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ABSTRACT

A test facility has been constructed and a test procedure developed to carry out full- and part-load performance tests on cooling and dehumidifying coils. Heat exchangers, up to 8 rows and having frontal face area of 0.61 by 0.61 m, can be tested in the working section. The air circulated in the closed loop duct can be heated and humidified after being cooled and dehumidified by the tested coil.

Dry and wet heat transfer test results of two coils are included. They are 4 and 8 rows, circular tubes, in staggered arrangement and held in space by flat continuous fins.

INSTALLATION DE MISE EN ESSAI DE SERPENTINS DE REFROIDISSEMENT ET DE DÉSHUMIDIFICATION: DESCRIPTION, MÉTHODES ET RÉSULTATS DE L'ESSAI

par A.H. Elmahdy et G.P. Mitalas

RÉSUMÉ

Un appareil d'essai a été fabriqué et une méthode d'essai mise au point pour effectuer des essais de charges complètes ou partielles sur des serpentins de refroidissement et de déshumidification. Des échangeurs de chaleur d'au plus 8 rangées et d'une surface de front mesurant 0.61 sur 0.61 m peuvent être mis à l'essai. L'air circulé dans le circuit en boucle fermé peut être chauffé et humidifié après son refroidissement et sa déshumidification dans le serpentin mis à l'essai.

Les résultats de l'essai de transfert de chaleur sèche et humide sur deux serpentins sont inclus. Ces serpentins tabulaires l'un de 4 et l'autre de 8 rangées disposées en chicane sont maintenus par une suite continue d'ailettes plates.

FACILITY FOR TESTING COOLING AND DEHUMIDIFYING COILS: DESCRIPTION, PROCEDURES AND TEST RESULTS

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A.H. Elmahdy and G.P. Mitalas

In recent years the need to save energy has prompted a great deal of interest in increasing the efficiency of heating, ventilating and air-conditioning (HVAC) systems. This has led to the development of sophisticated computer programs designed to determine the thermal performance of buildings and their associated HVAC systems under different loading conditions.

In a previous paper (1) the authors presented a mathematical model, followed by a computer program (2) simulating the thermal performance of finned-tube, multi-row heat exchangers at different cooling loads. A considerable amount of experimental heat transfer data about dry surface coil performance is available in the literature (3-6), but only a limited amount about wet surface coil performance (7 and 8). Consequently, a test facility capable of performing the required tests and of providing more heat transfer data, especially during dehumidification processes, has been constructed.

Detailed descriptions of the cooling coil test facility and test procedures are given in this paper. Two coils (4- and 8-row circular tubes with continuous fins) have been tested. The relevant heat transfer data, together with corresponding results obtained both analytically and experimentally are reported for comparison purposes.

DESCRIPTION OF TEST FACILITY

The test facility is located in the Environmental Engineering Laboratory of the Division of Building Research, National Research Council of Canada; a frontal view is shown in Figure 1. It is designed so that a coil with face dimensions 61 by 61 cm can be installed in the working section. For practical purposes air face velocity can be varied between about 0.7 to 6.0 m/s. Design capacities of sensible heat, latent heat and chilled water flow rate are 50 kW, 20 kW, and 3 dm^3/s, respectively. The test facility consists of an air circuit, chilled water circuit, and power supply.

Air Circuit

The air circuit is a closed loop duct, (a schematic diagram is shown in Figure 2). Air is circulated by means of a constant-speed axial flow fan (about 1.9 m^3/s at 490 Pa head). The air duct is made of galvanized steel sheets and has a square cross-section 61 by 61 cm, except for two transition sections upstream and downstream of the fan.

The air face velocity at the inlet to the cooling coil may be varied by dividing the air flow between the main air duct and a by-pass section by means of two sets of manually-controlled mechanical dampers. An electric heater (rated at 50 kW - 550 V) is located in the air duct to reheat the cooled air after it passes through the cooling coil; and a steam humidifier (rated at 13 g/s) is installed upstream of the fan to provide the required humidification. The steam supplied to the humidifier is generated by an electric boiler (rated power 30 kW at 550 V). During dehumidification tests the condensate from the cooling coil is collected from the duct section downstream of the coil. The bottom of this section is perforated so that the condensate passes through a false bottom into a funnel, which drains to a bottle that collects the condensate.

Chilled Water Circuit

Figure 3 is a schematic diagram showing the different components of the chilled water circuit. Chilled water is circulated in the system by means of a constant-speed pump (rated 3 dm^3/s at 200 kPa head) and the water flow rate varied manually by means of a throttling valve installed upstream of the cooling coil. The chilled water supply temperature is approximately 4.5°C, but the inlet water temperature to the coil is precisely controlled by a three-way diverter valve. This valve controls chilled water flow into the loop that includes the coil to maintain a constant temperature at the inlet to the coil.

An electric water heater (rated 12 kW at 550 V) is installed in the chilled water loop downstream of the cooling coil to determine the water flow rate through the coil. For calibration purposes, another electric water heater (rated 24 kW at 550 V) is installed in parallel with the cooling coil, but it is usually disconnected from the system unless calibration tests are to be performed (Figure 3).

Power Supply

Electric power is supplied to the fan, steam boiler and water heater from the main 550 V 3-phase supply in the Laboratory, whereas the power to the air (or calibration) heater is supplied from an electric generator driven by a gasoline engine. This engine-generator set provides a simple means of modulating power input to the air heater (or the calibration heater) by controlling the generator field excitation current.

INSTRUMENTATION AND MEASUREMENTS

Instrumentation is provided to measure air and chilled water conditions as well as power input to the heaters, fan and boilers. Most of these measurements are recorded on a punched paper tape for further processing on a digital computer.

Air dry-bulb temperature is measured at several locations in the air duct, i.e., upstream and downstream of both the cooling coil and the air heater. At each of these locations a 16-point thermocouple grid is used. Thermocouples of 24-gauge copper-constantan are shielded to minimize thermal radiation error. The grids are arranged according to the ARI 410-72 (9) and ASHRAE 33-64 Standards (10) (Figure 4).

Air dew-point temperature upstream of the cooling coil is measured by a Foxboro-Dewcel sensor. During the dehumidification tests condensate is collected (over 10-minute periods), weighed, and its temperature measured by a mercury-in-glass thermometer. Total and static pressures of the air stream are measured using grids of pitot tubes installed upstream and downstream of the cooling coil (Figure 4). These pressure measurements are used to determine the air velocity profile and the air flow rate through the cooling coil when $V_a > 1.5$ m/s. Over the air velocity range 1 to 5 m/s the velocity profile at entry to the coil (measured at 16 points over the air duct section) was found to be nearly flat, with a maximum velocity variation of about 0.05 m/s.

Chilled water temperature is measured by platinum resistance thermometers before and after the cooling coil and after the electric water heater, as shown in Figure 3. Water mixers are installed in the water circuit upstream of each temperature sensing element to ensure that temperature probes read the bulk mean water temperature at that cross-section of the pipe. Finally, the individual electric power input to air, reference (water) and calibration heaters are measured using separate Hall-Effect electric power transducers.

TEST PROCEDURES

As a first step the air fan is turned on and the air flow through the coil adjusted to the desired rate. The water circulating pump is then turned on and the desired water flow rate set by adjusting the by-pass valve. Electric power to the reference heater and air heater is switched on and the air temperature in the air duct is increased to the desired level by adjusting the field excitation of the engine-generator set. Readjustment on the water flow rate is usually necessary to keep the water temperature rise across the cooling coil within 1.5 to 5.5°C.

The steady-state condition is usually achieved after about 30 minutes. This may be checked by observing the different temperature readings as they are displayed on the DVM. In addition, the steam boiler and humidifier are switched on during dehumidification tests. In this case, steady-state is judged by the steadiness of the condensate flow rate. Once steady-state has been reached, the required measurements are recorded on punched paper tape for further processing. Six different sets of measurements are recorded for each test condition, each set requiring about 90 seconds for scanning.

TREATMENT OF EXPERIMENTAL DATA

Measurements, which were recorded at steady-state conditions, are processed on a computer using a special data reduction program to carry out logic and arithmetic operations. The program is based on the following equations and assumptions:

When air face velocity is less than about 1.5 m/s, calculation of air flow rate is based on the electric power input to the air heater and the corresponding air temperature rise as follows:

$$\dot{m}_a = \frac{P_a}{c_p(t_{a,4} - t_{a,3})} \tag{1}$$

where

P_a power input to the air duct heater, W

c_p air specific heat at constant pressure, J/(kg·°C)

$t_{a,3}$, $t_{a,4}$ average air temperature before and after the air heater, respectively, °C

$\dot{m}_a$ air mass flow rate, kg/s

At air face velocities greater than 1.5 m/s, the air flow rate calculation is based on dynamic and static pressure measurements, as described in Ref. 7.

Once the air flow rate has been determined, the average air face velocity at entry to the cooling coil section is calculated by:

$$V_a = \frac{\dot{m}_a}{\rho_a A} \tag{2}$$

where

ρ_a average air density, kg/m^3

A cross-section area of the duct, m^2

V_a average air velocity, m/s

The chilled water flow rate is calculated using the power input to the electric water heater as follows:

$$\dot{m}_w = \frac{P_w}{c_w(t_{w,3} - t_{w,2})} \tag{3}$$

where

P_w power input to the water heater, W

c_w average specific heat of water, J/(kg·°C)

$t_{w,2}$, $t_{w,3}$ average water temperature before and after the electric water heater, respectively, °C

$\dot{m}_w$ water flow rate, kg/s.

The total heat transfer rate from the air stream to the chilled water is calculated by:

$$Q_t = P_w \frac{t_{w,2} - t_{w,1}}{t_{w,3} - t_{w,2}} \tag{4}$$

where

Q_t total heat transfer rate, W

$t_{w,1}$ average chilled water temperature at inlet to the cooling coil °C

For dehumidification tests, the sensible and latent heat transfer rates are calculated as follows:

$$Q_s = \dot{m}_a \; c_p \; (t_{a,1} - t_{a,2}) \tag{5}$$

and

$$Q_l = Q_t - Q_s \tag{6}$$

where

Q_s, Q_l sensible and latent heat transfer rate, respectively, W

$t_{a,1}$, $t_{a,2}$ average air dry-bulb temperature before and after the cooling coil, respectively, °C.

A more detailed discussion of the reduction of experimental data and uncertainties associated with the measured quantities can be found in Ref. 7.

DESCRIPTION OF COILS TESTED

Two finned-tube heat exchangers (4- and 8-row) were tested. These coils have circular tubes in a staggered arrangement with continuous fins. A summary of physical data is given below. Figure 5 shows a schematic diagram of the heat exchanger together with other pertinent dimensions.

	4-row	8-row
Face area, m^2	0.372	0.372
Minimum air flow area/frontal area	0.54	0.54
Number of water circuits	16	16
Coil depth, m	0.14	0.29
Primary surface area, m^2	1.8	3.5
Secondary surface area, m^2	38.4	76.9

ANALYTICAL SIMULATION

The coil parameters required to describe the coil performance are C_1 and C_2. These are used to determine the relation between the air Reynolds number, Re, and the average heat transfer J-factor as follows:

$$J = C_1 \, Re^{C_2} \tag{7}$$

The J-factor is used to evaluate the value of f_o, the average heat transfer coefficient on the air side as follows:

$$f_o = G \, c_p \, J \, Pr^{-2/3} \tag{8}$$

where

- f_o average film heat transfer coefficient on the air side, $W/(m^2 \cdot K)$
- G maximum air mass flux (air flow rate per unit minimum area of flow), $kg/(m^2 \cdot s)$
- Pr Prandtl number.

For the two coils considered in this paper, the quantities for C_1 and C_2 are 0.104 and -0.366, respectively, as determined using the empirical relation given in Ref. 7 and based on the physical dimensions of the heat exchangers. The fin and the over-all heat exchanger efficiencies can be evaluated as described in Refs. 11 to 13. These data are needed to carry out calculations of outlet air and water conditions, as well as cooling load by the cooling coil simulation program. A detailed description of the computer program to carry out this analytical simulation program of cooling coil performance is given in Refs. 1 and 2.

EXPERIMENTAL RESULTS

In carrying out the heat transfer performance test on the two coils described in this paper, air face velocity and inlet conditions were varied over ranges that may be expected to occur in practical air-conditioning systems. For example, air dry-bulb temperature was varied between 20 and 38°C, combined with air relative humidity between 20 and 80 per cent. Air face velocity also was varied between 1 and 5 m/s. The chilled water flow rate was adjusted so that the water temperature rise across the cooling coil was in the range of 1.5 to 5.5°C, whereas the inlet chilled water temperature to the coil was maintained at about 6 to 8°C, depending on the supply from the main chiller.

Tables I through IV give a detailed listing of experimental results, with corresponding analytical quantities obtained by using the simulation program given in Ref. 2. Comparison of the different quantities shows good agreement between analytical and experimental values. For example, the maximum deviation of the predicted cooling load does not exceed 5 per cent of that determined experimentally. A similar percentage deviation is observed in the measured and calculated outlet chilled water temperatures. In terms of absolute values, the maximum difference between measured and calculated air dry-bulb temperature drop across the cooling coil is about 0.6°C, whereas the corresponding difference in chilled water temperature rise across the coil is within 0.1°C.

CONCLUSION

The test facility described in this paper was used to test one family of heat exchangers to determine their thermal performance with dry and wet surface conditions. The tests showed that the coil test facility can be used to carry out coil performance tests with reasonable accuracy, and to check the analytical methods of coil simulation.

NOMENCLATURE

All quantities used in this paper are in SI units (14).

Symbol	Description
c_p	Air specific heat at constant pressure
c_w	Water specific heat
C_1 and C_2	Coil thermal performance parameters
f_o	Average film heat transfer coefficient on the air side of the heat exchanger
G	Maximum air mass flux (mass rate flow of air per unit minimum area of flow)
J	Average heat transfer J-factor
$\dot{m}_a$, $\dot{m}_w$	Mass rate of flow of air and water respectively
P_a, P_w	Electric power input to the air and water heaters respectively
Pr	Prandtl number
Q_l, Q_s, Q_t	Latent, sensible, and total heat transfer rate respectively
Re	Reynolds number
t_a, t_w	Average air and water temperature respectively
V_a	Average air velocity
ρ	Air density

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TABLE I

SUMMARY OF DRY SURFACE TEST RESULTS AND THE CORRESPONDING ANALYTICAL RESULTS (8-ROW COIL)

Test No	Air Face Velocity ft/min	Inlet Conditions DBT °F	WBT °F	WT °F	Δ DBT °F Expt	Δ DBT °F Anal	ΔWT °F Expt	ΔWT °F Anal	Q_{total} $\frac{Btu/hr}{1000}$ Expt	Q_{total} $\frac{Btu/hr}{1000}$ Anal	Q_s $\frac{Btu/hr}{1000}$ Expt	Q_s $\frac{Btu/hr}{1000}$ Anal	Surface Condition
1	310	77.5	60.6	47.0	28.3	28.3	2.2	2.2	39.5	39.4	39.5	39.4	Dry
2	381	93.6	66.1	46.7	42.9	42.4	4.8	4.5	74.1	70.2	74.1	70.1	Dry
3	569	73.8	58.7	46.8	23.5	23.7	3.3	3.3	58.1	58.6	58.1	58.6	Dry
4	618	82.4	61.7	46.7	29.7	29.9	5.3	5.1	82.9	83.5	82.9	83.5	Dry
5	620	70.5	59.1	50.8	17.3	17.3	2.4	2.4	47.2	46.5	47.2	46.5	Dry
6	625	80.3	63.4	50.9	25.8	25.8	3.5	3.5	69.3	68.7	69.3	68.7	Dry
7	645	66.7	56.0	49.5	14.9	14.8	2.2	2.2	41.9	41.8	41.9	41.8	Dry
8	720	74.4	59.3	46.9	22.7	22.9	3.9	4.0	70.8	71.6	70.8	71.6	Dry
9	815	74.7	59.2	46.9	22.0	22.4	4.2	4.5	78.1	79.1	78.1	79.1	Dry
10	627	74.7	61.0	50.9	20.9	20.7	3.0	2.9	56.7	56.0	56.7	56.0	Dry
11	587	80.5	63.4	51.1	34.0	33.9	3.5	3.5	65.7	65.6	65.7	65.6	Dry
12	637	74.6	59.7	47.1	24.4	24.6	3.6	3.7	65.5	65.9	65.5	65.9	Dry
13	786	74.5	60.0	46.9	22.2	22.5	4.2	4.3	75.3	76.3	75.3	76.3	Dry
14	830	74.8	58.8	46.9	22.1	22.4	4.4	4.5	79.4	80.3	79.4	80.3	Dry
15	475	77.2	60.3	46.9	27.5	27.6	3.2	3.2	56.5	56.5	56.5	56.5	Dry
16	525	75.5	59.0	46.7	25.8	25.8	3.2	3.3	58.7	58.8	58.7	58.8	Dry

°C = (°F - 32)/1.8, Δ°C = Δ°F/1.8, W = 0.29307 Btu/hr and m/s = 0.00508 x fpm

DBT = Dry bulb temperature, WBT = Wet bulb temperature,

WT = Water Temperature, and Q_{sen} = Sensible cooling.

ΔDBT = Air dry bulb temperature drop

ΔWT = Water temperature rise

TABLE II

SUMMARY OF WET SURFACE TEST RESULTS AND THE CORRESPONDING ANALYTICAL RESULTS (8-ROW COIL).

Test No	Air Face Velocity ft/min	Inlet Conditions DBT °F	WBT °F	WT °F	Δ DBT °F Expt	Anal	ΔWT °F Expt	Anal	Q_{total} Expt	$\frac{Btu/hr}{1000}$ Anal	Q_s Expt	$\frac{Btu/hr}{1000}$ Anal	Surface Condition
1	250	97.7	75.9	47.3	47.2	47.8	4.6	4.7	84.3	85.4	50.3	50.9	Wet
2	285	86.6	71.0	47.2	36.3	36.8	4.2	4.2	75.0	74.6	44.0	44.6	Wet
3	286	93.1	77.4	46.8	41.1	41.4	6.6	6.5	102.5	100.9	50.8	51.1	Wet
4	300	79.5	68.6	47.1	29.6	30.3	3.9	3.9	68.1	67.8	37.1	37.9	Wet
5	336	78.1	63.9	47.0	28.5	28.2	3.7	3.6	56.5	54.9	41.8	41.3	Dry-Wet
6	382	92.1	72.5	47.3	39.1	39.8	5.6	5.9	101.6	106.5	67.7	68.9	Wet
7	445	99.0	73.4	47.2	45.0	45.4	6.3	6.4	115.3	117.4	83.0	84.3	Dry-Wet
8	455	79.9	67.0	46.8	27.8	28.8	4.8	4.9	86.7	87.7	54.6	56.6	Dry-Wet
9	470	78.7	69.5	46.8	24.3	24.0	6.4	6.2	100.3	97.2	52.1	51.6	Wet
10	503	73.5	61.8	47.0	21.9	21.8	4.1	4.0	64.2	62.0	49.0	48.1	Dry-Wet
11	575	83.7	67.8	47.1	29.6	30.2	5.6	5.7	103.3	103.8	72.0	73.3	Dry-Wet
12	579	93.9	71.6	47.2	38.4	38.6	6.8	7.1	126.1	129.0	93.4	93.8	Dry-Wet
13	590	84.4	68.0	47.6	29.5	30.3	5.9	5.9	106.4	106.2	75.0	76.2	Dry-Wet
14	675	70.1	62.3	47.2	16.9	17.9	4.5	4.5	79.8	80.1	50.0	52.7	Dry-Wet
15	682	87.7	69.4	47.2	31.8	31.8	6.7	7.0	123.5	126.8	91.9	92.1	Dry-Wet
16	688	84.9	68.0	47.6	28.8	29.4	6.4	6.5	116.4	117.3	84.5	86.1	Dry-Wet

°C = (°F - 32)/1.8, Δ°C = Δ°F/1.8, W = 0.29307 Btu/hr and m/s = 0.00508 x fpm

DBT = Dry bulb temperarure, WBT = Wet bulb temperature,

WT = Water temperature, and Q_{sen} = Sensible cooling.

ΔDBT = Air dry bulb temperature drop

ΔWT = Water temperature rise

TABLE II - (Con't)

Test No	Air Face Velocity ft/min	Inlet Conditions DBT °F	WBT °F	WT °F	Δ DBT °F Expt	Anal	ΔWT °F Expt	Anal	Q_{total} $\frac{Btu/hr}{1000}$ Expt	Anal	Q_s $\frac{Btu/hr}{1000}$ Expt	Anal	Surface Condition
17	680	92.5	71.2	47.2	35.8	35.8	7.4	7.7	133.2	137.6	100.6	100.5	Dry-Wet
18	802	84.2	67.6	47.6	27.0	27.4	6.9	6.9	125.0	125.5	92.2	93.9	Dry-Wet
19	810	88.8	69.7	47.6	30.2	30.0	7.9	8.2	146.0	148.9	112.4	111.4	Dry-Wet
20	819	84.7	68.2	47.3	27.5	27.5	7.1	7.3	127.6	130.7	96.3	96.0	Dry-Wet
21	820	78.4	65.6	47.1	22.5	23.2	5.2	5.4	111.2	113.8	80.1	82.5	Dry-Wet
22	830	100.5	74.2	47.1	40.3	39.0	9.6	10.1	176.4	183.9	142.0	137.5	Dry-Wet
23	850	91.7	70.9	47.2	33.0	32.4	8.3	8.6	150.0	155.6	118.2	116.0	Dry-Wet
24	857	89.1	69.8	47.3	30.9	31.4	7.8	8.1	143.7	147.5	111.4	110.0	Dry-Wet
25	476	83.6	67.4	47.6	27.2	27.7	6.5	6.5	118.2	118.4	86.6	87.9	Dry-Wet
26	339	90.7	71.9	47.5	38.9	39.7	4.9	5.0	87.9	88.8	55.4	56.5	Dry-Wet
27	441	84.9	68.7	47.5	32.0	32.9	5.1	5.0	93.0	91.1	60.1	61.9	Dry-Wet
28	485	85.2	68.2	47.3	31.9	32.8	5.4	5.3	98.5	95.7	65.9	67.6	Dry-Wet
29	798	84.1	68.1	47.2	27.5	27.5	6.8	7.1	124.5	128.7	93.5	93.4	Dry-Wet
30	564	88.9	69.6	47.1	34.1	34.6	6.3	6.5	112.9	114.9	80.9	82.0	Dry-Wet
31	616	75.2	64.5	47.2	21.6	22.7	5.0	4.4	89.9	89.2	57.6	60.6	Dry-Wet
32	766	78.0	65.5	47.2	22.6	23.5	5.8	6.0	106.4	109.0	74.7	77.7	Dry-Wet

°C = (°F - 32)/1.8, Δ°C = Δ°F/1.8, W = 0.29307 Btu/hr and m/s = 0.00508 x fpm

DBT = Dry bulb temperature, WBT = Wet bulb temperature,

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ΔWT = Water temperature rise

TABLE II - (CON'T)

Test No	Air Face Velocity ft/min	Inlet Conditions DBT °F	WBT °F	WT °F	Δ DBT °F Expt	Anal	ΔWT °F Expt	Anal	Q_{total} $\frac{Btu/hr}{1000}$ Expt	Anal	Q_s $\frac{Btu/hr}{1000}$ Expt	Anal	Surface Condition
33	666	80.1	66.3	47.2	25.2	25.9	5.7	5.7	104.6	104.6	72.4	74.4	Dry-Wet
34	190	79.6	72.5	47.1	31.1	31.2	3.2	3.3	56.9	57.5	25.5	25.6	Dry-Wet
35	198	83.4	73.8	47.2	34.8	34.6	3.4	3.5	62.3	62.6	29.4	29.2	Dry-Wet
36	436	84.3	69.4	47.3	32.0	32.2	4.9	5.2	90.8	94.5	59.4	59.9	Dry-Wet

°C = (°F - 32)/1.8, Δ°C = Δ°F/1.8, W = 0.29307 Btu/hr and m/s = 0.00508 x fpm

DBT = Dry bulb temperature, WBT = Wet bulb temperature,

WT = Water temperature, and Q_{sen} = Sensible cooling.

ΔDBT = Air dry bulb temperature drop

ΔWT = Water temperature rise

TABLE III

SUMMARY OF DRY SURFACE TEST RESULT AND THE CORRESPONDING ANALYTICAL RESULTS (4-ROW COIL)

Test No	Air Face Velocity ft/min	Inlet Conditions DBT °F	WBT °F	WT °F	Δ DBT °F Expt	Δ DBT °F Anal	ΔWT °F Expt	ΔWT °F Anal	Q_{total} $\frac{\text{Btu/hr}}{1000}$ Expt	Anal	Q_s $\frac{\text{Btu/hr}}{1000}$ Expt	Anal	Surface Condition
1	487	91.4	65.4	47.2	32.5	31.8	3.4	3.3	66.6	65.1	66.6	65.1	Dry
2	478	95.7	66.9	47.2	35.8	35.0	3.7	3.7	71.9	70.3	71.9	70.3	Dry
3	481	84.5	63.0	47.2	27.3	26.7	2.9	2.9	56.0	54.7	56.0	54.7	Dry
4	482	77.6	60.2	47.2	22.1	21.7	2.4	2.4	46.0	45.1	46.0	45.1	Dry
5	306	85.0	62.7	47.2	31.3	30.7	2.2	2.1	40.8	40.0	40.8	40.0	Dry
6	305	90.2	64.5	47.2	35.9	35.0	2.4	2.4	46.1	45.0	46.1	45.0	Dry
7	312	96.1	66.9	47.1	40.6	39.8	2.8	2.8	52.9	51.8	52.9	51.8	Dry
8	610	85.6	63.5	47.1	26.0	25.3	3.6	3.5	67.8	66.0	67.8	66.0	Dry
9	599	80.7	61.4	47.2	22.7	22.1	3.1	3.0	58.7	57.2	58.7	57.2	Dry
10	605	71.5	57.6	47.2	16.3	15.9	2.2	2.2	43.0	42.1	43.0	42.1	Dry
11	622	93.3	66.1	47.2	31.4	30.4	4.2	4.1	82.1	79.4	82.1	79.4	Dry
12	575	97.7	68.1	47.2	30.7	31.0	7.2	7.3	73.4	73.9	73.4	73.9	Dry
13	173	94.9	66.1	47.1	42.9	41.9	3.2	3.1	31.0	30.3	31.0	30.3	Dry
14	184	88.9	63.9	47.1	37.1	36.2	2.9	2.8	28.8	28.1	28.8	28.1	Dry
15	177	81.2	60.9	47.2	30.4	29.6	2.2	2.2	23.2	22.6	23.2	22.6	Dry
16	173	76.4	59.1	47.2	26.0	25.5	1.9	1.4	19.5	19.1	19.5	19.1	Dry

°C = (°F - 32)/1.8, Δ°C = Δ°F/1.8, W = 0.29307 Btu/hr and m/s = 0.00508 x fpm

DBT = Dry bulb temperature, WBT = Wet bulb temperature,

WT = Water temperature, and Q_{sen} = Sensible cooling

ΔDBT = Air dry bulb temperature drop

ΔWT = Water temperature rise

TABLE III (CON'T)

Test No	Air Face Velocity ft/min	Inlet Conditions DBT °F	WBT °F	WT °F	Δ DBT °F Expt	Δ DBT °F Anal	ΔWT °F Expt	ΔWT °F Anal	Q_{total} Expt	$\frac{Btu/hr}{1000}$ Anal	Q_s Expt	$\frac{Btu/hr}{1000}$ Anal	Surface Condition
17	781	92.0	67.1	47.1	23.6	23.7	7.6	7.7	77.7	77.8	77.7	77.8	Dry
18	770	83.4	63.1	47.2	19.0	19.1	6.0	6.1	62.4	62.9	62.4	62.9	Dry
19	762	75.0	59.5	47.2	14.5	14.7	4.6	4.7	47.8	48.3	47.8	48.3	Dry
20	757	70.1	57.4	47.2	11.9	12.1	3.8	3.9	39.4	39.8	39.4	39.8	Dry
21	755	100.2	68.9	47.2	28.7	28.8	8.7	8.7	90.2	90.3	90.2	90.3	Dry
22	877	96.1	67.2	47.2	24.6	24.6	8.5	8.6	89.7	89.7	89.7	89.7	Dry
23	870	88.6	64.4	47.2	20.5	20.6	7.3	7.4	75.7	76.0	75.7	76.0	Dry
24	874	79.4	61.2	47.2	15.9	16.0	5.8	5.9	60.0	60.4	60.0	60.4	Dry
25	908	73.9	58.7	47.2	13.0	12.9	4.8	4.8	50.8	50.6	50.8	50.6	Dry
26	386	75.1	59.3	47.2	19.7	19.8	3.1	3.2	32.8	33.0	32.8	33.0	Dry
27	378	79.7	61.0	47.2	23.0	23.2	3.6	3.7	37.5	37.8	37.5	37.8	Dry
28	404	84.5	63.1	47.2	25.9	26.1	4.3	4.4	44.7	44.9	44.7	44.9	Dry
29	399	92.2	65.8	47.3	31.6	31.7	5.2	5.2	53.2	53.2	53.2	53.2	Dry
30	403	96.5	67.6	47.1	34.8	35.0	5.6	5.7	58.7	58.9	58.7	58.9	Dry

°C = (°F - 32)/1.8, Δ°C = Δ°F/1.8, W = 0.29307 Btu/hr and m/s = 0.00508 x fpm

DBT = Dry bulb temperature, WBT = Wet bulb temperature,

WT = Water temperature, and Q_{sen} = Sensible cooling.

ΔDBT = Air dry bulb temperature drop

ΔWT = Water temperature rise

TABLE IV

SUMMARY OF WET SURFACE TEST RESULTS AND THE CORRESPONDING ANALYTICAL RESULTS (4-ROW COIL)

Test No	Air Face Velocity ft/min	Inlet Conditions DBT °F	WBT °F	WT °F	Δ DBT °F Expt	Anal	ΔWT °F Expt	Anal	Q_{total} Expt	$\frac{Btu/hr}{1000}$ Anal	Q_s Expt	$\frac{Btu/hr}{1000}$ Anal	Surface Condition
1	367	95.9	77.9	47.3	31.0	30.5	7.7	7.8	78.7	79.0	48.1	46.8	Wet
2	377	88.5	74.7	47.2	25.2	25.3	6.9	6.8	71.6	70.3	40.2	40.3	Wet
3	372	79.3	70.7	47.2	18.6	19.4	5.9	5.8	59.7	59.1	30.0	31.2	Wet
4	558	80.6	71.2	47.2	16.7	16.6	6.8	6.7	72.3	70.4	40.7	40.1	Wet
5	544	85.9	73.6	47.2	20.4	20.0	7.4	7.5	77.1	77.3	47.5	46.2	Wet
6	540	90.6	75.3	47.2	23.4	23.1	8.0	8.1	83.0	83.2	53.8	52.8	Wet
7	545	98.2	78.8	47.2	28.4	27.5	9.1	9.3	95.1	96.4	65.2	62.3	Wet
8	642	95.6	77.9	47.2	24.6	24.0	9.3	9.5	97.6	98.7	67.3	64.6	Wet
9	633	99.2	79.4	47.2	27.0	25.9	9.9	10.1	101.6	102.9	71.8	67.9	Dry-Wet
10	667	76.6	69.0	47.2	13.1	13.6	6.6	6.5	68.1	67.3	38.0	39.2	Wet
11	714	83.4	72.8	47.2	16.4	16.1	7.9	8.0	80.2	81.0	50.6	49.1	Wet
12	741	77.9	69.7	47.2	13.1	13.6	7.0	7.1	71.7	7.2	42.2	43.4	Wet
13	742	90.9	76.3	47.3	20.6	19.9	9.1	9.2	96.3	97.0	65.1	62.2	Wet
14	857	96.9	79.7	47.2	23.2	21.0	10.9	11.1	114.3	115.3	84.2	75.1	Dry-Wet
15	868	83.3	73.0	47.3	15.3	14.7	8.3	8.4	87.6	87.8	57.3	54.6	Wet
16	736	74.0	67.5	47.2	11.0	11.6	6.4	6.3	62.9	62.4	35.1	36.9	Wet

°C = (°F - 32)/1.8, Δ°C = Δ°F/1.8, W = 0.29307 Btu/hr and m/s = 0.00508 x fpm

DBT = Dry bulb temperature, WBT = Wet bulb temperature,

WT = Water temperature, and Q_{sen} = Sensible cooling

ΔDBT = Air dry bulb temperature drop

ΔWT = Water temperature rise

TABLE IV - (CON'T)

Test No	Air Face Velocity ft/min	Inlet Conditions DBT °F	WBT °F	WT °F	Δ DBT °F Expt	Anal	ΔWT °F Expt	Anal	Q_{total} $\frac{Btu/hr}{1000}$ Expt	Anal	Q_s $\frac{Btu/hr}{1000}$ Expt	Anal	Surface Condition
17	800	74.8	68.1	47.3	11.0	11.5	6.7	6.6	67.7	66.4	38.1	39.5	Wet
18	603	86.7	74.4	47.3	19.8	19.2	7.9	8.0	82.4	82.8	51.3	49.2	Wet
19	584	75.4	68.9	47.2	13.1	13.4	6.3	6.3	63.0	63.1	33.3	33.9	Wet
20	389	78.4	70.6	47.3	17.3	18.2	5.9	5.9	59.5	59.0	29.6	30.4	Wet
21	290	81.1	72.5	47.5	21.3	21.6	5.8	5.6	57.6	55.3	26.9	27.2	Wet
22	451	85.9	73.6	47.3	21.7	21.6	7.1	7.0	72.9	71.9	42.1	41.4	Wet
23	300	89.8	75.5	47.3	28.4	28.4	6.6	6.4	66.6	65.0	36.0	35.8	Wet
24	276	89.5	75.1	47.2	28.5	29.2	6.3	6.2	62.6	61.3	33.2	33.8	Wet
25	300	93.4	77.0	47.2	31.5	31.1	7.0	6.9	70.8	69.8	39.9	39.1	Wet
26	303	98.9	79.1	47.3	35.7	35.0	7.4	7.5	75.3	75.8	45.3	44.0	Wet
27	480	87.4	74.2	47.3	22.0	21.9	7.4	7.5	74.8	75.0	45.2	44.5	Wet
28	485	93.4	76.4	47.3	26.2	26.0	8.1	8.1	83.9	83.2	53.7	52.7	Wet
29	738	87.1	74.5	47.2	18.5	17.9	8.6	8.7	88.1	88.7	58.6	55.9	Wet
30	737	92.0	76.9	47.5	21.2	20.4	9.3	9.4	97.2	97.4	66.5	62.9	Wet
31	805	94.0	78.2	47.4	21.8	20.5	10.1	10.1	106.9	107.4	75.2	69.3	Wet
32	694	87.5	74.5	47.3	19.1	18.6	8.5	8.5	87.0	86.6	57.0	54.9	Wet

°C = (°F - 32)/1.8, Δ°C = Δ°F/1.8, W = 0.29307 Btu/hr and m/s = 0.00508 x fpm

DBT = Dry bulb temperature, WBT = Wet bulb temperature,

WT = Water temperature, and Q_{sen} = Sensible cooling

ΔDBT = Air dry bulb temperature drop

ΔWT = Water temperature rise

TABLE IV - (CON'T)

Test No	Air Face Velocity ft/min	Inlet Conditions DBT °F	WBT °F	WT °F	Δ DBT °F Expt	Anal	ΔWT °F Expt	Anal	Q_{total} Expt	$\frac{Btu/hr}{1000}$ Anal	Q_s Expt	$\frac{Btu/hr}{1000}$ Anal	Surface Condition
33	775	79.9	71.0	47.3	13.7	14.0	7.6	7.7	76.2	76.8	46.3	46.9	Wet
34	414	87.5	74.6	47.3	23.7	23.4	7.2	7.1	73.6	72.3	41.7	40.8	Wet
35	418	84.1	73.0	47.3	20.5	20.8	6.9	6.8	67.5	66.6	36.9	37.1	Wet
36	430	80.5	71.4	47.3	17.9	18.3	6.4	6.3	64.2	62.6	33.5	33.8	Wet
37	324	83.3	73.1	47.3	22.0	22.5	6.2	6.1	62.2	60.1	30.7	31.2	Wet
38	325	92.6	76.5	47.3	29.2	29.5	7.1	7.1	70.6	70.3	40.0	40.1	Wet

°C = (°F - 32) /1.8, Δ°C = Δ°F/1.8, W = 0.29307 Btu/hr and m/s = 0.00508 x fpm

DBT = Dry bulb temperature, WBT = Wet bulb temperature,

WT = Water temperature, and Q_{sen} = Sensible cooling.

ΔDBT = Air dry bulb temperature drop

ΔWT = Water temperature rise

Figure 1

Frontal view of the cooling coil testing facility showing the closed loop air duct

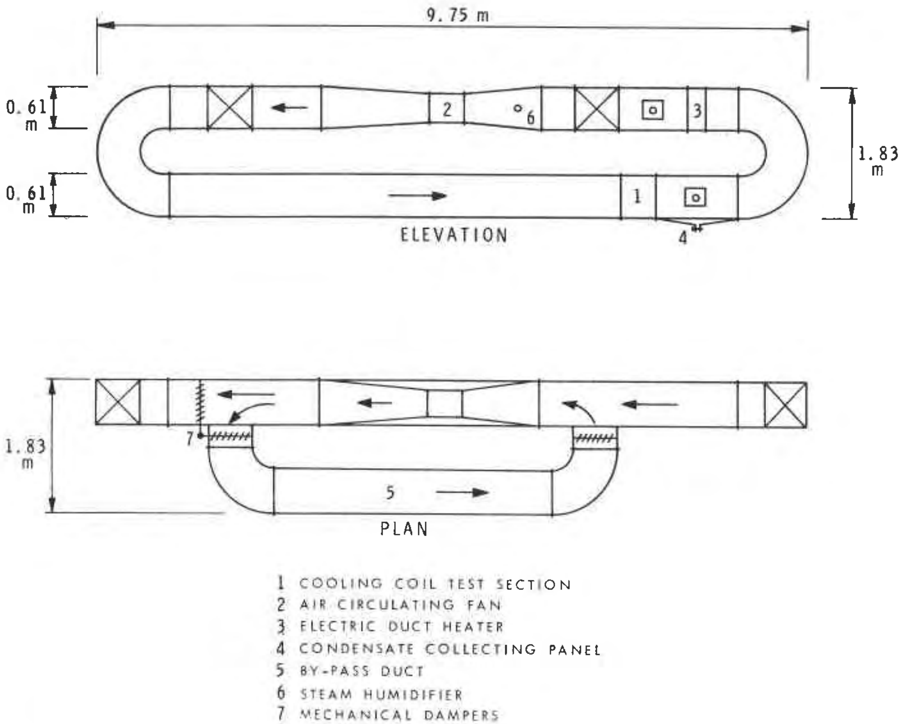


FIGURE 2
SCHEMATIC DIAGRAM OF AIR DUCT

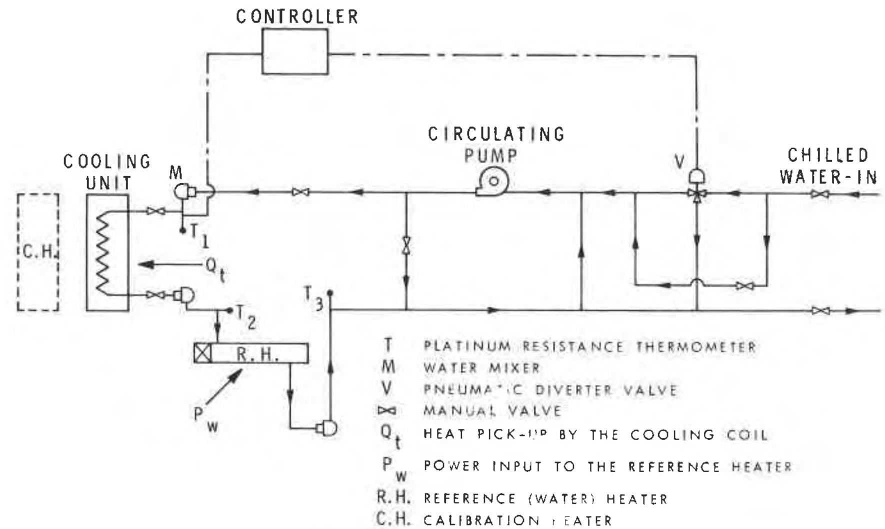
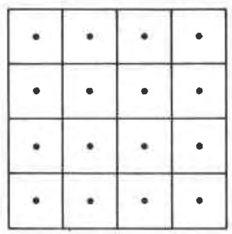


FIGURE 3
SCHEMATIC DIAGRAM OF CHILLED WATER/CIRCUIT

1 COOLING COIL
2 AIR CIRCULATING FAN
3 ELECTRIC DUCT HEATER
4 BY-PASS BRANCHING
5 DEWCEL ELEMENT
6 16-POINT THERMOCOUPLE GRID
7 16-POINT TOTAL PRESSURE PROBE GRID
8 CONDENSATE COLLECTING ARRANGEMENT
• THERMOCOUPLE OR PRESSURE PROBE

16 AREAS EACH
$0.023\ m^2$



CROSS-SECTION
OF AIR DUCT

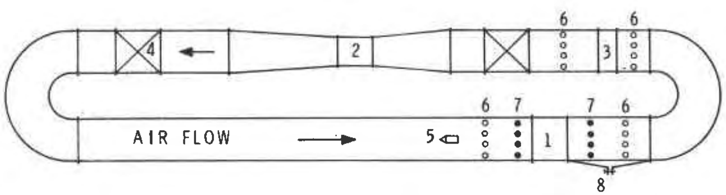
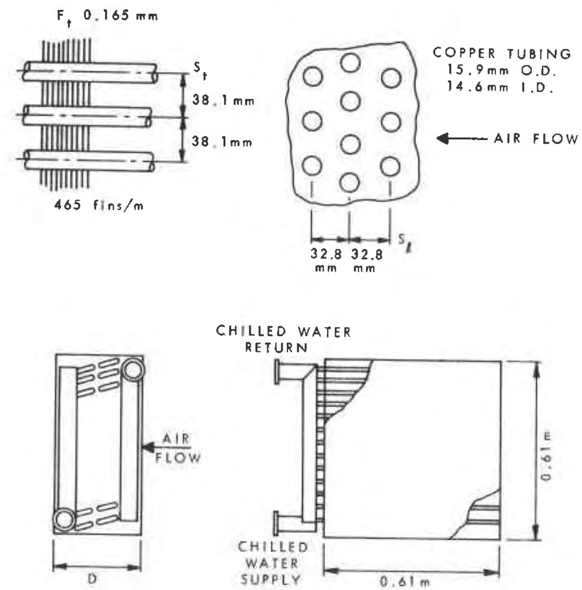


FIGURE 4

LOCATION OF THERMOCOUPLE AND TOTAL PRESSURE PROBE GRIDS IN MAIN AIR DUCT



D = 0.14 m 4-ROW COIL
= 0.29 m 8-ROW COIL

FIGURE 5

PERTINENT DIMENSIONS OF COOLING COIL UNDER INVESTIGATION