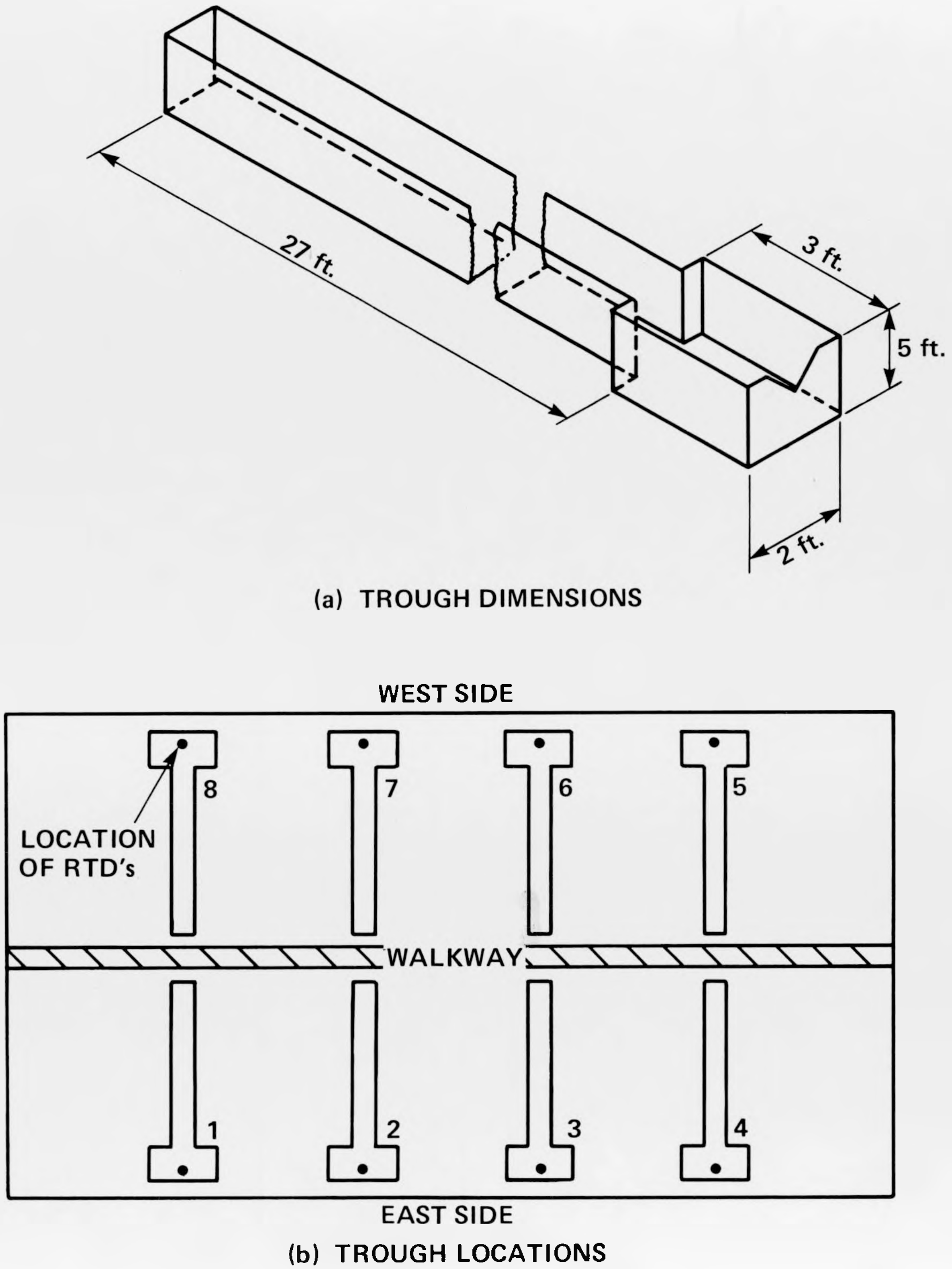

Figure 11
TROUGH DIMENSIONS AND LOCATIONS IN CELL 2H

Testing Procedures

A typical test took the greater part of an entire working day to complete. Normally, work began prior to 8 a.m. and concluded at 3:30 p.m. During the morning hours, two traverses were made on one of the fan stacks. In addition, the pressure differential across each fan stack was measured.

During the morning hours when wet-bulb temperatures were usually low, the Leeds and Northrup scanner-programmer was used at 30-min intervals to scan the inlet wet-bulb temperatures at the louvers, the hot water temperature in the riser, and the cold water temperature in each trough. Although these readings were never used to determine tower performance, they were invaluable as indicators of potential instrumentation problems and measurement difficulties, such as faulty wiring and dry wicks. Many of these problems could be solved during the morning hours so that data collection in the afternoon went smoothly.

From approximately 1 p.m. until 3:30 p.m., the bulk of the tower performance data was acquired. During this period, the following data were obtained at 15-min intervals:

1. Air inlet wet-bulb temperatures at louvers,
2. Air ambient wet-bulb temperature at each station,
3. Air ambient dry-bulb temperature at each station,
4. Tower-on water temperatures,
5. Tower-off water temperature in each trough,
6. Water flow onto Cell 2H, and
7. Power required by fan motors.

In addition to obtaining these data, the second fan stack was traversed twice.

As a point of interest, it should be noted that every attempt was made to ensure that steady state tower conditions had been achieved prior to testing. Furthermore, the two cells adjacent to Cell 2H were operational during all tests to closely approximate actual operating conditions.

DISCUSSION OF RESULTS

GENERAL OBSERVATIONS

Weather Data

A summary of the weather data obtained during each test is given in Table VIII. With the possible exception of Test 3, the wind velocities were normally calm. In no cases were wind velocities outside the guidelines recommended by both CTI[4] and ASME[2]. The ambient dry-bulb temperature varied some 27°F between tests; however, it changed very little during a single test. The weather instrumentation was obtained from and maintained by the National Oceanic and Atmospheric Administration (NOAA).

Table VIII

SUMMARY OF WEATHER DATA OBTAINED DURING EACH TEST

Test No.	Date	Dry-Bulb Temperature, °F	Relative Humidity, %	Average Wind Speed, mph	Maximum Wind Speed, mph	General Direction
1	9/29/75	70-74	46-50	4.0	11.0	East
2	9/30/75	70-76	46-65	3.5	9.0	North
3	10/02/75	60-64	36-40	9.0	15.5	Southwest
4	10/09/75	70-76	42-60	4.0	11.0	East
5	8/10/76	80-83	46-50	5.0	9.0	Southeast
6	8/11/76	84-87	53-57	7.0	9.0	West
7	8/12/76	88-90	48-52	5.0	9.0	Southwest
8	8/13/76	86-90	45-49	5.0	9.0	West
9	8/23/76		Malfunctioning Instrumentation			
10	8/24/76		Malfunctioning Instrumentation			
11	9/17/76	77-81	42-52	6.0	11.0	Northeast
12	9/24/76	78-80	38-42	4.0	6.0	Southwest
13	9/29/76	68-72	78-82	5.0	6.0	Southwest
14	10/01/76	69-73	40-44	7.0	11.0	North

Water Flow Rate Range

The range of water flow rates covered by the tests (approximately 4,000 to 6,000 gpm) was not as large as originally planned. Utilities Operations controlled this parameter by varying the number of cooling tower cells in operation during the test days. Since their primary objective was to satisfy the plant heat removal requirements, they were unable to provide the flexibility needed for changing water flows, especially on the warmer days. Due to the limited range, extrapolation to Cascade Uprating Program (CUP) operating conditions was required, as detailed later. This does not appear to be a serious limitation; however, uncertainty does exist.

Volumetric Air Flow Rate

For each of the 11 tests in which fan stack traverses were made, the volumetric flow rate of the exhaust air was determined by the method outlined in Part I. No traverses were made during Tests 9, 10, and 11 because of instrumentation difficulties during this period.

During the period between Tests 8 and 12, the vane anemometer and associated instrumentation were returned to the supplier, R. M. Young Company, for repair and recalibration as an entire unit. The results of this calibration agreed with the initial calibration performed by the Company before the instrumentation was purchased. Thus, it can be stated with confidence that the performance of the vane anemometer changed very little during the testing period.

Table IX shows a comparison of the volumetric air flow rates and average air outlet temperatures obtained during each test. The air flow rate shown for Tests 9 and 10 is the arithmetic average of the flow rates obtained from Tests 5 through 8, while the air flow rate shown for Test 11 is the arithmetic average of that obtained from Tests 12 through 14. The average air outlet temperature was determined by weighing the individual measurements with the air velocity at the same location. Since traverses were not conducted during Tests 9 through 11, air outlet temperatures could not be recorded.

Tower-Off Water Temperature

The tower-off water temperatures was measured with RTD's located in troughs beneath the fill in Cell 2H. Considering the east and west sides of the tower separately, variations of from 5 to 10°F between individual trough readings on the same side of the tower were not uncommon. This was probably due to poor water flow distribution. Normally, the average water-off temperature on the west side was some 5 to 10°F colder than that of the east side. This was also caused, in all likelihood, by poor water flow distribution. For purposes of analysis, the arithmetic average of the eight readings was taken as the best indication of water-off temperature for the cell during each time interval. The uncertainty associated with the measuring technique was discussed earlier. A daily averaged cold water temperature was obtained by averaging the readings at each time interval.

TABLE IX

COMPARISON OF THE VOLUMETRIC AIR FLOW RATES AND THE AIR OUTLET TEMPERATURES OBTAINED FROM THE TEST DATA

Test No.	Total Volumetric Air Flow Rate, cfm	Average Air Outlet Temperature, °F
1	700,190	108.16
2	739,500	108.31
3	704,600	98.88
4	756,400	115.94
5	679,200	98.85
6	659,800	100.60
7	663,600	99.39
8	664,400	99.22
9	666,750	---
10	666,750	---
11	617,300	---
12	596,900	106.37
13	614,200	102.42
14	640,800	103.32

Ambient Wet-Bulb Temperature Determination

Ambient wet-bulb temperatures were determined for each of the 15-min time intervals. These temperatures were then averaged to obtain a daily average. For a given time interval, the wind speed and direction as well as the temperatures on each side of the tower were noted. When possible, readings taken on the *windward* or *upwind* side of the tower were averaged to obtain each instantaneous reading. In nearly every case, the windward readings were both more consistent and lower than the readings obtained from the *leeward* or *downwind* side of the tower. When the wind direction was predominantly either north or south so that windward readings could not be obtained, an average of all 6 readings was used for that time interval. On the days when weather data were not recorded, the consistency and magnitude of the temperatures were used to estimate the true ambient wet-bulb temperature for each interval.

Inlet Wet-Bulb Temperatures at Louvers

Sixteen RTD's at two elevations were monitored in order to determine these temperatures. Normally, all 16 readings were averaged to give the best indication of the inlet wet-bulb temperature at the louvers. By using this procedure, it was reasoned that the full effect of recirculation would be reflected in the measurements. Reisman and Ovard[7] addressed this problem as it related to crossflow towers in a 1972 publication. In a few cases, one or more readings were discarded since they indicated potential wick dryness.

TOWER PERFORMANCE RESULTS

General

The best method of presenting cooling tower performance data is in the from of graphs showing the tower characteristic, KaV/L, as a function of the ratio of water to air mass flow rate, L/G. Discussion of the required assumptions and development of the necessary equations to determine these parameters are given in publications by Baker and Shryock[8] and by Fraas.[9] Normally, K is referred to as the overall unit mass transfer conductance between saturated air at the water temperature and the main air stream, a is the area of the water interface, V is the active tower volume, L is the mass flow rate of the water, and G is the mass flow rate of the air.

An improved method for evaluating KaV/L and L/G has recently been developed by Cross, et al.[10] at ORGDP. The computer program resulting from this effort was used an as aid in analyzing the data acquired during the experimental program.

Since the purpose of this report is to evaluate the effect of RBTC on tower performance, the experimental data were separated into two groups--those which represented the effect of RBTC (Tests 5 through 10) and those which depicted existing conditions (Tests 1 through 4 and 11 through 14). Only daily averaged data were considered.

Summaries of the performance results obtained from the computer program are given in Tables X and XI.

For the most part, test dates were selected with several factors in mind. First, weather conditions such as wind velocity and precipitation were carefully considered to minimize potential recirculation. Furthermore, testing was done during the summer months so that actual ambient, wet-bulb temperatures would approach the design value of 78°F. In most cases these conditions were closely approximated; however, some tests were run on relatively cold days, and wind conditions were not always ideal. For the most part, good conditions did exist and little or no recirculation was evident. This is reflected in Tables X and XI where the performance based on inlet wet-bulb temperature differs by only a few percentage points from the performance based on ambient wet-bulb temperatures. There is a definite trend toward higher performance with inlet wet-bulb temperatures as would be expected (Tests 1 through 8, 10, and 14). The results from the remaining tests seem to invalidate this premise; however, in all cases the two wet-bulb temperatures (inlet and ambient) are in the neighborhood of 0.5°F apart. Such a small deviation can be explained by changing wind conditions and it is safe to say that little or nor recirculation was present on those test dates.

Effect of RBTC on Tower Performance

Since test conditions were not comparable, the information in Tables 10 and 11 does not provide a complete evaluation of the merits of RBTC. This evaluation is best done by fitting an equation to the performance data using the least squares method. Because of the relatively small range of L/G data available, this was done in two steps. First, equations of the form:

$$KaV/L = A(L/G)^n, \tag{7}$$

where A and n are constants determined by the fitting procedures, were fit to each of four sets of data (RBTC - inlet wet-bulb, RBTC - ambient wet-bulb, EBTC - inlet wet-bulb, EBTC - ambient wet-bulb). The comparison plots are shown in Figures 12 through 15. These plots also contain a two-standard deviation (2σ), Monte-Carlo error band (developed by the Computer Sciences Division, ORGDP). The resulting equations are given below:

For Set 1 (RBTC - inlet wet-bulb)

$$KaV/L = 2.011\ (L/G)^{-0.973} \tag{8}$$

For set 2 (RBTC - ambient wet- bulb)

$$KaV/L = 1.986\ (L/G)^{-1.015} \tag{9}$$

TABLE X

SUMMARY OF TOWER PERFORMANCE RESULTS
BASED ON INLET WET-BULB TEMPERATURES AT LOUVERS

Test No.	L/G	KaV/L	Calculated Temperature of Exit Air, °F	Measured Wet-Bulb Temperature of Inlet Air, °F
1	1.206	1.984	116.34	66.46
2	1.122	1.710	113.70	67.54
3	0.816	2.128	104.31	54.44
4	1.319	1.338	117.21	67.71
5	1.096	1.773	106.24	67.69
6	1.149	1.957	108.06	72.82
7	0.962	2.056	106.28	74.51
8	1.010	2.007	104.97	71.86
9	1.210	1.479	109.36	72.26
10	1.212	1.672	108.28	74.88
11	1.155	1.714	109.56	62.04
12	1.260	1.200	109.39	61.92
13	1.351	1.495	112.40	65.07
14	1.208	1.575	109.25	58.75

TABLE XI

SUMMARY OF TOWER PERFORMANCE RESULTS
BASED ON AMBIENT WET-BULB TEMPERATURES

Test No.	L/G	KaV/L	Calculated Temperature of Exit Air, °F	Measured Ambient Wet-Bulb Temperature of Inlet Air, °F
1	1.202	1.820	115.58	64.10
2	1.119	1.625	113.15	65.96
3	0.813	1.938	103.22	51.20
4	1.315	1.270	116.54	65.65
5	1.095	1.731	105.98	67.09
6	1.147	1.880	107.70	72.03
7	0.962	2.041	106.21	74.38
8	1.010	1.989	104.89	71.70
9	1.212	1.527	109.71	73.05
10	1.211	1.632	108.05	74.38
11	1.156	1.740	109.73	62.52
12	1.262	1.235	109.81	63.10
13	1.351	1.502	112.45	65.22
14	1.208	1.574	109.24	58.72

For Set 3 (EBTC - inlet wet-bulb)

$$KaV/L = 1.881\ (L/G)^{-0.973} \qquad (10)$$

For Set 4 (EBTC - ambient wet-bulb)

$$KaV/L = 1.815\ (L/G)^{-1.015} \qquad (11)$$

Equations (10) and (11) were developed by constraining the exponents to the values shown. Uncertainty in the data obtained during 1975 was the determining factor which led to this decision. Statistically superior fits also resulted from this procedure. From Figures 12 through 15 the degree of agreement between the equations and the experimental data can be considered good, as borne out by the large number of data points which lie within the Monte-Carlo error band.

Some interesting features result when Equations (8) through (11) are applied to the proposed CUP design conditions. The design conditions are summarized in Table XII while the results from applying the equations

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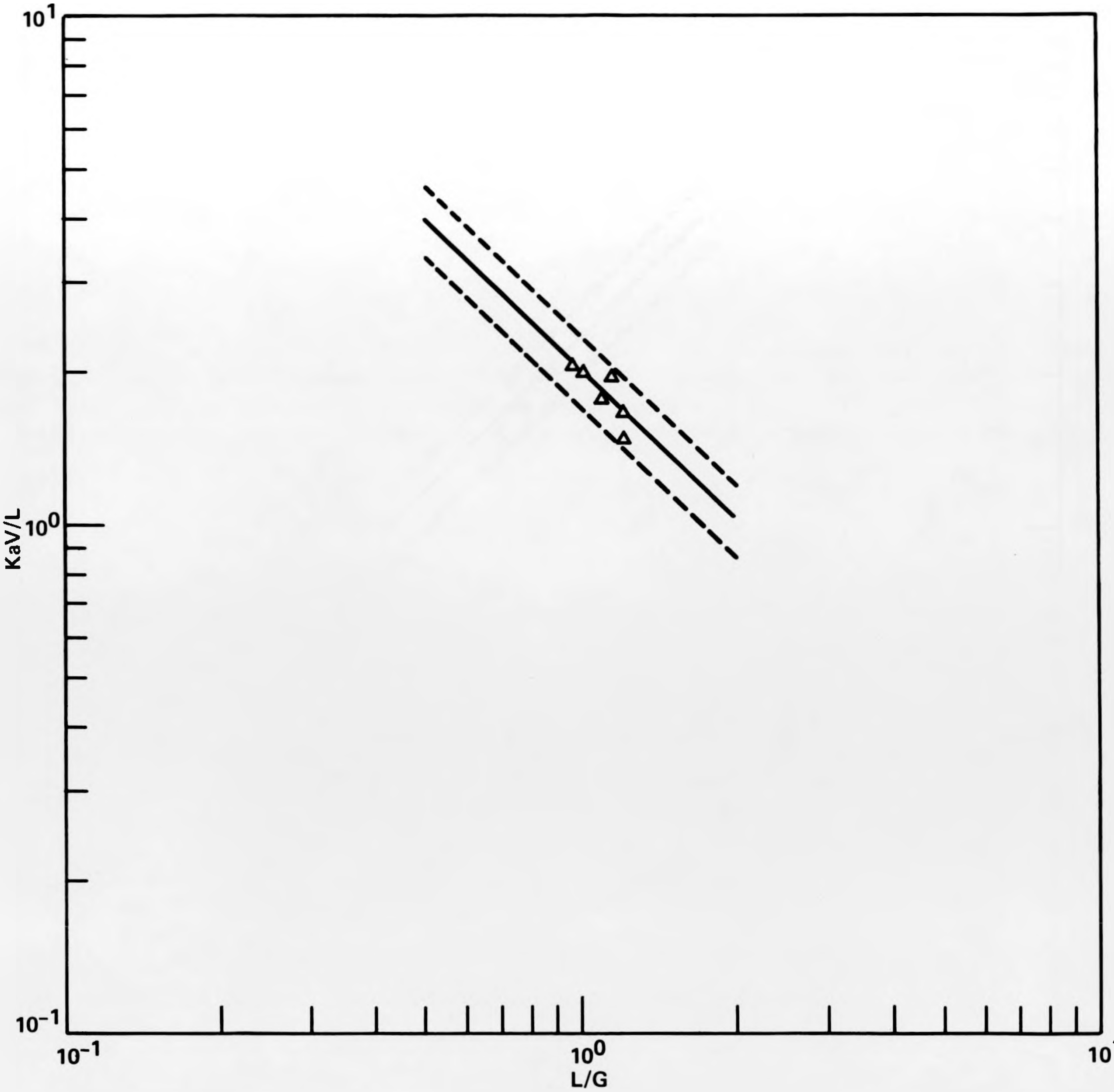

Figure 12

PERFORMANCE OF CELL 2H BASED ON INLET WET-BULB TEMPERATURE, WITH RBTC

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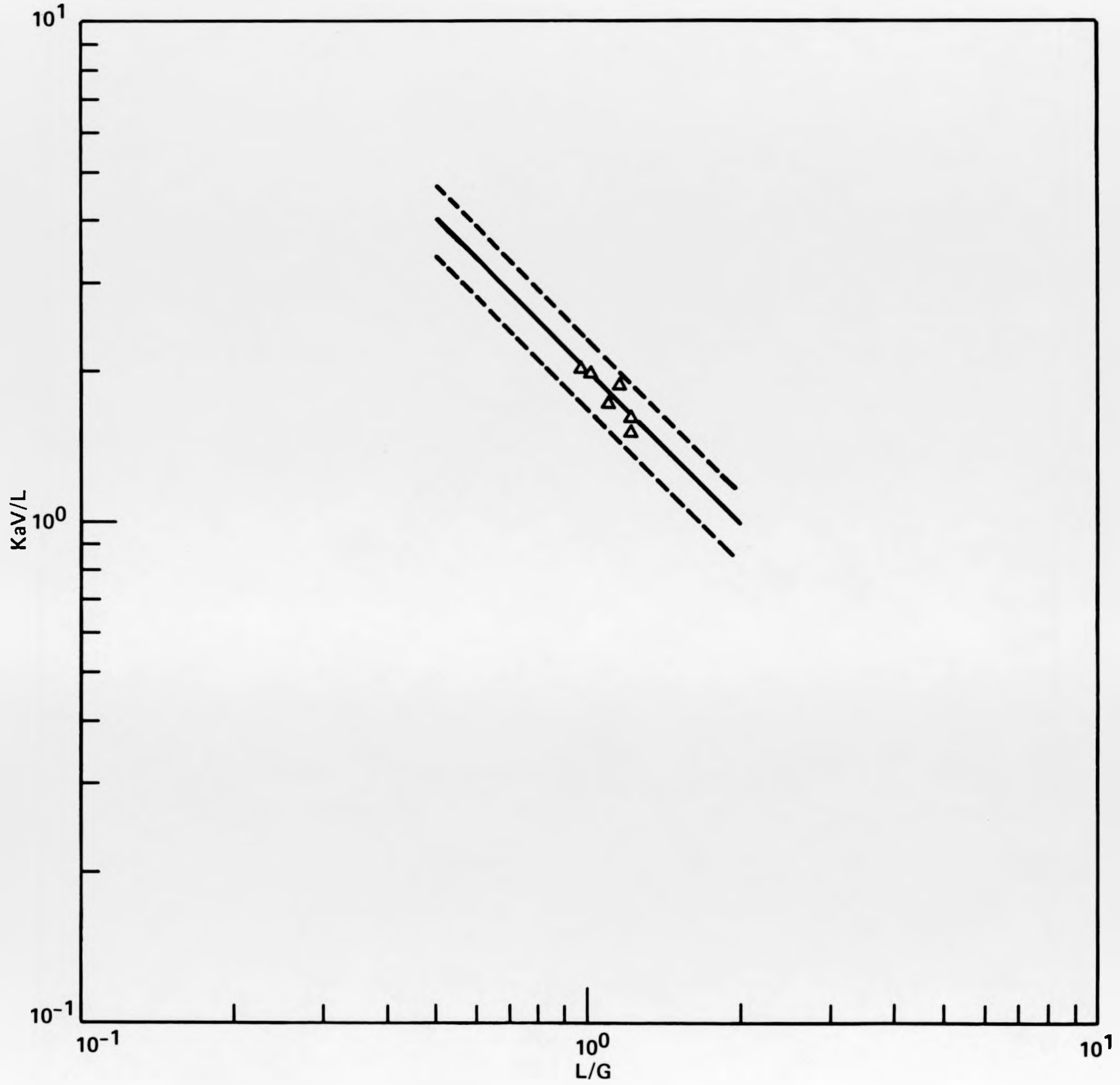

Figure 13
PERFORMANCE OF CELL 2H BASED ON AMBIENT WET-BULB TEMPERATURE, WITH RBTC

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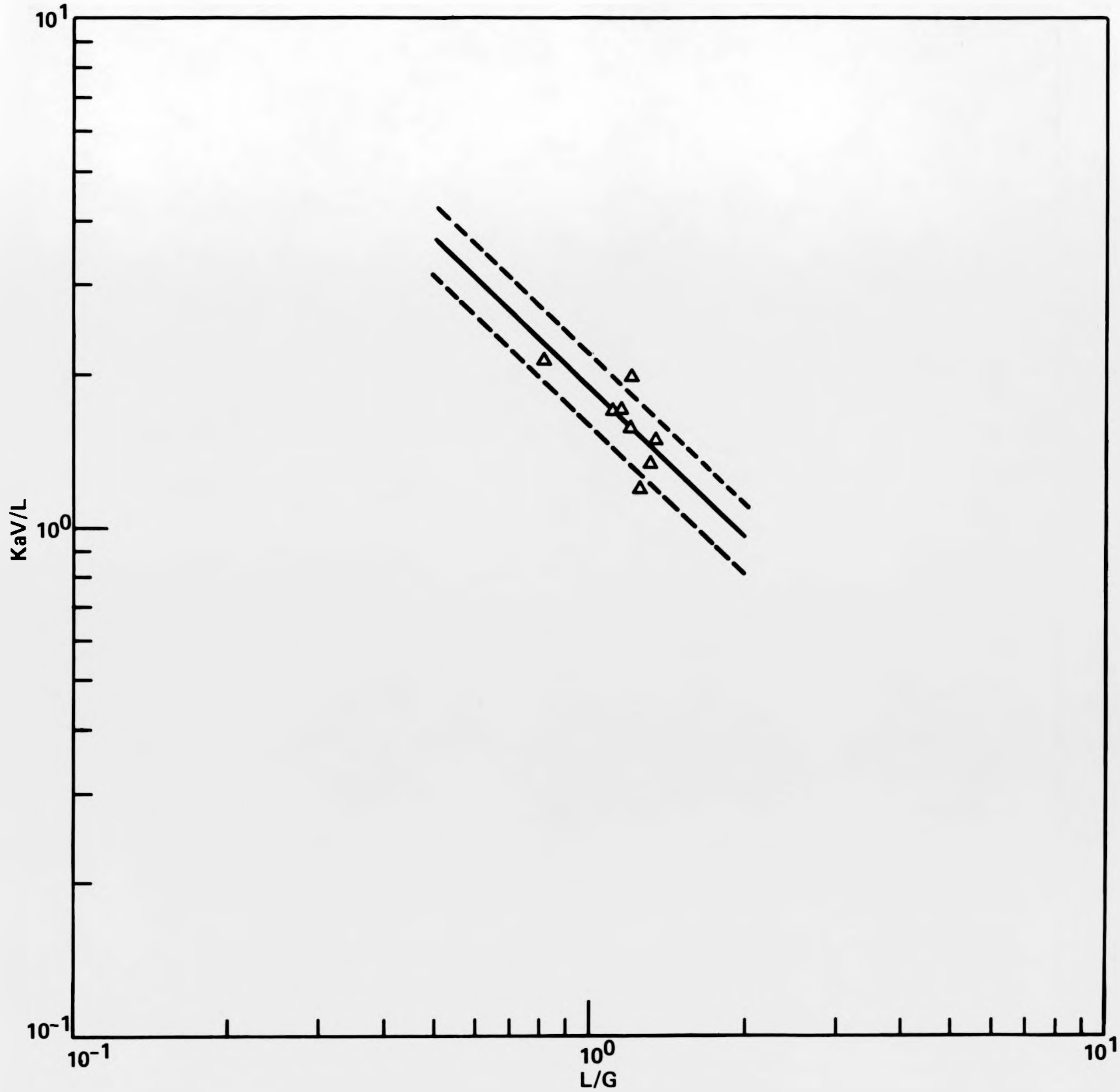

Figure 14
PERFORMANCE OF CELL 2H BASED ON INLET WET-BULB TEMPERATURE, WITHOUT RBTC

DWG. NO. G-77-1913
(U)

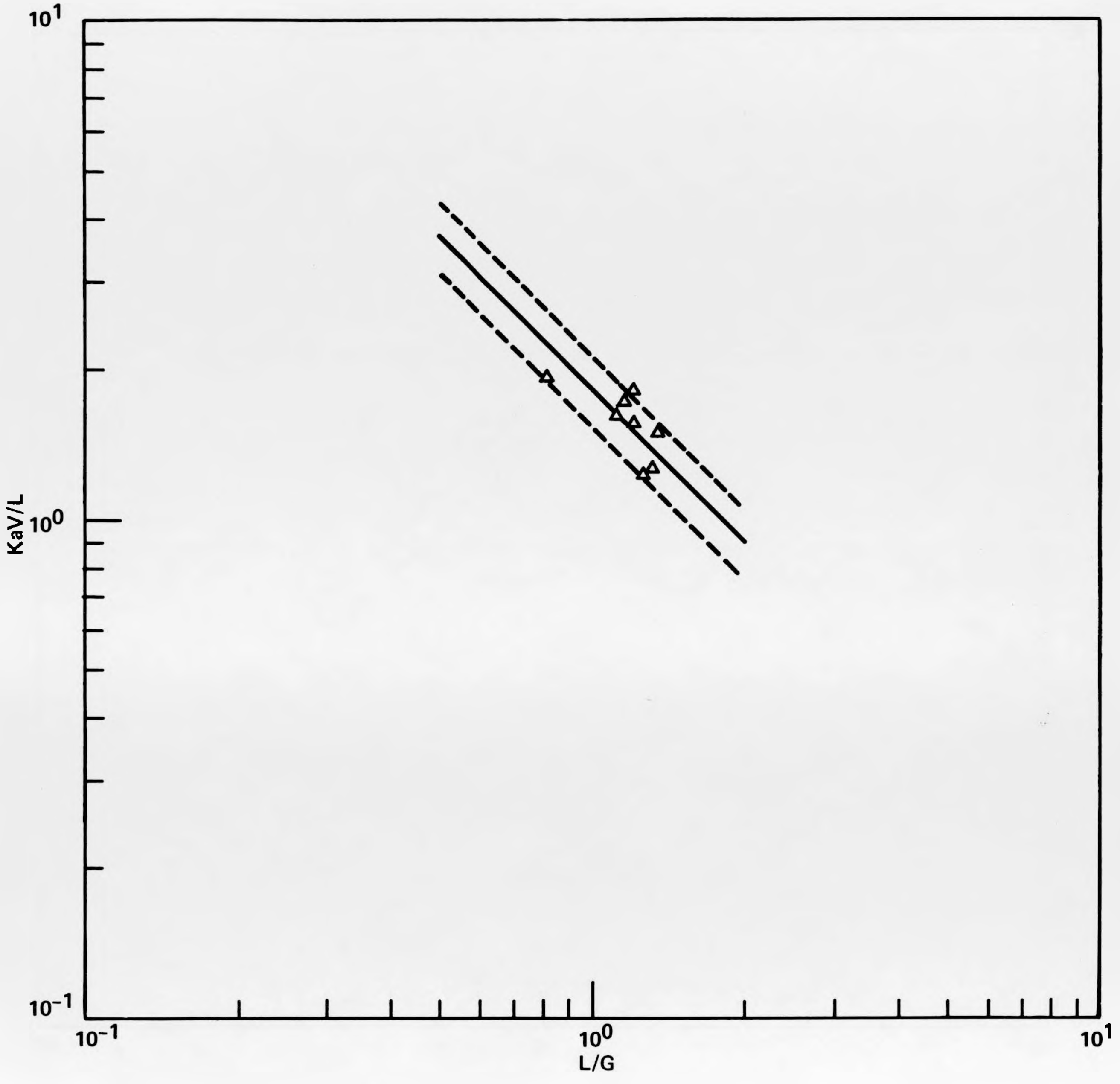

Figure 15
PERFORMANCE OF CELL 2H BASED ON AMBIENT WET-BULB TEMPERATURE,
WITHOUT RBTC

TABLE XII

PROPOSED CASCADE UPRATING PROGRAM
DESIGN CONDITIONS FOR K-892 TOWER

Water Flow Rate, gpm/cell	7,100
Inlet Wet-Bulb Temperature, °F	81
Tower-On Water Temperature, °F	140
Tower-Off Water Temperature, °F	95
L/G Ratio With RBTC[a]	1.442
L/G Ratio With EBTC[a]	1.575

[a]These parameters are based on the same water loading rate.

are given in Table XIII. It may be concluded from these tables that RBTC can result in an improvement in tower performance ranging from 16.5 to 19.7% for a fixed water mass flow rate. This is believed to be due to: (1) improved air flow distribution resulting in more uniform water cooling, and (2) a net increase in the mass of air flowing through the tower per unit time.

TABLE XIII

EFFECT OF RBTC ON TOWER PERFORMANCE
FROM EQUATIONS (8) THROUGH (11)

Based on Inlet Wet-Bulb Temperatures	
KaV/L With RBTC From Equation (8)	1.408
KaV/L With EBTC From Equation (10)	1.209
Percentage Increase Due to RBTC, %	16.5
Based on Ambient Wet-Bulb Temperatures	
KaV/L With RBTC From Equation (9)	1.370
KaV/L With EBTC From Equation (11)	1.145
Percentage Increase Due to RBTC, %	19.7

Earlier in this report, it was stated that the experimental data considered were obtained over a relatively small L/G range and that extrapolation would be required for CUP conditions. It is therefore possible that the equations developed previously do not truly represent tower performance at values of L/G approaching CUP conditions. In other words, if experimental data could have been collected for L/G values ranging from approximately 0.75 to 2.0, then the resulting equations might have possessed different A and n values. To evaluate this possibility, a second step of data analysis was undertaken. In this step, an equation of the same form as Equation (7) but with n constrained to a value of -0.60 was used. This value of n was selected because it is representative of a large quantity of cooling tower performance data as pointed out in the paper by Baker and Shryock and in a paper by Kelly and Swenson[11]. The following relationships resulted:

For Set 1 (RBTC - inlet wet-bulb)

$$KaV/L = 1.962\ (L/G)^{-0.60} \tag{12}$$

For Set 2 (RBTC - ambient wet-bulb)

$$KaV/L = 1.928\ (L/G)^{-0.60} \tag{13}$$

For Set 3 (EBTC - inlet wet-bulb)

$$KaV/L = 1.826\ (L/G)^{-0.60} \tag{14}$$

For Set 4 (EBTC - ambient wet-bulb)

$$KaV/L = 1.752\ (L/G)^{-0.60} \tag{15}$$

The experimental data are plotted along with these equations in Figures 16 through 19. The 2σ Monte-Carlo error band also appears on the graphs. The degree of agreement between data and equation still appears to be quite good.

Using Equations (12) through (15), Table XIV was constructed which gives another indication of the effect of RBTC on tower performance. The appropriate L/G ratios from Table VI were used in the calculations.

Based on information in this table, RBTC can result in a 13.3 to 16.0% improvement in tower performance at the same water loading rate.

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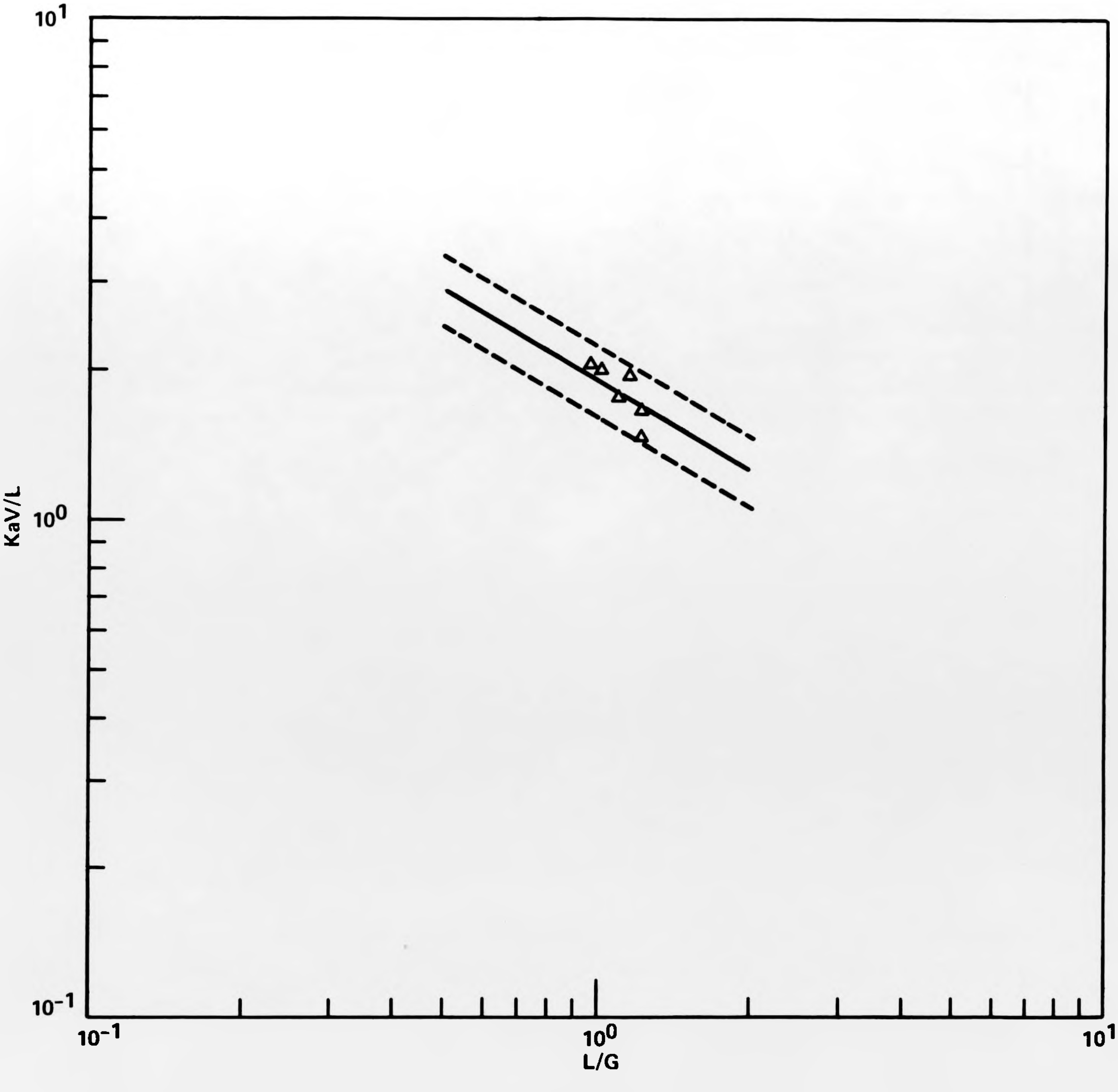

Figure 16
PERFORMANCE OF CELL 2H BASED ON INLET WET-BULB TEMPERATURE, WITH RBTC, n = –0.60

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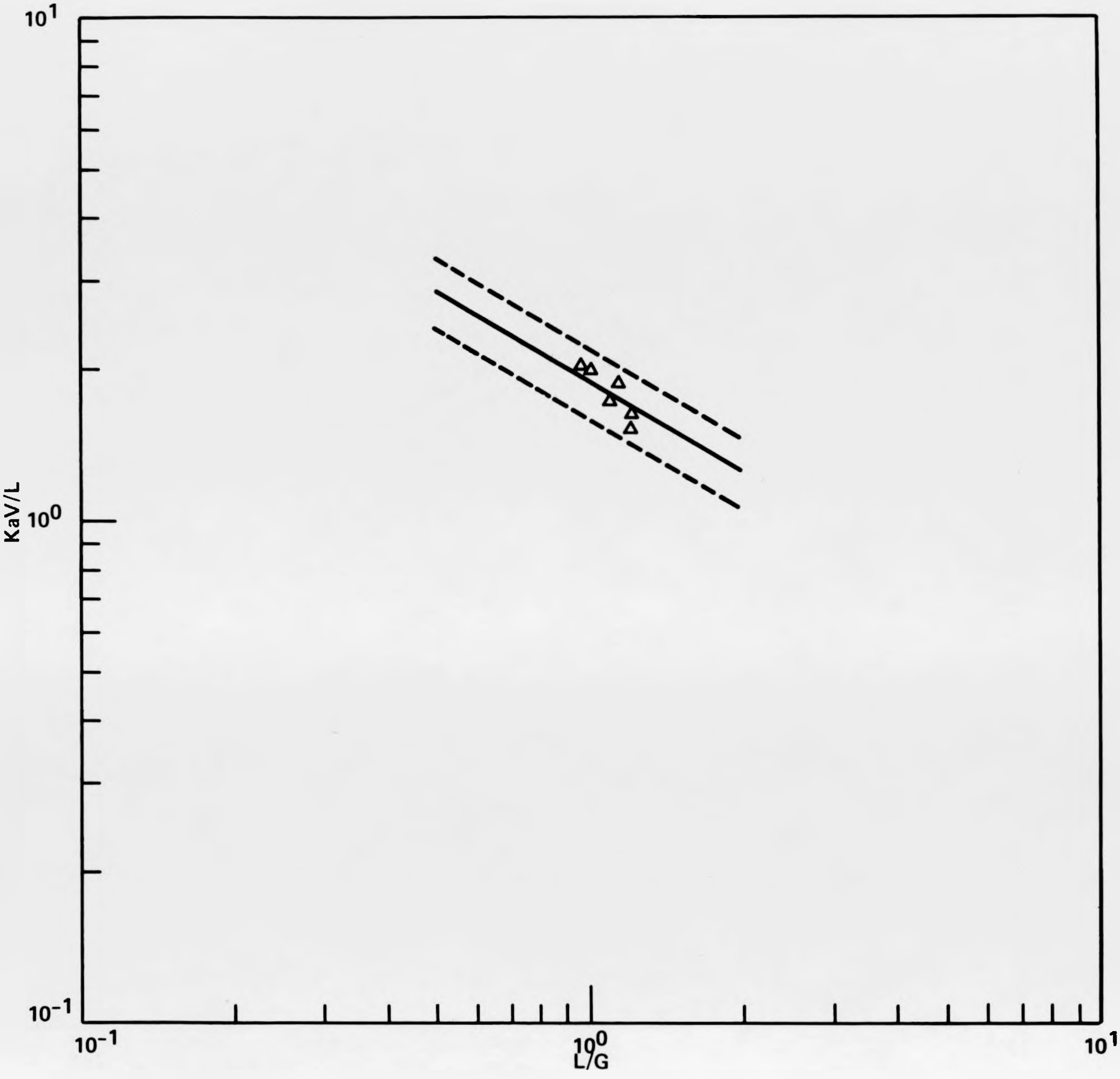

Figure 17
PERFORMANCE OF CELL 2H BASED ON AMBIENT WET-BULB TEMPERATURE, WITHOUT RBTC, n = –0.60

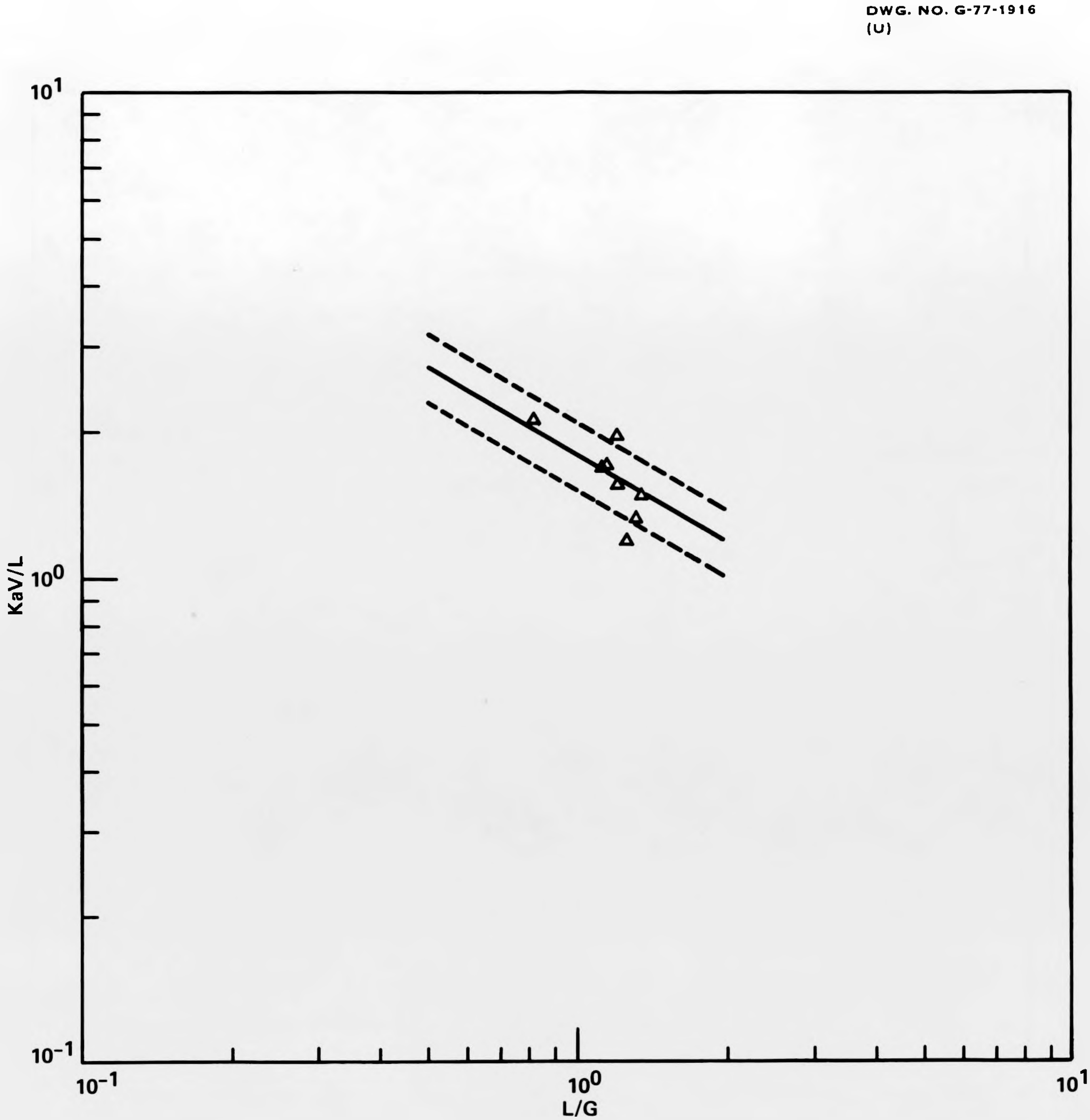

Figure 18

PERFORMANCE OF CELL 2H BASED ON INLET WET-BULB TEMPERATURE, WITHOUT RBTC, n = −0.60

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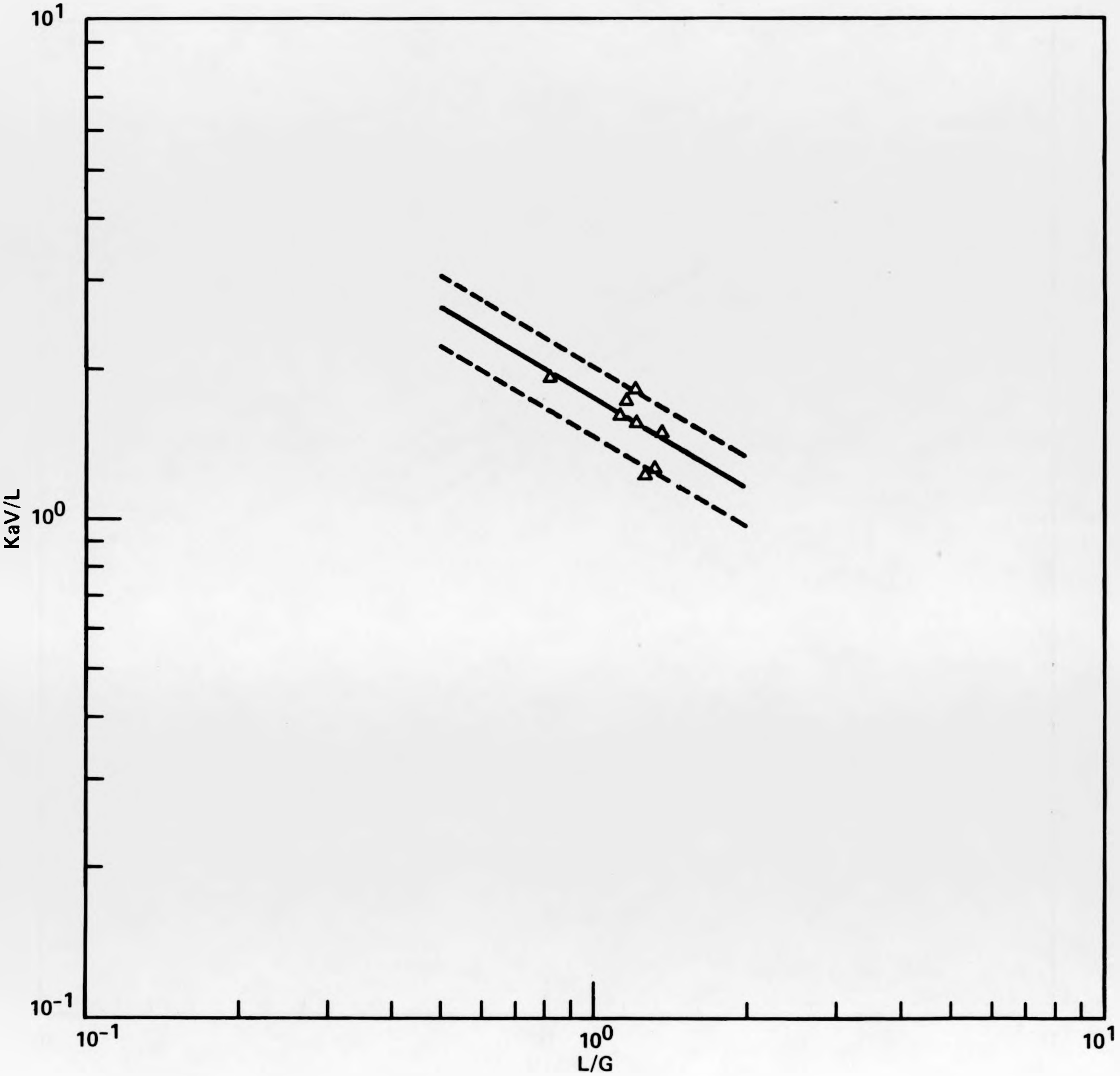

Figure 19
PERFORMANCE OF CELL 2H BASED ON AMBIENT WET-BULB TEMPERATURE,
WITHOUT RBTC, n = −0.60

TABLE XIV

EFFECT OF RBTC ON TOWER PERFORMANCE FROM EQUATIONS (12) THROUGH (15)

Based on Inlet Wet-Bulb Temperatures	
KaV/L With RBTC From Equation (12)	1.575
KaV/L With EBTC From Equation (14)	1.390
Percentage Increase Due to RBTC, %	13.3
Based on Ambient Wet-Bulb Temperatures	
KaV/L With RBTC From Equation (13)	1.548
KaV/L With EBTC From Equation (15)	1.334
Percentage Increase Due to RBTC, %	16.0

Figure 20 was constructed as a visual aid in quantifying the merits of RBTC. In the figure, Equations (9), (11), (13), and (15) are plotted to show the performance improvement resulting from RBTC.

CONCLUSIONS

Based on the results of this experimental program several conclusions are evident. They are listed below:

1. By minimizing air recirculation between the blade tips and shroud at a given blade angle, RBTC causes an increase in volumetric air flow ranging from 6.4% to 9.6%.

2. For a given blade angle, RBTC causes the amount of power required to drive fan motors to increase from 9.3% to 9.7%.

3. RBTC results in increased total fan efficiencies averaging from 24% on Fan 3H to 56% on Fan 4H.

4. Installation of tip seals instead of changing blade angles to obtain a given increase in volumetric air flow can result in an attractive annual cost savings as high as 1400 $/Fan-yr as well as reduced power consumption.

5. At a given blade angle RBTC causes increased cooling capability ranging from 13.3% to 19.7% at a fixed water flow rate.

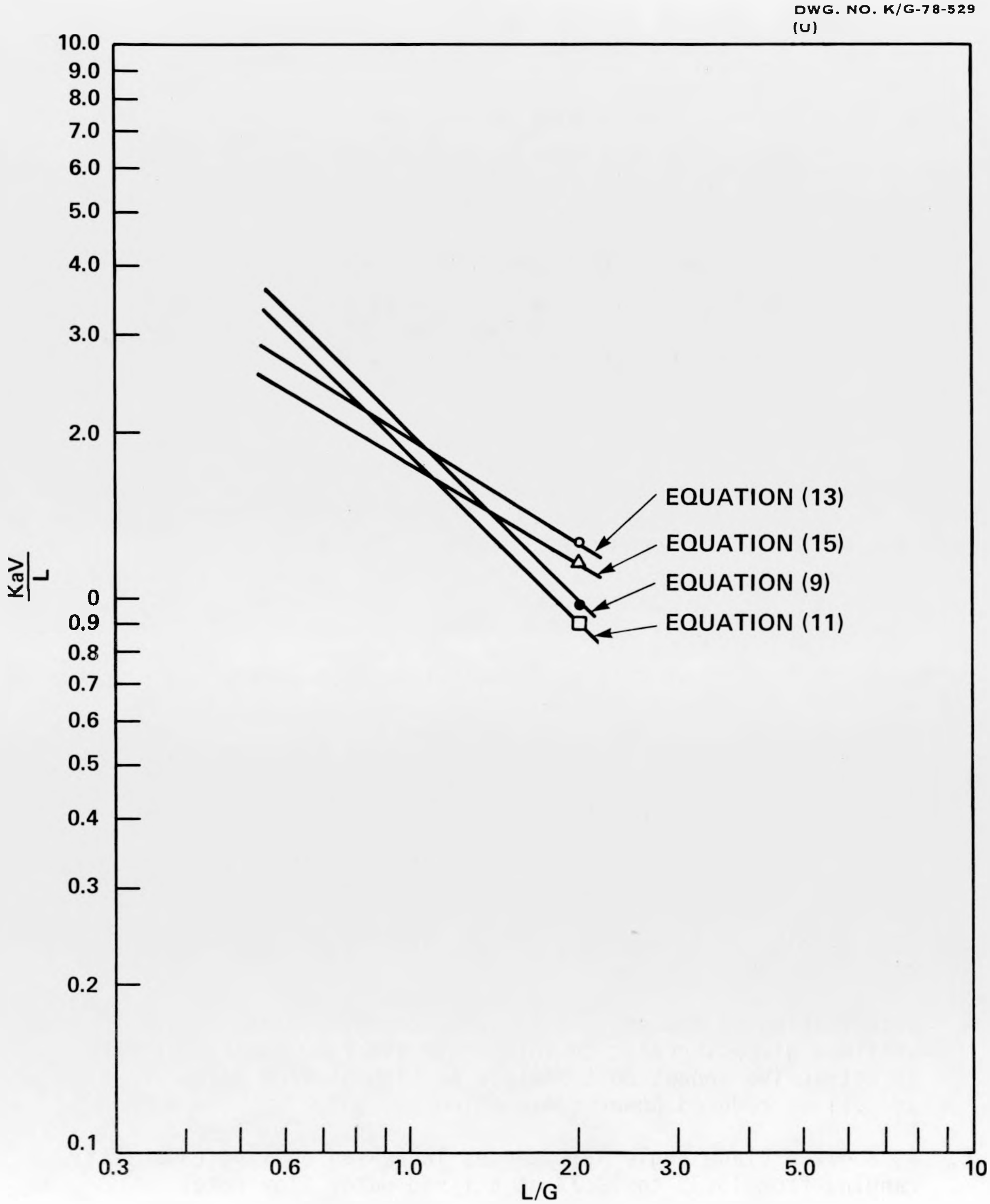

Figure 20

A SUMMARY OF THE PERFORMANCE IMPROVEMENT RESULTING FROM RBTC

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