ACKNOWLEDGEMENTS

I would like to acknowledge, with thanks, my project supervisor, Mr. Mal Turaga, for his advice and guidance in this endeavor.

Special gratitude is also due to my wife, Charlotte, whose efforts permitted me the time to complete this work. Credit for typing this manuscript must also go to her. No ammount of thanks is adequate to repay her continuing assistance, patience and understanding.

TABLE OF CONTENTS

TABLE OF CONTENTS

NOMENCLATURE

Q = the rate of heat transfer

U_1 = the coefficient of heat transfer between the air and the moist surface (BTU/HR/SQ FT/BTU OF ENTHALPY DIFF)

A = the total area of the external surface of the coil including any fins (SQ FT)

A_f = the surface area of the fins of the coil (SQ FT)

A_t = the free surface area of the tubes of the coil (SQ FT)

U_A = the apparent coefficient of heat transfer on side 1 (BTU/HR/SQ FT/BTU OF ENTHALPY DIFF)

t_m = mean difference in temperature between air and refrigerant (^{O}F)

a = a numerical coefficient. See Table 1

B = ratio of external to internal surface area of coil

c = humid specific heat of air (average $c = 0.243$)

d_L = large enthalpy (total heat) difference, BTU

d_M = mean enthalpy (total heat) difference, BTU

d_s = small enthalpy (total heat) difference, BTU

D = wet bulb depression of the air, deg Fahr.

$D'' = t_1'' - t_r$

$D_b = t_b - t_r$

D_L = large temperature difference, deg Fahr.

D_M = mean temperature difference, deg Fahr.

D_S = small temperature difference, deg Fahr.

f_g = coefficient of heat transfer through air film, BTU/HR/SQ FT/degree of temperature difference

NOMENCLATURE

f_r = coefficient of heat transfer through refrigerant film, BTU/HR/SQ FT/degree of temperature difference

G = mass of air, LB/HR

$R = \frac{B \times f_g}{f_r}$

h = enthalpy (total heat) of humid air, BTU/LB of dry air

h_R = enthalpy (total heat) of saturated air at a temperature equal to the refrigerant temperature, BTU/LB of dry air

h_s = enthalpy (total heat) of saturated air at a temperature equal to the surface temperature, BTU/LB of dry air

H_w = total heat lost by the air flowing past the wetted surface, BTU/HR

M = a numerical factor. See Equations 15 and 19

t = dry bulb temperature, deg Fahr

t' = wet bulb temperature, deg Fahr

t" = dew point temperature, deg Fahr

t_b = boundary dry bulb temperature, deg Fahr

t_b' = boundary wet bulb temperature, deg Fahr

t_r = refrigerant temperature, deg Fahr

t_s = surface temperature of coil, deg Fahr

U_D = overall coefficient of heat transfer through dry surface of coil, BTU/HR/SQ FT/ degree of temperature difference

U_w = overall coefficient of heat transfer through wetted surface of coil, BTU/HR/SQ FT/BTU of enthalpy difference

Notes:

The subscript 1 refers to the initial condition of the air or refrigerant entering the coil. The subscript 2 refers to the final condition of the air or refrigerant leaving the coil.

NOMENCLATURE

Q_w = heat loss through duct walls, BTU/Hr

U_d = overall coefficient of heat transfer for the duct wall, BTU/HR/SQ FT/$^\circ$F

d_c = equivalent circular diameter of a rectangular duct, ft

P = perimeter of the duct, feet

L = length of duct, feet

t_e = temperature of air entering duct, $^\circ$F

t_l = temperature of air leaving duct, $^\circ$F

t_a = temperature of air surrounding duct, $^\circ$F

A_d = cross sectional area of duct, ft^2

V = mean velocity of fluid, feet per minute

ρ = density of air, pounds per cubic foot

R.E. = refrigerating effect

q_c = heat loss to the air from the condenser jacket

C.O.P. = coefficient of performance

C_N = coefficient of discharge for the nozzle

A_N = area of nozzle - ft^2

V_N = specific volume of air at the nozzle - ft^3/lb of dry air

W_N = humidity ratio of air at the nozzle - lb of water/lb of dry air

Δ_{PN} = static pressure difference across the nozzle - inch of water

R_e = Reynold's number

q_{ta} = total air side cooling capacity

NOMENCLATURE

W_a = mass rate of air flow – lb per min

ΔW = difference in humidity ratio across coil, lb water/lb dry air

t_4' = wet bulb air temperature leaving mixing chamber

h = enthalpy designation

W_r = flow of refrigerant

W_w = flow of water – lb per hour

t_{w2} = temperature of water leaving the condenser – ^oF

t_{w1} = temperature of water entering the condenser – ^oF

q_c = heat loss to the air from the condenser jacket

CHAPTER I

INTRODUCTION

The volume of production of direct expansion cooling coils used in the air conditioning and refrigeration industry is very high compared to the alternate choices currently available. These coils constitute the "heart" of an air conditioning system and their performance has a considerable effect on the overall system performance. This provides a strong incentive for improving the performance of these coils and thereby reducing the systems operating costs. Soaring energy prices and increasing energy demands have magnified the benefit which would be gained from these improvements. These factors are the reason for the present need for a closer examination of the heat transfer phenomena in air conditioning system components. This paper will hopefully add to the volume of knowledge currently available in the field.

A number of theoretical and experimental studies have been conducted on heat transfer phenomena in wetted heat exchanges. (see references) Most of the recent work, however, is based on chilled water coil performance and the majority of existing experimental studies on direct expansion coils dates back to 1930-1940. Since substantial improvements have taken place in measurement techniques during the period 1940-1970, it is now possible to update and expand on these previous works.

Coils for volatile refrigerants present more complex problems for heat transfer analysis compared with water or brine coils. In chilled water coils, most of the heat transfer occurs with the water in a liquid state with very little vaporization involved. In the direct expansion coils, the evaporation which takes place introduces many added complexities to the heat transfer phenomenon. The oil mixed with the refrigerant during this change in state presents additional problems in the analysis of the heat transfer process.

Characteristics of individual coil design play a major role in the coil's performance. This factor combined with the difficulties already mentioned regarding the generalized theoretical methods for coil performance evaluation make a proper experimental verification of the coils a basic necessity. To aid in the testing of coils, the American Society of Heating, Refrigeration and Air Conditioning Engineers (ASHRAE) developed ASHRAE Standard 33-64 entitled "Methods of Testing and Rating Forced-Circulation Air Cooling and Heating Coils".

The need for a testing facility in the evaluation of DX cooling coils, combined with the lack of any such facility in the Montreal area prompted Concordia University's Center for Building Studies to build a test facility which would satisfy the ASHRAE and ARI standards for the testing of coils. The closest test facility of this kind is presently situated in Ottawa at the National Research Council. This latter facility, however, is geared to test chilled water coils and has a limited air volume control. Local coil manufacturers have expressed their interest in a test facility which would be capable of testing to the ASHRAE standard and provide them with performance evaluations.

The test facility was built with sufficient flexibility to enable its use in applications other than the testing of coils. For example, with small modifications, the system lends itself easily to air filter testing, odour filter testing, or the testing of heat recovery systems such as 'Z'- DUCT. The refrigeration and hot water/steam systems can be used as a source of supply for other studies at the center. One area already under study at the center which can and will make use of the facility is the heat transfer performance evaluation of solar energy storage units.

At the present time, heat exchanger manufacturers are not obligated to publicize experimentally verified performance data, particularly at off peak loads. However, it's very likely that stringent energy conservation regulations will force manufacturers to make this data available in the near future. It has been found that the interpolated values for off peak performance used by most manufacturers are quite inadequate (ref. 21, pages 6-8) and a more precise evaluation is obviously required. The unit designed and constructed by Concordia's Center for Building Studies has the unique capacity to verify experimentally, the part loading performance of direct expansion refrigerant cooling coils as well as hot water and steam heating coils, up to 10 ton or 120,000 BTUH capacity.

The main objective of this paper is to describe the existing test facility and test methods and summarize the conventional theoretical models for predicting the performance of a DX coil so that comparisons can be made with test results. Since commercial air conditioning and refrigeration systems operate mostly at a part-load condition, the information generated from this project would be of considerable value to the system designers and energy conservation analysts. Future work is anticipated in this area. Specifically, the comparison of theoretical and experimental results will be forthcoming.

CHAPTER II

THE THEORECTICAL PERFORMANCE OF DIRECT EXPANSION COOLING COILS

1. Introduction:

The information and analysis which follows is based on the work of previous authors. The papers by Goodman, (28) and Brown, (10) in particular, have been very helpful. The purpose of the following discussion is to summarize the methods used by these authors to predict the performance of cooling and dehumidifying coils. The majority of the sections which follow have been extracted from the work by Goodman.

2. Heat Transfer Across A Coil:

Any coil which dehumidifies the air must do so by condensing the water vapor on the cold surfaces of the coil. A portion of the coil will usually operate dry and a portion will operate wet. On the wet portion of the coil, there will be a film of water on the coil surface. In addition, a thin layer of stagnant air will be present on the coil surface creating a surface film. On the refrigerant side, direct expansion coils have characteristics not found in water or brine coils, namely a temperature of boiling refrigerant which is essentially constant (due to evaporation) and a film of oil on the refrigerant side of the coil walls.

shown below is a diagrammatic representation of the surface films for a DX coil.

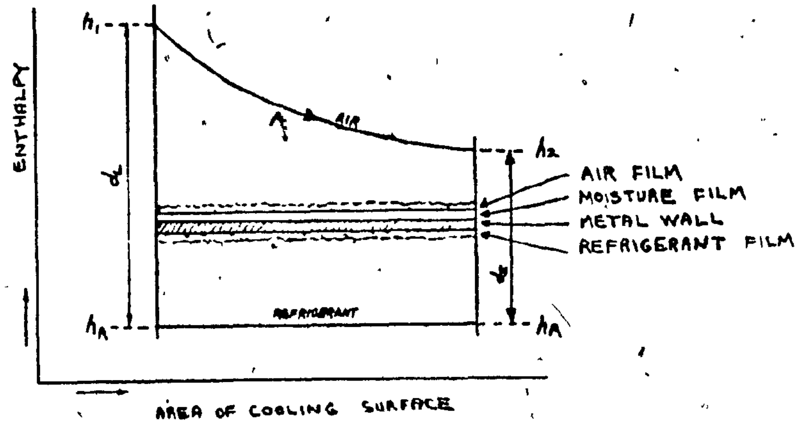


Figure 1

The general equation for the heat transfer across a wetted surface is

$$H_w = A\ U_w\ d_m \tag{1}$$

where H_w is the total heat removed by the coil (it is the sum of sensible and latent heat lost by the air) (BTU/HR)

A is the external area of the coil including any finned surface area (SQ FT)

The equation above applies equally to wet and dry surfaces, however U_w is determined differently for a wet and dry surface.

The general equation is not universal in its application because it is derived by using an empirical linear relationship between h_s and t_s. It does yield accurate results for the range of conditions ordinarily encountered in air conditioning. For application on a dry surface the equation becomes

$$H_d = A\ U_D\ t_m \tag{2}$$

When a coil causes dehumidification to take place, the water vapor in the air, which exists in the form of superheated steam, must be cooled and condensed by the cold surface. The vapor can reach the surface only by travelling through the surface air film, which it does by diffusion and convection. When the vapor reaches the cold surface, it condenses and releases its latent heat. Simultaneously, the surrounding air is cooled sensibly.

3. <u>The Overall Coefficient Of Heat Transfer Through Surfaces-</u>
<u>Method 1:</u>

The value of U_W, the overall coefficient for a wet surface, is computed from the air and refrigerant film coefficients by the same equation used to determine U_D, the overall coefficient for a dry surface.

$$\frac{1}{U_W} = \frac{c}{f_g} + B\ \frac{a}{f_r} \tag{3}$$

$$\frac{1}{U_D} = \frac{1}{f_g} + B\ \frac{1}{f_r} \tag{4}$$

The same values of the film coefficients f_g and f_r are used in both equations. The numerical value of these coefficients may be found from the available literature or from tests on actual coils. For a finned coil, the values of U_D and U_W represent the heat transferred per square foot of external surface. The value of the numerical coefficient a might be called the specific heat of saturated air since it represents the average total heat removed in cooling one pound of saturated air one degree and condensing moisture. The values of a depend only on the refrigerant temperature and are tabulated below. The values of c and a differ because the quantity of heat transferred through the air film and refrigerant are different. The reason for this is that the latent heat travels through the air film as superheated steam, as explained previously, and does not release its heat until it reaches the cold surface.

Table 1-Values of a for Use in Equation 3

t_r	a	t_r	a
20	0.372	15	0.561
21	0.377	16	0.571
22	0.382	17	0.582
23	0.388	18	0.593
24	0.393	19	0.603
25	0.398	50	0.613
26	0.405	51	0.623
27	0.411	52	0.635
28	0.417	53	0.645
29	0.423	54	0.657
30	0.430	55	0.668
31	0.438	56	0.679
32	0.445	57	0.690
33	0.452	58	0.703
34	0.460	59	0.713
35	0.468	60	0.725
36	0.476	61	0.738
37	0.485	62	0.749
38	0.493	63	0.762
39	0.503	64	0.775
40	0.512	65	0.788
41	0.522	66	0.800
42	0.531	67	0.813
43	0.541	68	0.827
44	0.551	69	0.840

The resistance of the wall of the metal tube to the flow of heat has been neglected in the previous equations. The conductivity of the wall is so high compared to the conductivities of the surface films, and the wall tube is so thin, that its resistance to the flow of heat is negligibly small compared to the resistance of the two surface films, thus justifying this assumption.

4. The Overall Coefficient Of Heat Transfer Through Surfaces-Method 2:

In dry air heat exchangers, with plain or extended surfaces, the overall coefficient of heat transfer U_D can be expressed as a function of the respective velocities v and v_r of the fluids.

$$\frac{1}{U_D} = \frac{1}{f_1} + \frac{1}{fv^p} + \frac{1}{f_r v_r^u} \qquad (5)$$

The thermal resistance of the film between the fluid on the side 2 and the wall of the heat exchanger is represented by the last term in Eq. (5). The first two terms express the respective thermal resistances of the film on the side 1 and of the fins, if any (on the side 1; the surface on the side 2 is always assumed to be plain).

Tests are made in order to determine U_D as a function of the air velocity v and the velocity v_r of the fluid on the side 2. When the exponents p and u are known, the constants f, f_1 and f_r can be determined by plotting the test results in graphs with appropriate scales on the co-ordinate axes.

If the heat exchanger is of the plain surface type, or if the temperature drop in the fins is small, then the first term in Eq. (5) represents the thermal resistance of the wall, while the second term expresses the thermal resistance of the film on the side 1.

On the other hand, if the heat exchanger is of the extended surface type, then both the thermal resistance of the film on the side 1 and the thermal resistance of the fins vary with the air velocity. In this case, a correct value of U_1 can be determined as follows.

The total surface area A on the side 1 is given by the sum of the surface area of the fins A_f and the free surface area of the tubes A_t. The temperatures at the edge of a fin and at the tube are not equal. If t_m denotes the mean value of the difference between the temperature of the surrounding air and the temperature of the fin, and if t_o designates the difference between the temperature of the surrounding air and the temperature of the tube, then we have

$$Q = U_1 (A_f \times t_m + A_t \times t_o) \quad (6)$$

on condition that the coefficient of heat transfer U_1 is assumed to be applicable both to the fin surface and to the tube surface. This assumption is to be regarded as justifiable, see e.g. Th. E. Schmidt, Die Warmeleistung von berippten Oberflachen, Karlsruhe 1950.

If we introduce an apparent coefficient of heat transfer

U_A on the side 1 which is defined by

$$Q = U_A \times A \times t_o \tag{7}$$

then U_A is the reciprocal of the thermal resistance of the film on the side 1 and of the fins. In this case, as has been shown in the above, U_A can be calculated from:

$$\frac{1}{U_A} = \frac{1}{f_1} + \frac{1}{fv^p} \tag{8}$$

Eqs. 6 and 7 yield:

$$U_A = U_1\left(\frac{Af \times t_m}{A \times t_o} + \frac{At}{A}\right) \tag{9}$$

The ratio t_m/t_o expresses the thermal efficiency of the fins. This efficiency is dependent not only on the dimensions of the fins and on the thermal conductivity of their material, but also on the coefficient of heat transfer U_1. The thermal efficiency of the common types of fins can be found in manuals, and will be denoted by η in what follows.

Now if U_1 is to be determined as a function of the air velocity v, then, to begin with, a series of values of U_A is calculated from Eq. (9) at given values of U_1. After that, Eq. (8) is used to compute v from the calculated values of U_A. The results are plotted in a graph representing U_1 as a function of v.

We consider an extended surface tube which is provided with fins consisting of a steel strip wound on the edge in a spiral around the tube (see e.g. McAdams, Heat Transmission, 2nd Edition, p.231, Fig. 116). The tube is supposed to be made of steel, and to be 20 mm in outside diameter and 17 mm in inside diameter. The thickness of the fins s is 0.0005 m and their height h is 0.0115 m. The fin tube is assumed to be galvanised, the zinc layer being 0.00014 m thick.

In a fin tube of this type, the thermal efficiency of the fins is equal to that of a straight fin. We have:

$$\eta = \frac{tgh \beta h}{\beta h} \qquad (10)$$

where

$$\beta = \sqrt{\frac{2U_1}{\lambda s}} \qquad (11)$$

(λ is the thermal conductivity of the material of the fins).

Since λ is 97 for zinc and 50 for iron, we obtain:

$$s = 97 \times 0.00028 + 50 \times 0.0005 = 0.052$$

If the pitch of the fins is 5 mm, we get $F = 0.518\ m^2$, $F_f = 0.455\ m^2$, and $F_t = 0.063\ m^2$ per running metre of tube length.

Inserting these values in Eqs. (9) and (10) gives

$$U = 12.4 \sqrt{U_1}\ tgh\ (0.071 \sqrt{U_1}) + 0.122\ U_1$$

Tests have shown that $f_1 = 0, f = 31$, and $u = 0.55$ in Eq. (5) at normal air velocities. Consequently, Eq. (8) yields

$$U = 31\ v^{0.55}$$

In this formula, v is the air velocity in the total cross-sectional area at the inlet of the heat exchanger. The distance from centre to centre of two adjacent fin tubes at right angles to the direction of air flow is 50 mm.

By using the last two equations, we obtain graphically:

$$U_1 \approx 32v^{0.6}$$

5. Mean Enthalpy Difference:

The value of d_m, the mean enthalpy difference, is found in exactly the same way as the mean temperature difference for a dry surface namely, by means of the following commonly used equation:

$$d_m = \frac{d_L - d_s}{\log_e \frac{d_L}{d_s}} \tag{12}$$

The only difference is that the enthalpy (total heat) of the air is used instead of its dry bulb temperature. In order to compute d_m, the values of d_L and d_s must first be found. Referring to Fig. 1 which has been drawn for a direct expansion coil in which the temperature of the boiling refrigerant is theoretically constant throughout the coil.

$$d_L = h_1 - h_r \tag{13}$$

$$d_s = h_2 - h_r \tag{14}$$

In these equations, h_1 is the initial enthalpy of the air entering the wet coil, h_2 is the enthalpy of the air leaving the wet coil, and h_r is the enthalpy of saturated air at a temperature equal to the temperature of the refrigerant.

If the pressure were the same throughout a direct expansion coil, the temperature of the boiling refrigerant would be constant throughout the coil. However, because of the small pressure drop of the liquid while flowing through the tubes, the temperature of the boiling refrigerant usually varies slightly. For computing the mean enthalpy difference of a direct expansion coil, the saturation temperature of the refrigerant corresponding to the average pressure inside the tubes of the coil is used. When finding this average pressure, the pressure drop in the liquid header or other liquid distributing device should be ignored: only the change in pressure while the liquid is actually inside the tubes forming the coil should be considered.

Any superheating of the refrigerant that takes place inside the coil should be ignored when computing the mean enthalpy difference. Even if some superheating does occur, only the average temperature inside the tubes of the coil should be used. In a counterflow coil, the computation of the mean enthalpy difference is based on the temperatures of the water (or other refrigerant) entering and leaving the coil - not on the average temperature of the refrigerant. The reason for the difference in computation procedure for direct expansion coils and counterflow coils is this. As the water in a counterflow coil rises in temperature, it absorbs an equal amount of heat for each degree of temperature rise. Thus if water rises 10°F in temperature while flowing through a counterflow coil, it absorbs 10% of the total heat removed by the coil for each degree rise in temperature. On the other hand, if a refrigerant vaporizes inside a coil at substantially constant temperature and is then superheated 10°F, only a very small percentage of the total heat

Table 2

Factors for Finding Log Mean Temperature Differences

$\frac{D_S}{D_L}$	$\frac{D_M}{D_L}$	$\frac{D_M}{D_S}$	M	$\frac{D_S}{D_L}$	$\frac{D_M}{D_L}$	$\frac{D_M}{D_S}$	M
1	2	3	4	1	2	3	4
0.00	0.0000			0.25	0.5440	2.164	1.3863
0.01	0.2150	21.500	4.6052	0.26	0.5494	2.113	1.3470
0.02	0.2505	12.525	3.9420	0.27	0.5575	2.065	1.3094
0.03	0.2766	9.220	3.5065	0.28	0.5657	2.020	1.2729
0.04	0.2982	7.455	3.2189	0.29	0.5736	1.978	1.2378
0.05	0.3171	6.312	2.9957	0.30	0.5845	1.938	1.2039
0.06	0.3311	5.568	2.8136	0.31	0.5891	1.900	1.1712
0.07	0.3497	4.996	2.6596	0.32	0.5968	1.865	1.1394
0.08	0.3643	4.554	2.5257	0.33	0.6014	1.832	1.1086
0.09	0.3779	4.199	2.4079	0.34	0.6118	1.799	1.0788
0.10	0.3909	3.909	2.3026	0.35	0.6192	1.769	1.0498
0.11	0.4032	3.666	2.2072	0.36	0.6264	1.740	1.0217
0.12	0.4451	3.459	2.1199	0.37	0.6336	1.712	0.9944
0.13	0.4265	3.281	2.0399	0.38	0.6407	1.686	0.9677
0.14	0.4375	3.125	1.9657	0.39	0.6479	1.661	0.9416
0.15	0.4479	2.986	1.8976	0.40	0.6548	1.637	0.9163
0.16	0.4581	2.865	1.8326	0.41	0.6617	1.614	0.8916
0.17	0.4685	2.756	1.7716	0.42	0.6686	1.592	0.8675
0.18	0.4780	2.656	1.7156	0.43	0.6752	1.570	0.8442
0.19	0.4879	2.568	1.6601	0.44	0.6820	1.550	0.8211
0.20	0.4971	2.486	1.6094	0.45	0.6889	1.531	0.7984
0.21	0.5062	2.411	1.5687	0.46	0.6954	1.512	0.7766
0.22	0.5152	2.342	1.5140	0.47	0.7018	1.493	0.7552
0.23	0.5239	2.278	1.4697	0.48	0.7086	1.476	0.7338
0.24	0.5325	2.219	1.4272	0.49	0.7148	1.459	0.7134

Table 2 Cont'd.

Factors for Finding Log Mean Temperature Differences

$\frac{D_S}{D_L}$	$\frac{D_M}{D_L}$	$\frac{D_M}{D_S}$	M	$\frac{D_S}{D_L}$	$\frac{D_M}{D_L}$	$\frac{D_M}{D_S}$	M
1	2	3	4	1	2	3	4
0.50	0.7213	1.443	0.6932	0.75	0.8698	1.160	0.2874
0.51	0.7276	1.427	0.6735	0.76	0.8740	1.150	0.2746
0.52	0.7341	1.412	0.6539	0.77	0.8792	1.142	0.2616
0.53	0.7402	1.397	0.6350	0.78	0.8856	1.135	0.2484
0.54	0.7464	1.382	0.6163	0.79	0.8904	1.127	0.2359
0.55	0.7528	1.369	0.5977	0.80	0.8963	1.120	0.2231
0.56	0.7586	1.355	0.5800	0.81	0.9015	1.113	0.2108
0.57	0.7653	1.343	0.5619	0.82	0.9071	1.106	0.1984
0.58	0.7711	1.330	0.5447	0.83	0.9125	1.099	0.1863
0.59	0.7770	1.317	0.5277	0.84	0.9176	1.091	0.1744
0.60	0.7827	1.305	0.5110	0.85	0.9228	1.086	0.1626
0.61	0.7893	1.294	0.4941	0.86	0.9282	1.079	0.1508
0.62	0.7948	1.282	0.4781	0.87	0.9336	1.073	0.1392
0.63	0.8011	1.272	0.4619	0.88	0.9385	1.067	0.1279
0.64	0.8061	1.260	0.4466	0.89	0.9440	1.061	0.1165
0.65	0.8130	1.251	0.4305	0.90	0.9492	1.055	0.1054
0.66	0.8184	1.240	0.4154	0.91	0.9552	1.050	0.0942
0.67	0.8234	1.229	0.4008	0.92	0.9590	1.042	0.0834
0.68	0.8291	1.219	0.3859	0.93	0.9642	1.037	0.0726
0.69	0.8359	1.212	0.3709	0.94	0.9701	1.032	0.0619
0.70	0.8404	1.201	0.3570	0.95	0.9794	1.027	0.0513
0.71	0.8475	1.194	0.3422	0.96	0.9792	1.020	0.0409
0.72	0.8522	1.184	0.3286	0.97	0.9859	1.016	0.0304
0.73	0.8577	1.175	0.3148	0.98	0.9906	1.011	0.0202
0.74	0.8642	1.168	0.3009	0.99	0.9950	1.005	0.0101
				1.00	1.0000	1.000	0.0000

removed by the coil will be absorbed by the superheated vapor. By far the major part of the heat transfer takes place while the refrigerant is boiling at constant temperature: the amount absorbed by the superheating of the vapor is negligibly small by comparison. Hence, inasmuch as the refrigerant absorbs practically all of the heat while its temperature is constant, only this constant temperature should be used in computing either the mean temperature difference or the mean enthalpy difference. Furthermore, the derivation of the equations given is based on the heat transfer process just described.

6. <u>Area of Wetted Surface Required For A Direct Expansion Coil</u>:

After finding the values of U_w and d_m as described in the preceeding sections, the general equation can be used to find the area of wetted surface needed to cool and dehumidify a given quantity of air.

<u>Example</u>: Air at an initial wet bulb temperature of $72^{O}F$ is to be cooled to a final wet bulb temperature of $58^{O}F$ with a refrigerant at $42^{O}F$. The total quantity of air to be cooled is 12,500 cfm. For the particular coil in question, $f_g = 12$, $f_r = 300$, and $B = 17$. Find the total area of wetted surface required to cool and dehumidify the air.

Solution

From Table 1, the value of a is .531

$$\frac{1}{U_w} = \frac{.243}{12} + 17 \times \frac{.531}{300} = .05034$$

U_w = 19.9 BTU/HR/SQ FT/BTU of enthalpy difference.

Using the given data and a psychrometric chart:

$$d_L = 35.77 - 16.15 = 19.62 \text{ BTU}$$

$$d_s = 25.09 - 16.15 = 8.94 \text{ BTU}$$

$$d_m = \frac{d_L - d_s}{\log_e \frac{d_L}{d_s}} = \frac{19.62 - 8.94}{\log_e \frac{19.62}{8.94}} = 13.60 \text{ BTU, mean enthalpy difference}$$

$$G = 12{,}500 \text{ CFM} \times 4.5 = 56{,}250$$

Using a psychrometric chart to yield h_1 and h_2

$$H_w = G(h_1 - h_2)$$

$$= 56{,}250\ (35.77 - 25.09) = 600{,}800 \text{ BTU per hr.}$$

$$A = \frac{H_w}{U_w\ d_m}$$

$$= \frac{600{,}800}{19.9 \times 13.60} = 2220 \text{ sq. ft. of wetted external surface}$$

7. Final Wet Bulb and Dry Bulb Temperature of Air Leaving a Direct Expansion Coil:

The general equation can also be used to find the final wet bulb temperature of the air leaving a coil of a given area. For this purpose, the value of the factor M defined by the following equation must first be found:

$$M = \frac{A\ U_w}{G} \qquad (15)$$

After finding the value of M for a given coil, the value of d_s/d_L can be found by referring to Table 2.

From this the final enthalpy and, therefore, the final wet bulb temperature of the air can be determined.

For any given area of wetted surface and any given air velocity the final wet bulb depression of the air is always a definite fraction of the initial wet bulb depression as shown in equation (16). The following equations can be used to determine the final dry bulb temperature of air leaving DX coil.

$$\frac{D_s}{D_L} = e^M \qquad (16)$$

$$D_L = t_1 - t_1^1 \qquad (17)$$

$$D_s = t_2 - t_2^1 \qquad (18)$$

$$M = \frac{A \times f_g}{G \times c} \qquad (19)$$

Also from (16) and (19)

$$\log \frac{D_L}{D_s} = \frac{A \times f_g}{G \times c} = M \tag{20}$$

Using these equations, the value of t_2, the final dry bulb temperature of the air leaving the coil, can be found.

Example: Air at an initial wet bulb temperature of $72^{o}F$ and a dry bulb temperature of $90^{o}F$ is to be cooled by a coil with 1600 sq. ft. of wetted external area. The refrigerant temperature is $42^{o}F$. The total quantity of air to be cooled is 12,500 cfm. For the particular coil in question, $f_g = 12$, $f_r = 300$, and $B = 17$. Find the final wet bulb and dry bulb temperatures.

$$M = \frac{A \times U_w}{G} = \frac{1600 \times 19.9}{56,250} = 0.566$$

(U_w and G determined in previous example)

Referring to Table 2, for $M = 0.566$, $d_m/d_L = 0.7636$.
From the previous example $d_L = 19.62$.
Therefore, $d_m = 14.98$ BTU.

$$H_w = A\ U_w\ d_m = 1600 \times 19.9 \times 14.98$$
$$= 477,000 \text{ BTU/HR.}$$

The heat removed per lb. of air flowing through the coil is

$$\frac{H_w}{G} = \frac{477,000}{56,250} = 8.48 \text{ BTU per lb. of air.}$$

From the previous example h_1 = 35.77 BTU per LB of Air.

Therefore, h_2 = 35.77 - 8.48 = 27.29 BTU Per LB of Air.

From the psychrometric chart for h_2 = 27.29 BTU Per LB of Air.

t_2^1 = 61.3 ^{O}F, the final wet bulb temperature of the air leaving the coil.

$$D_L = t_1 - t_1^1 = 90 - 72 = 18^{O}F$$

$$M = \frac{A \times f_g}{G \times c} = \frac{1600 \times 12}{56,250 \times .243} = 1.40$$

Referring to Table 2, or Equation (15), for $M = 1.40$, $D_s/D_L = 0.2466$

Therefore, for $D_L = 18^{O}F$, $D_s = 0.2466 \times 18 = 4.4^{O}F$

and $t_2 = D_s + t_2^1 = 61.3 + 4.4 = 65.7^{O}F$

8. The Condition Curve:

If all of the conditions of the example given previously had been kept constant except that the area of the coil was varied by increasing the rows of tubes composing the coil, the final dry and wet bulb temperatures of the air leaving these various rows could be found by the methods illustrated previously. Then if the final condition of the air leaving these coils of different numbers of rows of tubes were plotted on a psychrometric chart, all of these points would fall on a curve of the form illustrated in Fig. 2. This curve is called the "condition curve" because it shows the condition of the air at any point in its passage over a cold wetted surface. As the number of rows of tubes is increased, the point representing the final condition of the air will move down along the condition curve toward the saturation curve.

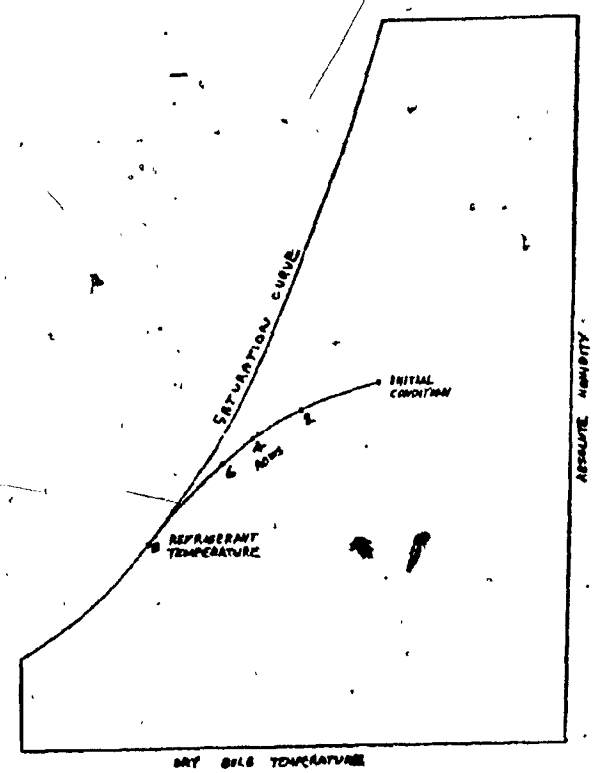


FIGURE 2

A second method exists for drawing the condition curve which illustrates the basic physical process involved in the cooling and dehumidifying of air.

The change in the condition of the air as it flows over a cold, wetted surface can be illustrated by considering the microscopically thin film of air in contact with the wetted surface. This thin film of air in contact with the surface is stationary even though the main air stream flows past the surface with a turbulent motion. Continuous mixing takes place between the moving air stream and the stationary surface film because the turbulent movement of the main air stream literally tears away portions of the surface film. Also because of this turbulent movement, infinitesimal portions of the main air stream are driven into the surface film to replace the portions of the film carried away. This mixing process takes place in addition to the processes of heat transfer and moisture diffusion.

Now if the air film in contact with the wetted surface is considered to be saturated, the temperature of this microscopically thin film of saturated air at a point on the surface is equal to the surface temperature of the metal at that point. Inasmuch as the surface temperature varies throughout the coil, the temperature of the saturated surface film also varies. Therefore, as the air to be cooled travels through the coil, it is constantly being mixed with colder air from the saturated surface film. This mixing process can be used to trace the condition curve on the psychrometric chart.

For a constant refrigerant temperature, the temperature of the surface and therefore, the temperature of the stationary, saturated film of air at any point on the coil depends only upon the wet bulb temperature of the main air stream at that point. The surface temperature corresponding to these various wet bulb temperatures can be computed. The condition curve plotted by this second method is shown in Figure 3.

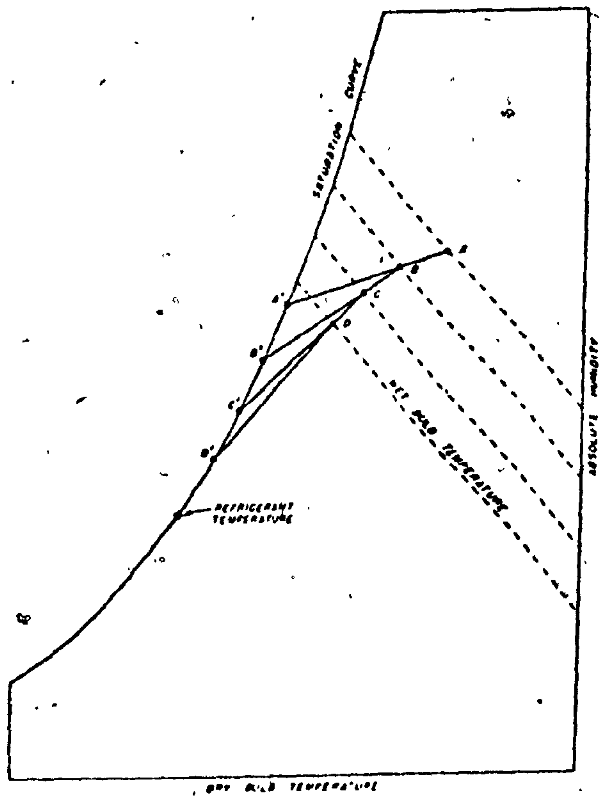


FIGURE 3

The methods used to determine the surface temperature and to draw the condition curve are illustrated in the work by Goodman. The results of these two methods yield substantially the same condition curve.

9. The Boundary Condition:

All of the preceeding discussion has been based on the assumption that the entire surface of the coil will be covered with a thin film of moisture condensed from the air. However, under some conditions, dehumidification of the air cannot begin until after the air has penetrated some distance through the coil.

The temperature of a point on the dry surface of a coil will be at some point between the dry bulb temperature of the air flowing over the coil and the refrigerant temperature. Inasmuch as the dry bulb temperature of the air falls as it flows over a dry coil, the surface temperature of the coil must also fall in the direction of air flow.

When drawing the condition curve on the psychrometric chart, the condition of the air flowing over the dry part of the coil is represented by a horizontal line drawn from point A to point B in Fig. 4 because no condensation of moisture takes place until the boundary is reached. The dry bulb temperature through the point marked B is called the boundary dry bulb

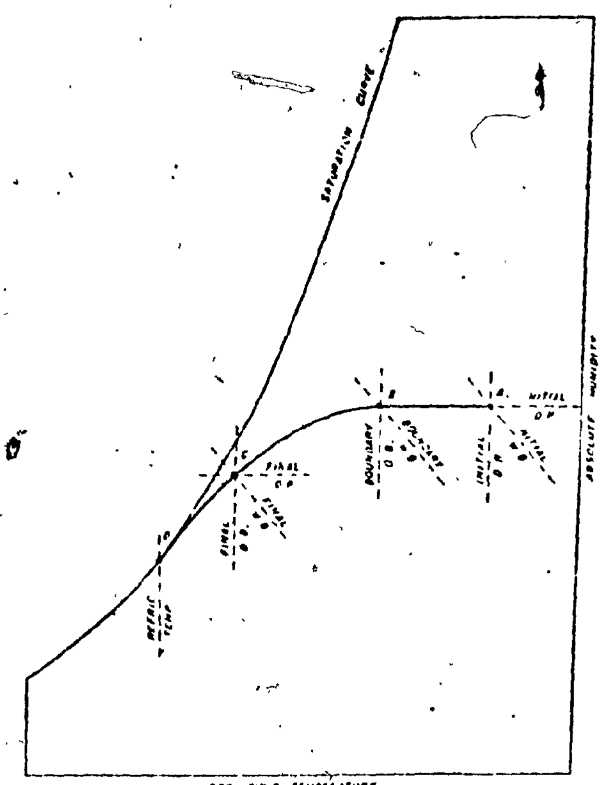


FIGURE 4

temperature because it is the dry bulb temperature of the air when it flows over the imaginary boundary line between the dry and wet portions of the coil. From point B onward, the curve drops as the air is dehumidified by the cold surface.

The following equation is used to find the boundary dry bulb temperature of the air flowing through a direct expansion coil:

$$D_b = \left(1 + \frac{1}{R}\right) D'' \qquad (21)$$

or

$$t_b = \frac{1}{R}(t''_1 - t_r) + t''_1 \qquad (22)$$

If the initial dry bulb temperature of the air entering a coil is higher than the boundary dry bulb temperature found by Equation 21 above, part of the coil will be dry. On the other hand, if the initial dry bulb temperature of the air is equal to or lower than the boundary dry bulb temperature, the entire surface of the coil will be wet; there will be no dry surface.

For a given coil and a given refrigerant temperature, Equation 22 can be used to find the boundary dry bulb temperature corresponding to different initial dew point temperatures. Plotting these boundary dry bulb temperatures and dew point temperatures on the psychrometric chart for different refrigerent temperatures results in a set of curves. Once plotted, these curves of constant refrigerant temperatures are of considerable value because they show at a glance the conditions under which the coil will be entirely wet, or partially wet and partially dry. If the initial condition of the air is represented by a point on one side of the curve for the given refrigerant temperature, the entire surface will be wet. On the other hand, if the initial condition of the air is represented by a point on the other side of the temperature curve, a portion of the surface will be dry.

If the final dry and wet bulb temperatures of the air leaving a coil must be found, only the area of the wetted portion of the coil should be used. Furthermore, the wet bulb temperature of the air entering the wetted portion of the coil will be lower than the initial wet bulb temperature of the air because of the sensible heat removed in cooling the air from the initial dry bulb temperature to the boundary dry bulb temperature. The temperatures and enthalpies at various points in a coil that is partly wet and partly dry are illustrated in Fig. 5.

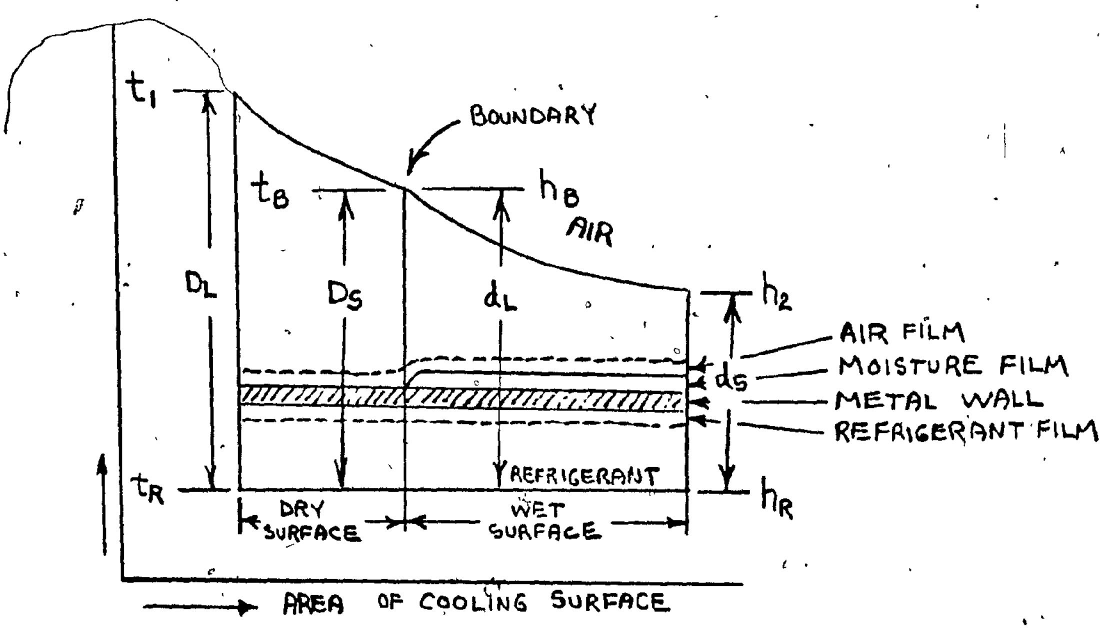


FIGURE 5

10. Determining Film Coefficients:

The values of the film coefficients can be determined by tests on coils without any difficult measurements of surface temperature. For this purpose, tests are first run on completely wetted coils- there should be no dry surface on the coil during the test. Dry surface can be detected easily as will be explained later. After running a series of tests with wet surfaces, the values of U_w can be found by means of Equation 1.

After the values of U_w have been determined, either one of two methods can be used to determine the individual film coefficients. The first method requires that a second series of tests be run on the coil with the surface completely dry. From such tests the values of U_D can be computes by means of Equation 2. Knowing the values of U_D and U_w for the different air and refrigerant velocities for which the coil was tested, the values of f_r can be determined by means of the following equation which is obtained by eliminating f_g between Equations 3 and 4.

$$f_r = \frac{B(a-c)}{\frac{1}{U_w} - \frac{c}{U_D}} \qquad (23)$$

After the values of f_r have been evaluated for the various tests, they can be used in Equation 4 to solve for f_g. If the coefficients are obtained in this manner, the results can be applied accurately to coils of various areas which differ markedly from the area of the coil tested.

The second method used to evaluate the film coefficients does not require any dry surface tests. Equation 16 is used to determine the values of f_g from the results of the wet surface tests alone. With the values of both f_g and U_w known, Equation 3 can then be used to find the values of f_r.

The results obtained by the two methods are in fair agreement, although tests analyzed by the second method do not give results that are as accurate as those obtained by running dry surface tests in addition to the wet surface tests.

11. Limitations of Goodman's Analysis:

The previous analysis has been challenged by a number of people, including Mr. Gosta Brown of the Royal Institute of Technology, Stockholm (1954). Goodman's analysis provides a simple, straightforward presentation of the heat transfer phenomenon. For that reason it proves useful to understanding the heat transfer process. The assumptions which have been questioned include the following:

1. The superheating of refrigerant in the coil is not considered. The justification is that the ammount of heat absorbed to do the superheating is only a very small percentage of the total heat and can be ignored.
2. Enthalpy changes due to the pressure drop in the coil are resolved by assuming a refrigerant temperature which corresponds to the average pressure inside the tubes of the coil.

3. The conductivity of the metal walls and fins is assumed infinite. The great difference between the transmission of the air and refrigerant films and the transmission of the metal walls and fins was used by Goodman to justify the omission of these factors. This assumption does not permit any consideration of the efficiencies of the fins under wet and dry conditions. Section 4 of this chapter presents a method of evaluation which permits incorporation of the fin efficiencies.

4. The effect of the boiling phenomenon of the refrigerant was oversimplified.

5. The effect of the refrigerant oil film on the inside of the tube walls was not considered in the analysis.

CHAPTER III

THE TEST FACILITY AND DESIGN CRITERIA

1. General Description of Test Chamber and Testing Procedure:

As mentioned in Chapter 1, Concordia University has constructed a facility for testing heat exchangers. This chapter describes that facility. The test chamber consists of a closed-loop air circulation system driven by a variable volume air handling unit. Air at a pre-determined rate (velocity) is passed through a flow adjustment device, where the velocity is kept at a desired value before approaching the test coil. The flow adjustment device helps to maintain a constant approach velocity profile as required by the A.R.I. and A.S.H.R.A.E. Standards (maximum allowable velocity variation at the coil surface is ± 2% of the test velocity).

To represent various loads on the coil, as recommended by the standards, the four test velocities of 800, 600, 400, and 200 F.P.M. at the upstream face of coil are chosen. The rationale for selection of these velocities is discussed in later sections.

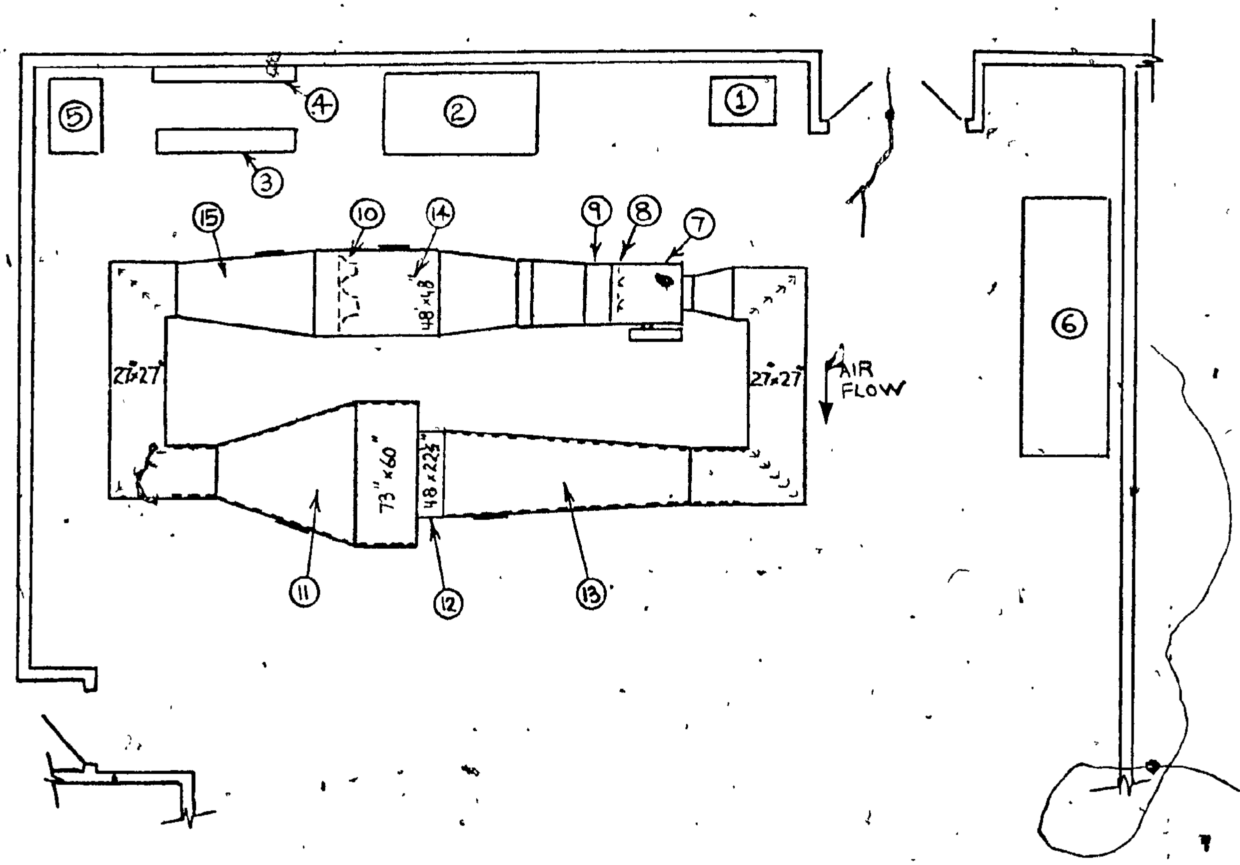
The test coil cools and de-humidifies the incoming air. To maintain constant test conditions in the closed loop system a heating coil and a humidifier of appropriate size are located down-stream of the test coil to bring the incoming air back to the original psychrometric conditions. After a steady state condition is reached for a given face velocity at the test coil face, these conditions are maintained for a period of 45 minutes. During this period, the air mass flow and the heat transfer rates are monitored at two-minute intervals by appropriate instrumentation and the heat transfer efficiency of the heat exchanger is calculated from the observed data for the 45 minute test period. The procedure is repeated for the four test velocities under standard test conditions and the results are tabulated on the forms recommended by the A.R.I. standard. An Automatic Data Acquisition System is used to collect the required test data from all the measuring instruments and to record the data on a magnetic tape for computer analysis.

2. General Design Criteria For Test Equipment:

2.1 Test Chamber Layout:

In most of the previous studies on the performance of cooling and heating coils, an open loop system was used for testing purposes (11) and (12). One of the main problems with an open ended system is the difficulty in controlling the incoming air psychrometric conditions accurately due to fluctuations of temperature and humidity in the room during the test period. Dependency on an exterior environment beyond the control of the test equipment is a major factor in the control of the test conditions.

Based on this criteria, a closed loop system was selected for this test unit. With proper insulation and proper sizing of heating, cooling and humidifying equipment, the incoming air psychrometric conditions can be controlled much

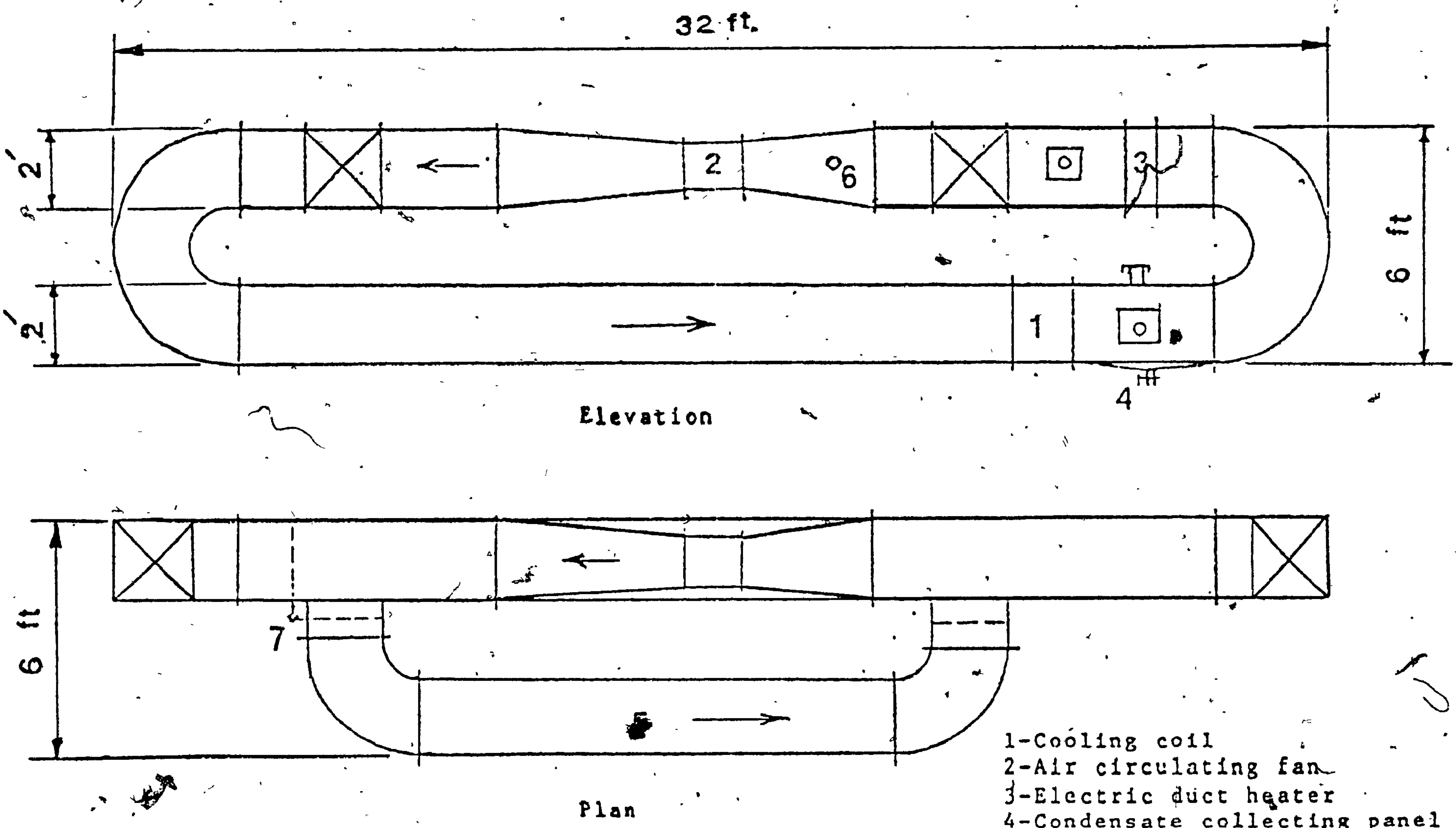


LEGEND

1.	60 KW Boiler	9.	Hot Water Coil
2.	10 Ton Refrigeration Unit	10.	Discharge Nozzles
3.	Control Panel	11.	Insulated Mixing Chamber
4.	Electrical Panels	12.	Test Coil
5.	Data Acquisition System	13.	Insulated Intake Chamber
6.	Heat Exchanger Rack	14.	Discharge Chamber
7.	Variable Volume Fan	15.	Intake Chamber
8.	Humidifier Jets		

CONCORDIA'S HORIZONTAL TEST FACILITY LAYOUT

FIGURE 6



1-Cooling coil
2-Air circulating fan
3-Electric duct heater
4-Condensate collecting panel
5-By-pass duct
6-Location of the steam humidifier
7-Mechanical dampers

A TYPICAL VERTICAL CLOSED LOOP LAYOUT

FIGURE 7

more closely with a closed loop system, as compared to an open loop system. With a closed loop system, the designer has two possible options: a vertical loop or a horizontal loop as shown in Figures 6 and 7, each with its own advantages and disadvantages. The vertical loop, for example, occupies less floor space provided the ceiling height in the test room is adequate for proper installation. On the other hand, a horizontal loop provides easy access to all parts of the loop. In this particular case, because of the limited ceiling height (9 feet) and the requirement to test both heating and cooling coils, a horizontal loop was selected.

2.2. Cooling Equipment:

Some of the criteria used for selecting the maximum capacity of the cooling equipment were:

- Available budget
- Available space
- Cost and performance of instrumentation required for control and monitoring functions of the system
- Range of capacity of industrial equipment where there is a 'gap' in the available performance data.
- Ease of operation of overall system and controls.

Taking into account the considerations listed above, a capacity limit of 10 tons was chosen.

This limit of 10 tons was selected because it included a majority of the coils produced for commercial air conditioning purposes by a majority of the local coil manufacturers.

The initial research activity will examine three coils of different configurations.

A. 10 ton 4 row straight fin coil

B. 10 ton 3 row straight fin coil

C. 5 ton 4 row straight fin coil, manufactured by Blanchard-Ness Inc. of St. Hubert, Quebec.

REFRIGERATION SYSTEM COMPONENT LIST:

1 Refrigeration compressor, open type belt driven nominal capacity: 10TR at 40° F ET, 110° FCT with crankcase heater

1 Compressor steel base

1 Open drip proof electrical motor 15 HP, 575 V - 3 - 60 for across-the-line start

1 High pressure cut-out

1 Low pressure cut-out

1 Oil failure switch

1 Discharge line oil separator

1 High pressure gauge

1 Low pressure gauge

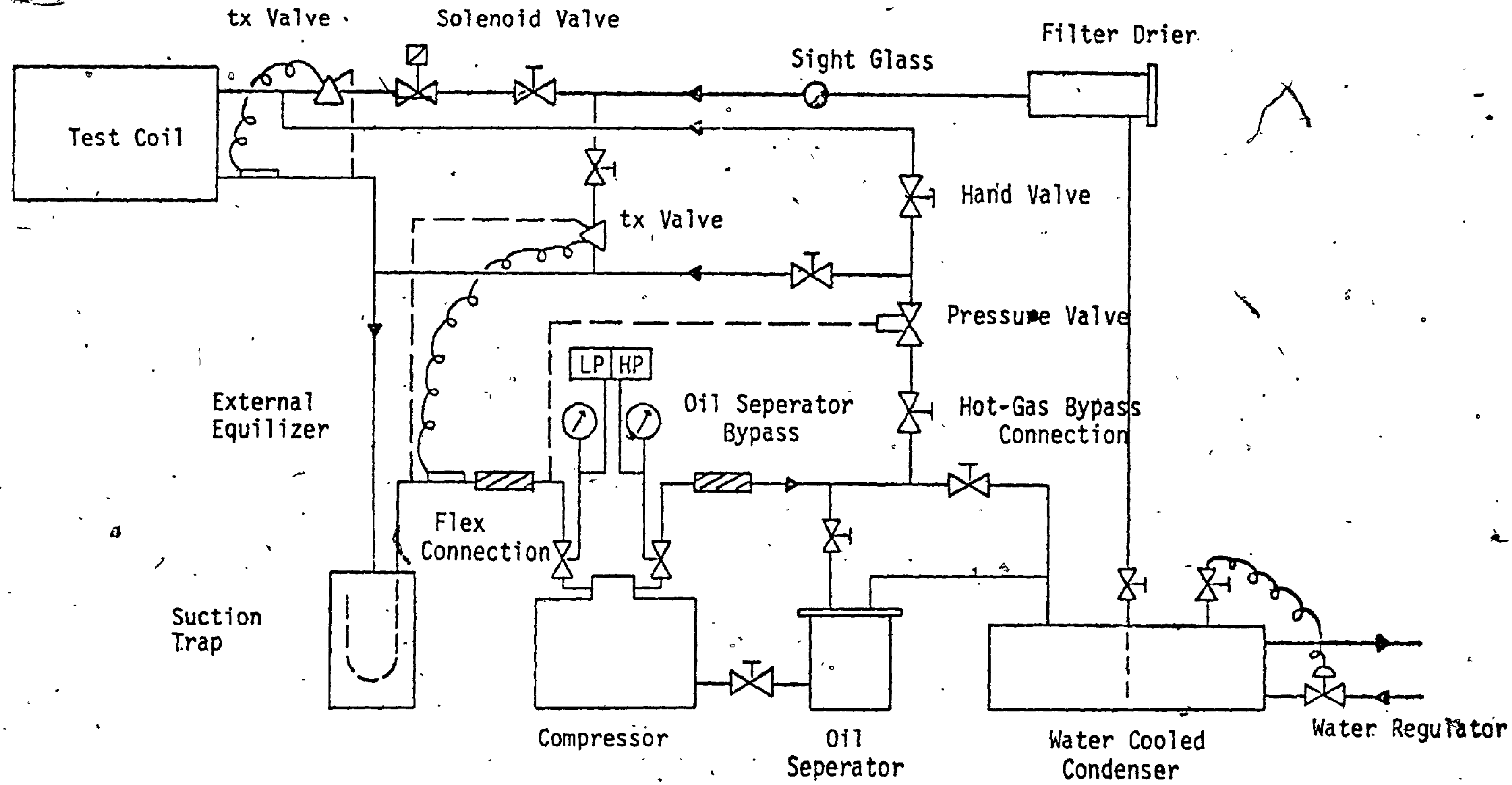
1 Oil pressure gauge

1 Discharge line globe-valves 1 1/8" for oil separator and hot gas bypass

1 Water cooled shell and tube condenser

1 2-way water regulating valve 1 1/4" NPT

1 Liquid line shut-off valves 5/8"



REFRIGERATION FLOW DIAGRAM

FIGURE 9

REFRIGERATION SYSTEM COMPONENT LIST CONT'D.

1 Filter-drier shell 5/8" for replaceable cores

2 High efficient filter-drier cores

1 Liquid and moisture indicator 5/8"

1 Suction line liquid accumulator 1 3/8"

1 Liquid line solenoid valve 5/8" 110 V coil

1 Steam boiler - safety valve
- sight glass
- blow down
- level control
- . . .

6 gate valves - 3 x 3/4"
- 3 x 1/2"

2.3 Air Handling Unit:

The basic capability of the required air handling unit is to supply air at various load conditions (0 to 100%) to match the 10 ton refrigeration system. The air handling unit must have the capability of varying the volume to facilitate testing at part loads.

Some of the methods of air flow variation are:

- A variable volume fan with
  a) inlet vane control
  b) outlet vane control
- Constant speed fan with air-by-pass circuit
- Variable speed motor drive
- Variable speed drive (belt driven)

In this particular case after considering price, delivery time, availability of service and available performance data, a 'Mark-Hot' - vari-mark, outlet damper controled backward-curved-vane centrifugal fan, manufactured by a local manufacturer was selected.

3. Major Steps In Test Chamber Design:

Step 1:

For cooling and de-humidifying purposes, a nominal capacity of 400 C.F.M./ ton of refrigeration is recommended by the A.S.H.R.A.E. 1975 Equipment Handbook Pg.6.8. The minimum air required for the 10 ton coil is therefore, 10 x 400 = 4000 C.F.M. Allowing for possible leakage and better flexibility a maximum of 6000 C.F.M. was selected.

Step 2:

The maximum velocity across the face of the cooling and de-humidifying coil without condensed-water-droplet-entrainment is about 550 C.F.M. per A.S.H.R.A.E. 1975 Equipment Handbook Pg.6.6. From the previous data, the maximum coil surface which can be tested for the selected C.F.M. of 6000 C.F.M. is 6000 ÷ 550 = 11 sq. ft.

Step 3:

The next step is to establish the configuration of the test chamber. Following the specifications of A.S.H.R.A.E. standard 33-64 and A.R.I. standard 410-67, the test chamber must incorporate the following components:

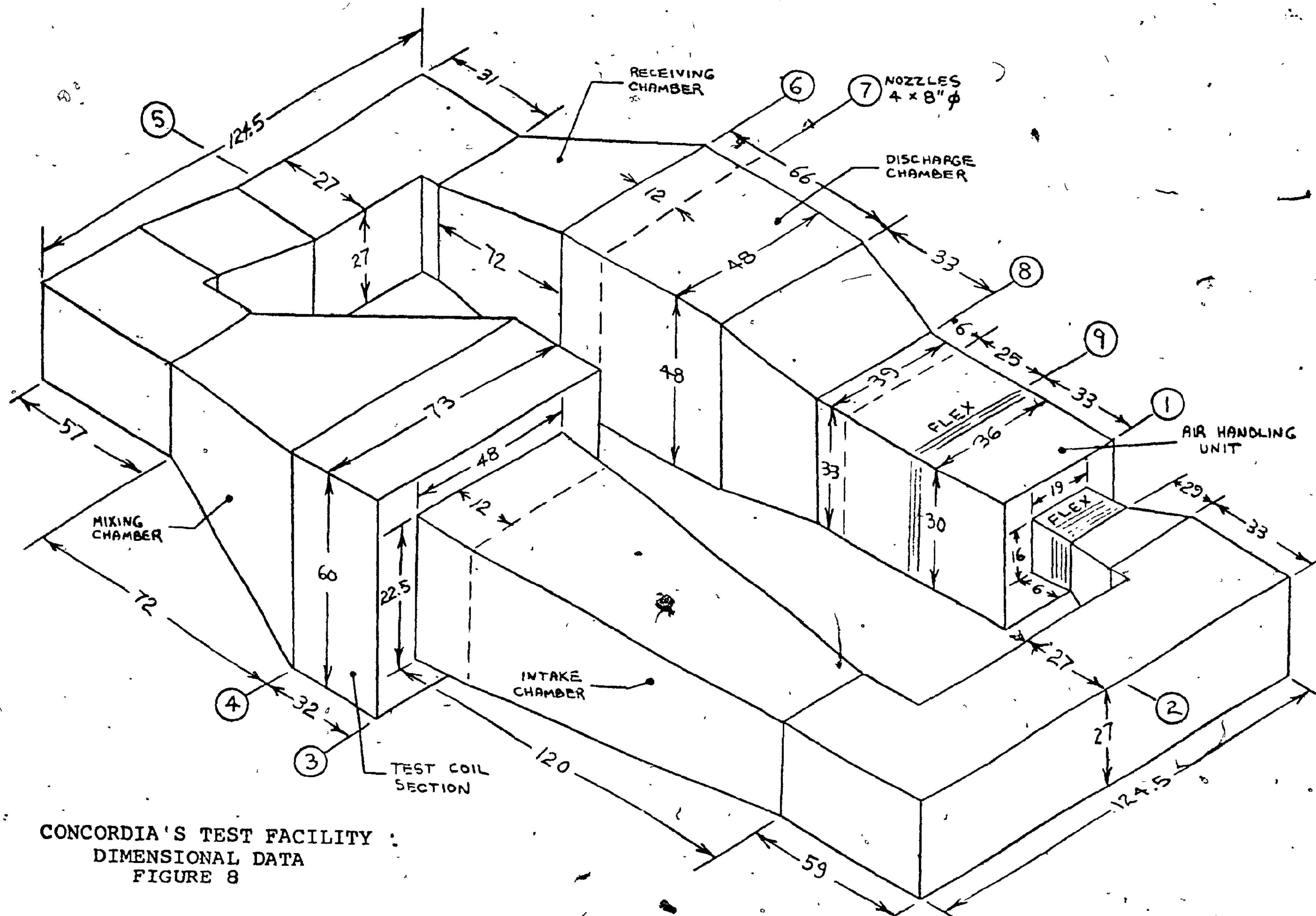
- Air handling unit
- In-take chamber
- Test chamber
- Mixing chamber
- Receiving chamber
- Flow measuring nozzles
- Discharge chamber

After a number of trials, and taking into account the flexibility necessary for the installation of new cooling and heating coils in the future, the configuration was selected (see Fig. 8). The selected configuration incorporates the following features:

1) The modular structure facilitates easy change-over of test coils.
2) Provides easy access to all parts of the chamber for observation and service.
3) Facilitates easy mounting and observation of measuring instruments and controls.

Step 4:

In addition to supplying the necessary C.F.M., the circulating fan must generate enough static pressure to overcome friction caused by the test coil, heating coil, flow control device and flow measuring nozzle section. The following pressure drop limits were selected for various sections in the closed-loop.



CONCORDIA'S TEST FACILITY :
DIMENSIONAL DATA
FIGURE 8

Test coil:	0.75 in water gauge
Flow measurement nozzles:	0.50
Heating coil:	0.50
Duct friction:	0.25
Flow regulating device:	1.00
	3.00
Allowance for future testing:	1.00
	4.00

Recommended fan static pressure: 4" WG

Step 5:

Heating coil: To balance the 10 ton refrigeration load, a 120,000 BTUH, 9 F.P.I. hot water coil was selected which is installed in the selected air handling unit. Allowing a temperature drop of 20°F (max.) in the heating coil, the necessary hot water flow rate is 120,000/ (8.33 x 60) x 20 = 12 USGPM (8.33 lbs/ USGPM and 60 min./ hour). A temperature drop of 20°F is standard practice in heating coils for commercial installations. Therefore, the hot water circulation pump must have a minimum 12 GPM capacity.

Step 6:

Humidification: Assuming a maximum latent load of 70% of coil capacity, the humidification requirement is

$$\frac{0.7 \times 120{,}000}{970} = 86.5\text{lb/hr. of steam.}$$

Maximum humidification load = 0.7 x 120,000 = 84,000 BTUH

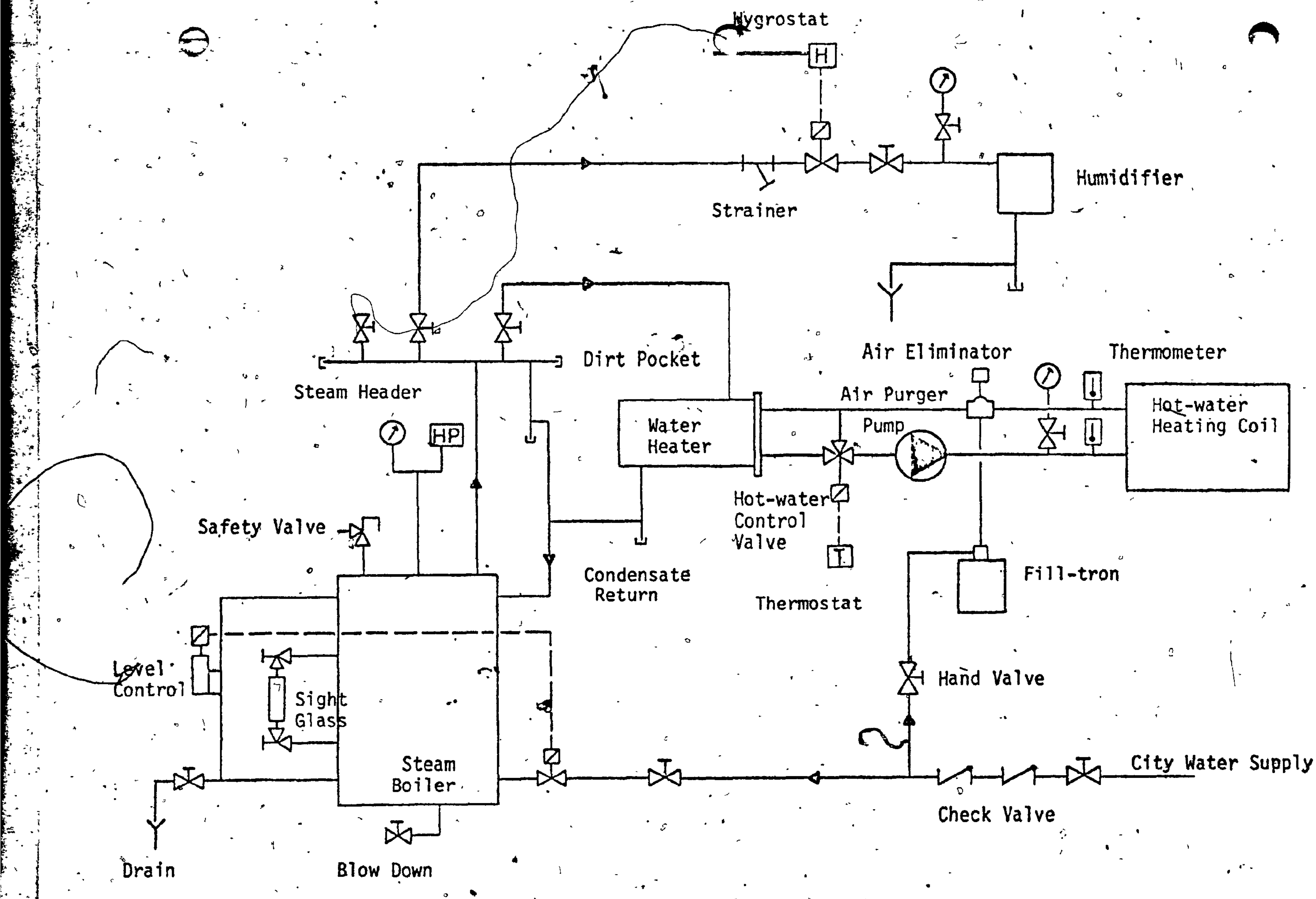
Step 7:

Boiler Size: The boiler selected has to meet the requirements of heating coil and the humidifier. Therefore, the total boiler capacity = 120,000 + 84,000 = 204,000 BTUH

= 204,000 x 0.000293 = 60 KW

Due to the space implications and fire-protection regulations in the laboratory, a 60 KW electric boiler with a 200 lbs/hr steam capacity and 100 PSIG pressure was selected.

The steam is fed to the steam-jet humidifier located on the intake side of the fan. The hot water is circulated by a pump from a heat exchanger to which steam is supplied from the boiler and water is supplied from the city water supply.



HEATING & HUMIDIFYING FLOW DIAGRAM

FIGURE 10

FIGURE 10 CONT'D.

HEATING SYSTEM COMPONENT LIST:

1 water heater

1 pump

1 ex-trol unit - purger
- float vent
- auto. feed valve
- exp. tank

1 3-way hot water control valve

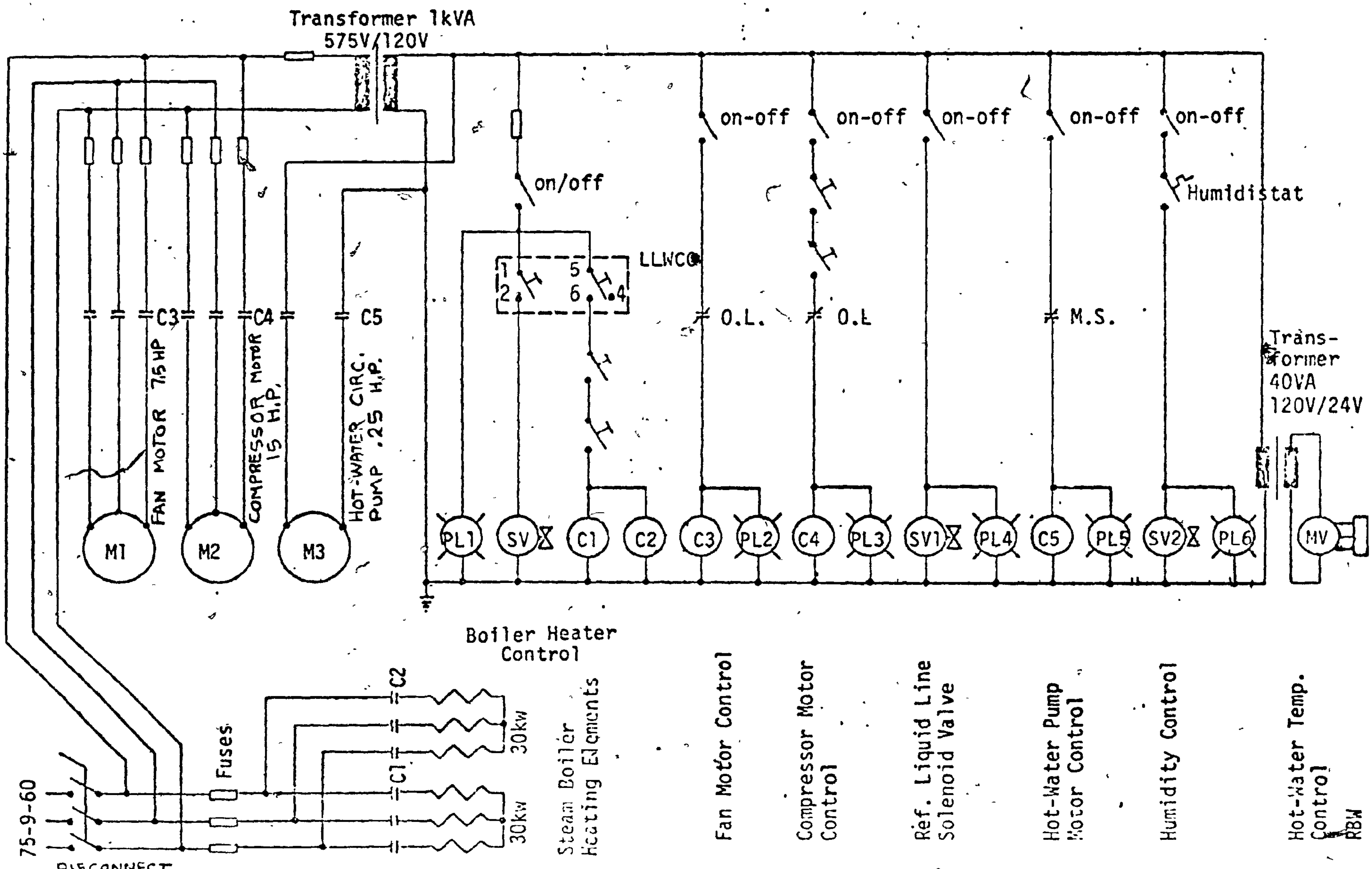
3 strainers - 1 x 3/4"
- 2 x 1/2"

2 check valves - 2 x 1/2"

2 stem type thermometers with separate wells

1 steam trap

1 hygrostat



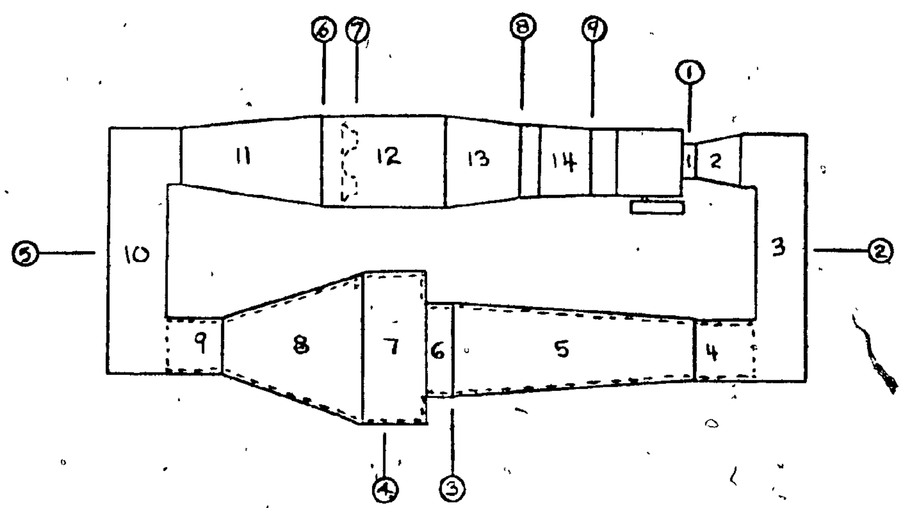
POWER AND CONTROL DIAGRAM
FIGURE 11

CHAPTER IV

THEORETICAL PERFORMANCE ANALYSIS OF TEST EQUIPMENT

1. Air Handling Unit Performance:

As stated in the previous section, a variable volume fan capable of producing 6000 CFM @ 4" S.P. drives the air around the closed loop. The chosen test coil face velocities are 200 FPM; 400 FPM; 600 FPM and 800 FPM. The first coil selected for testing has the face dimensions of 48" x 22½" and an area of 7.5 ft.^{2}. The corresponding air volumes are 1500 CFM; 3000 CFM; 4500 CFM and 6000 CFM -(area in Sq. Ft. x test face velocity). Figure 12 illustrates the reference points selected for the theoretical velocity, pressure, and temperature profiles which are covered in the following pages. Table 3 gives the tabulated results for the air side velocities and Figure 13 presents these results graphically.

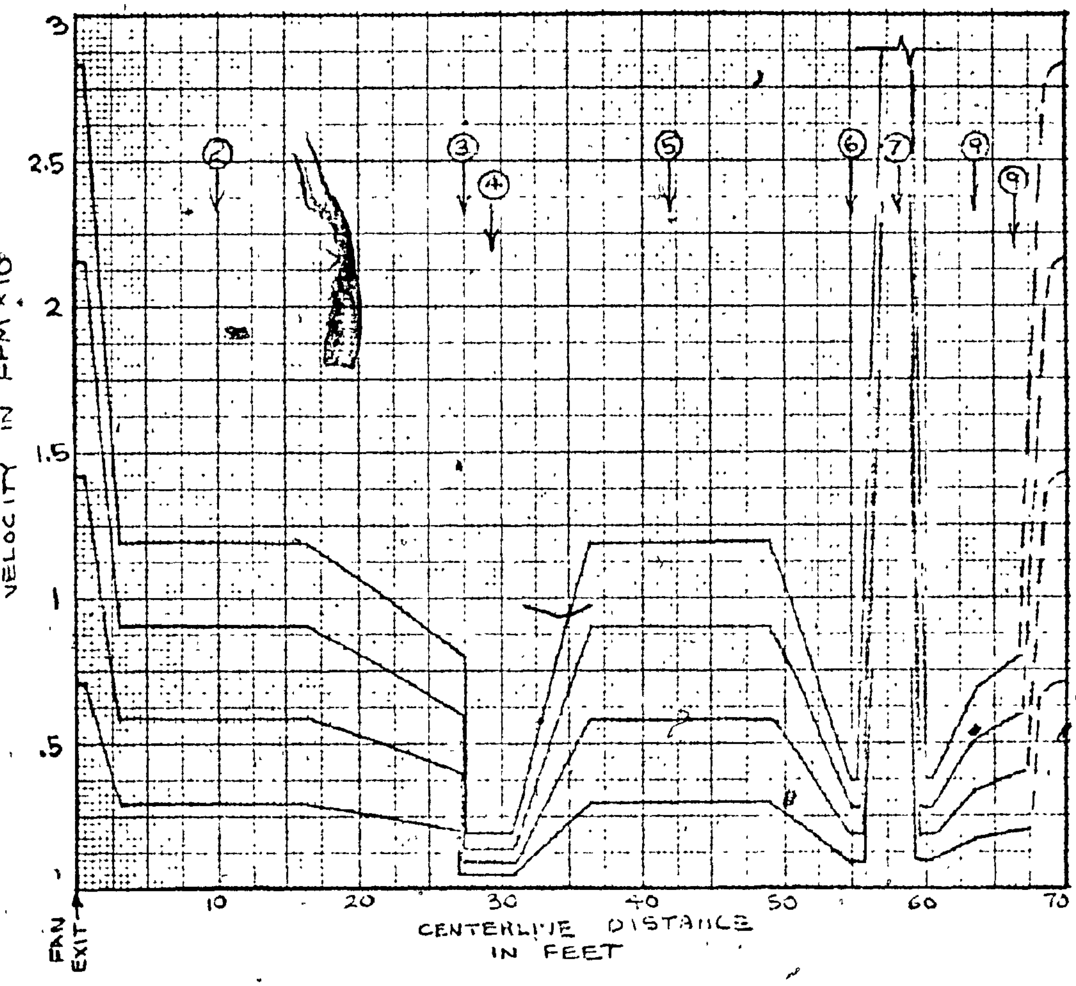


EQUIPMENT LAYOUT AND REFERENCE SECTIONS
FIGURE 12

			VELOCITY FPM			
	SECTION	SQ FT AREA	200 FPM 1500 CFM	400 FPM 3000 CFM	600 FPM 4500 CFM	800 FPM 6000 CFM
1	16" × 19"	2.11	711	1422	2133	2844
2	27" × 27"	5.06	296	593	889	1186
3	22½" × 48"	7.50	200	400	600	800
4	60" × 73"	30.40	49	99	148	197
5	27" × 27"	5.06	296	593	889	1186
6	48" × 48"	16.00	94	188	281	375
7	4 × 8ø	1.40	1075	2149	3214	4286
8	33 × 39"	8.94	168	336	503	671
9	30" × 36"	7.50	200	400	600	800

Table 3

Air Velocity At The Reference Sections



THEORETICAL VELOCITY PROFILE IN THE TEST CIRCUIT
FIGURE 13

2. Pressure Variation Of Air Handling Unit:

The variable volume fan must also overcome the static pressure present in the system as a result of duct friction and the various obstructions in the circuit. The predicted pressures were tabulated and calculated to enable comparison with actual results. Again, the results are given for each of the four test runs. To evaluate the duct friction effects, it was necessary to calculate the circular equivalents of each duct section and from these determine the duct friction. The circular equivalents are determined from the ASHRAE tables using the following formula. The duct friction has been extrapolated from the tables at low velocities

$$d_c = 1.30^8 \frac{(ab)^5}{(a+b)^2} \tag{24}$$

			STATIC PRESSURE IN/100 FT			
			200 FPM	400 FPM	600 FPM	800 FPM
	SECTION	CIRC EQUIV.	1500 CFM	3000 CFM	4[illegible]00 CFM	6000 CFM
1	16" × 19"	18.6"	.042	.159	.343	.600
2	27" × 27"	29.3"	.005	.019	.040	.068
3	22⅛" × 48"	35.7"	.002	.007	.015	.026
4	60" × 73"	71.5"	—	—	—	—
5	27" × 27"	29.3"	.005	.019	.040	.068
6	48" × 48"	52.6"	—	—	.002	.004
7	4 × 8"ø	8.0"	.240	.880	1.900	3.400
8	33 × 39"	39.2	.001	.004	.009	.016
9	30" × 36"	35.9	.002	.007	.015	.025

Table 4

Static Pressures/100 Ft. For The Reference Sections

Using the centerline lengths from the test chamber drawings, (Figure 8) the friction contributed by the various sections of duct is tabulated in Table 5. Factors which have a negligible contribution have been ignored.

	FITTING	LENGTH	C_o	REMARKS	200 FPM	400 FPM	600 FPM	800 FPM
1	DUCT	~~[illegible]~~ 3" DUCT 3"	—	ADD 15" FOR TRANS.	.001	.003	.007	.012
	TRANSITION	29"	.10	$\frac{A_o}{A_1} = .4$; $\theta = 16^\circ$	.003	.013	.028	.051
2	DUCT	233"	—		.001	.004	.008	.013
	ELBOW	—	.26	DOUBLE THICKNESS VANES	.003	.005	.013	.023
	ELBOW	—	.26	DOUBLE THICKNESS VANES	.003	.005	.013	.023
3	DUCT	72"	—		—	—	.001	.002
	COIL	—	—	MFG	.060	.190	.380	.625
	TRANSITION	120"	.02	$\frac{A_o}{A_1} = .6$; $\theta = 8^\circ$	—	—	.001	.002
	ABRUPT TRANSITION	—	.62	$\frac{A_o}{A_1} = .25$	.002	.006	.013	.024
4	TRANSITION	72"	.03	$\frac{A_o}{A_1} = .2$; $\theta = 36^\circ$	—	—	.001	.001
5	DUCT	230.5"	—		.001	.004	.008	.013
	ELBOWS	—	.26	TWO AS ABOVE	.006	.010	.026	.046
	TRANSITION	72"	.05	$\frac{A_o}{A_1} = .32$; $\theta = 8^\circ$	—	.001	.002	.004
6	BELLMOUTH	—	.03	$\frac{r}{D} > .2$	.002	.009	.019	.034
7	EXIT	—	.85	$\frac{A_o}{A_1} = .088$; $\theta = 180^\circ$	.061	.245	.547	.973
	NOZZLES	12"	—		.002	.009	.019	.034
8	DUCT	38"	—		—	—	—	.001
9	DUCT	35"	—		—	—	—	.001
				TOTAL	.15"	.50"	1.09"	1.88"

Table 5
Calculation Of Duct Friction Losses

3. Prediction Of Temperature Variation:

During any particular test, the temperatures before and after the test coil are measured and recorded. The value at the entrance to the test coil is the controlled quantity and the heating coil in the air handling unit should heat the air to the desired value, losses being taken into consideration. The heat losses for the ductwork are calculated with the following equations (ASHRAE handbook of fundamentals 1972-Pg. 479).

$$Qw = UPL\left(\frac{(te + t_1)}{2} - ta\right) \tag{25}$$

$$t_e = \frac{t_1\ (y + 1) - 2ta}{(y - 1)} \tag{26}$$

$$t_1 = \frac{te\ (y - 1) + 2ta}{(y + 1)} \tag{27}$$

$$y = 28.8\ AV_e/UPL \tag{28}$$

These equations require the circuit to be subdivided into appropriate sections with constant values for P, L, A, & U. Table 6 gives these parameters for the circuit.

	1	2	3	4	5	6	7	8	9	10	11	12	13	14
	DUCT	TRANSITION	DUCT	DUCT	TRANSITION	COIL	DUCT	TRANSITION	DUCT	DUCT	TRANSITION	DUCT	TRANSITION	DUCT
P	5.83	7.42	9.00	9.00	10.33	11.75	22.17	15.58	9.00	9.00	12.50	16.00	14.00	12.00
L.	.5	2.42	11.33	2.17	10.	1.	2.67	6.00	2.17	11.04	6.00	5.50	3.17	2.83
A	2.11	3.59	5.06	5.06	6.23	7.5	30.4	17.73	5.06	5.06	10.53	16.00	12.47	8.94
U	1.64	1.64	1.64	.25	.25	.25	.25	.25	.25	1.64	1.64	1.64	1.64	1.64

Table 6

Using the equations given previously, the temperature variation and heat loss can be calculated for the entire circuit. The values used in this calculation (i.e. $\dot{V}$, t_e) are variables. The velocity is dependent on which of the four loading conditions is chosen, whereas ρ and t_e are set by the initial conditions which are desired at the coil entrance. For this analysis assume that the ambiant air is maintained at $70^{\circ}F$ by the buildings' mechanical systems. The coil loading was simulated by adjusting the circulating air volume. This implies that the temperature drop across the coil remains constant at $18.52^{\circ}F$ for test runs on coils used in variable volume constant temperature systems. In order to demonstrate the heat loss calculations for a particular run, an entering dry bulb temperature of $75^{\circ}F$ and an air density of 0.73 lb/ft^3 with a completely dry coil are assumed. In Table 7, the temperatures have been tabulated at those conditions and at 200 FPM coil face velocity.

	1	2	3	4	5	6	7	8	9	10	11	12	13	14
V	711	504	296	296	248	200	49	72	296	296	195	94	131	168
Y	—	—	—	—	—	—	202.93	119.56	671.44	20.12	36.54	22.81	49.13	59.03
ρ	—	—	—	—	—	—	.076	.076	.076	.076	.076	.076	.076	.076
t_e	75°	—	—	—	—	75°	56.48	56.66	56.97	57.02	58.72	59.59	60.89	61.45
t_1	—	—	—	—	—	56.48	56.66	56.97	57.02	58.72	59.59	60.89	61.45	61.90

Table 7

Using equation (25), the heat loss can be calculated for each of the fourteen sections tabulated above. By appropriate addition of the different sections, the heat loss of the various chambers can be determined. For the particular conditions mentioned, the following heat losses were calculated.

		HEAT LOSS	
Fan exit to intake chamber	-	1007	BTUH
Intake chamber	-	167	BTUH
Mixing chamber	-	570	BTUH
Mixing chamber to fan entrance	-	5825	BTUH
	TOTAL	7569	BTUH

The 7.5 H.P. fan motor is outside the airstream and its contribution to the heat gain is given by the formula:

$$Q = \text{HORSEPOWER RATING} \times \text{LOAD FACTOR} \times 2545 \qquad (29)$$

For air circulating at 1500 CFM, the load factor is about .2. At this test condition, Q = 3718 BTUH.

4. Predicted Performance Of Refrigeration System:

The quantity of heat that each pound of refrigerant absorbs ~~while~~ flowing through the evaporator is known as the refrigerating effect. The heat added to each pound of refrigerant in the evaporator is the difference between the enthalpy of the vapor leaving the evaporator and the enthalpy of the liquid approaching the expansion valve. The R.E. is influenced by any superheating or subcooling produced in the system. The standard (4) recommended is 10°F superheating and 10°F subcooling.

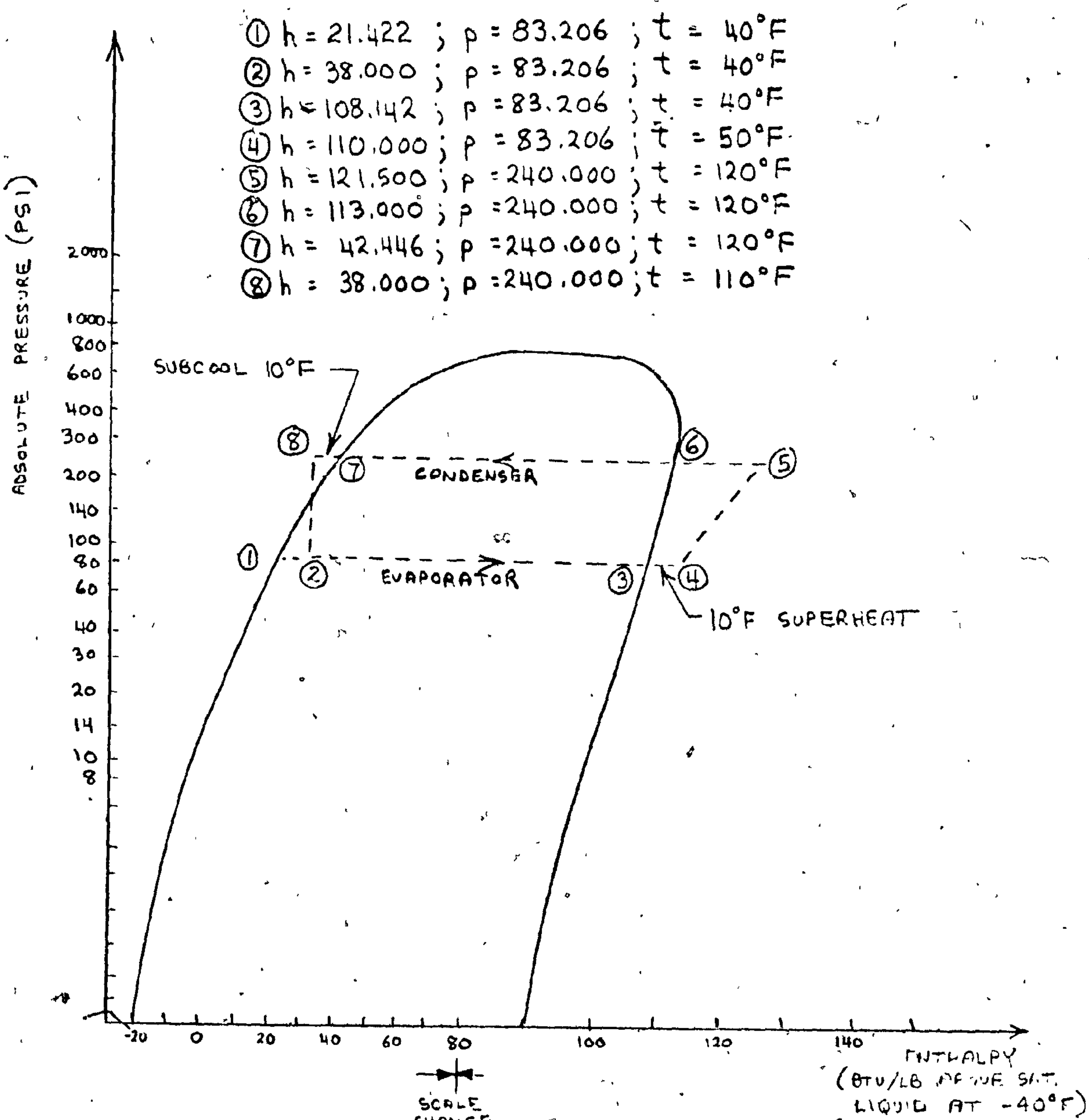
The R.E. is calculated as follows with the appropriate adjustments for subcooling and superheating. The process is illustrated on the P.H. chart in Figure 14.

$$RE = h(\text{ENG EVAP}) - h(\text{LEAVING EVAP})$$

$$= 108.142 - 42.446 + 3.25$$

$$= 67.56 \text{ BTU/LB}$$

① h = 21.422 ; p = 83.206 ; t = 40°F
② h = 38.000 ; p = 83.206 ; t = 40°F
③ h = 108.142 ; p = 83.206 ; t = 40°F
④ h = 110.000 ; p = 83.206 ; t = 50°F
⑤ h = 121.500 ; p = 240.000 ; t = 120°F
⑥ h = 113.000 ; p = 240.000 ; t = 120°F
⑦ h = 42.446 ; p = 240.000 ; t = 120°F
⑧ h = 38.000 ; p = 240.000 ; t = 110°F



P-H Chart Of System Performance

FIGURE 14

The flow of refrigerant can easily be tabulated at any loading condition from the relationship:

$$\text{FLOW} = \frac{Q}{\text{R.E.}} \tag{30}$$

In the part load test, evaluations of the refrigerant circuit will be made at three different loads - full load, 2/3 load and 1/3 load. Using the previous formula, the amount of refrigerant required to be circulated at each load can be found.

	REFRIGERANT FLOW
FULL - 120,000 BTUH	29.62 LB/MIN
2/3 - 80,000 BTUH	19.75 LB/MIN
1/3 - 40,000 BTUH	9.87 LB/MIN

The coefficient of performance for the refrigeration system can also be determined. The COP is the ratio of the refrigeration effect to work supplied. The work supplied is equal to the change in enthalpy during compression. Assuming that the compression occurs at constant entropy, the work is 121.5 - 110 = 11.5 BTU/LB. from the P.H. chart

$$\text{Therefore C.O.P.} = \frac{67.56}{11.5} = 5.87$$

The horsepower requirements can also be calculated as follows:

$$\text{bhp} = \frac{\text{WORK X FLOW}}{\text{42.4 X OVERALL EFFICIENCY}} \tag{31}$$

$$\text{COMPRESSION RATIO} = \frac{\text{HIGH PRESS}}{\text{LOW PRESS}} = \frac{240}{83.206} = 2.88$$

For the compression ratio of 2.88, we can assume an efficiency of 77%. Using the above results for flow at the different loads and the WORK of 11.5 BTU/LB the horsepower requirements can be tabulated. Based on the results, a 15 HP compressor motor was provided.

	REQUIRED HORSEPOWER
FULL - 120,000 BTUH	10.4
2/3 - 80,000 BTUH	7.0
1/3 - 40,000 BTUH	3.5

The condenser is water cooled and if we assume a temperature rise in the water of $10^{o}F$, the quantity of cooling water required can be determined.

The change in enthalpy across the condenser is:

$$h_c = 121.5 - 42.4 = 79.10 \text{ BTU/LB} \tag{32}$$

$$Q = \text{G.P.M.} \times 500 \times \Delta T$$

$$\therefore \text{G.P.M.} = \frac{(h_c \times \text{FLOW})}{500 \times \Delta T}$$

	GPM OF WATER REQUIRED
FULL - 140,577 BTUH	28.12
2/3 - 93,734 BTUH	18.75
1/3 - 46,843 BTUH	9.37

The piping for the various circuits was sized based on common practice and the projected loading of the systems. The load variation is 3.4 - 10 tons and the Table 8 shows the pipe sizes selected and the associated velocities.

		SIZE	EQUIV. LENGTH	VELOCITY (FPM) FULL LOAD	2/3 LOAD	1/3 LOAD	RECOMMENDED VELOCITY (FPM) MAX	MIN. HOR.	MIN. VERT.
R-22	REFRIGERANT SUCTION	$1\frac{1}{8}$"	30'	3900	2500	1200	4000	500	1000
	REFRIGERANT DISCHARGE	$\frac{7}{8}$"	30'	2200	1700	840	4000	500	1000
	REFRIGERANT LIQUID	$\frac{5}{8}$"	50'	262	174	87	300	—	—
WATER	CONDENSER INLET / OUTLET	2"	—	165	115	55	240	—	—
	HOT WATER INLET/ OUTLET	$1\frac{1}{4}$"	—	150	100	50	240	—	—

SYSTEM PIPING

TABLE 8

5. The Heating Coil:

At a temperature drop across the coil of $20^{O}F$, the quantities of hot water required can be determined from Equation 32.

	HOT WATER REQ'D
FULL LOAD	12 GPM
2/3 LOAD	8 GPM
1/3 LOAD	4 GPM

The water piping for the heating coil and condenser were sized based on the recommendations of ASHRAE (ref. Handbook of Fundamentals Ch. 32). The water velocities for the chosen pipe sizes are included in Table 8 on the previous page.

CHAPTER V

MEASUREMENT AND INSTRUMENTATION

1. Introduction:

Included in this section of the paper is a description of the instrumentation installed on the testing facility and the measurement techniques involved in a test. In selecting the instrumentation, in addition to best performance characteristics and cost, particular consideration was given to the adaptability of selected instruments to automated data acquisition systems. Where necessary or economically possible, certain readings were automated and these readings are handled by a data acquisition system which is discussed in more detail in the following pages. In this section will be found a detailed description of the instruments, their location, purpose and criteria for selection. A summary of measurement standards as per ASHRAE 33-64 is also included.

2. Temperature Measurement Standard:

MEASUREMENT	HEATING COIL TEST	COOLING COIL TEST	REMARKS
AIR-WET OR DRY BULB	$\pm$ .50°F	$\pm$ .10°F	AIR VELOCITY 700 TO 2000 FPM, PREFERABLY 1000 FPM
WATER	$\pm$ .50°F	$\pm$ .10°F	OR 2% OF WATER TEMP DROP IF SMALLER
OTHER	$\pm$ .50°F	$\pm$ .50°F	

NOTE: USE MERCURY-IN-GLASS, THERMOCOUPLES OR ELECTRIC RESISTANCE TYPE.

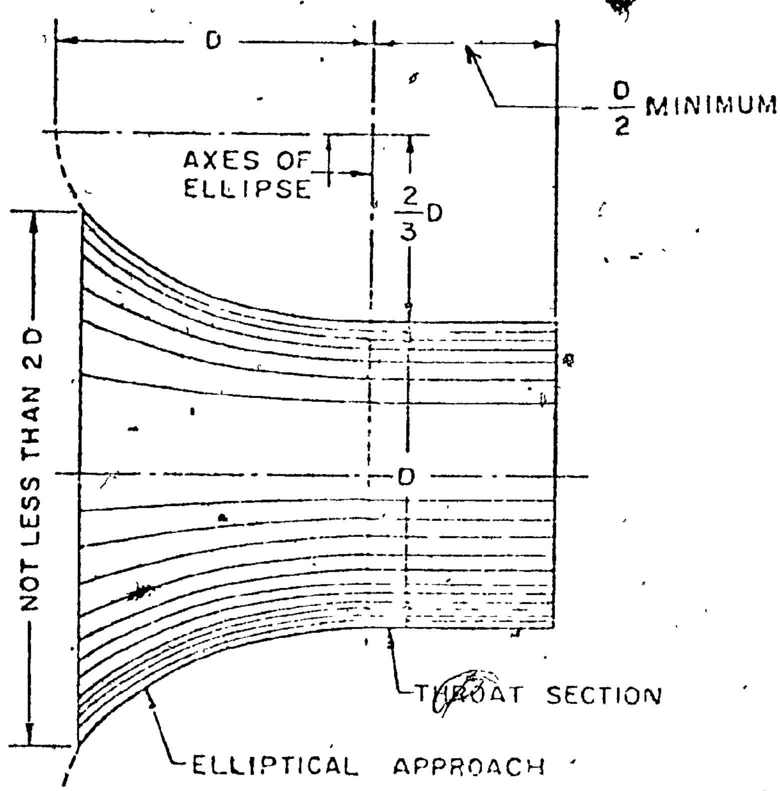
3. Pressure Measurement Standard:

MEASUREMENT	ACCURACY	REMARKS
REFRIGERANT SUCTION	± .5% ABSOLUTE	MERCURY OR BOURDON TUBE
OTHER	± 2.0% ABSOLUTE	

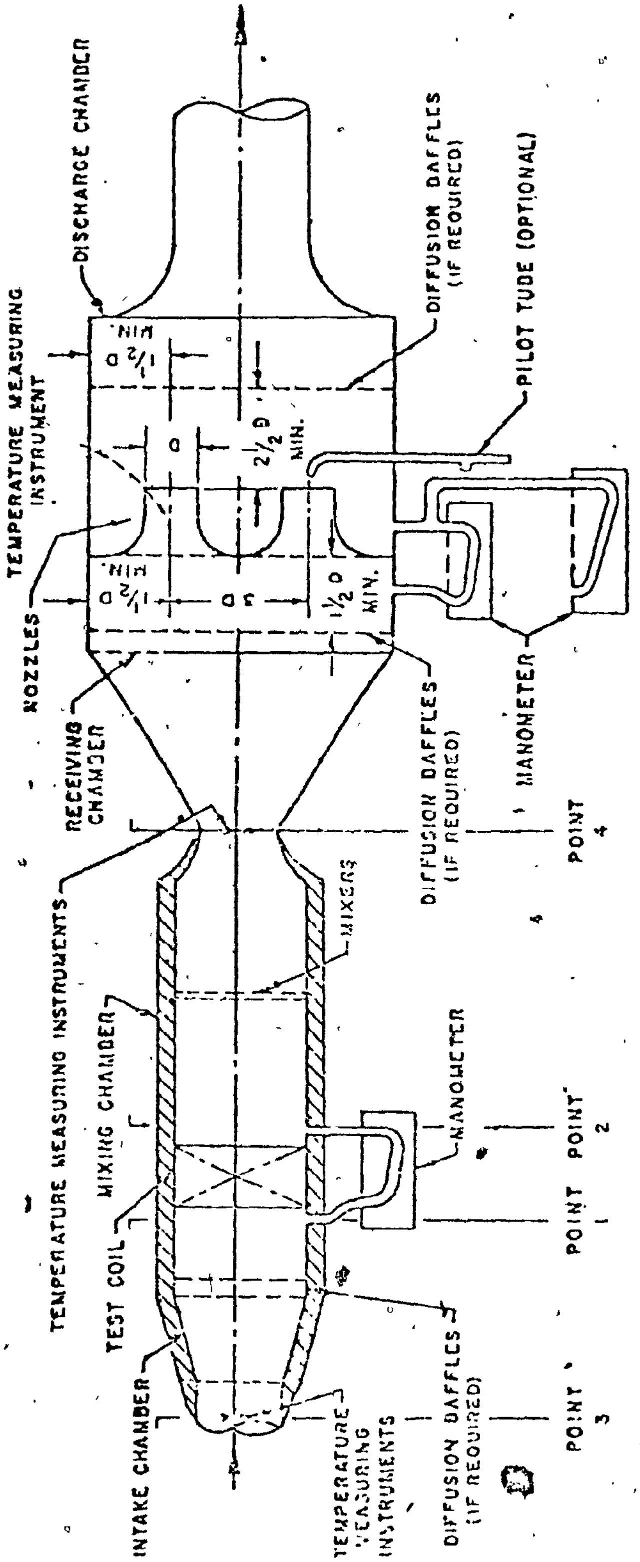
NOTE: Use Mercury Column, Bourdon Tube Gauge, Manometer or Draft Glass Types Manometers or Draft Glasses calibrated to 1% of reading or .005" H_2O, whichever is greater.

4. Flow Measurement Standard:

Flow Nozzles constructed in accordance with ASHRAE standard 33-64 are required for air flow measurement. The velocity can be obtained by reading the static pressure drop across the nozzles or by measuring the velocity pressure with pitot tubes and manometers.



AIR FLOW MEASURING NOZZLE
FIGURE 15



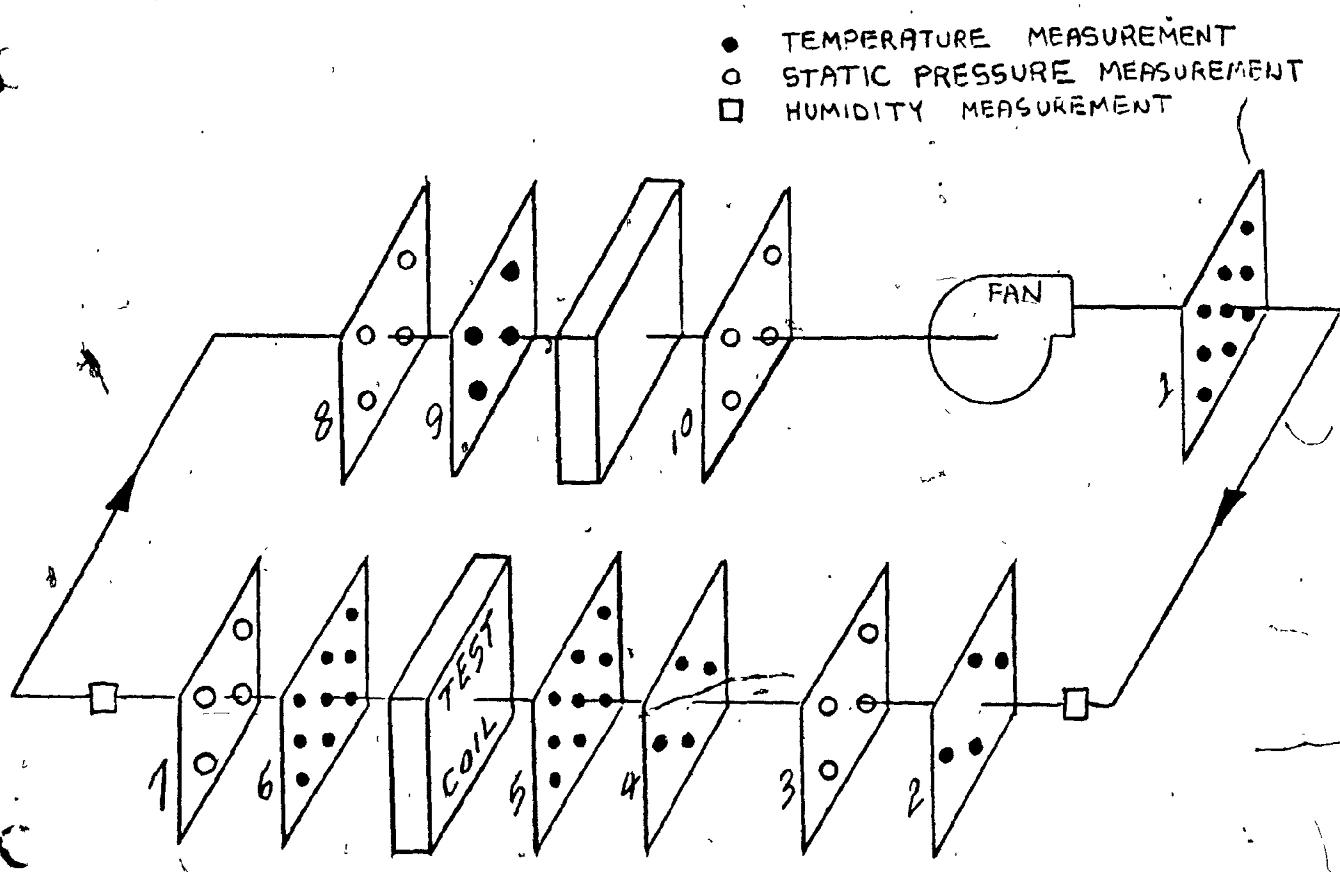
AIR FLOW AND TEMPERATURE MEASURING APPARATUS RECOMMENDED BY ASHRAE

FIGURE 16

Refrigerant flow measurements can be made by A) the flow meter method B) the condenser water method or C) the calibrated compressor method. The accuracy should be ± 2% of the quantity measured.

5. Description Of The Air Side Instruments:

In this section, the instruments will be described, discussed and the symbology associated with them will be introduced. A listing of the instruments is also given. In Figure 17 below, the various air side points of measurement have been identified. The measurement locations have been indicated by numbers, from 1 to 10, to facilitate the references and descriptions which follow.



A SYMBOLIC REPRESENTATION OF THE AIR SIDE MEASUREMENTS
FIGURE 17

At locations 1, 5 & 6 of Figure 17 a grid of nine thermocouples was installed in order to obtain an average temperature reading at each point. All twenty seven of these thermocouples have been automated by linking them to the data acquisition system. The symbolic designation of each of these thermocouples is given below.

<u>NOTE</u>: ALL LOCATIONS VISUALIZED FROM UPSTREAM SIDE LOOKING DOWNSTREAM.

T1	T2	T3
T4	T5	T6
T7	T8	T9

LOCATION 1

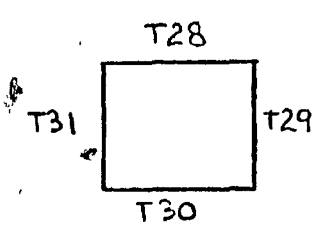
T10	T11	T12
T13	T14	T15
T16	T17	T18

LOCATION 5

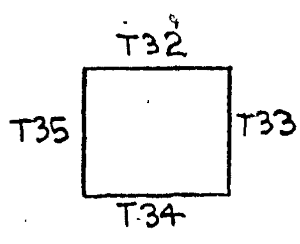
T19	T20	T21
T22	T23	T24
T25	T26	T27

LOCATION 6

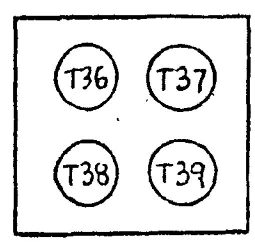
At locations 2 & 4, thermocouples are installed outside the intake chamber to measure the temperature of the ambiant air. At location 9, thermocouples are installed at the outlet of each of the four velocity measurement nozzles. All twelve of these temperature measurements are automated by using the data acquisition system. The symbolic designation of these measurements follows.



LOCATION 2

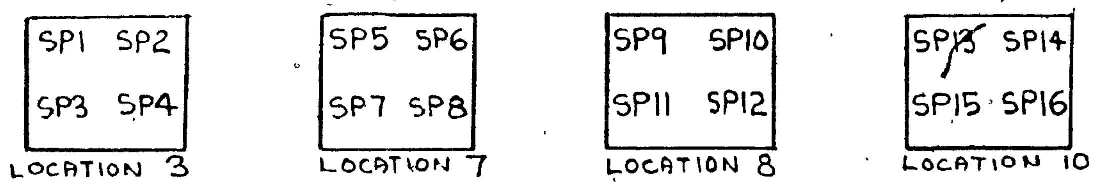


LOCATION 4



LOCATION 9

At locations 3,7,8 &10 static pressure measurements are made. All sixteen of these measurements have been automated along with the two relative humidity sensors situated before and after the test coil. The sensor upstream of the test coil has been designated RH 1 and the downstream one RH 2. The static pressure taps are designated as follows.



The equipment which was selected to record the air side measurements is listed in the following table. Further details on these instruments is included in the specification sheets attached as ANNEX A.

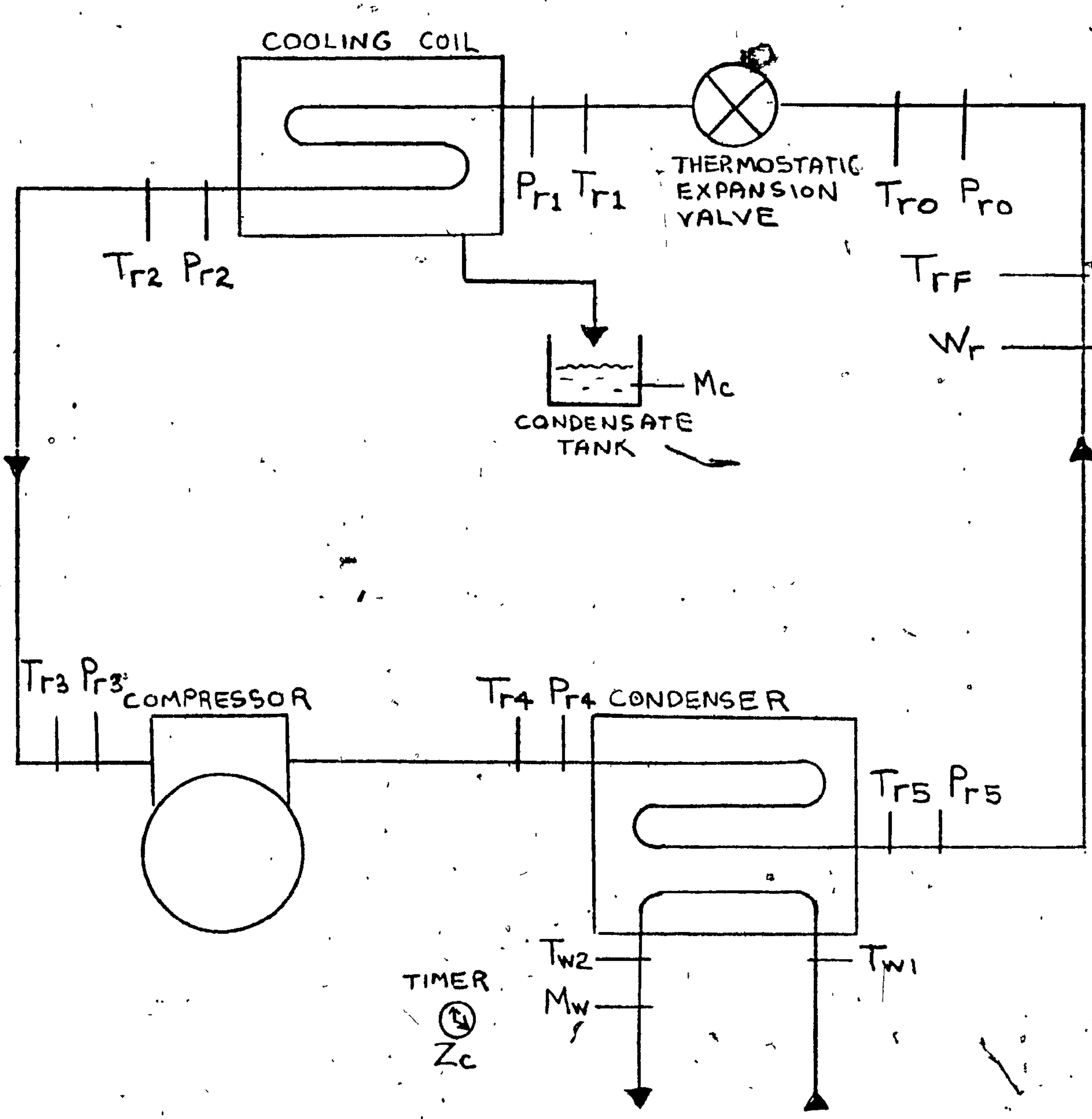
INSTRUMENT	TYPE	MODEL	QUANTITY
T1 → T39	ZIG-ZAG THERMOCOUPLE	HYCAL TC-112-7-A-24-180	39
SP1 → SP16	PRESSURE TRANSDUCER 0-20" WATER	INTERTECH MODEL 215 2MV/V	16
RH1 → RH2	RELATIVE HUMIDITY SENSORS 0-100%	HYCAL HS-3553-B-8-F18-120 WITH TRANSMITTERS RH-515-A-C-0-100%	2

Table 9

In addition to the automatic measurements mentioned above, a manually read velocity pressure tap is located at each of the four nozzle outlets.

6. Description Of Refrigerant Side Instruments:

Various temperature, pressure and flow measurements are required in the refrigerant circuit in order to properly monitor its performance and permit an analysis of the test coil according to the ASHRAE and ARI standards. These measurements are illustrated in Figure 18.



A SYMBOLIC REPRESENTATION OF THE REFRIGERANT SIDE MEASUREMENTS

FIGURE 18

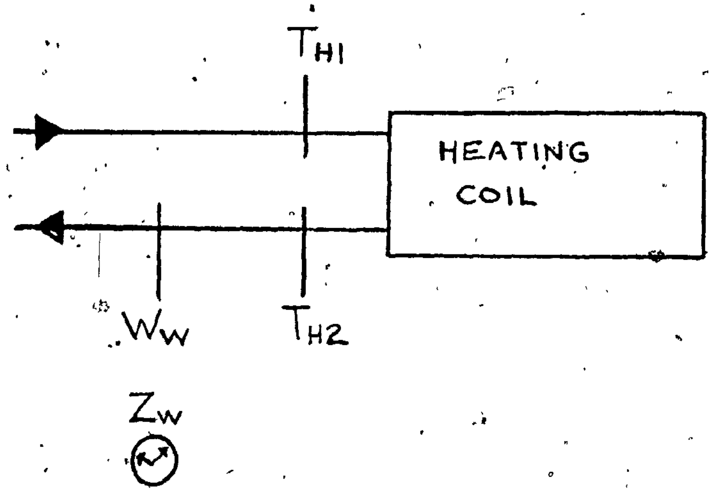
Not all of the refrigerant circuit measurements were automated. The reason for this was the high cost involved in converting the sensor outputs to a form acceptable to the data acquisition system. An evaluation was undertaken to determine which of the measurements would provide the greatest advantage by being automated. Based on the importance of the measurement, the frequency with which it would require reading and the cost of automating the observation, the decision to automate or not was made. The results have been tabulated in Table 10 below with additional data on the instruments also given.

INSTRUMENT	TYPE	MANUFACTURER	QUANTITY
Tr1; Tr2; TrF	RTD 100 OHM @ $0^{o}C$	HYCAL RTS-34-[illegible]-100-C-3-120	3
Pr1; Pr2	PRESSURE TRANSDUCER 0-300 psi	HYCAL DP 2-1-500	2
Wr			1
Tr0; Tr3; Tr4; Tr5	TEMPERATURE GAUGE		4
Pr0; Pr3; Pr4; Pr5	PRESSURE GAUGE		4
Tw1; Tw2	TEMPERATURE GAUGE		2
Zc	TIMER		1
Mw	FLOW METER		1

TABLE 10

7. Hot Water Measurements:

Manual measurements which permit an evaluation of the performance of the hot water heating coil are made possible through the installation of two thermometers and a flow measuring device. These are illustrated along with their designating symbols in Figure 19.



HOT WATER CIRCUIT MEASUREMENTS

FIGURE 19

8. <u>The Data Acquisition System</u>:

In order to properly record the large number of measurements involved in any particular test, a data acquisition system was purchased for the testing facility. Our requirements were for a portable 80-channel data logger with an instantaneous tape output and a facility to record data on a magnetic tape. A DORIC DIGITREND 220 was purchased to satisfy these requirements.

The DIGITREND 220 accepts 80 channels of analog inputs. Its flexibility allows complete control of the data which is to be monitored. A continuous scan feature combined with a printer provides a record of all desired data on a continuous basis. For example, a reading can be obtained on each sensor every 30 seconds. This time frame can be altered using a periodic single scan feature which gives a reading every 2, 5, 10, 15, 30 minutes as desired. The 2 minute time interval is used for our test purposes. The data logger is equipped with front panel controls which permit isolation of a single piece of data and its display on the logger. Many other features and capabilities enable the automation of most of the data from any test and the complete control of the monitoring function.

The Doric Scientific 220 microprocessor based Data Acquisition System is specified as:

Model 220-80-05-11(1)-66-25-06(2)

The instrument is described as follows:

220-80	- This is the basic system with 80 channels of analog inputs.
05(1)	- Option 05 is a high sensitivity analog/ digital converter which provides a resolution of 1.0 microvolt or 0.1 degrees.
66	- Multiple program option - 3 functions.
25	- Multiple program extension - 6 functions.
06(2)	- Special scaling for 2 scales

The Doric 220 has provision for adding a magnetic tape coupler at a future date and/or can be interfaced with computer and/or calculators.

CHAPTER VI

UNCERTAINTY ANALYSIS

1. Introduction:

In this section, the uncertainties involved in establishing the test results are discussed. The uncertainties associated with the test results are determined by the well known methods of statistics as described in ASHRAE standard 41.5-75. The uncertainty is expressed in terms of the standard deviation σ, where

$$\sigma^2 = \frac{1}{N-1} \cdot \sum_{i=1}^{N} (X_i - X_m)^2 \tag{33}$$

X_m is the mean value of the variable X_i taken over N measurements.

Assuming a normal distribution of error, then the 95 percent confidence limit is equal to 2σ. For this analysis, uncertainties are often known only in terms of absolute limits. For these cases, the 2σ confidence limit corresponds to the absolute limit of the uncertainties.

In the following calculations the absolute limits of uncertainty specified by ASHRAE standard 33-64 are used. Our instruments were selected to achieve a better accuracy than the standard requires. As a result we can expect our results to be within the requirements of the standard as determined on the next pages.

The basic equation used in the calculation of uncertainties is given below. For a quantity, Q, calculated from several other independent quantities, a, b, c..., when Q = f (a, b, c,..) then the combined uncertainty associated with Q is given by:

$$\sigma_Q^2 = \left(\frac{\sigma_Q}{\sigma_a}\right)^2 * \sigma_a^2 + \left(\frac{\sigma_Q}{\sigma_b}\right)^2 * \sigma_b^2 + \left(\frac{\sigma_Q}{\sigma_c}\right)^2 * \sigma_c^2 + \ldots \qquad (34)$$

2. <u>Air Flow Rate</u>:

The air mass flow rate through a single nozzle is calculated by the following equation when effects due to air compressibility, thermal expansion and contraction of nozzles, and approach velocity are negligible

$$W_a = 200.4 \quad C_N A_N \left[\frac{\Delta p_N \, p_N}{V_N \, (1+W_N)}\right]^{0.5} \qquad (35)$$

ref: ASHRAE 33-64

where C_N = coefficient of discharge for the nozzle

A_N = area of nozzle $-ft^2$

V_N = specific volume of air $-ft^3/lba$

W_N = humidity ratio -lbw/lba

Δp_N = static pressure difference across nozzle - in. water

P_N = absolute pressure at nozzle throat - in. mercury

Step 1:

The nozzle area is determined from an average of the measured diameters

$$A_N = \frac{\pi}{4}\left[\left(\sum_{N=1}^{4} d_N\right)\Big/ 4\right]^2 \qquad (36)$$

where $d = 8" \pm .2\%$ (ref. ASHRAE standard 33-64 para. 4.3.3)

$$A_N = 50.265 \text{ in}^2 \pm .071086 \text{ in}^2$$

$$= .34906 \pm .49365 \times 10^{-3} \text{ ft}^2$$

Step 2:

The coefficient of discharge of the nozzles can be taken as .99 (ASHRAE 33-64 - para 4.3.2). To determine the confidence limits of this coefficient, we find that the ASHRAE handbook of fundamentals is useful (Pg 212-1972 edition). The coefficient is based on the nozzle geometry and Reynolds Number.

For our nozzle $\beta = \frac{d}{D}$

$$= \frac{8''}{(24 \times 24 \times 4)/96} \quad \text{(ref. standard 33-64 para 9.10.1)}$$

$$= .333$$

The flow variation is from 1500 CFM to 6000 CFM

and $$R_e = \frac{\rho UD}{\mu}$$

where

$$\rho = .075 \text{ lbm/ft}^3$$

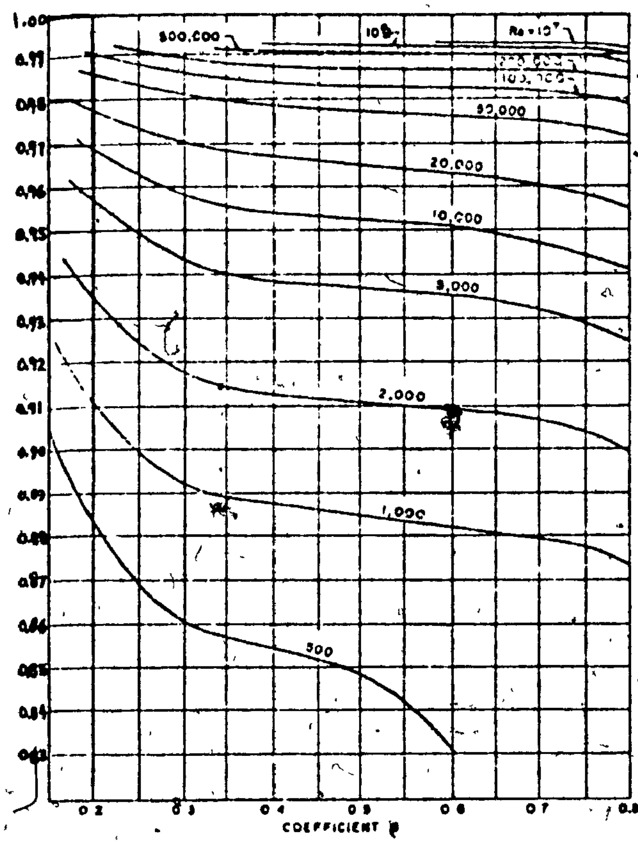
$$\mu = 124 \times 10^{-7} \text{ lbm/sec - ft.}$$

$$D = .67 \text{ ft}$$

$$U = 1075 \text{ ft/min} \rightarrow 4286 \text{ ft/min}$$

Therefore $$R_e = .44 \times 10^7 \text{ to } 1.737 \times 10^7$$

and C can be obtained from Figure 20 reproduced from the handbook of fundamentals. The confidence limit is estimated as .995 ± .005



DISCHARGE COEFFICIENTS FOR ASME LONG RADIUS FLOW NOZZLES WITH PRESSURE TAPS LOCATED ONE PIPE DIAMETER PRECEDING AND ONE-HALF PIPE DIAMETER FOLLOWING THE INLET FACE OF THE NOZZLE

FIGURE 20

Step 3:

Find P_{WS} (t) from the following equation -

$$\log_{10}\left(\frac{P_{WS}}{218.167}\right) = -\frac{\beta}{T}\left(\frac{a + b\beta + c\beta^3}{1 + d\beta}\right) \quad (37)$$

where P_{WS} = saturation pressure

β = 647.27 -T

T = absolute temperature, Kelvin

a = 3.2437814

b = 5.86826 x 10^{-3}

c = 1.1702379 x 10^{-8}

d = 2.1878462 x 10^{-3}

The propagation of uncertainty in this equation is negligible and P_{WS} (t) can be assumed constant for a particular temperature. At t = 60°F $\pm$ .1F , P_{WS} (t) = .52160" H_g

Step 4:

Find P_W from

$$P_W = \phi \; P_{WS} \quad (38)$$

at ϕ = 75% $\pm$ 2% (accuracy of instrument chosen for the tests)

P_W = .3912 $\pm$ .010432" H_g $\doteq$ ($\pm$ 2.66%)

Step 5:

Find W from

$$W = .62198 \frac{P_W}{P-P_W} \tag{39}$$

$$\text{at } p = 29.921 \pm .59842'' \ H_g \qquad \text{(standard 33-64)}$$

$$W = .00824 \pm 2.7597 \times 10^{-4} \text{ lbw/lba} = (\pm 3.35\%)$$

Step 6:

Find W_S from

$$W_S = .62198 \frac{P_{WS}}{P-P_{WS}} \tag{40}$$

using same values as previously

$$W_S = .011035 \pm .00022462 \text{ lbw/lba} = (\pm 2.0353\%)$$

Step 7:

Find u from

$$u = \frac{W}{W_S} \tag{41}$$

$$u = .74671 \pm .029265 = (\pm 3.9191\%)$$

Step 8:

Find v from

$$v = \frac{R_a T}{p} (1 + 1.6078W) \tag{42}$$

$$= \frac{53.352 \times (519.67 \pm .1) \times (1.0132 \pm 4.437 \times 10^{-4})}{(2116.2 \pm 42.324)}$$

$$= 13.274 \pm .26558 = (\pm 2\%)$$

Step 9:

Find W_a from the previous equation and calculated data

Given $\Delta p_N = 1.16 \pm .005"\ H\ O$ (Standard 33-64)

$P_N = 29.921 \pm .59842$ = in. Mercury (Standard 33-64)

$V_N = 13.274 \pm 2.1242 \times 10^{-6}\ ft^3/lba$ (From Step 8)

$W_N = .00824 \pm 4.3909 \times 10^{-9}$ lbw/lba (From Step 5)

$C_N = .995 \pm .005$ (From Step 2)

$A_N = .34906 \pm .49365 \times 10^{-3}\ ft^2$ (From Step 1)

Simplifying:

$$W_a = 200.4x(.995\pm.005)x(A_N)\left[\frac{(1.16\pm.005)x(29.921\pm.59842)}{(13.274\pm2.1242x10^{-6})(1+.00824\pm4.3909x10^{-9})}\right]^{\frac{1}{2}}$$

$$= 69.602\pm.36334 \left[\frac{34.708\pm.71011}{13.383\pm2.1424x10^{-6}}\right]^{\frac{1}{2}}$$

$$= 69.602\pm.36334 \left[2.5934\pm.053061\right]^{\frac{1}{2}}$$

$$= 69.602\pm.36334 \left[1.6104\pm.032949\right]$$

$$= 112.09\pm2.3668$$

Since $Q = \dfrac{W_a}{.075335}$ (43)

$= 1487.9 \pm 31.418$

ERROR = $\pm$ 2.1%

The confidence limit required by standard 33-64 for a test run at T = 60°F; R.H. = 75%; and 6000 CFM is = 2.1%. This accuracy is largely influenced by the confidence limit set on the atmospheric pressure ($\pm$ 2%). The accuracy allowable for W_a according to the standard is $\pm$ 5%.

3. Air Side Cooling Capacity:

The air side cooling is given by

$$q_{ta} = 60\ W_a \left[(h_1 - h_2) - \Delta W\ (t_4' - 32) \right] \tag{44}$$

$W_a = 112.09 \pm 2.3668$ (Calculated Previously)

for a dry coil $\Delta W = 0$

$$\text{and } q_{sa} = 60\ W_a C_p\ (t_1 - t_2) \tag{45}$$

$$= 60 \times (112.09 \pm 2.3668) \times (.243 \pm .005) \times (18.52 \pm .14142)$$
$$= 30226.7 \pm 921.8\ \frac{\text{BTUH}}{\text{NOZZLE}} = (\pm\ 3.05\%)$$

therefore $q_{sa} = 120907 \pm 3687.2$ BTUH

4. Volatile Refrigerant Flow:

The flow of refrigerant is found using the condenser water method

$$W_r = \frac{W_w\ (t_{W2} - t_{W1}) + q_c}{h_{c_1} - h_{c_2}}$$

$q_c = K_c \times \Delta t = U \times A \times \Delta t = .2\ \text{BTU/HR/}^\circ\text{F/FT}^2 \times 8\text{ft}^2 \times 35^\circ\text{F}$
$= 56$ BTU/HR (The error can be ignored)

$t_{w_2} - t_{w_1} = 20^\circ\text{F} \pm .14142$

$W_w = 7021.9$ lbw/hr $\pm$ 1%

The enthalpy of the entering and leaving refrigerant is determined from the temperature measurements ($\pm .1^\circ$F) and refrigerant tables. The error in enthalpy is small and can be ignored. From the previously given P-H chart

$$h_{c_1} - h_{c_2} = 121.5 - 42.45 = 79.054 \text{ BTU/lbr}$$

$$W_r = \left[(7021.9 \pm 70.219)(20 \pm .14142) + 56\right]/79.054$$

$$= \left[(140440 \pm 1720) + 56\right]/79.054$$

$$= 1777.2 \pm 21.757 \text{ lbr/hr}$$

$$= 29.62 \pm 1.2\% \text{ lbr/min}$$

$$= (\pm 1.2242\%)$$

5. Tube Side Cooling Capacity:

For the volatile refrigerant coil

$$q_{tz} = W_r\ (hr2 - hr1) \qquad (47)$$

If hr2 and hrl are obtained from the tables and the measured values for temperature as before, the confidence limits are small and they can be ignored.

since $W_r = 1777.2 \pm 21.757$ lbr/hr

$$q_{t2} = (1777.2 \pm 21.757)(110 - 42.446)$$

$$= 120060 \pm 1.2\%$$

CHAPTER VII

CONCLUDING REMARKS

The objective of this project was to report on Concordia University's heat exchanger testing facility and to summarize conventional theories on the heat transfer process for heat exchangers. As mentioned in the introduction, the proper theoretical evaluation of a coil requires that testing be done to backup the theoretical analysis. The first part of this report discussed the theory of the heat transfer process. The latter part of the report discussed the test facility which will eventually be used to verify the validity of that theory. Using the testing facility the coefficients for air and refrigerant films can be determined and the theory which is detailed can be evaluated. The heat transfer process is a complex one and as a result it was necessary to make various assumptions in the theory. Using the test facility these assumptions can be checked.

REFERENCES

1. Abdelmessih, A.H., and Al-Kherdaji, N., "Isotherms and Effectiveness of Annular Fins in Cross Flow", ASHRAE paper No. 2249 presented at the Annual Meeting at Nassau, Bahamas, June 1972.

2. Air Conditioning and Refrigeration Institute, "Standards of Forced-Circulation Air-Cooling and Air-Heating Coils", ARI Standards 410-72, Arlington, Va., 1972.

3. Ashley, C., "A Method of Analyzing Finned Coil Heat Transfer Performance", Journal of the A.S.R.E., June 1946.

4. ASHRAE 33-64 Standard "Methods of Testing for Rating Forced Circulation Air Cooling and Air Heating Coils", ASHRAE, Jan. 1964.

5. "ASHRAE Handbook of Fundamentals", Chapter 12, Measurements and Instruments, 1972.

6. Brooks, D.B., and Doe, J., "Energy Conservation: How Big a Target?", ASHRAE Journal, Vol. 16, No. 8, August 1974.

7. Brousoon, D.M., "New Challenges for Engineers in a Brave New World", ASHRAE Journal, Vol. 15, No. 8, August, 1973.

8. Berg, C.A., "Energy Conservation Through Effective Utilization", Science, Vol. 181, 13 July 1973, Pg. 128-138.

REFERENCES

9. Briggs, D.E., and Young, E.H., "Convection Heat Transfer and Pressure Drop of Air Flowing Across Triangular Pitch Banks of Finned Tubes", Chem. Eng. Progress Symposium Series - Heat Transfer, Vol. 59, No. 41, 1963, p. 1-10.

10. Brown, G., "Theory of Moist Air Heat Exchangers", Transactions of the Royal Institute of Technology, Stockholm, Sweden, 1954.

11. Bryan, W.L., "Heat and Mass Transfer in Dehumidifying Surface Coils", ASHRAE Journal, Sept. 1961, p. 51.

12. Bryan, W.L., "Heat and Mass Transfer in Dehumidifying Extended Surface Coils", ASHRAE Journal, April 1962, pp. 60-63.

13. Bump, T.R., "Average Temperatures in Simple Heat Exchangers", ASME Trans. Vol. 85, Series C, No. 2, 1963, Pg. 182-183.

14. "Computer Program for Analysis of Energy Utilization in Postal Facilities", Vols. I, II and III, General American Transportation Corp., Niles, III, 1967.

15. Colburn, A.P., Ind. Eng. Chem., Vol. 25, Pg. 873-877, 1933.

16. Chen, S.Y., and Zyskowski, G.L., "Steady State Heat Conduction in a Straight Fin With Variable Film Coefficient", ASME paper No. 63 - HT - 12, 1963.

REFERENCES

17. Carrier, W.H., and Anderson, S.W., "The Resistance to Heat Flow Through Finned Tubing", Heating, Piping and Air-Conditioning, Vol. 16, 1944, Pg. 304-320.

18. Cook, N.H., and Robinowicz, E., "Physical Measurements and Analysis", Addison-Wesley Publishing Co., Inc., 1963.

19. Draper, N.P., and Smith, H., "Applied Regression Analysis", John Wiley & Sons, N.Y., 1968.

20. Eckert, E.R.G., and Drake, "Analysis of Heat and Mass Transfer", McGraw-Hill Co., 1972, Pg. 408.

21. Elmahdy, A.H., "Analytical and Experimental Multi-Row, Finned-Tube Heat Exchanger Performance During Cooling and Dehumidification Processes " Doctoral Thesis, Carleton University, 1975.

22. Fleming, W.S.F., "Energy Conservation: Outlook and State of the Art", ASHRAE Symposium on Energy Conservation, Chicago, U.S.A., Jan. 28- Feb. 1, 1973.

23. Foxboro, Co., DEWCEL Equipment for Dew Point Measurement System", Report No. GS 1-2C1A, Mass., U.S.A., 1972.

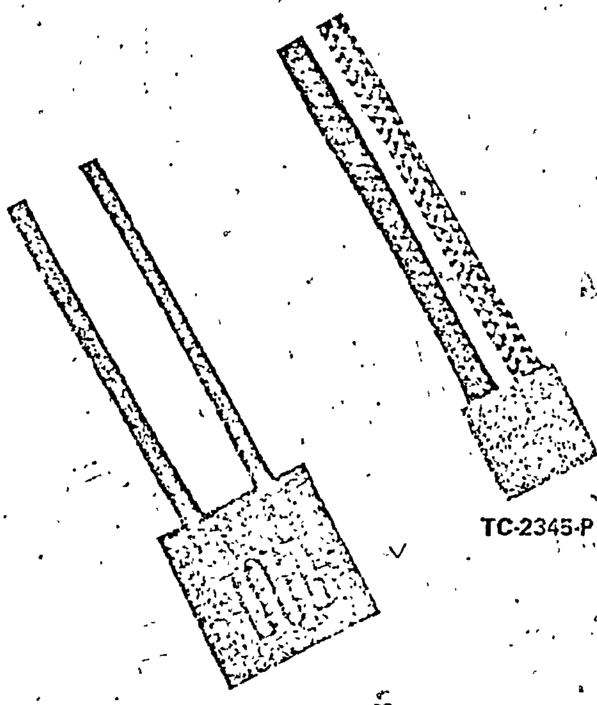
24. Grimison, E., "Correlation and Utilization of New Data on Flow Resistance and Heat Transfer for Cross Flow of Gases Over Tube Banks", ASME Trans., Vol. 59, 1937, Pg. 583-594.

ANNEX A

MINIATURE ZIG/ZAG THERMOCOUPLES

FOR SURFACE TEMPERATURES

TC-2345 SERIES



TC-2345-P

TC-2345-D

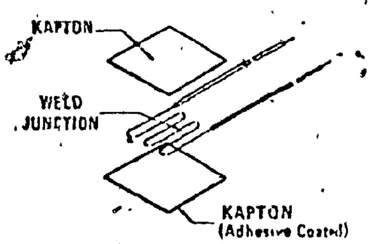
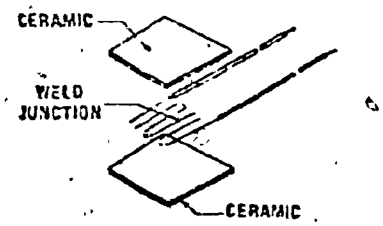
TYPICAL APPLICATIONS
Surface Temperatures in the ELECTRONICS INDUSTRY . . . Transistors Diodes IC Packages Switches Solenoids Components In the PROCESS INDUSTRY . . . Pipelines Boilers Heaters Extruders Motors Generators

See PD-546-1 for Additional ZIG/ZAG Thermocouples

Featuring . . . .

- SMALL IN SIZE
- EASILY ATTACHED TO SMALL SURFACES
- FROM -320°F TO +850°F
- FAST RESPONSE - HIGH ACCURACY
- FINGER-TIP MOUNTING
- AVAILABLE FOR TYPES J, K, T, & E
- "NO-LIE" JUNCTION

...FICATIONS

	LO-TEMP Encased in Kapton	VERY HI-TEMP Encased in Ceramic
MINIATURE ZIG/ZAG THERMOCOUPLE CONSTRUCTION		
MODEL NUMBER	TC-2345 – D – ☐ – ☐ – ☐	TC-2345 – P – ☐ – ☐ – ☐
TEMPERATURE RANGE	-320° F to +350° F	-150° F to + 850° F
RESPONSE TIME	20 milliseconds	50 milliseconds
ACCURACY	Meets ISA Standard limits of error	Meets ISA Standard limits of error
INSTALLATION	Mounts on any metallic or non-metallic surface. Not for use in a vacuum.	Mounts on any metallic or non-metallic surface. Not for use in a vacuum.
T/C LEADWIRE	6 inches of Fiberglass sleeved thermocouple wire is supplied.	6 inches of Fiberglass sleeved thermocouple wire is supplied.
OPTIONAL LEADWIRE	For longer leads, 28 AWG FEP - Teflon is permanently joined to TC wire.	Optional 28 AWG Fiberglass insulated leadwire may be specified, permanently joined to the T/C wire.
MOUNTING	STICK IT ON.... Special pressure sensitive silicon adhesive backing. Remove paper backing and mount to any clean surface.	CEMENT IT ON.... Commercially available "Sauereisen #1," general purpose cement is recommended for most applications.

ORDERING INFORMATION

PATCH SIZE 1st dash	D 1/4" x 1/4" x .015"	P 3/16" x 3/16" x .032"
THERMOCOUPLE TYPE 2nd dash	E Chromel/Constantan J Iron/Constantan K Chromel/Alumel T Copper/Constantan	E Chromel/Constantan J Iron/Constantan K Chromel/Alumel
THERMOCOUPLE WIRE SIZE 3rd dash	3 3 mil 5 5 mil	3 3 mil 5 5 mil
OPTIONAL LEADWIRE 4th dash	☐ Specify in inches. Leadwire is permanently attached to the T/C wire.	☐ Specify in inches. Leadwire is permanently attached to the T/C wire.
EXAMPLE:	TC-2345 – P – K – 5 – 72	3/16" x 3/16" Hi-Temp patch, Type K, 5 mil T/C wire, 72 inch leads.

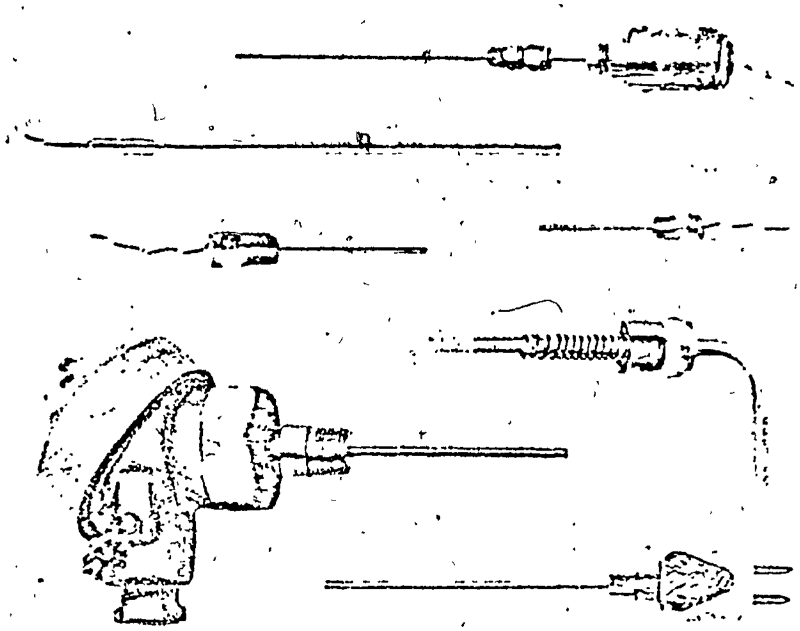
The ZIG/ZAG Thermocouple provides true surface temperature measurements. Normally a thermocouple extracts heat from the point of measurement through the leadwires. This conduction process has a cooling effect on the thermocouple junction, resulting in erroneous readings. The unique configuration of the ZIG/ZAG thermocouple prevents this condition. The ZIG/ZAG is heated to the same temperature as the surface to which it is attached, independent of the junction, to insure against temperature losses.

TEMPERATURE INSTRUMENTATION FOR INDUSTRY
HY-CAL ENGINEERING
12105 LOS NIETOS ROAD, SANTA FE SPRINGS, CALIFORNIA 90670
PHONE (213) 693-7785

Hy Cal

PLATINUM RESISTANCE TEMPERATURE SENSORS*

ANNEX A



RTS – 4135 & 4140 SERIES

TYPICAL APPLICATIONS

Temperature Measurement and Control in . . .

- Pipelines
- Refineries
- Textiles
- Chemicals
- Metals
- Pulp & Paper
- Food & Drug
- Environmental

and other Major INDUSTRIES

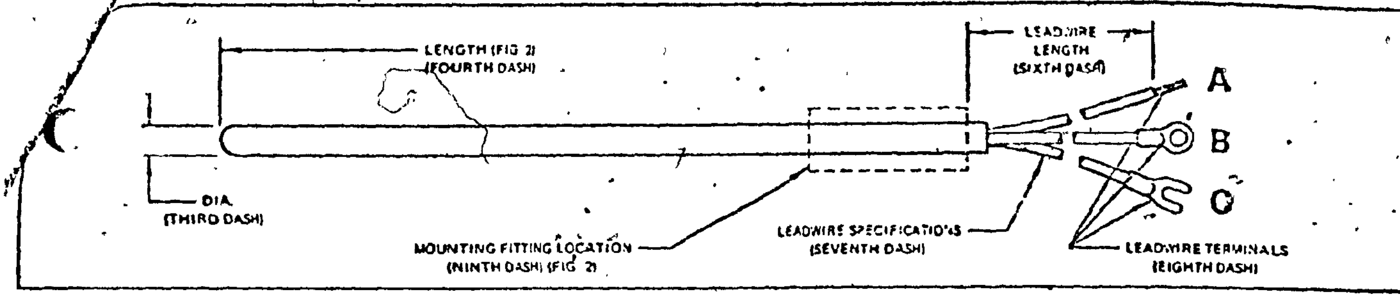
* For Copper, Nickel and Balco Sensors see PD-552

Featuring . . .!

- FROM –320 TO +1500° F
- HIGH ACCURACY & STABILITY
- FAST RESPONSE
- NUMEROUS CONFIGURATIONS
- PROCESS MOUNTINGS
- EXCELLENT REPEATABILITY
- 2, 3 OR 4-WIRE

[DESCR]IPTIVE DRAWING

FIG. 1



ORDERING INFORMATION

SERIES	DESCRIPTION	TEMPERATURE RANGE
RTS-4135	Platinum Resistance Sensor – Low Temperature	–320 to + 500°F
RTS-4140	Platinum Resistance Sensor – High Temperature	–60° to + 1000°F

FIRST DASH	TEMPERATURE COEFFICIENT	
A	385 Platinum	See Note ①
B	392 Platinum	

SECOND DASH	RESISTANCE AT 0°C (R_0) (Less Lead Resistance)	
100	100 ± 0.1 ohm	A silver-tip may be specified for high tip sensitivity & fast response. Add "ST" after the second dash No.
200	200 ± 0.2 ohms	
500	500 ± 0.5 ohms	

THIRD DASH	SHEATH DIAMETER	MATERIAL	TIME CONSTANT
A	3/16"	300 Series Stainless	4.5 sec. (Nom.)
B		Inconel	
C	1/4"	300 Series Stainless	7.5 sec. (Nom.)
D		Inconel	
E	3/8"	300 Series Stainless	12.0 sec. (Nom.)
F		Inconel	

FOURTH DASH	IMMERSION LENGTH (Reference Fig. 1 & 2.)
☐	Specify in one inch increments. (2" Minimum)

FIFTH DASH	LEADWIRE CONFIGURATION	
2	2-wire	Refer to "SENSOR CONNECTIONS" On Page 4.
3	3-wire	
4	4-wire	

SIXTH DASH	LEADWIRE LENGTH (Reference Fig. 3)
☐	Specify in one inch increments. (For Mtg. Ftg H5, H6, H7, C1, C4, C8 or C9 Specify [0])

SEVENTH DASH	LEADWIRE SPECIFICATIONS
☐	Select wire and insulation from Fig. 3 (If "6th Dash" [0], Specify [0]) See Notes ② & ⑤

EIGHTH DASH	LEADWIRE TERMINALS
☐	Select from Fig. 1 above. (If "7th Dash" [0], Specify [0]).

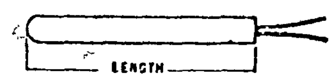
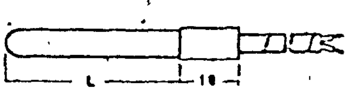
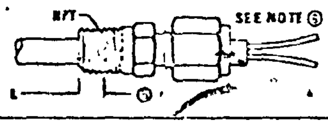
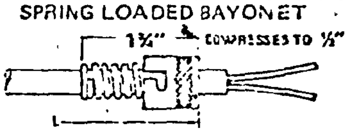
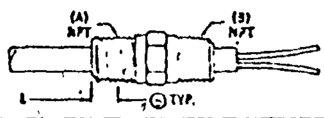
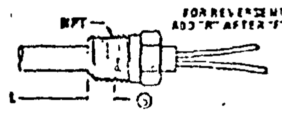
NINTH DASH	MOUNTING FITTINGS
☐	Select from Fig. 2 See note ③

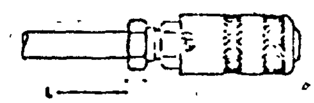
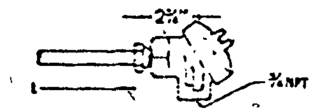
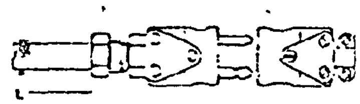
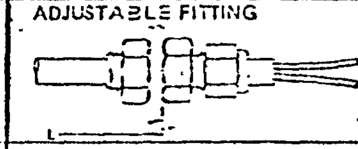
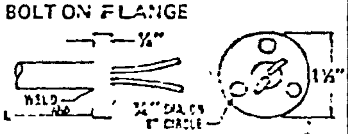
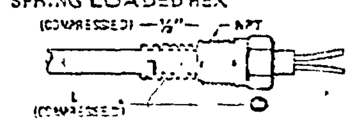
RTS-☐-☐-☐-☐-☐-☐-☐-☐-☐-☐	← Your Model No.
RTS-4140 – A – 100ST – D – 12 – 3 – 0 – 0 – 0 – F4ATHG	← Sample Model No.

SPECIAL REQUIREMENTS: Add "X" after specific Dash Number and specify details.

MOUNTING FITTINGS

FIG. 2

TYPE	NINTH DASH		
NONE 	FO		
LEADWIRE ADAPTER 	FA		
COMPRESSION FITTING 		BRASS	S.S.
	1/8 NPT	F 1	F 5
	1/4 NPT	F 3	F 6
	1/2 NPT	F 4	F 7
SPRING LOADED BAYONET 	F 2 (Std. for 3/16" Dia. Sheath)		
FIXED DOUBLE HEX NIPPLE 	A x B	BRASS	S.S.
	1/4 x 1/4 NPT	F 8	F 11
	1/4 x 1/2 NPT	F 9	F 12
	1/2 x 1/2 NPT	F 10	F 13
FIXED SINGLE HEX 		BRASS	S.S.
	1/8 NPT	F 14	F 17
	1/4 NPT	F 15	F 18
	1/2 NPT	F 16	F 19

TYPE	NINTH DASH		
MINIATURE HEAD 	H5		
CAST HEAD & FIXED HEX 	(Terminal Board Included)		
	IRON	H 6	
	ALUMINUM	H 7	
QUICK DISCONNECT 		MALE	FEMALE
	2 PIN	C 1	C 4
	3 PIN	C 8	C 9
ADJUSTABLE FITTING 	F 20		
BOLT ON FLANGE 	F 21		
SPRING LOADED HEX 		BRASS	S.S.
	1/8 NPT	F 22	F 25
	1/4 NPT	F 23	F 26
	1/2 NPT	F 24	F 27

LEADWIRE SPECIFICATIONS

FIG. 3

TYPE (SEVENTH DASH)	WIRE MATERIAL	SEE NOTE	INSULATION	INSULATION TEMPERATURE LIMITS	WIRE GAGE SUPPLIED	STD. LENGTH SUPPLIED	WIRE DATA
A	Platinum	②	Bare	----	.010" Dia.	1"	• High Temperature • Short Leads — Cost
D	Nickel Clad Copper	②	Fiberglass	+32 to +900°F	24 AWG	18"	• Medium Temperature • Medium Leads • Extremly Rugged
E		②	TFE Teflon	-150 to +500°F	22 AWG	18"	
F	Stranded Copper	② ④	FEP Teflon	-150 to +300°F	22 AWG	18"	• Low Temperature
G		② ④	PVC	-50 to +220°F	22 AWG	18"	• Long Leads
H		② ④	Nylon	-50 to +220°F	22 AWG	18"	• Very Rugged

NOTES

① TEMPERATURE COEFFICIENT: ... Hy-Cal features a temperature coefficient of .00385 or .00392 ohms/ohm/°C in all of our Platinum sensors. For industrial applications, Platinum 385 is generally used since it offers a substantial cost savings without loss of quality, ruggedness or utility. Platinum 385 is also recommended for new applications and large quantity requirements.

② LEAD WIRE INSULATION: .... The selection of lead wire insulation shall be determined by the maximum and minimum temperature conditions at the lead wire end of the probe. See insulation temperature limits in Figure 3.

③ MOUNTING FITTINGS: .... More than one mounting fitting may be specified, provided they are compatible with the sheath diameter. Group mounting fitting numbers for the "Ninth Dash".

④ COPPER LEAD WIRE: .... May be specified when lead wire temperatures are below 300°F.

⑤ NPT ENGAGEMENT: .... Maximum thread engagement for 1/8 NPT = .26"; 1/4 NPT = .40"; 1/2 NPT = .53".

⑥ COMPRESSION FITTING: .... Fixed position only after initial installation. For Adjustable Compression Fitting, specify "A" after specific "F" number and add one of the following Ferrule material letters. N = neoprene (-40 to +200°F), T = teflon (-300 to +500°F), or L = lava (-300 to +1800°F). Note: Lava seal is not reuseable. Example: F5AT = 1/8 NPT - S.S. Adj. Comp. Ftg. with teflon seal.

...ICATIONS

...RATURE RANGE ...	RTS-4135 Series: -320 to +500°F. (-200 to +250°C). RTS-4140 Series: -60 to +1000°F (0 to 500°C)
...CONSTANT:	The time required for the sensor to reach 63.2% of a step change in temperature in water flowing at 3 feet per second at 140°F (60°C). Specifications are outlined for each probe diameter under "Ordering Information".
SE... ...ATING:	Defined as the rise in the indicated temperature due to power dissipated in the sensor. Self-heating is derived from the sensor immersed in water flowing at 3 feet per second at 68°F (20°C). The maximum self-heating effect is 25 mw/°F (50 mw/°C) for all sensors.
MAXIMUM CURRENT:	5 MA recommended maximum for all sensors.
INTERCHANGEABILITY: .	±1.0°F at 32°F. (± 0.5°C at 0°C).
REPEATABILITY:	After 10 consecutive thermal shocks over the specified range, the sensor will not shift R₀ by more than an amount equivalent to 0.25°C (0.5°F) or 0.1% of the total temperature range, whichever is greater.
VIBRATION:	The sensor shall be undamaged by a sinusoidal vibration of 25 g's from 20 to 2000 Hz along each of the three mutually perpendicular axes for a period of 15 minutes.
STABILITY:	R_o will not change more than 0.25°C (0.5°F) or 0.1%, whichever is greater, after one year of service.
SHEATH MATERIAL:	Inconel or 300 Series Stainless Steel, as specified.

SENSOR CONNECTIONS

For Industrial applications a 2-wire sensor may be used for distances up to 1 or 2 feet, terminating into a junction-box or connector. The connector provides the transition for small diameter sensor wires to larger diameter leadwires and allows the addition of a third compensating wire. This 3-wire configuration is now basically a 3-wire sensor, and provides almost total compensation for long leadwire runs. For extremely long runs a 2-Wire Transmitter may be used with either a 2 or 3-wire sensor to provide an output of 4-20 or 10-50 MA. Sensors incorporating 4-wires are generally employed in the laboratory when extremely high precision is required. General information on sensor connections is outlined on Resistance-Element data sheet PD-550.

ADVANTAGES

- Industrial or Reference Grade Platinum
- Controlled accuracy & excellent repeatability
- Low Cost — Rugged construction
- L... service life – Extremely durable
- Fast response
- Excellent stability
- High output
- No reference required
- Sensitive to small temperature changes
- Silver-tip sensitivity option
- Fail-safe protection
- Assured interchangeability

GENERAL INFORMATION

The platinum resistance sensors described in this bulletin are designed and manufactured by Hy-Cal to meet the industrial requirements for ruggedness, stability, interchangeability and standardization. The sensors are constructed of extremely fine platinum wire wound on a mandrel and imbedded into a metallic sheath. The resistance of the sensor varies precisely with temperature and provides a relatively linear output. They feature fast response, high accuracy and excellent repeatability over a wide temperature range. Strain-free windings, rugged lead and sheath materials are employed permitting the sensor to withstand rigorous vibration and thermal shock. The sensors provide an output signal approximately ten times higher than thermocouples, thereby eliminating the need for high gain instrument amplifiers and greatly reducing the noise susceptibility and zero drift associated with low-level input signals.

HY-CAL

Every facility for quality assured manufacturing of precision platinum sensors is provided by Hy-Cal . . . . specialists in temperature instrumentation. Our production engineering staff has the experience and expertise to assure consistently reliable products. Complete testing facilities coupled with unparalleled manufacturing methods provide significant performance benefits from every sensor.

STOP ! ASK FOR ADDITIONAL INFORMATION ON ...

- Copper, Nickel & Balco Sensors
- Bridge Networks
- Transmitters
- Thermocouple References
- Universal Transducer Indicators
- RTD Elements
- RTD Amplifiers & Linearizers
- Digital RTD Indicators
- T/C Amplifiers & Linearizers
- Temperature Calibration Sys.
- RTD Air, Gas & Surface Sensors
- Signal Conditioners
- Thermocouples
- Analog & Digital T/C Indicators
- Platinum Standards

TEMPERATURE INSTRUMENTATION FOR INDUSTRY
HY-CAL ENGINEERING
12105 LOS NIETOS ROAD, SANTA FE SPRINGS, CALIFORNIA 90670
PHONE (213) 698-7785

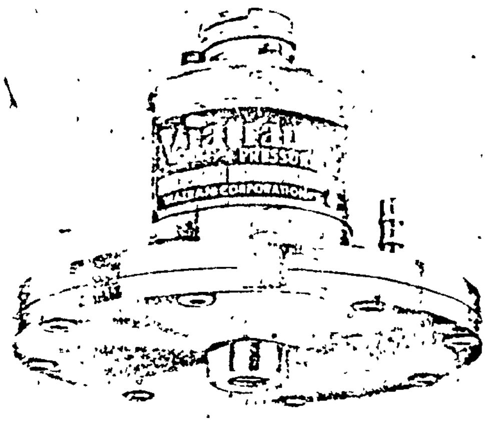
ANNEX A

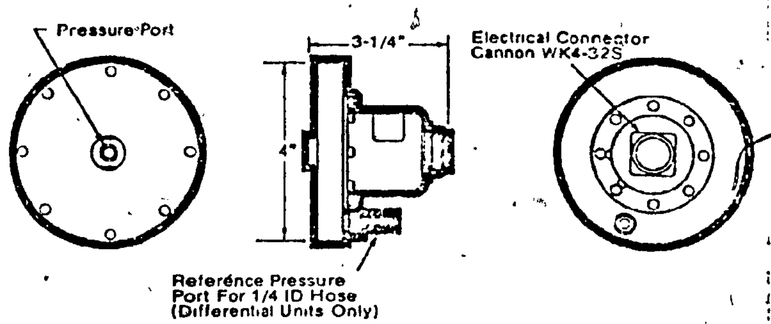
Pressure Transducer

Model 215

SPECIFICATION SHEET No. 52

Output: 215: 3mv/v DC
Ranges: 0-5" W.C. thru 0-150" W.C.





Specifications

*Pressure Ranges	0 5, 10, 15, 20, 25, 30, 50, 100, 150 inches of water, gage and differential
Differential Reference	100 PSI maximum
*Measureand Fluids	Any fluids compatible with 17-7 ph & 303 stainless Differential units require a clean dry gas as a reference media
*Static Error Band	The maximum deviation of any pressure point from a straight line drawn between the end points shall not exceed ±0 4%
Linearity (terminal)	Within ±0 4% FSO
Hysteresis	Within ±0 2% FSO
Repeatability	Within ±0 1% FSO
Resolution	Infinite
Zero Balance @ 70°F	Within ±2% FSO
Full Scale Output	0-5" W C 1mv/v minimum 0-10" thru 0-30" W C 2mv/v minimum 0-50" & Up 3mv/v minimum
Excitation Voltage	10V AC or DC (15V max)
Bridge Resistance	350 ohm
Insulation Resistance	Greater than 5000 megohms at 50V DC @ 70°F
Operating Temperature Range	−100°F to +250°F
Insulation Resistance	Greater than 100 megohms @ 50V DC and 70°F

*Compensated Temperature Range	−65°F to −250°F
Storage Temperature Limits	−100°F to −275°F
*Temperature Effect on Zero	Less than 2%/100°F
*Temperature Effect on Span	Less than 1%/100°F
*Proof Pressure	1½ times rated range
Natural Frequency	10 Hz nominal for 0-5" W C range extended up to 125 Hz nominal for 0-150" W C Range
Response Time	Less than 5 milliseconds to reach 90% of full scale
*Pressure Connection	¼" NPT Female
Differential Reference Pressure Connection	Fitting for ¼" I D hose
Electrical Connection	Cannon WK4-32S (mating connector WK4-21C)
Enclosure	High quality stainless steel All electrical components are protected against adverse environmental conditions
Identification	Model number serial number, pressure range and manufacturers name are provided on a stainless steel nameplate welded to transducer body
Weight	22 ounces nominal

*Indicates specifications that can be improved or changed by standard modifications listed in Catalog No 48

Ordering: Specify model pressure range and indicate modifications or accessories required Refer to Catalog No 48 for details

Via INTERTECHNOLOGY

ANNEX A

PD 593

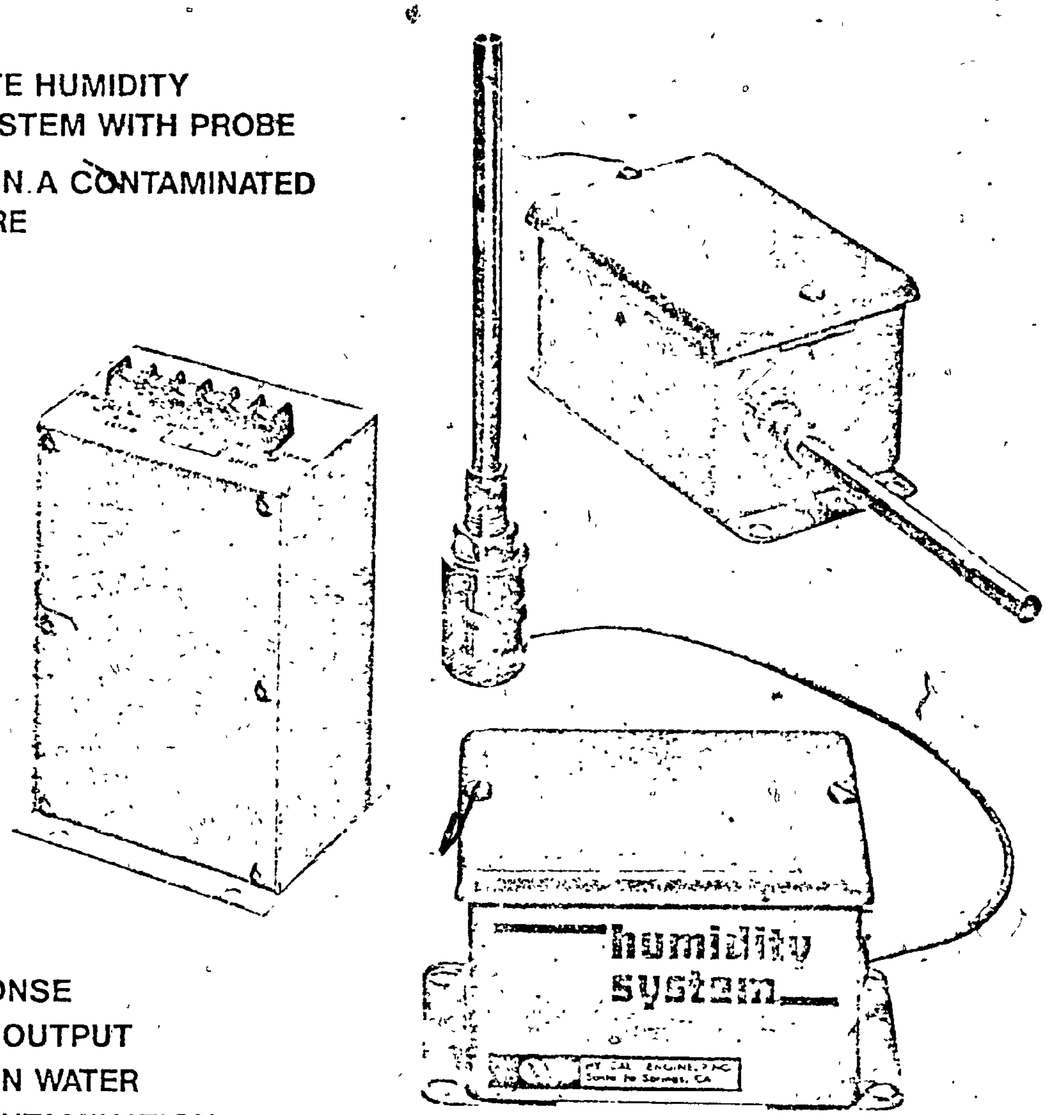
PRODUCT DATA

Humidity System

4 to 20 milliamp

A COMPLETE HUMIDITY SENSOR SYSTEM WITH PROBE

ACCURATE IN A CONTAMINATED ATMOSPHERE



LINEAR
FAST RESPONSE
4 TO 20 mA OUTPUT
WASHABLE IN WATER
RESISTS CONTAMINATION
WIDE TEMPERATURE/PRESSURE RANGE
RELATIVE HUMIDITY READOUT FROM 0% TO 100%

HY-CAL ENGINEERING