AN ENGINEERING ANALYSIS OF PHYTOTRON "ELLIE"

A design for a plant growth cabinet developed to satisfy broad operating conditions of temperature and humidity for the requirements of the

Department of Agronomy,

Waite Agricultural Research Institute,

University of Adelaide.

by

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3rd February, 1966.

Mr. H.E. Wesley Smith,
Academic Registrar,
The University of Adelaide,
ADELAIDE. SOUTH AUSTRALIA.

Dear Mr. Wesley Smith,

I authorise the University of Adelaide to make available for loan and for photo-copying my thesis on "An Engineering Analysis of Phytotron 'Ellie'."

Yours faithfully.

ALLAN SHAW

I dedicate this paper to my wife, and in her name may Phytotron Ellie serve well as an

E xtraordinary

L ight-box for

L eguminous

I nternal

E nvironment

A.S.

STATEMENT OF AUTHORSHIP

This thesis contains no material which has been accepted for the award of any other degree or diploma in any University and, to the best of my knowledge and belief, contains no material previously published or written by another person, except when due reference is made in the text of the thesis.

The design presented here is registered for Commonwealth Provisional Protection, Application No. 45567 dated 9th June, 1964.

ALLAN SHAW
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1. SUMMARY.

This paper describes the system developed to satisfy the design requirements for an artificially lit plant growth cabinet that will operate over a broad range of simulated climatic conditions, maintain to close tolerances the desired temperature and humidity and provide automatic change-over between day-night conditions including change in daily temperature range. The system presented here can serve to meet the design requirements of many research organizations in agriculture, genetics and allied fields.

The C.S.I.R.O. has devoted much time to this area - it has been a matter of growing interest to those doing research in agriculture, genetics, etc. The 1964 edition of the American Society of Heating, Refrigerating and Air Conditioning Engineers, "Guide," devotes an entire chapter to this subject. The "Guide" refers to work done in England, America, Belgium, Sweden, Russia, South Africa, Australia, etc.

A study of the problems associated with existing phytotron design at home and abroad preceded this paper. The international meeting held in Melbourne - the 1962 C.S.I.R.O. Symposium on Engineering Aspects of Environment Control for Plant Growth - served to highlight many of these problems.

The engineering of a temperature-humidity phytotron may, at first glance, appear quite simple to the uninitiated. The loads are relatively constant. In the artificially lit cabinet the light load does not vary significantly. The transmission load

is very low, particularly if the walls are well-insulated. The moisture and heat loads from the plants are also very low and do not vary greatly. Nevertheless, extensive enquiry indicated that a simple, economically feasible unit meeting all of the Waite Institute requirements did not exist.

Growth cabinets of this size range produced in Australia and abroad have been designed to be self-contained units. Emphasis has been placed on the production of a single package that will operate immediately after delivery. One of the basic changes in this design is to reject this unitary approach as being incompatible with the design requirements. The unitary approach used in the design of existing cabinets fails to take full account of the thermodynamic and fluid mechanic principles involved. In the "Waite" design the cabinet proper is separated from the air conditioning system and linked with a system of ductwork. As a result it was possible to eliminate some of the major problems associated with existing designs.

Performance of temperature-humidity cabinets to date has been unreliable and their use has been discouraged. The automatic control of the dehumidifying process as a means to maintain the humidity setting is a major factor responsible for the problem. This has resulted in unnecessarily intricate and expensive designs.

The control system of this design is extremely simple and will operate to close tolerances free of the dehumidification function. A system is presented here that makes temperature-humidity control cabinets available at a reasonable additional cost when compared with temperature-only cabinets. The conclusion to limit the use of temperature-humidity cabinets [1*] no longer is valid.

This paper presents an original design which attempts to overcome some of the defects of existing temperature-humidity growth cabinets.

* Superscript numerals refer to List of References.

2. INTRODUCTION.

2.1 Equations Forming Background To The Air-Conditioning Calculations

There are several relationships and simplifying assumptions that have become common usage under certain circumstances in the very specialized branch of engineering associated with air conditioning.

1. Air quantities are frequently expressed in volume rather than mass flow terms.
2. When the static pressure of the air system is within several inches water gauge of atmospheric pressure the variation in specific volume of the air in the common temperature ranges is frequently ignored and in such cases it is assumed that the specific volume remains constant at a standard condition of 13.5 cubic feet per pound mass of dry air.
3. Enthalpy deviation is ignored.

To one unfamiliar with this specialized branch it may appear incorrect to operate with a hybrid system of units including cubic feet of air per minute, Btu per hour and pound mass of dry air. The reason for this approach is that it agrees with the terminology adopted by the industry in its literature and standards, it simplifies the calculations and it is adapted to the use of the psychrometric chart. The significant figures that are associated with the calculations permit these simplifying assumptions since the results can deviate as much as 10 per cent from the true value. (See section 3.4.2(d)). A classical approach using 16 equations as presented by Morris [2] would not be warranted in this paper. Instead 3 fundamental equations used in air conditioning will describe the various process paths of the air system. On the other hand, the nature of this design does not permit the simplifying assumption that Morris [2] made that the air leaving the cooling unit is saturated with water vapour.

Each of the 3 air conditioning equations are expressed in volume flow terms that are based on the mass of the dry air component, (non-condensing). Water vapour is the other component, (condensing), that makes up the atmospheric air mixture. In all 3 equations, $\dot{m}$ represents the pounds of <u>dry air</u> per hour and lb, without a subscript, represents the pounds of dry air. (It is to be noted that in equation number 2 the product of the latent heat of vapourization in Btu per pound of water vapour and the humidity ratio in pounds of water vapour per pound of dry air results in the enthalpy units of Btu per pound of dry air).

<u>Equation for sensible heat change is developed as follows:</u>

$$Q_S = \dot{m} \times c_p \times \Delta t$$

$$\text{Btu/hr} = \text{lb/hr} \times \frac{\text{Btu}}{\text{lb.}^\circ\text{F}} \times {}^\circ\text{F}$$

let $\dot{m}$ = lb/hr (pounds of dry air per hour)

let $\bar{v} = 13.5\ \text{ft}^3/\text{lb}$ (of dry air)

let $c_p = 0.24 \dfrac{\text{Btu}}{\text{lb.}^\circ\text{F}}$

then $Q_S = \text{cfm} \times 1.08 \times \Delta t$ ....(1)

A dimensional check to verify the constant of 1.08:-

$$Q_S = \frac{\text{ft}^3}{\text{min}} \times \frac{60\ \text{min}}{\text{hr}} \times \frac{1}{13.5\ \text{ft}^3/\text{lb}} \times 0.24 \frac{\text{Btu}}{\text{lb.}^\circ\text{F}} \times {}^\circ\text{F} = \frac{\text{Btu}}{\text{hr}}$$

whence $\dfrac{60 \times 0.24}{13.5} = 1.08$

Equation for latent heat change is developed as follows:-

$Q_{LH} = \dot{m} \times h_a$

Btu/hr = lb/hr x Btu/lb

let $\dot{m}$ = lb/hr = pounds of dry air per hour

let lb_{wv} = pounds of water vapour

let $\bar{v} = 13.5\ ft^3/lb$ (of dry air)

let h_a = enthalpy in Btu per pound of dry air

let h_{wv} = 1055 Btu per pound of water vapour
(latent heat of vapourization)

let w = grains of water vapour per pound of dry air

7000 grains of water vapour = 1 lb_{wv}

then $Q_{LH} = cfm \times 0.67 \times \Delta w$ ....(2)

A dimensional check to verify the constant of 0.67:-

$$Q_{LH} = \frac{ft^3}{min} \times \frac{60\ min}{hr} \times \frac{1}{13.5\ ft^3/lb} \times \frac{1\ lb_{wv}}{7000\ grains} \times \frac{1055\ Btu}{lb_{wv}} \times \frac{grains\ of\ wv}{lb}$$

$$= \frac{Btu}{hr}$$

whence $\frac{60 \times 1055}{13.5 \times 7000} = 0.67$

Equation for total heat change is developed as follows:-

$Q_{Tot} = \dot{m} \times h_a$

Btu/hr = lb/hr x Btu/lb

let $\dot{m} = \frac{lb}{hr}$ = pounds of dry air per hour

let $\bar{v} = 13.5\ ft^3/lb$ (of dry air)

let h_a = enthalpy in Btu per pound of dry air

then $Q_{Tot} = cfm \times 4.45 \times \Delta h_a$ ....(3)

A dimensional check to verify the constant of 4.45:-

$$Q_{Tot} = \frac{ft^3}{min} \times \frac{60\ min}{hr} \times \frac{1}{13.5\ ft^3/lb} \times \frac{Btu}{lb} = \frac{Btu}{hr}$$

whence $\frac{60}{13.5} = 4.45$

2.2. Background To Design

2.2.1 Existing designs having humidity control

In the size and cost range considered, only the L.B.H. environmental plant growth cabinet developed by the Commonwealth Scientific and Industrial Research Organisation, Highett, Victoria, Australia, was designed to meet both the temperature and humidity function over a wide range of settings. At the inception of the "Waite" project, I visited the Engineering Section at Highett.

The part of the L.B.H. working section which includes the hand-operated winch that positions the platform, and its plant tray arrangement allowing for easy withdrawal from the compartment, has been applied with some modifications to the "Waite" design. The outer structural frame dimensions are similar to the L.B.H. since it is fabricated by the same contractor, whose plant is tooled for the production of that particular size. The "Waite" design does not include a concealed duct in the wall and as a result, although the exterior cross-section of the plant compartment is identical, the "Waite" interior cross-section is 20 square feet, or 14 per cent larger than the L.B.H.

The systems of numerous commercially available self-contained cabinets were studied; none of them offered humidity control over a full range. Many of the commercial manufacturers offered humidity control as optional additional equipment. Humidity control as an adjunct to a temperature control system has limitations. In all the cases examined, the humidity control feature was limited. For example, many of these commercially available units, because of their self-contained feature, remove all the sensible heat gains from lights by means of the refrigeration system. They thus eliminate the need for a separate ventilation system which would involve problems of air intake and exhaust, and connections to the outside. (Fig.37A, p.372 [3]). In these cabinets it would be difficult to add humidity control in the low humidity ratio region since the coil performance characteristic accompanying a high dry bulb temperature and a low humidity ratio entering condition will approach dry performance; that is, the load ratio line has almost a zero slope. Thus the additional sensible heat removed from the lights serves to prevent the cooling coil from meeting the dehumidifying requirement when the design setting is at a low humidity ratio. Both the L.B.H. cabinet and the "Waite" design partition the light section from the plant compartment, separately ventilating the light section.

2.2.2 The problem: "Range" of operating settings

In the design of a phytotron a clear distinction should be made between the cooling load variation for any one operating setting and the variation desired in the range of operating settings. As pointed out in section 1, the former is minimal (and, in fact, is an advantage that is utilised in this design) whereas the latter, as opposed to conventional air conditioning applications, is very large. A system to cope with this range when both temperature-humidity control is specified poses many difficulties when a cooling coil is used to achieve the dehumidifying process.

One solution, expensive and impractical, is the dehumidification of that design set-point of maximum humidity ratio and dry bulb temperature to a condition far enough below the dew point of the lowest humidity ratio and lowest dry bulb setting to meet the total cooling load of the latter condition. Such a solution would involve not only an immense cooling system of 17 tons of refrigeration (in the case of the "Waite" requirements), but also large and wasteful reheat and humidification processes. As a consequence, the many designs using this approach or modifications of it, have limited their range of humidity settings to high humidity ratios only. [4, 5, 6] This is unfortunate since it excludes a range of weather conditions that occur frequently.

2.2.3 The L.B.H. cabinet

Reference will be made to the L.B.H. cabinet since it is a pioneer development in this field. It represents one method of providing a growth cabinet that covers a wide range of both temperature and humidity control. The L.B.H. cabinet offers a complex solution focusing its attention on economy and maintaining, as do the commercial units, a self-contained single package unit. Basically, a very different approach was taken in the "Waite" design.

2.2.4 Brief description of L.B.H. cycles

The L.B.H. system utilises 2 separate air circuits. In one circuit approximately 1/3 of the return air combines with the fresh-air intake supply, passing, in the order listed, through:- one path of a cross-flow air-to-air heat-exchanger, an auxiliary cooling coil, a dehumidifier, the other path of the air-to-air heat-exchanger and, finally, a refrigerant condenser, where it is reheated. The other 2/3 of the return air is sensibly cooled and then combined with the first circuit at a point upstream of the fan. Downstream of the fan it is humidified and/or reheated if necessary. One on-off temperature controller is associated with the dehumidification circuit and another on-off temperature controller with the sensible cooling circuit. In addition, there are two other controllers associated with reheat and humidification downstream of the fan. The 2 refrigeration circuits are not independent, the auxiliary cooling coil of one air circuit being cooled by the refrigeration system of the other air circuit. (Fig. 11 [7], repeated in Appendix IX). The use of restrictors or capillary tubing rather than thermal expansion valves and the use of hermetic compressors, point to a uniform rate of refrigerant vapor and operating conditions which are inflexibly fixed and established. (pp. 352-3, [8]). Thus, a system is presented which is dependent on the complex interaction of two on-off refrigeration plants responsive to temperature, associated with reheat and humidification to maintain a single design setting. (See Appendices VIII and IX for schematic comparison of the L.B.H. and "Waite" cycles).

2.2.5 Problems associated with L.B.H. - The "Waite" solution

The following problems have been reported by the designers:

2.2.5.(a) Temperature and humidity are dependent variables

"The temperature control and humidity control systems are not independent since there is always some interaction, particularly from the dehumidification system (as described above) which causes some heating .... a departure from the control point in a vertical direction on the chart would result in corrective action by both the temperature and humidity systems .... there is an inbuilt instability which must be allowed for in the design." [1]

At "Waite" the major control system's corrective action is to sensibly heat and/or humidify. The design accentuates maximum separation of these two functions. The sensing element of the controller is responsive to dew point temperature or humidity ratio; thus the "independent variable" of moisture content per pound of dry air is measured rather than the dependent variable, relative humidity. As is detailed under the section "The Role of the Humidifier in the System," at "Waite" the humidifier is selected to minimise variation in dry bulb temperature that accompanies the humidifying process. During automatic day-night change-over the humidifier action is not affected.

2.2.5.(b) Time function

"The time constant of the dehumidifying process is different from the humidifying process and different again from heating and cooling. These are some of the reasons why temperature and humidity control become so much more complicated and expensive in the control of environment than the control of temperature alone." [1]

It should be noted that these remarks are associated with the on-off control system. At "Waite" the effects of the dehumidification time functions are eliminated. The heating and the humidification time functions proceed with negligible interaction. The modulating controls serve to maintain a constant load compatible with the operating setting and the system performance.

2.2.5.(c) Climatic noise, plant ripple, hunting

"Where heating and cooling is required, such as for the glasshouse itself or for artificially illuminated cabinets, a double-acting controller is necessary which must include some form of feedback compensation to prevent hunting. Methods of achieving this, and, more particularly, means of reducing the width of the neutral zone and off-set to negligible proportions have also been described by Riordan.

"This method of control does lead to a series of ripples in the temperature time curve with a frequency varying from half a minute to about three minutes. The effect of these high-frequency changes on the plant is not known, but it is believed that it can be regarded as climatic noise. This and other aspects of the significance of rapid fluctuations of short duration will no doubt be discussed in a later session." [1]

At "Waite" the modulating control system with 2-mode control (proportional and reset), associated with a continuously operating plant, avoids this problem.

2.2.5.(d) Temperature gradients in supply ducts

"It is an advantage to reduce the dry bulb .... to avoid large gradients in the supply duct. This can be achieved by means of an auxiliary cooling coil ...." [1]

At "Waite" a continuously operating single air-cycle and single refrigeration cycle maintain a constant setting. Again, this L.B.H. problem is associated with on-off control.

2.2.5.(e) Air distribution fails to meet the major design aim

(This is very important and is treated in section 3.2 of this paper).

2.2.5.(f) <u>Unsuitability for naturally lit cabinets</u>

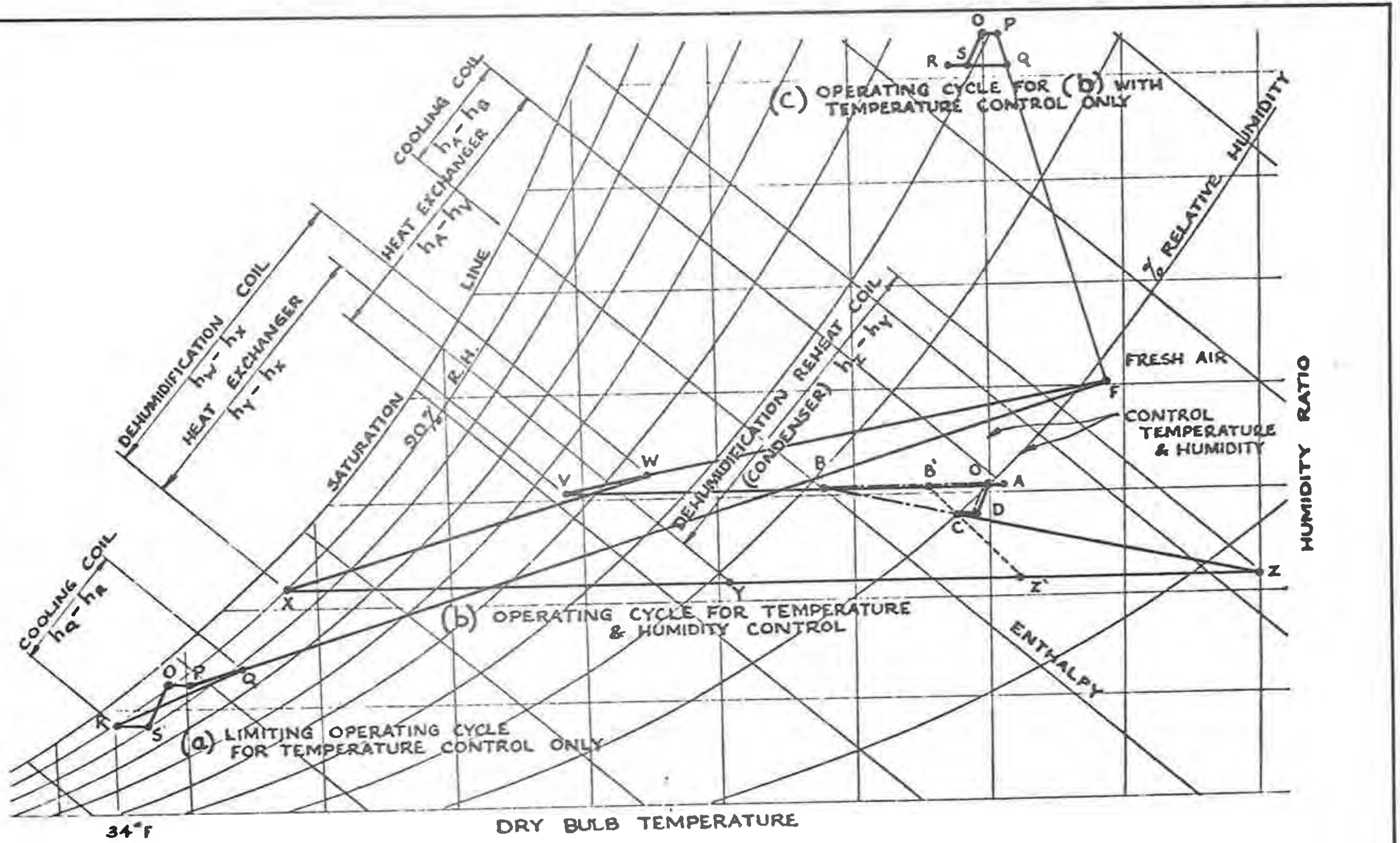
"If in addition to these inbuilt instabilities, the load itself fluctuates rapidly, the problem becomes a very difficult one indeed. This is precisely the case when the humidity control of a naturally lit enclosure is attempted. A sudden change in solar radiation caused by say, a cloud, not only rapidly changes plant transpiration and, therefore, the latent-heat load, but also alters the sensible load, affecting both the humidity control and temperature control systems simultaneously. Furthermore, the magnitude of the sensible heat load in this case is so high that the problem is a difficult one even without taking into account these sudden changes. For these reasons, it is believed that accurate humidity control of naturally lit enclosures should not be attempted unless there are very compelling reasons for doing so." [1]

The "Waite" system is so devised as to avoid an excess artificial load. The statement that the magnitude and fluctuation of the loads are high is not quite correct. An examination of the air paths drawn on a psychrometric chart clearly indicates this point. Furthermore, had the L.B.H. system been able to utilize the full air-flow, rather than one-third, to accomplish dehumidification, the change in the intensive property of humidity ratio would have been reduced by one-third. Note in Figs. 5 [1] and 6, [1] (see Fig. 1 on the following page repeated from Fig. 5 [1]), the distance between D and A is very short and represents the small sensible and latent heat gains within the plant compartment. It is not the load or fluctuation of load that is high. Both in the L.B.H. and "Waite" designs (lines 8 to 1D Appendix VI), the temperature and humidity gradients have been kept low by the use of large air-flow quantities (sect. 1). On the contrary, the loads are minimal. It is the on-off L.B.H. system of control that causes large fluctuations.

The "Waite" design excludes the dehumidification and sensible cooling processes from the control system and balances the minor load variations by means of modulating controllers in association with continuously operating air and refrigeration cycles. The system will maintain the set-point to close tolerances without fluctuation even in the case of naturally lit cabinets.

2.2.5.(g) <u>Dehumidification and frosting</u>

In an "on-off" refrigeration cycle, where the set-point temperature of the refrigerated air is above 32°F, it is not uncommon to depend on the "off" period of the cycle for defrost to occur. Thus an inexpensive cabinet [9] costing $250 (£A110) is an example of such a design. Here "plant ripple" is also considered acceptable. Where close control including humidity is concerned, it should be avoided. The heat transfer to the direct expansion coil is varying during the "on" period as the coil is increasingly "insulated" with accumulating ice. Heat transfer during this period



OPERATING CYCLES

(NOTE: (b) REPRESENTS THE LBH CYCLE)

(See section 2.2.4 for a brief description of the L.B.H. cycle).

FIG. I

does not follow a smooth diminishing pattern with formation of frost and there is one point at the start of frosting where heat transfer is actually improved. Thus the "Waite" design would not permit the condition of frosting even if it were an on-off operating refrigeration cycle, and since it is a continuously operating cycle there is no question that the design must not permit frosting.

Three fundamental "air-conditioning" equations have been presented in section 2.1, representing sensible, latent and total heat variation as related to air-flow volume quantities. As pointed out by Morse [1] in connection with the internal plant compartment temperature gradient, the higher the air-flow, the lower the temperature gradient for the same heat transfer. The sensible heat equation No.1 clearly demonstrates this inverse proportionality. Let us apply the same consideration to dehumidification, that is, the latent heat equation No.2. (Section 2.1). Here the variation in rate of air-flow is inversely proportional to the variation in humidity ratio.

In the L.B.H. cabinet

> "approximately one-third of the main air-flow passes through the dehumidification system ...... two-thirds through the cooling circuit ...... the coil running dry." (Figs. 5, 6 [1]).

Thus, in the L.B.H., the problem of dehumidifying at a temperature high enough to prevent frosting is aggravated since, with the use of only one-third the air-flow, three times more moisture per pound of dry air must be removed. This problem may lead to frosting or failure to maintain the design setting under full-load conditions. To counteract this, it may be necessary to eliminate the fresh air intake quantity. [1] (Sections 2.2.5.(i) and 3.5.1).

In L.B.H., the precooling of one-third of the main return air quantity in the first pass through an air-to-air heat exchanger assists in dehumidification. As the entry condition to a dehumidification coil moves to the left (to a lower dry bulb temperature) along a horizontal line, (humidity ratio line), dehumidification increases on a psychrometric chart; that is, the slope of the load ratio line increases. However, with the reduction of cooling load (as one moves to the left) the direct expansion coil acts to reduce the refrigeration temperature in the conventional cycle employing a thermal expansion valve to maintain a constant superheat, (as in the case of "Waite"). Consequently, with the reduction of heat transfer due to the lower entering dry bulb, the thermal expansion valve throttles and the refrigerant temperature approaches the condition of frosting. In so doing, however, it achieves two functions that are particularly essential to a coil which is employed over a wide range of operating settings:

1. It keeps the evaporator active without permitting wet compression;
2. It keeps the rate of refrigerant flow compatible with the rate of heat removal by the evaporator.

When a restrictor (capillary or expansion tube) is used, as in the L.B.H. system, there is no adjustment for varying operating conditions and the system must function within the limitations imposed by the resistance to flow across its fixed length and bore. It must seek other means to adjust to load variations; accumulators on the downstream side of the restrictors, low side receivers, and a hot gas by-pass line with a reducing valve for suction pressure control are the correctives employed in the L.B.H. system. (Fig. 11, [7], repeated as upper figure in Appendix IX). Though this is an often-used solution, this valve introduces another unstable element in the refrigeration cycle. It is to be noted that although this corrective action prevents frosting, it also may prevent meeting the dehumidification requirement. It should also be noted that the hot gas bypass process, similar to reheating, is wasteful.

At "Waite," the direct expansion coil is selected to dehumidify 3.7 grains of moisture per pound of dry air, or the equivalent of 3,850 Btu/hour at that design setting in the operating range where dehumidification is minimal. This exceeds by 50 per cent the total latent heat design load, including latent heat associated with fresh air, of the L.B.H. system.[7] If an unusually high moisture content in the outside fresh air supply is encountered, further dehumidification is possible by means of alternate scheme No.2, (Section 3.5.2), which utilises an air-to-air cross-flow heat exchanger to dehumidify the fresh air intake supply at the expense of the sensible heat of the <u>full</u> 1600 c.f.m. directly downstream of the direct expansion coil. The coil is selected to perform with a safety margin of about 2°F above the refrigerant temperature that would result in frosting. The minimal permissible temperature setting downstream of the preheat coil is associated with frost prevention.

2.2.5.(h) L.B.H. restrictor and "Waite" thermostatic expansion valve

Read, Cunliffe, Chapman and Kowslczewski [10] state:

> ".... As the expansion tube, as distinct from the thermal expansion valve, can be selected to operate without superheat, and is in addition reliable and trouble-free in operation, it has been chosen as means of expansion in the cabinets. ..."

The use of the expansion tube (restrictor or capillary tube), in the L.B.H. application may be satisfactory. In fact, when used in the L.B.H. on-off system the expansion tube allows for low starting torque since the high and low sides of the system equalize during the off period. However, the criticism of the thermostatic expansion valve is not justified.

Let us examine the first objection - the fact that the presence of superheat means that there must be a temperature variation in the air leaving the face of the coil. This condition is minimized by counterflow piping of the coil, the entering condition of the air meeting the leaving condition of the refrigerant. In this manner only a minimal portion of coil surface is used to obtain the superheat. (p. 865 [3]). Furthermore, at "Waite" a slight variation downstream of the coil would not affect the uniform entering air condition to the growth cabinet.

The second objection implies that the thermostatic expansion valve is unreliable and troublesome as compared to the capillary tube. Reference is probably being made to the reputation of the valve for cycling. When selected properly the thermostatic expansion valve is a reliable precision device. The ASHRAE "Guide," p. 605, [11] has this to say about cycling:

> "Although hunting is commonly attributed to the action of the thermostatic expansion valve, the valve alone is seldom responsible."

and in general the ASHRAE "Guide," p. 615, [11] states:

> "Inherently a capillary does not operate as efficiently over a wide range of conditions as does a thermostatic expansion valve but due to counterbalancing factors in most applications, its performance is generally very good. The simplicity of the capillary gives it the advantage of greatly reduced cost. It is seen that there are many compelling reasons for using capillaries in systems mass-produced for a highly competitive market."

With a restrictor there is only one condition for a balanced steady flow system. To quote Jordan and Priester p.324, [12] :

> "A capillary tube, with fixed dimensions and fixed resistance to flow, increases in ability to supply liquid refrigerant as the condenser pressure increases and/or the evaporator pressure decreases and the pressure differential thus becomes greater.

Conversely, the ability of the remainder of the system to supply liquid refrigerant to the capillary tube decreases as the evaporator pressure drops and/or the condensing pressure increases. Thus there is a point termed "capacity balance" at which the ability of the condensing unit to supply liquid refrigerant is just matched by the flow of the refrigerant through the capillary tube and with a liquid seal at the entrance to the capillary. On either side of this point of capacity balance there will exist conditions under which either refrigerant vapor enters the capillary tube along with the liquid refrigerant or liquid refrigerant will build up in the condenser. Only one point of capacity balance will exist for any one set of operating conditions."

There is still another draw-back which prevents the use of a restrictor in the "Waite" design. It is dependent on the system's maintaining a fixed refrigerant charge and a uniform rate of flow of oil with the refrigerant. At "Waite" flexibility for adjustment and modification is considered an important design requirement and the system uses a continuous operating, variable speed, open type compressor. This is not compatible with the unsteady flow characteristic of restrictors and with the changes in refrigerant charge due to losses at the open type compressor's shaft seals.

2.2.5.(i) The L.B.H. internal latent heat load

The estimate of latent heat load as presented by Pescod, Read and Cunliffe [7] appears to be in error. They itemise only 2 sources of latent heat:

170 Btu/hr for plant watering

2,550 Btu/hr for fresh air latent heat

Of the two latent heat loads enumerated the fresh air load is removed before being introduced into the plant section and therefore only the 170 Btu/hr forms the basis of the internal plant compartment latent heat addition. When 470 c.f.m. is flowing across the dehumidification coil this amounts to .54 grains of water vapor per pound of dry air.

Calculations are based on Equation 2 of section 2.1 :

$$Q_{LH} = \text{c.f.m.} \times .67 \times \Delta w$$

$$Q_{LH} = 170 \text{ Btu/hr}$$

$$\text{c.f.m.} = 470$$

$$\therefore \Delta w = .54 \text{ grains of water vapour per pound of dry air.}$$

At "Waite," the latent heat design load due to transpiration and evaporation including plant watering was estimated, on the basis of empirical data gathered from the operation of 2 existing growth cabinets, to be 40 ounces of water vapor per hour. Since the datawere based on short plants operating at 90°F dry bulb temperature and 50 per cent relative humidity within cabinets of approximately the same size as L.B.H. and this "Waite" design, the above figure includes an allowance for taller plants and lower relative humidities. Forty ounces of water vapor an hour is equivalent to a latent heat load of about 2700 Btu per hour. When this is substituted in equation 2, this load is equivalent (for 1,600 c.f.m.), to the dehumidification of 2.5 grains of water vapour per pound of dry air at "Waite." For the 470 c.f.m. associated with the dehumidification process in the L.B.H. system it is equivalent to a dehumidification of 8.6 grains of water vapour per pound of dry air. If the tabulation by Pescod, Read and Cunliffe [7] is correct they have ignored 8.6 - .54 (allowed for plant watering) or 8.0 grains of water vapour per pound of dry air in their design. This is a very serious omission since it is the dehumidification load that is the limiting factor determining the coil selection, not the sensible. The plant compartment latent "heat" addition reflects an equivalent sensible "heat" reduction. Actually an <u>internal</u> energy interchange takes place. However, this is no advantage. The critical design setting to achieve dehumidification occurs during the highest dry bulb and lowest humidity ratio setting and at this critical setting the dehumidification coil performs to remove more sensible heat than is necessary in the process of meeting the dehumidification requirements. The dehumidification setting is the first critical point that is analysed in the "Waite" design and is represented as condition No.1 in Appendices IV and V. Pescod, Read and Cunliffe [7] are also aware of the importance of this condition since they too indicate the highest dry bulb temperature and lowest humidity ratio in their design range as critical for dehumidification. They state:

> "The dehumidification of air at 113°F to maintain a dew point of 40°F was a bigger problem. Water vapor would be added continuously by incoming fresh air and by transpiration by the plants."

If, in the L.B.H. design, coil selection is based on the assumption that the internal latent heat load is only 170 Btu per hour rather than 2,700 Btu per hour, it is a mistake that cannot be readily corrected. If the L.B.H. cabinet's total (internal and fresh air) dehumidifying design load were equal to that of "Waite's" for the Alternative 1 scheme (see Step 20, section 3.4.2.g) 12 grains per lb. of dry air would have to be removed. For the Alternative 2 scheme, (see sections 3.1.3 and 3.1.4), the dehumidification requirement would be 24 grains per lb. of dry air. A coil occupying the restricted area of the L.B.H. existing dehumidification coil would be incapable of dehumidifying the total load. Consequently, a serious error exists in the L.B.H. design.

2.2.6 Cost comparison

The economics of a system involve many factors beyond first cost. Maintenance, serviceability, life expectancy, running costs, reliability, the quality of the system - all must be considered. The designer must constantly make decisions which may result in the cost of a particular component rising in order to gain an important advantage. Thus, the cost of the sensing element alone, the dew point temperature probe at "Waite," is about £100. This expense was considered warranted in order to measure the independent variable, humidity ratio, rather than the dependent variable, relative humidity.

The design profitably utilises the additional sensible cooling associated with the dehumidification processes in a simplified, inexpensive, automatic, change-over arrangement between day and night temperature settings. Simplicity is a prime factor in the "Waite" design and this is related to economy. The costs of a system must be reasonable and commensurate with the results obtained. However, it is not the prime consideration in the design of a complex laboratory device.

2.2.6.(a) <u>Comparison of cost of "Waite" system with estimate by the 1964 ASHRAE "Guide"</u>

The cost of the first model of this design is estimated to be approximately £5,800. The cost of the engineering investigation and the developmental work at "Waite" is £1,500. Thus, the cost of a second model would be in the neighbourhood of £5,800 less £1,500, or £4,300. In order to compare this figure with a 1964 ASHRAE estimate for a cabinet of this nature, pp. 376-7, [3] the cost of the structure must be subtracted. In this case, it is a simple thing to do since this structure is being fabricated separately and costs £1,400. Thus, less structure, the design is estimated to cost £2,900 or £A145 per square foot of cross-sectional area.

The 1964 ASHRAE "Guide" estimates that the cost for such a system is, (in the United States), less structure, \$500 to \$1,000 per square foot of cross-sectional area, or £A225 to £A450. Thus, it is quite clear, disregarding a 50 per cent mark-up in price for such laboratory facilities in Australia, that the design presented falls well below the lowest 1964 estimate made by the American Society for Heating Refrigerating and Air Conditioning Engineers and that essentially a unique design is presented here which costs very little more than a high quality temperature-only cabinet.

2.2.6.(b) Comparison of the "Waite" system with L.B.H. on a component basis

Let us examine the "Waite" phytotron with the C.S.I.R.O. L.B.H. phytotron on a component basis. This comparison is particularly valid since both systems handle approximately the same air quantity. The heat load from the lights of the mercury lamp system at "Waite" is larger than in the L.B.H. and the "Waite" system has an additional 14 per cent plant space. (Sect. 2.2.4).

The air side of the L.B.H. system requires the circuitous bypassing of return air between 2 separate refrigeration plants and includes a specially constructed air to air heat exchanger. Air from the dehumidifying plant passes through a heat exchanger, auxiliary cooling coil, dehumidification coil, other pass of same heat exchanger, and, finally, an air condenser. It then mixes with air leaving the second refrigeration plant cooling coil and is introduced to the cabinet after passing through a humidifier and a reheater.

The air side of the "Waite" system requires a preheater, a cooling coil, a reheater and a humidifier, an inexpensive preheat controller and a reheat and humidifier controller.

The L.B.H. refrigeration plant consists of:

2 compressors

3 condensers: 1 air, 2 water

3 cooling coils

2 restrictors

2 low pressure and 1 high pressure receivers

1 reducing valve for suction pressure control in bypass line

4 controllers: 2 for refrigeration, 1 for humidification, 1 for reheat

The "Waite" plant consists of:

1 compressor

1 condenser

1 cooling coil

(no suction pressure control valves and

no bypass required)

(no controllers on the refrigeration plant)

1 thermostatic expansion valve

3 controllers on the air cycle

The simplicity of the "Waite" design results in a considerable saving. (See Appendices VIII and IX for schematic comparison of the L.B.H. and "Waite" cycles. The upper diagram in Appendix IX is repeated from Fig. 11. [7]).

3. DESIGN

The broad applicability of this design is emphasized in this presentation. It is suitable to multi-unit as well as single unit installations, to variations in size, light intensity and range, to naturally as well as artificially lit cabinets. Paralleling this broad approach will be the specifications for a growth cabinet meeting the needs of the

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Specific component selection will be used demonstrating by example the reality and significance of the design solution presented here.

The design will be discussed in 3 parts:

1. Design Requirements
2. The Systems of Air Distribution
3. The System of Air Treatment, Refrigeration and Control

3.1. Design Requirements

The design must cope with the following conditions:

3.1.1 Sensible heat gains associated with temperature gradient (see definition of "temperature gradient in section 3.2.9)

At "Waite," it is equal to 5,200 Btu per hour for a temperature gradient of 3°F and a volume of 1,600 c.f.m.

3.1.2 Sensible heat gains not associated with temperature gradient

This includes loads due to outside air intake, duct heat gains, fan heat losses and miscellaneous losses. At "Waite," it is equal to 4,300 Btu per hour, corresponding to a temperature rise of 2.5°F when supplying 1,600 c.f.m.

3.1.3 Internal latent heat gains

At "Waite," this is estimated to be about 2,700 Btu per hour corresponding to a humidity ratio change of 2.5 grains of water vapor per pound of dry air when 1,600 c.f.m. is considered.

3.1.4 Latent heat gains due to outside air intake

At "Waite," this is estimated to be about 5,000 Btu per hour corresponding to a humidity ratio change of 4.6 grains of water vapor per pound of dry air when 1,600 c.f.m. is considered. To meet this requirement, this design presents two alternative schemes. (Sections 3.5 and 3.5.1).

3.1.5 Operating range

The phytotron must be capable of maintaining any one state condition within a wide designated range. At "Waite," the range is designated on a psychrometric chart. (Appendix No. I). These bounds are as follows: left border 55°F dry bulb temperature; bottom border 38 grains of moisture per pound of dry air humidity ratio between the dry bulb temperatures of 55°F and 74°F; lower right hand border 30 per cent relative humidity between the dry bulb temperatures of 74°F and 85°F; right hand border 85°F dry bulb temperature; upper border 85 per cent relative humidity between the dry bulb temperatures of 55°F and 85°F.

3.1.6 Automatic day-night temperature change-over requirement

The system is required to change over automatically between day-night conditions, simulating a daily temperature range. By definition, this change-over requirement was to be in dry bulb temperature only with the humidity ratio remaining constant.

3.1.7 Avoidance of problems associated with existing phytotrons

A further aim of this design is to avoid the following problems associated with existing phytotrons:

3.1.7.(a) Frosting on the direct expansion coil

See sections 2.2.5.(g); 3.4.1.(a); 3.4.2.(d); 3.4.2.(e).

3.1.7.(b) Plant "shake" or vibration

Plant "shake" or vibration affects growth.[13] The self-contained unit design must include isolators to separate the plant compartment from the effects of on-off action of controllers; the reciprocating compressor; and fan rotation. Maximum isolation is possible when the plant compartment is separated from the mechanical equipment, as is the case at "Waite."

3.1.7.(c) Climatic noise, plant ripple, or hunting

See sections 2.2.5.(a) to 2.2.5.(f) inclusive and 3.6.

3.1.7.(d) Unstable air flow patterns

Air will be introduced to the plant growth cabinet at a uniform temperature and velocity at each increment of cross-sectional area. (Sections 3.2.8; 3.2.9; 3.2.10; 3.2.11).

3.1.7.(e) Unstable elements in the air and refrigeration cycles

E.g: backpressure, suction, bypass and solenoid valves are not employed in the "Waite" system. (Sections 3.3.2.(e); 3.3.2.(f).

3.2 The System of Air Distribution

3.2.1 The aim of the design

In the artificially lit phytotron, the most critical part of the cabinet is that cross-sectional area which corresponds to the top of the leaves. It is here where the design temperature and humidity conditions must be uniformly maintained to close tolerance. This area is usually located at a fixed distance below the lights; at "Waite" it is 2 feet. As the plants grow, this fixed distance is kept constant by the lowering of a platform. The design aim is clearly expressed by R.N. Morse. [1]

> "..... the aim of the designer should be to ensure that each element in the same horizontal plane has the same gradient, a necessary condition for the micro-climate to be the same."

3.2.2 Air distribution in the self-contained phytotron

In the self-contained phytotron, space conditions dictate that the supply air be introduced through top or bottom located wall grilles after returning from the plant compartment via a hollow wall of the cabinet. This results in the air entering the cabinet in an uncontrollable fashion and assuming a pattern of air flow contrary to the design aim expressed above. Fig. 2 [10] repeated on the following page, which is based on tests run for naturally lit plant growth cabinets (these cabinets have a similar configuration to the artificially lit ones) indicates the uneven air velocities, particularly in any one cross-sectional area. The analysis of the results of Fig. 2 is aptly described [10]:

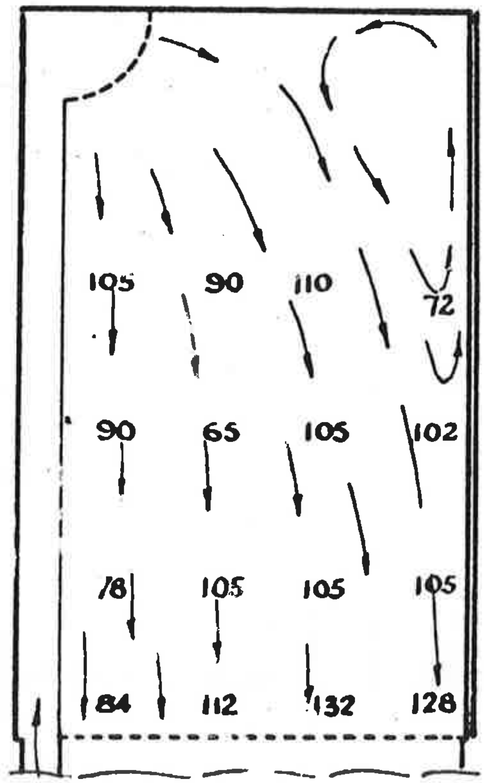


Fig. 2 Air flow pattern - Type B cabinet

"The rather large variation in air temperature in the plane 12 in. below the top of the cabinet ..... is attributed to the uneven air velocities which cause vortices and eddies in that zone. This is due to the difficulty in reversing the direction from vertical up in the back duct to vertical down in the cabinet with a simultaneous reduction in velocity."

These variations also apply to self-contained cabinets having bottom wall supply grilles.

3.2.3 Temperature difference and air velocity

The fundamental air conditioning relationship between flow and temperature difference points to the limitations of this method of air introduction.

$$Q = \text{c.f.m.} \times 1.08 \times (t_{design} - t_{inlet})$$

(See equation 1 section 2.1)

For any constant heat gain Q, $\Delta t \propto \frac{1}{\text{c.f.m.}}$

i.e. Δt is inversely proportional to the c.f.m. Since the cabinet has a constant cross-sectional area this relationship can be written as

$$\Delta t \propto \frac{1}{V}$$

i.e. for a constant heat gain Δt is inversely proportional to the velocity.

3.2.4 Effect of velocity variation

If the quantity of fluid passing a given horizontal cross-section per unit time is visualised as composed of an infinite number of stream tubes and one stream tube is moving at 72 f.p.m. and another stream tube passing through the same cross-section is moving at 110 f.p.m. as per Fig. 2,[10] the decrease in velocity is 72/110 or 0.65 of its maximum value.

Since change of $\Delta t \propto \frac{1}{v}$, substituting $\Delta t \propto \frac{1}{0.65}$, there is a 153 per cent temperature difference between the specific stream-lines considered in a single cross-sectional area.

This in itself would not result in too serious a temperature difference at the particular cross-section studied if it were not compounded with other factors.

3.2.5 Variations in length of path of stream tubes

The stream tubes travel very different paths. One stream tube moving directly down from the grille will travel approximately 1 foot to reach the critical area representing the top of the leaves; another will cross the entire cabinet in an arc before turning down, thus travelling 4 or 5 times the distance, frequently at a lower velocity than the first stream tube and thus having a longer period to pick up the cabinet heat gains.

3.2.6 The heat gains to the stream tubes are not uniform

The heat gains to the stream tubes are not constant, as assumed in developing the equations above, not only because of the differences in lengths of path they take, but also because of the nature of the paths. In one case, a stream tube taking the shortest, i.e. vertical path, down from the grille, will have no contact with heat transferred by conduction through the ceiling where the lights are located. In another case, the longest stream tube path will "wash" against the ceiling and a considerable temperature rise will result.

The consequences are obvious, and this method of air introduction is very poor. In the self-contained unit design there is very little that can be done to improve these conditions and achieve the design aim.

3.2.7 Evaluation of factors involved to determine whether air should be introduced from above or below the plants

Considerable discussion has taken place on the question of upward or downward air flow within the plant compartment. There appears to be no agreement and the empirical findings vary.

The following are the factors that have led me to conclude that upward flow of air is more desirable. This conclusion is dependent on the method of introducing the air into the cabinet. It is linked to the immutable fact that the lights occupy the entire "ceiling" of the phytotron. Where air is introduced to the bottom of a cabinet from a side located grille there is very little to be said in favour of the upward flow versus the top wall grille method of air introduction resulting in downward flow. Both are undesirable and result in air configurations contrary to the design aims. When such methods are compared empirically, Fig. 3 [1] or Fig. 32 p.369 [3] results have indicated an insignificant advantage to the downward flow method. Probably the plant trays, benches, pots and platform structure may add further turbulence to the upward air pattern. On the other hand, there would probably be a slight advantage to the upward flow pattern where some of the conducted heat from the light section would not affect the plant-occupied area.

There are other methods of introducing the air from below such as from an air diffuser centred on the bottom of the cabinet. This method is a little better but would have stream tubes moving out to the sides, up the wall, with induction down the centre and again would fail to achieve the design aim of even velocities and temperature gradients at all points in a horizontal cross-section.

3.2.8 The system of air distribution at "Waite"

In this system air is introduced into the cabinet by essentially a "displacement" method using the entire under-floor of the plant compartment as a plenum. Air is introduced through two series of perforated grids. Specifically, the lower grid is composed of ten perforated diffusers equipped with manual volume control. The second or upper grid consists of a flat plate evenly perforated. The grids fill the entire floor area.

In the "Waite" cabinet, air is removed from the plant compartment at very low velocities by two 60 x 10 inch return air registers located high above the top of the plants, along the entire length of the long walls of the cabinet. The return registers provide flexibility for volume control, being equipped with opposed blade dampers and both vertical and horizontal adjustment through formed deflection blades.

This method of air distribution provides for uniform velocities at each increment of cross-sectional area unobtainable by means of top or bottom wall-located supply grilles. To quote an authority referring to air conditioning of rooms, Severns and Fellows [14] state:

> "Perfect distribution would be achieved by a system in which the air is introduced at room temperature through one entire room surface, uniformly perforated, and removed through the opposite room surface perforated in a similar manner."

3.2.9 Temperature gradient redefined

The uniform "displacement" upward flow air pattern suggests a more precise definition of the term "temperature gradient." In this case, it is incorrect to consider the temperature gradient as the temperature difference between the air entering and leaving the plant growth compartment. The temperature gradient should be defined as the temperature difference between the air leaving the top of the leaves and the air entering the cabinet.

This system appreciably reduces the temperature gradient. The air leaving the top of the plants towards the return registers will rise in temperature due to the conducted heat through the glass partition at the roof of the plant growth cabinet, directly below the lights. This conducted heat will affect the capacity of the system but not the temperature gradient.

The temperature gradient in the "Waite" design is reduced approximately 25 per cent due to this method of air distribution.

The system of air distribution also eliminates one of the major sources of uneven temperatures within the plant-occupied area. There is of necessity an unevenly heated air stream, since the plane of stream tubes in closest contact with the underside of the high temperature glass of the light section will rise to a higher temperature than those air streams that are furthest away. The "displacement" upward flow method of distribution isolates this unevenly heated air from the plant areas below since it is moving at very low velocities, 30 f.p.m., without significant back pressure effects, away from the plant-occupied area.

3.2.10 Temperature gradient across the length of vertical stream tubes

This method of air distribution would result in an undesirable temperature difference equal to the gradient between the air entering and leaving the plant area were it not for the nature of the heat load. The major load is due to the radiant energy from lights. (Conducted heat from the lights, as explained above, does not appreciably affect the temperature gradient). This radiant energy does not become converted to heat until it strikes an object. Thus, even when the plants are crowded more than half the internal sensible heat gains are picked up at the floor of the plant compartment. Because of this circumstance, not only is there a negligible temperature gradient between elements in a horizontal plane but also along elements in a vertical plane.

3.2.11 Streamlining

To fully utilize the system of upward air flow, all flat obstructions in the path of the air stream will be eliminated. All structural members will, where necessary, be covered with air foil sections or fairings to prevent turbulence. The aim will be to modify all obstructions in the air stream so that the maximum angle with the vertical is 7-1/2 degrees. Furthermore, the base of all pots will have the shape of a nose cone. These pots will be supported in wire holders. All obstructions to the air stream will form an even pattern. Thus, should 11 pots be under test a 12th "dummy" pot would be added to permit a symmetrical configuration on the plant trays.

3.3 The System of Air Conditioning, Refrigeration and Control

3.3.1 Introduction

The Air Conditioning, Refrigeration and Control Systems are considered together in this section because of their close interdependence in this continuously operating system.

Within the refrigeration cycle, the inter-relation of the condenser, the direct expansion coil and the compressor performances further affects the design. Thus, this system consists of numerous components, and the variation of any one of them affects the performance characteristics.

The sum total of this interaction results in a system meeting all the design requirements. An important feature in this system is that the major components are tried and tested and can be purchased competitively. However, in order to present the system, specific reliable components with known engineering performance characteristics have been selected. Equivalent equipment of reliable manufacturers may also be approved.

3.3.2 Factors upon which system performance depends

This system is dependent on the maintenance of several conditions. These are as follows:-

3.3.2.(a) Constant volume air flow

The volume of air circulated shall be constant. (In the case of the "Waite" design, 1600 c.f.m.). An orifice plate and gauge shall indicate this volume in terms of c.f.m.

3.3.2.(b) Full volume air flow through evaporator

There shall be full volume flow through the direct expansion coil with no provision for bypass. (The fresh air intake quantity or CO_2 metered supply shall be kept constant; an orifice plate and gauge shall indicate the volume flow quantities).

3.3.2.(c) Constant condensing temperature

The condensing temperature shall be constant, (in "Waite" design at 105°F with 5°F subcooling).

3.3.2.(d) Preheat coil schedule of settings

The preheat coil shall function to maintain a constant temperature setting upstream of the direct expansion coil. The temperature setting shall be as per a schedule which has been established for a particular range of dew point and dry bulb operating conditions. (This schedule shall be posted next to the control system and be similar to Appendix II of this paper, selected for the equipment specified for the "Waite" phytotron conditions).

3.3.2.(e) Refrigerant flow control

A thermostatic expansion valve shall maintain a constant superheat setting. (In "Waite" design, 6°F). The thermostatic expansion valve specified must be engineered to operate for the variable range of evaporator temperatures. There are several manufacturers who are equipped to supply modern,reliable, modulating, standard engineered cross-charged valves for this system.

An examination of the Freon-12 saturation curve of Fig.14.5 [8] indicates that the pressure differential that must be supplied to the power element to change the needle valve from the open to closed position results in a relatively constant superheat for the range of evaporator temperatures, at "Waite" 27°F to 53°F, as per Appendix IV.

3.3.2.(f) Compressor operation

The compressor shall, during the use of the phytotron, operate continuously, (never in the "off" position), without solenoid valves, backpressure control valves, suction pressure control valves, hot gas bypass control valve or unloading devices. The only controls on the refrigeration cycle shall be the thermostatic expansion valve indicated in 3.3.2.(e) above, and conventional safety devices that shall not perform except during emergencies. These devices shall include high-low pressure cut-off and low voltage protection.

3.4 The Role of Individual Components In Satisfying The Design Requirements

3.4.1 The role of the preheater in the system

3.4.1.(a) Frost prevention

The preheat coil maintains sufficient load on the direct expansion coil to prevent the occurrence of frosting. For example, in the "Waite" design, the preheat schedule of Appendix II specifies a minimum temperature upstream of the direct expansion coil of 63°F. This critical condition to prevent frosting was determined on the basis of an analysis of conditions 2, 2a and 2b. (Appendices IV and V).

3.4.1.(b) Association with automatic day-night change-over

The preheat coil functions to permit automatic change-over between "day" and "night" conditions. Without any change to the preheat temperature setting of its controller, the preheat setting is established on the basis of the day setting. As such, the preheat schedule of Appendix II indicates specific temperatures equal or greater than the day setting for any particular design condition. The maximum daily temperature range, or the minimum night setting, is limited by the difference between the dry bulb temperature of the day setting and the temperature of the air leaving the direct expansion coil after it has picked up the cabinet, duct and miscellaneous system heat gains. These gains are only small at night since the phytotron lights are off. Consequently, a wide daily temperature range is possible without the need of a preheat controller having a different setting for the night condition of operation. Condition No. 9 (Appendix IV) is traced in its complete air path of operation for both conditions of operation. Appendix VI traces the day cycle, Appendix VII the night cycle and Appendix VIII both. It is to be noted that a maximum daily temperature range of $14^{\circ}F$ is possible for this specific condition. It is to be noted also that the operating requirements are such that during the low humidity ratio setting a higher daily range of temperature may be required. The shallower slope of the load ratio line coincides with this requirement and permits wider daily temperature ranges where they are needed.

3.4.1.(c) Limits size of humidifier

The preheat coil functions to limit the size of the humidifier. The extent of humidification required affects both the initial cost of the humidifier, the cost of the automatic modulating controller and the operating costs. In all cases where the preheat coil schedule calls for a temperature setting greater than the "day" operating condition, there is a reduction of the amount of humidification required. The critical point determining the size of the humidifier is condition No.3 in Appendices IV and V. It is to be noted that for an 85^{o}F dbt and an 80^{o}F dpt a considerably larger humidifier would have been required due to a steeper slope of the load ratio line, were the preheater to maintain a temperature of 85^{o} rather than 90^{o}. (See preheat coil temperature setting schedule, Appendix II).

3.4.2 The direct expansion coil

3.4.2.(a) Introduction

The determination of direct expansion coil performance is very complex, particularly when the air is both sensibly cooled and dehumidified. Both heat and mass transfer are involved. The condition of the air varies as it is being cooled. The temperature of the coil surface varies throughout the coil. Furthermore, there are individual variations between similar coils due to the variations in manufacturing technique, such as in the bonding of the fins to the tubes or in the methods of installation with relation to direction of air flow.

The design requirements of growth cabinets differ from those of conventional air conditioning applications.

The wide range of operating settings required in growth cabinets result in different entering and leaving air conditions at the direct expansion coil, making still more difficult the selection of coils and prediction of their performance. In this design, the refrigerant temperature is also permitted to vary widely with variation of the operating setting.

In the conventional application of reciprocating compressors selected to operate with on-off control, selection is not too critical. An over-designed compressor will simply remain for a longer period of time in its off condition. In the "Waite" design, the compressor is operating continuously and with varying capacity dependent on the varying suction pressure which, in turn, is dependent on the varying coil capacity. Every design setting yields a new relationship, a new capacity and a new condition of compatibility between the coil and compressor performance. The designer must select the most suitable coil from a large choice of face areas, fins per inch, diameter of tubing and depth of coil. In this design, simplicity of the final system was a foremost consideration and the goal was to use a single coil to meet the "Waite" requirements at every setting in the design range.

A new system of selection that breaks down the multitude of variations in a manner which leads to a solution of the coil selection problem is presented here.

3.4.2.(b) Methods of selection of direct expansion coils

There are several methods of determining the performance of direct expansion coils.

One system, still in use, was first presented by W.H. Carrier [15] in 1936. This method takes into account the fact that some particles in an air stream come in direct contact with the coil. These particles are assumed to reach the same temperature and moisture content as the film at the coil surface. The remaining air that did not come in physical contact with the coil is assumed to pass through the coil unchanged and is classed as "bypassed air." Coil performance has been described using this bypass method.

Goodman [16] derives from the temperature difference between coil surface and the refrigerant an enthalpy potential. It is an approximate method and ignores the effect of dehumidification on fin efficiency.

A recent method [17] presents an approach which utilizes data derived from sensible heat transfer tests. It is a difficult system to apply. The authors are planning to simplify their solution.

McElgin and Wiley [18] present a "three-line method" utilizing enthalpy potential. This method ignores the resistance of the condensate film between the interface and the outside surface and involves trial and error.

Brown [19] includes the effect of fin efficiency. This method involves complex trial and error solutions.

Many coil manufacturers present methods based on coil performance tests for particular coils under various operating conditions. These basic coil data are extended to other coil configurations and operating conditions on the basis of relationships established from further test data. This is the case in the McQuay [22] direct expansion coil selection described in this paper. The manufacturer used as a basic coil one that had a 1/2" tube, 2 rows deep and 12 fins per inch operating with an airstream face area velocity of 500 f.p.m. The manufacturer then provided additional data to permit corrections. The particular coil that was selected for "Waite" is a one-half inch o.d. tube, 2 rows deep, 6 fins per inch, operating with an air stream velocity across the face area of the coil of 229 f.p.m. (See section 3.4.2.(g), where a step-by-step presentation of the method is given for condition 1 Appendices IV and V).

3.4.2.(c) The role of the direct expansion coil in the system

The most important single component in this system is the direct expansion coil. The successful operation of the system depends on this coil meeting all the sensible cooling and dehumidifying requirements for all of the design operating conditions. At "Waite," a single 1/2" o.d., 6 fins per inch, 2 rows deep, 7 square feet face area coil meets these requirements.

For purposes of analysis these heat gains are presented in the following subdivisions:-

1. the phytotron cabinet internal sensible heat gains associated with the plant temperature gradient.

2. the phytotron cabinet internal latent heat gains.

3. all other sensible heat gains

   (a) the phytotron cabinet internal sensible heat gains not associated with plant temperature gradient. (Section 3.2.9).

(b) sensible heat gains associated with fan losses and the duct system.

(c) sensible heat gains associated with the intake of outside air.

4. latent heat gains associated with the intake of outside air. Two alternative schemes are presented.

In addition, the cooling coil functions in the system of daily temperature range changeover to sensibly cool the air for the "day" setting to a point which also satisfies the sensible cooling requirements of the "night" setting.

3.4.2.(d) <u>Selection of direct expansion coil</u>

The direct expansion coils (along with complete engineering performance data) of several reputable manufacturers represented in Australia were studied. For the purpose of establishing a specific example, one of these manufacturer's standard coils was analysed and selected for this "Waite" system according to the manufacturer's data. (Section 3.4.2.(b)).

In line with the specifications listed in section 3.3.2, a coil selection was made for a critical design condition. This condition was selected from the infinite number of possible operating state points within the bounds of the design operating range as indicated in Appendix I. This critical point is characterised by the fact that it not only will economically meet the design requirements for its particular condition and serve as a basis for compressor selection, but will result in this single coil operating in the refrigeration and air conditioning cycles to satisfy all other design operating state points. This point is labelled as condition number 1 in Appendices IV and V. It is the condition under which both dehumidification and the load ratio line slope are minimal. At "Waite," it is 74°F dbt and 56°F wbt.

The variables in coil selection are many. After determining the one critical condition that governed coil selection for the system, the next step was to determine the coil specifications. A qualitative and then a quantitative comparison of the numerous variables resulted in a final coil selection. This is a 1/2" o.d. 6-fin per inch, 2-row deep, 7 square foot face area coil, which has a capacity of 3 tons of refrigeration for this operating condition. With point number 1 established and its evaporator temperature calculated by the method outlined in section 2.2.5.(g), an allowance was made for the pressure drop between the evaporator and the suction to the compressor. For this suction pressure, a compressor was selected and from the compressor performance data of the manufacturer (and for the condition of constant condensing pressure) a capacity versus compressor suction pressure curve was drawn as per Figure 12.5. [8] (Appendix III). The corresponding evaporator temperatures were also plotted along the abscissa of this curve to simplify the analysis of the other critical points in the system. For the purpose of gaining flexibility in compressor performance an open type, variable speed, belt-driven, Freon 12 compressor was selected. (Section 3.4.2.(f)).

The next points analysed were 2, 2a and 2b

(Appendices IV and V). These points represent operating conditions under which the minimum refrigerant temperature of the system is determined. For this particular cycle, a temperature of 27°F within the direct expansion coil is a safe minimum for frost prevention. Thus, Point 2b determines the minimum dry bulb entry condition of 63°F (for "Waite") upstream of the direct expansion coil. The preheat coil schedule incorporates this information. (See top row, 3rd column in Appendix II). Frosting must be completely avoided in this design. The compressor is continuously operating while the phytotron is in use and there is no "off" period or any other means provided to permit defrosting of coils. (Section 2.2.5.(g)).

The analysis of points 2, 2a and 2b and all other points representing the air state condition downstream of the preheat coil other than the original point 1 requires a trial and error method of solution. Though the entering state point is known, the capacity of the coil in equilibrium with the compressor that was selected for point 1 is not known. Several trial capacities must be assumed and the method outlined in Section 2.2.5.(g) followed. However, only when the refrigerant temperature is compatible with the resulting suction pressure of the compressor will the selection be valid. This operation can be reduced to only 2 trials. The plotting of these evaporator temperatures on the capacity-evaporator temperature curve of Appendix III and connecting the points derived from the 2 trial capacities with a straight line that crosses the compressor capacity-pressure curve gives the correct refrigerant temperature and capacity at the point of intersection. This method is sufficiently accurate for the purposes of this design and essentially represents the simultaneous solution of the evaporator and compressor performance equations.

Eleven points were analysed by these methods and the reason for their selection listed in the heading of their respective columns. (Appendix IV). In addition, all 11 points were plotted on a psychrometric chart indicating the coil entry and leaving condition, the load ratio line, the apparatus dew point and the sensible heat ratio. (Appendix V). This analysis determined the preheat, reheat and humidifying requirements of the system. In conjunction with the known cooling loads, it also indicated the maximum daily temperature range that can be used for a particular "day" operating condition.

3.4.2.(e) Relationships which guided selection procedure

The following are some of the relationships which guided the selection procedure. An examination of Appendix V indicates that with dry bulb temperature constant, on increasing the humidity ratio, the slopes of the corresponding load ratio lines will increase. For example, the slope of condition 5 is greater than condition 6 which is greater than condition 7.

Appendix V also indicates that with humidity ratio constant, on decreasing the dry bulb temperature the slopes of the corresponding load ratio lines will increase. For example, the slope of condition 2b is greater than the slope of condition 2a which is greater than the slope of condition 2 which is greater than the slope of condition 1.

Since the compressor is a volume oriented machine, the higher the suction pressure, the greater the capacity. The evaporator performs in the opposite direction. The higher the evaporator pressure (temperature) the lower the capacity.

In selecting the coil, the factors that will depress the minimum refrigerant temperature for frost prevention must be considered. The shallower the coil (the number of rows deep) and the less fins per inch, the lower the minimum temperature for frost prevention. These relationships are clearly indicated in the manufacturer's tables [20] repeated as Fig. 3 on the following page. Note that a coil with 8 fins per inch and 4 rows deep has a minimum temperature of 26°F for a coil face velocity of 300 feet per minute and an entering wet bulb of 65°F, whereas a coil of 14 fins per inch and 8 rows deep for the same face velocity and entering wet bulb has a minimum temperature of 32°F.

As indicated, it is important to select a shallower coil with fewer fins per inch in order to depress the minimum refrigerant temperature for frost prevention. The question now arises: Will this shallower coil be a serious limitation in meeting the critical dehumidification requirement? The shallower the coil, the lower the minimum limits of the refrigerant temperature. The lower the refrigerant temperature, the greater the dehumidification. Thus, the "Waite" system with its varying refrigerant temperature is particularly suited to meet the dehumidification requirements since, during the higher humidity ratio settings, a higher refrigerant temperature limits the dehumidification process. During the lower humidity ratio setting, the system permits the refrigerant temperature to drop, thus achieving dehumidification

RECOMMENDED MINIMUM REFRIGERANT TEMPERATURES FOR FROST PREVENTION (F)

Direct Expansion Coils – 8 Fins/Inch

EWB (F)	Coil Face Velocity (fpm)														
	300			400			500			600			700		
	Number of Rows														
	4	6	8	4	6	8	4	6	8	4	6	8	4	6	8
65	26	30	30	26	28	29	25	25	26	25	25	25	25	25	25
70	26	29	29	25	27	27	25	25	25	25	25	25	25	25	25
75	25	27	28	25	25	25	25	25	25	25	25	25	25	25	25
80	25	25	26	25	25	25	25	25	25	25	25	25	25	25	25

Direct Expansion Coils – 14 Fins/Inch

EWB (F)	Coil Face Velocity (fpm)														
	300			400			500			600			700		
	Number of Rows														
	4	6	8	4	6	8	4	6	8	4	6	8	4	6	8
65	27	31	32	26	29	32	25	27	31	25	25	30	25	25	28
70	27	30	32	25	28	31	25	26	30	25	25	28	25	25	26
75	26	29	32	25	26	30	25	25	27	25	25	26	25	25	25
80	25	28	31	25	25	29	25	25	26	25	25	25	25	25	25

NOTES: 1. Coil selection below loadings shown in the rating tables may result in unsatisfactory oil return. Selections at loadings above those shown may result in excessive pressure drops.

2. Selections at refrigerant temperatures below 32 F may result in frost formation. See tables above for minimum recommended refrigerant temperatures at various coil loadings. This warning also applies to any selection where continuous operation at partial load is required. Under such conditions, the need for a back pressure regulator for compressor capacity control should be investigated.

Fig. 3

where it is most critical. A shallow coil is accompanied by reduced heat and mass transfer and might fail to dehumidify sufficiently during the lowest critical humidity ratio settings in spite of the lower refrigerant temperature at which frosting would occur. Too deep a coil would raise the minimum refrigerant temperature permissible to prevent frosting and again would fail to dehumidify sufficiently to meet the design requirements. Thus a compromise is required.

Some of these variations are illustrated in Figure 4 repeated on the following page from J.H. Carpenter Figure 9.[21] Air side coil performance curves for three different refrigerant temperatures are presented here. The numbered points represent the leaving condition and the number of rows deep of coil used to obtain this condition. All the coils have the same entering condition, D, and the same coil velocity. Note that a 2-row coil with a 40°F refrigerant temperature will dehumidify more with less associated sensible cooling and less tonnage than a 4-row coil at a 50°F refrigerant temperature.

Figure 4 illustrates another point. The load ratio lines drawn in Appendix V are straight lines connecting the points representing the entering and leaving conditions of the coil. They fail to depict the actual curved process path through the coil. The approximate nature of the calculations is further illustrated by the failure of the load ratio lines in the case of conditions 3, 4 and 8 to intersect the saturation curve. The results presented here are sufficiently reliable to form a basis for coil and compressor selection with the provision for minor adjustment in the

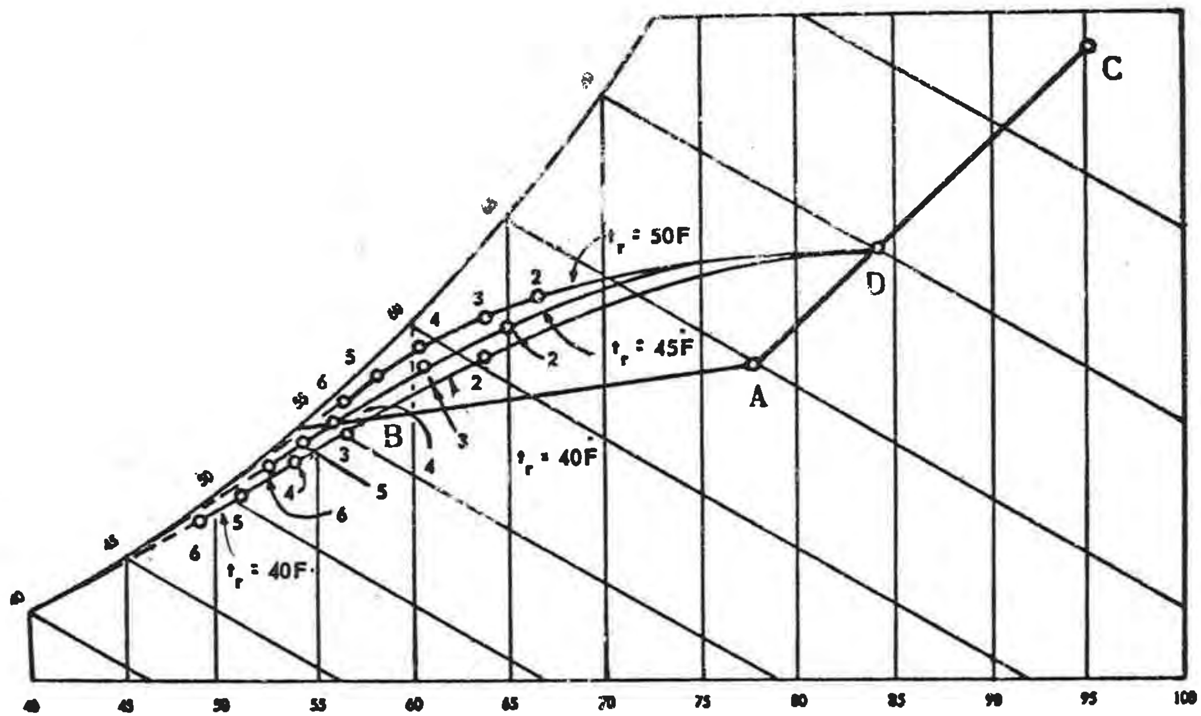


Fig. 4 Air side coil performance

system via compressor speed and/or air volume.

3.4.2.(f) Variable speed compressor

By using a variable speed compressor within the permissible limits of proper performance, the range of the phytotron is increased and the system can be operated more economically for certain design settings. To design for different speeds it is necessary to add the parallel compressor performance curves for the range of permissible minimum to maximum operating speeds to the compressor curve of Appendix III. In establishing the limits of the range the rpm must not be reduced to a point where oil return is unsatisfactory. Manufacturers of the coil should be consulted to insure that the load does not fall below a prescribed minimum. It would be possible to select the minimum operable rpm for a particular set point that would meet the design setting, including the provision for day-night automatic change-over. Appendices IV and V would then be reconstructed for a range of compressor speeds and by selecting the load ratio line that equals or just exceeds the load for the particular setting, the system can be operated most efficiently. Since changes in rpm involve engineering analysis they should be carried out only under supervision and instruction of the designer.

3.4.2.(g) Method used to determine coil performance data of Appendices IV and V

Step 1. Based on a preliminary study, a tonnage for the system was assumed for condition number 1: 3 tons or 36,000 Btu per hour.

Step 2. As per the heading of column 1 of Appendix IV the entering dry bulb temperature of that operating setting which represents "Condition of Minimum Dehumidification ----," $74^{o}F$, is entered here.

Step 3. Similarly the entering wet bulb temperature for state point of step 2, $56^{o}F$, is entered here.

Step 4. The corresponding enthalpy (ignoring enthalpy deviation) for the condition of step 3, 23.84 Btu per lb_m of dry air, is entered here. (See A.S.H.V.E. Guide 1949 or McQuay Engineering Manual [22] Sheet EDSH 195-3).

Step 5. Coil cooling capacity of system per square foot of coil face area: Since the face area of the coil selected based on preliminary studies is 7 square feet, the value of step 1 divided by 7 is entered here:

$$\frac{36000 \text{ Btu per hr.}}{7 \text{ sq. ft.}} = 5143 \text{ Btu per hr. per sq. ft. of face area}$$

Step 6. The entering depression is defined as the temperature in degrees between the dry and wet bulb entering condition. In this case it is the difference between step 2, 74^{o} dbt, and step 3, 56^{o} wbt, or $18^{o}F$.

Step 7. As pointed out in section 3.4.2.(b), the manufacturer has established relationships between a variety of coils that he fabricates and a particular basic coil used to established empirical performance data. Thus in this step a t.c.f. or total correction factor is applied to convert the selected coil rating to the condition of the basic empirical coil data. The basic coil has an average face area velocity of 500 feet per minute, is 2 rows deep and has 12 fins per inch. The coil selected has an average face area velocity of 229 feet per minute, is 2 rows deep and has 6 fins per inch. In this step, the latter, the selected coil condition, is corrected to the former, the basic coil condition. Both coils have $\frac{1}{2}$" o.d. and the correction data are presented in the form of several graphs and a table in the McQuay Engineering Manual, [22] Sheet EPS No. 2251. The velocity and row correction factor from this source is 0.73 and the fin correction factor is 0.765. The product of these 2 factors is the total correction factor of 0.558.

Step 8. The selected coil rating of step 5, 5143 Btu per hour per square foot of face area is divided by the correction factor of step 7 to give the coil rating for the basic test coil used by the manufacturer, thus

$$\frac{5143}{.558} = 9200 \text{ Btu per hour per square foot of face area per basic coil.}$$

Step 9. From a graph in the McQuay Engineering Manual, [22] Sheet EPS No.2251, plotting basic coil rating, step 8 above, versus entering enthalpy, step 4, the refrigerant temperature of the selected coil is obtained, 30.7°F.

Step 10. The change in enthalpy is derived from equation 3 of this paper (section 2.1):

$$Q = \text{cfm} \times 4.45 \times \Delta h_a$$

$$36000 = 1600 \times 4.45 \times \Delta h_a$$

$$\Delta h_a = 5.00$$

Step 11. The leaving enthalpy is the difference between the entering enthalpy of step 6, 23.84 Btu per lb. of dry air, and the difference in enthalpy of step 10, 5 Btu per lb. of dry air:

$$lh_a = eh_a - \Delta h_a$$

$$lh_a = 23.84 - 5.00 = 18.84 \text{ Btu per lb. of dry air.}$$

Step 12. From the table used in step 4 we can find the corresponding leaving wet bulb temperature for the leaving enthalpy of step 11:

$$l\ wbt = 47.3^{\circ}F$$

Step 13. The leaving depression ratio, l dep r, is obtained from the McQuay Engineering Manual,[22] No. 2252 entitled "Wet Bulb Depression Ratio for McQuay $\frac{1}{2}$" Tube Surface." By entering the abscissa of the graph (for 6 fins per inch) at the face velocity of the selected coil, 228 feet per minute (standard air), and moving vertically up to the curve for a 2-row deep coil, the wet bulb depression ratio can be obtained from the ordinate, 0.492.

Step 14. The leaving depression is equal to the product of the wet bulb depression ratio, step 13, and the entering depression, step 6.

$$l\ dep = dep\ r \times e\ dep$$

$$l\ dep = 0.492 \times 18 = 8.86^{\circ}F$$

Step 15. The leaving dry bulb temperature, ldbt, is equal to the leaving wet bulb temperature, l wbt, step 12, plus the leaving depression, ldep, step 14.

$$l\ dbt = l\ wbt + l\ dep$$

$$l\ dbt = 47.3 + 8.86 = 56.16^{\circ}F$$

Step 16. The sensible heat ratio, the ratio of sensible to total heat removed by the coil, can be obtained from the Carrier Psychrometric Chart by extension of a line parallel to the load ratio line drawn through the 80°F dbt and 50% relative humidity state point to the upper right hand margin of the chart. This reads 0.87 (See upper right hand margin Appendix V).

Step 17. The <u>Apparatus dew point</u>, ADP, can be obtained from a Psychrometric Chart by extending the coil load ratio line until it intersects the saturated air line of the chart. This reads 31.8°F (Appendix V).

Steps 18 & 19. The <u>entering and leaving humidity ratios</u>, e w and l w, are obtained from the Psychrometric Chart by entering horizontally towards the right hand margin from the entering and leaving state point conditions respectively. This reads 38 and 34.3 grains of water vapour per pound of dry air.

Step 20. The <u>change in humidity ratio</u>, Δ w, is the difference between steps 18 and 19:

$$\Delta w = e w - l w$$

$$\Delta w = 38 - 34.3 = 3.7 \text{ grains of water vapour per lb. of dry air.}$$

Step 21. The <u>change in dry bulb temperature</u>, Δ t, is the difference between step 2, e dbt, and step 15, l dbt:

$$\Delta t = e\ dbt - l\ dbt$$

$$\Delta t = 74 - 56.16 = 17.8°F.$$

The method described above represents entering condition 1. All other entering conditions followed the same procedure. (However, see remarks in section 3.4.2.(d) regarding trial and error solution that ran concurrently with this procedure for these points).

3.4.3 The role of the reheat coil in the system

3.4.3.(a) Control agent maintaining dry bulb temperature settings

In this system the reheat coil is the exclusive control agent by means of which the desired dry bulb temperature operating condition is maintained. As previously indicated, the direct expansion coil operates freely without controllers for all design state points and has a cooling capacity equal to or greater than the summation of all the sensible heat gains to the system. Thus, if the cooling capacity is equal to the sensible heat gains the reheat coil will be in its off position and where the cooling capacity is greater the reheat coil controller will modulate to maintain the design setting dry bulb temperature.

3.4.3.(b) Reheat during day-night settings

During night temperature settings the extent of reheat is affected by the summation of two opposing variations from the day operating condition. Reheat is reduced because of the lower night temperature setting and at the same time it is increased because of the reduction of heat gains from lights, transmission and outside air.

3.4.4 The role of the humidifier in the system

3.4.4.(a) Exclusive control agent maintaining humidity ratio setting

In this system the humidifier is the exclusive control agent by means of which the desired humidity ratio is maintained. As previously indicated, the expansion coil operates freely without controllers for all design state points and has a dehumidifying capacity equal to or greater than the summation of all the latent heat gains to the system. Thus, if the dehumidifying capacity is equal to the latent heat gains, the humidifier would be in its off position and when the dehumidifying capacity is greater the humidifier controller will modulate to maintain the design setting dew point temperature or humidity ratio.

3.4.4.(b) Humidity ratio does not vary during automatic day-night change-over

In this design, the day-night change-over is defined as occurring under constant humidity ratio conditions and consequently the day and night dew point temperatures are equal for any particular operating condition. Thus, during the automatic day-night change-over, no control action is required to change the dew point setting of the humidifier.

3.4.4.(c) Aim to keep humidity ratio changes independent of dry bulb temperature

It would be most desirable to accomplish the addition of moisture with a minimal change in dry bulb temperature so that the humidity controls would behave independently. A steam jet humidifier introducing low pressure steam by way of a modulating steam valve would best serve this purpose since the process is almost isothermal. Unfortunately, there is no steam available at the Waite Institute and the untreated condition of Adelaide water poses additional problems to the introduction of steam. Therefore, the method of humidification selected is that of a modulating pan humidifier, since quick response is not an essential feature and once the system reaches equilibrium the load changes will be slow and small. Thus, accurate control of dew point temperature is possible. In order to minimize the increase in dry bulb temperature accompanying the humidifying process, the pan humidifier should have an efficiency factor approaching unity. This represents the ratio of total heat delivered to that which is converted to latent heat. The efficiency factor varies with the design of the pan since it is a function of the relative areas of the metal surface of the pan (which delivers sensible heat) and the surface of the water in contact with the air (which delivers latent heat). Consequently, a relatively large surface area pan humidifier will be specified.

Eliminators to prevent the carry over of water will be installed downstream of the humidifier. Aluminium ducts pitching in the direction of a drain will be used to minimize corrosion.

3.5 Designs To Cope With Phytotron Outside Air Latent Heat Load

An important design approach was to analyze the cooling and dehumidifying loads without considering the effect of the 4 per cent outside air supply required to meet the CO_2 needs of the plants. The additional effect of the various critical outside air loads in system performance were then studied. In most cases they did not prevent the system from maintaining the design conditions.

In the case of the outside air latent heat load the critical condition is based on a combination of a very humid hot day, a critical operating setting within the cabinet during which minimum dehumidification occurs (see Condition 1 Appendices IV and V) and a plant filled cabinet at full growth.

The system is able to meet the design requirements for this condition provided the outside air humidity ratio does not exceed 65 grains of moisture per pound of dry air.

To satisfy the design requirements, two alternative solutions are offered here.

3.5.1 Alternative 1

The phytotron would operate satisfactorily on the basis of a 4 per cent outside air supply provided that a pre-scheduled outside air humidity ratio (based on the cabinet dew point temperature operating setting) is not exceeded. For the rare simultaneous occurrence of conditions where the outside air latent heat load would be excessive the system would operate on a 100 per cent recirculating basis. During this period a CO_2 metered supply would substitute for the outside air requirement. At "Waite," this alternative was accepted, since the nature of the experiments favour the higher specific humidity settings for cabinet operation and 100 per cent recirculation would rarely be necessary. This solution is used in existing phytotrons.[1]

3.5.2 Alternative 2

This scheme takes advantage of the fact that the coil load ratio line has a very flat slope during the critical setting. As a consequence the supply air leaving the direct expansion coil has an excess sensible cooling capacity which can be used advantageously (before it enters the phytotron cabinet) to pre-dehumidify the 4 per cent outside air supply. Thus a closed air to air cross-flow heat exchanger would be introduced. To construct this heat exchanger, the fresh air supply should be conducted through a closed system of passages or tubes through the section of the air conditioning cabinet immediately downstream of the direct expansion coil. This, in effect, would create the heat exchanger. A pre-dehumidified fresh air supply would then flow directly to the intake of the air conditioning unit. The pressure drop of the fresh air through the pre-dehumidifier must not be too great. If for economy or other reasons it is large enough to prevent the flow of 4 per cent outside air, at "Waite," 64 c.f.m., then a small booster fan should be included in the system. Of course, the intake manifold to the fresh air pre-dehumidifier, located at a low point, would pitch to drain facilities and all aluminium construction would be specified. An air filter would be located at the intake opening.

It is to be noted that the specifications for this heat exchanger not only must include the requirements for the cabinet critical condition where minimum dehumidification occurs, but also, due to the system preheat temperature settings, the heat exchanger should be specified not to exceed for the lowest dbt and dpt operating settings a specified maximum sensible rise in temperature. This maximum would be established on the basis that the total sensible cooling load in the cabinet, plus miscellaneous duct and fan heat gains, must be satisfied for the dry bulb settings after the heat gain from the pre-dehumidifier. This is important if the pre-dehumidifier is to operate freely without adjustment for different settings. It is to be noted that at "Waite" during the critical condition for minimum dehumidification (point 1 of Appendices IV and V) the air entry temperature to the pre-dehumidifier is 56.2°F.

For condition 2b it would be 48.4°F. If there is no adjustment made to air quantities of the supply and fresh air circuits with varying cabinet operating set points, the heat transfer to the 1600 c.f.m. circuit would increase, reducing the sensible cooling capacity of the supply air. As a result, the system might fail to satisfy the sensible cooling design requirement for condition 2b. The use of alternative 2 calls for careful specification and empirical testing.

Neither solution (Alternatives 1 and 2) require the addition of a refrigeration plant. Alternative 2 eliminates change-over to CO_2 metering and requires no controls or variation of settings. Alternative 2 broadens the scope of application of this design to phytotrons that must maintain a minimum fresh air supply.

3.6 The Control System

3.6.1 Introduction

It has been difficult and expensive to control both humidity and temperature in existing phytotrons and the consensus of opinion has been to limit the use of humidity control. J.J. Kowalczewski [23] at the 1962 C.S.I.R.O. Symposium on Engineering Aspects of Environment Control for Plant Growth pointed out the problem:

> "I would also like to stress that humidity control at moisture contents above that of the air surrounding the cabinet is relatively simple and inexpensive as it requires humidification only. This can be achieved by addition of, for example,

a simple electrode steam humidifier, a sensing device and a relay. Dehumidification on the other hand requires additional moisture removing equipment, which is bulky and costly, and a substantially more complicated control system. This could possibly double the cost of the cabinet and could necessitate an increase of the cabinet dimensions.

"It is therefore advisable to conduct humidity effect investigations in a few expensive cabinets, preferably under artificial lighting, provided with full humidity control and, should this be required, to provide means of increasing humidity only on a wider basis."

Also see comments by R.N. Morse [1] in section "Control of Temperature and Humidity" and "Summary of Conclusions."

The "Waite" design eliminates the objections raised.

3.6.2 General

The preheater, the reheater and the humidifier are the major control agents of the modulating automatic control system. As pointed out in sections 3.4.3.(a) and 3.4.4.(a), the system is engineered to require reheat to maintain sensible conditions and humidification to maintain latent conditions. The controls have been simplified. They are independent of the sensible cooling and dehumidification processes of the refrigeration cycle. They complement each other to maintain the fixed set point condition.

The automatic temperature and humidity control system will be interlocked with the supply fan in a manner which will prevent the preheater, the reheater, the humidifier and the light section from operating when the supply fan is off.

The controls will have thermal system and electronic resistance bulb actuation. At "Waite" the control function is pneumatic since compressed air at 60 p.s.i.g. will be available 24 hours a day. The control rig will contain a pressure reducing station including filters and driers to provide clean air at 20 p.s.i.g. to the control system.

3.6.3 Preheat coil controller

The preheat coil located in the air conditioning cabinet will be controlled with a duct type temperature sensing primary element located downstream of the preheat coil.

On a drop in dry bulb temperature a modulating controller will gradually increase the heat output of the preheat coil to maintain a fixed dry bulb temperature. The fixed design temperature represents one of a range of design temperatures. A conveniently adjustable knob will be provided on the control panel for selecting the temperature for a specific operating condition. At "Waite," the preheat temperature setting will be within the range of $50^{\circ}F$ to $100^{\circ}F$ inclusive. It will be selected according to a schedule determined by the operating dry bulb and dew point temperatures of the growth cabinet and the engineering performance data of the direct expansion cooling surface in the system. The preheat coil temperature setting schedule will be framed and fixed to the control panel. (See Appendix II).

At "Waite," preheater electrical resistance elements are to be controlled by a filled thermal system type of temperature indicator-controller including capillary compensated actuation. The control is to be proportioning and will act on a motorized variable transformer of a specified rating, arranged for 3-phase operation to proportion the input voltage to the heating element.

3.6.4 Reheat coil controller

The reheat coil located in the air conditioning cabinet will be controlled with a temperature sensing element located as required within the plant compartment.

On a drop in dry bulb temperature a modulating controller will gradually increase the heat output of the reheat coil to maintain a fixed dry bulb temperature. The fixed design temperature represents one of a range of design temperatures. A conveniently adjustable knob will be provided on the control panel for selecting the temperature for a specific operating condition. At "Waite," the reheat temperature setting will be within the range of $50^{o}F$ to $100^{o}F$ inclusive. It will represent the operating dry bulb temperature at which the plants are to be maintained.

At "Waite," the sensing element will be suitable for mounting on a movable platform and fitted with a flexible connection to permit the raising and lowering of the platform without fracture. In addition, the sensing element will be fastened to a platform bracket so that it can easily be adjusted in height to the growth and size of the plants. The sensing element will be shielded to prevent error that may be introduced by radiation from the light section above. The resistance thermometer recorder-controller will have an accuracy of ± 1% or better of scale, a sensitivity of 0.25% of scale and be provided with upscale drive of pen on open circuit of sensor. The chart of the recorder will be electrically driven and will have a 24 hour revolution. The chart will have a 12 inch diameter and its scale will be from 0-100°F with a tolerance of ± 1/2°F. The control will be a two mode pneumatic system having adjustable proportioning and reset action. The controlled output of the instrument will be used to operate a motorized variable transformer of a specified rating which is arranged for 3-phase operation to proportion the input voltage to the electrical resistance heating elements.

3.6.5 Pan humidifier controller

The pan humidifier located in the air duct system outside of the phytotron cabinet, will be controlled with a dew point temperature primary sensing element located as required within the plant compartment. On a drop in dew point temperature a modulating controller will gradually increase the water vapour output of the pan humidifier to maintain a fixed dew point temperature setting. The fixed design dew point represents one of a range of design temperatures. A conveniently adjustable knob will be provided on the control panel for selecting the dew point temperature for a particular operating condition. The dew point setting at "Waite" will be within the range of 41°F to 81°F. It will represent the operating dew point temperature setting (the humidity ratio) at which the plants are to be maintained.

At "Waite" the primary humidifier sensing element will be suitable for mounting in the same manner as the reheater sensing element. (See final paragraph of section 3.6.4). The housing will be arranged to prevent condensation on the element or terminals, which may occur during a change from one set point to another.

The recorder-controller will have an accuracy of $\pm$ 1-1/2°F dew point temperature; it will be similar to that employed for the reheat coil control. The pan humidifier electrical resistance elements will be controlled so as to maintain the dew point temperature within a tolerance of $\pm$ 1/2°F. The controlled output of the instrument will be used to operate a motorized variable transformer of a specified rating which is arranged for 3-phase operation to proportion the input voltage to the electrical resistance heating elements.

(See Appendix VIII for schematic of control system).

4. PSYCHROMETRIC ANALYSIS OF THE AIR CYCLE FOR A SPECIFIC DAY-NIGHT OPERATING SETTING.

The system performance will be examined by tracing on a psychrometric chart the air cycle for a specific operating point at "Waite." This analysis will include the day and night settings. The outside air latent load will be analysed for both alternative 1 and alternative 2 design solutions.

For the purpose of this analysis the following conditions will be assumed to exist. (Though all points shown in the cycle exist, some of the path changes are so small that they have been exaggerated in order to be identifiable on the psychrometric chart).

Design dry bulb temperature		
day setting	$74^{\circ}F$	
night setting	$62^{\circ}F$	
Design dew point temperature	$53^{\circ}F$	
Design humidity ratio	60 grains wv/lb.d.a.	
	DAY	NIGHT
Fresh air intake dry bulb temperature	$80^{\circ}F$ $94.2^{\circ}F$	$59.9^{\circ}F$ for Alternate 1 $59.9^{\circ}F$ for Alternate 2
Fresh air intake wet bulb temperature	$72^{\circ}F$ $78.4^{\circ}F$	$58.2^{\circ}F$ for Alternate 1 $58.2^{\circ}F$ for Alternate 2

4% Outside Air Intake (64 c.f.m.)

The points indicated on Appendices VI and VII are identified as follows:- (Note correspondence with points on air cycle diagram Appendix VIII).

Point	Position in air cycle
1D	Day operating setting of phytotron
IN	Night operating setting of phytotron
2	State of return air at inlet to air conditioning cabinet.
2'	State of fresh air assumed for analysis of alternative 1. (4% fresh air without a pre-dehumidifier).
2'	State of fresh air at inlet to air conditioning cabinet for both alternatives 1 and 2 (with and without predehumidification).
2"	State of fresh air assumed for analysis of alternative 2 (4% fresh air with a pre-dehumidifier).
3	Mixture condition of 4% fresh air and 96% return air.
4	State of air downstream of preheater.
5	State of air downstream of direct expansion coil.
5'	State of main supply air leaving pre-dehumidifier (for alternative 2 only).
6	State of air downstream of reheater.
7	State of air downstream of pan humidifier.
8	State of air after addition of miscellaneous air conditioning cabinet

and supply ductwork heat gains
due to transmission, fan losses,
leakage, etc.

The path taken by the Air Cycle for Alternative 1 is

ID or IN - 2; 2, 2' - 3; 3 - 4 - 5 - 6 - 7 - 8.

The path for Alternative 2 is

ID or IN - 2; 2" - 2'; 2, 2' - 3, 3 - 4 - 5 - 5' - 6 - 7 - 8.

(The path of the air is detailed on the following page).

	SYSTEM AIR CYCLE
PATH	PROCESS
ID, IN-2	Total plant compartment heat gains not associated with temperature gradient.
2" - 2'	Path of fresh air intake supply through pre-dehumidifier for Alternative 2 only.
2, 2' - 3	Fresh air and return air mixing process.
3 - 4	Path through preheater. [Note: In this example, an examination of Appendix II indicates that for the "day" design setting of this problem 74°F dbt and 53°F dpt the preheat coil is set to maintain a temperature of 74°F dbt. In the case of the day cycle the temperature downstream of the preheat coil is greater than 74°F and the preheat coil is in the "off" position. Thus point 3 equals point 4 in the case of the day cycle].
4 - 5	Path through direct expansion coil (load ratio line).
5 - 5'	Path of main supply air through dehumidifier (for alternative 2 only).
5 or 5' - 6	Path through reheater.
6 - 7	Path through humidifier.
7 - 8	Miscellaneous heat gains - (Process occurring simultaneously with paths 3 to 7).
8 - 1	Plant compartment heat gains associated with temperature gradient.

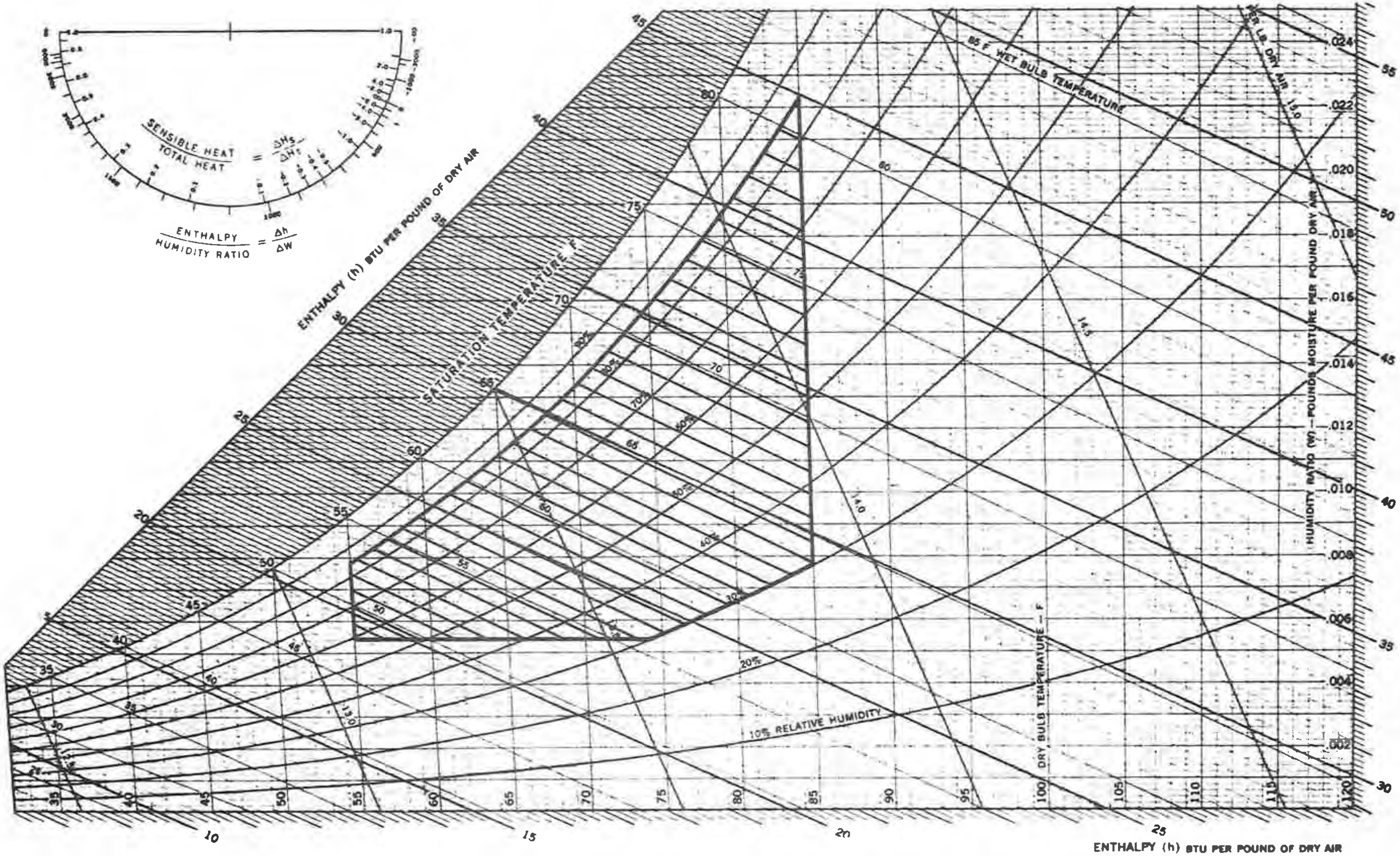
5. CONCLUSION

A new system of phytotron design is presented here. The use of a temperature-humidity controlled phytotron has been largely avoided to date due to its expense, its limited range and its failure to maintain conditions to reasonable tolerances.

This design points to a new outlook. It should now be economically feasible to satisfy the design requirements over a broad range of operating conditions. It gives the agronomist a wider area for experimentation. The system applies to naturally as well as artificially lit units.

The design decisions leading to the system presented here differ from those taken by others. The reasons for these differences have been presented. The unitary form of existing phytotrons has been replaced by a dual package. Emphasis has been placed on the thermodynamic and fluid-flow principles that relate to phytotron performance.

Reliable standard equipment with simple conventional air and refrigeration cycles and controls is engineered into an integrated system that satisfies the requirements of the agronomist and eliminates the major problems that beset existing designs.



RANGE OF PHYTOTRON OPERATING SETTINGS

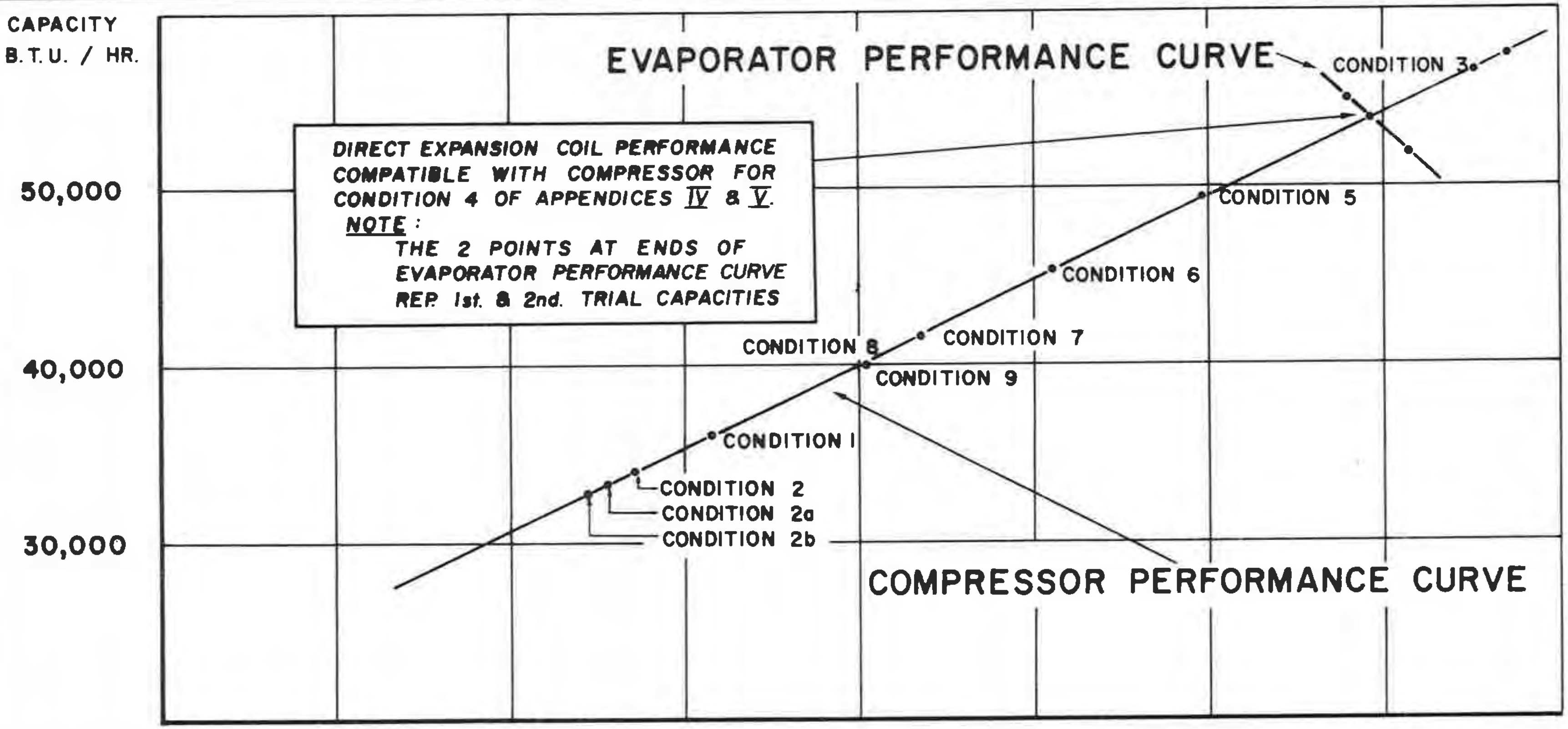
APPENDIX I

ASHRAE PSYCHROMETRIC CHART NO. 1

APPENDIX II

SCHEDULE OF PREHEAT SETTINGS FOR RANGE OF OPERATING CONDITIONS

OPERATING DEW POINT TEMPERATURE °F	OPERATING DRY BULB TEMPERATURE °F	REQUIRED TEMPERATURE SETTING UPSTREAM OF COOLING COIL TO BE MAINTAINED BY PRE-HEATER °F
41° to 71°	55° to 85°	Same as day dry bulb setting but not less than 63° dbt
71° to 76.5°	76° to 85°	2 degrees higher than day dry bulb setting
76.5° to 80°	81.5° to 85°	5 degrees higher than day dry bulb setting



EVAPORATOR TEMPERATURE	°F	20	25	30	35	40	45	50	55
EVAPORATOR PRESSURE	PSIA	35·7	39·3	43·1	47·3	51·7	56·4	61·4	66·7
COMPRESSOR SUCTION	PSIA	34·7	38·3	42·1	46·3	50·7	55·4	60·4	65·7

EVAPORATOR TEMPERATURES COMPATIBLE WITH COMPRESSOR PERFORMANCE FOR CONDITIONS OF APPENDICES IV AND V

APPENDIX III

APPENDIX IV

TABULATION OF DIRECT EXPANSION COIL PERFORMANCE FOR VARIOUS ENTRY CONDITIONS

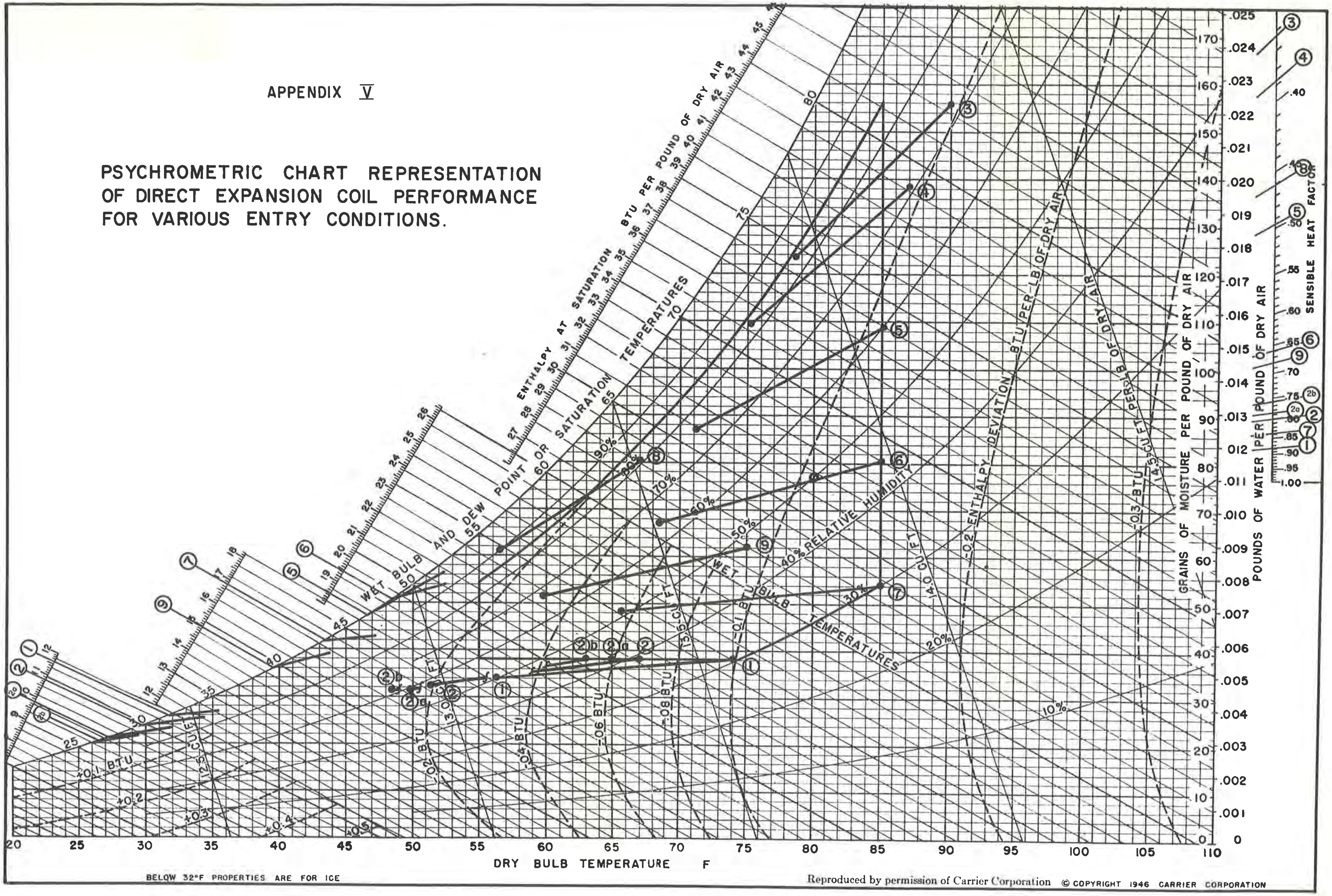
NOTE: The analysis presented here is based on the use of the engineering performance data for the McQuay 1/2" Direct Expansion Coil. (Section 3.4.2 (g))

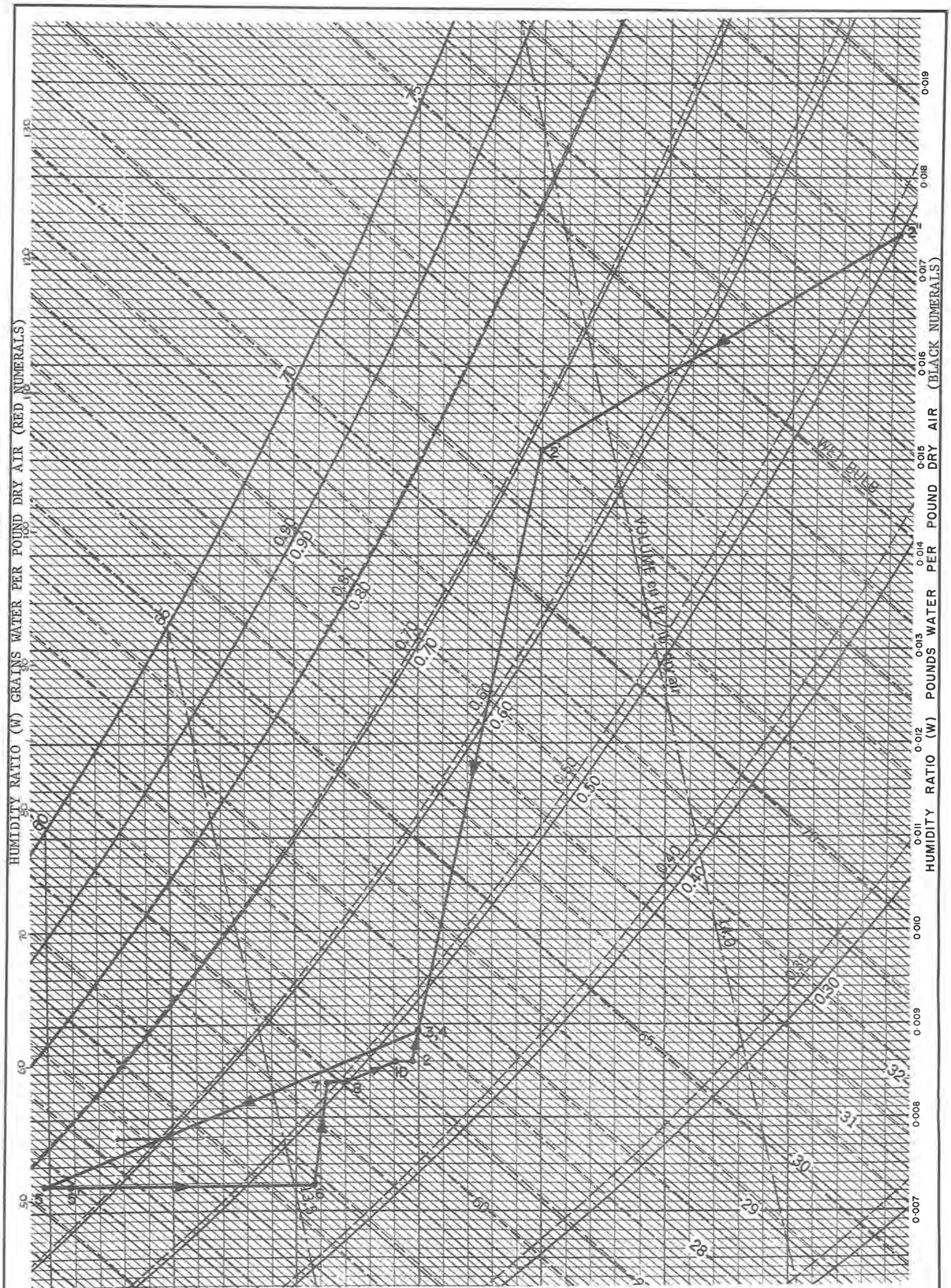
		Condition of Minimum Dehumidification and Basis for Compressor Selection	Condition where Frosting is most likely to occur			Condition Determining Maximum Humidification and Tonnage	Points analyzed to determine preheat schedule settings			Condition Determining Maximum Preheat and Reheat	Random Points	
SYMBOL *	UNITS	1	2	2a	2b	3	4	5	6	7	8	9
Q_a	Btu/hr	36000	34000	33400	33000	56500	54000	49800	45600	41350	41000	40000
e dbt	°F	74	67	65	63	90	87	85	85	85	67	75
e wbt	°F	56.0	53.15	52.25	51.4	82.3	79.1	74.0	69.0	63.65	63.15	62.0
e h_a	Btu/lb d a	23.84	22.11	21.58	21.09	46.24	42.73	37.66	33.25	29.05	28.68	27.85
Q_a/f a (actual)	Btu/hr/sq ft	5143	4857	4771	4714	8071	7714	7114	6515	5890	5857	5714
e dep	°F	18.0	13.85	12.75	11.6	7.7	7.9	11.0	16.0	21.35	3.85	13.0
t c f	-	0.558	0.558	0.558	0.558	0.558	0.558	0.558	0.558	0.558	0.558	0.558
Q_b/f a (basic)	Btu/hr/sq ft	9200	8690	8570	8440	14470	13820	12740	11680	10550	10500	10250
t_r	°F	30.7	28.5	27.7	27.25	52.6	49.6	44.9	40.5	36.8	36.1	35.2
Δh_a	Btu/lb d a	5.00	4.73	4.63	4.58	7.85	7.5	6.93	6.34	5.74	5.70	5.56
l h_a	Btu/lb d a	18.84	17.38	16.95	16.51	38.39	35.23	30.73	26.91	23.31	22.98	22.29
l wbt	°F	47.3	44.46	43.6	42.7	74.77	71.32	65.87	60.66	55.15	54.6	53.45
dep r	-	0.492	0.492	0.492	0.492	0.492	0.492	0.492	0.492	0.492	0.492	0.492
l dep	°F	8.86	6.82	6.29	5.72	3.79	3.89	5.41	7.88	10.49	1.89	6.4
l dbt	°F	56.16	51.3	49.89	48.4	78.56	75.21	71.28	68.54	65.64	56.5	59.85
S H R	Btu sensible/ Btu total	0.87	0.80	0.78	0.76	0.35	0.39	0.50	0.66	0.84	0.46	0.68
A D P	°F	31.8	29.9	27.3	27.1	-	-	46.5	48.3	43.4	-	40.0
e w	grains of wv/ lb d a	38.0	38.0	38.0	38.0	15.60	138.0	109.0	80.5	54.0	80.5	62.0
l w	" " "	34.3	32.3	31.6	31.2	123.4	109.3	86.7	66.6	48.3	61.2	51.0
Δ w	" " "	3.7	5.7	6.4	6.8	32.6	28.7	22.3	13.9	5.7	19.3	11.0
Δ t	°F	17.8	15.7	15.1	14.6	11.4	11.8	13.7	16.5	19.4	10.5	15.1

* See Appendix X, Key to Symbols.

APPENDIX V

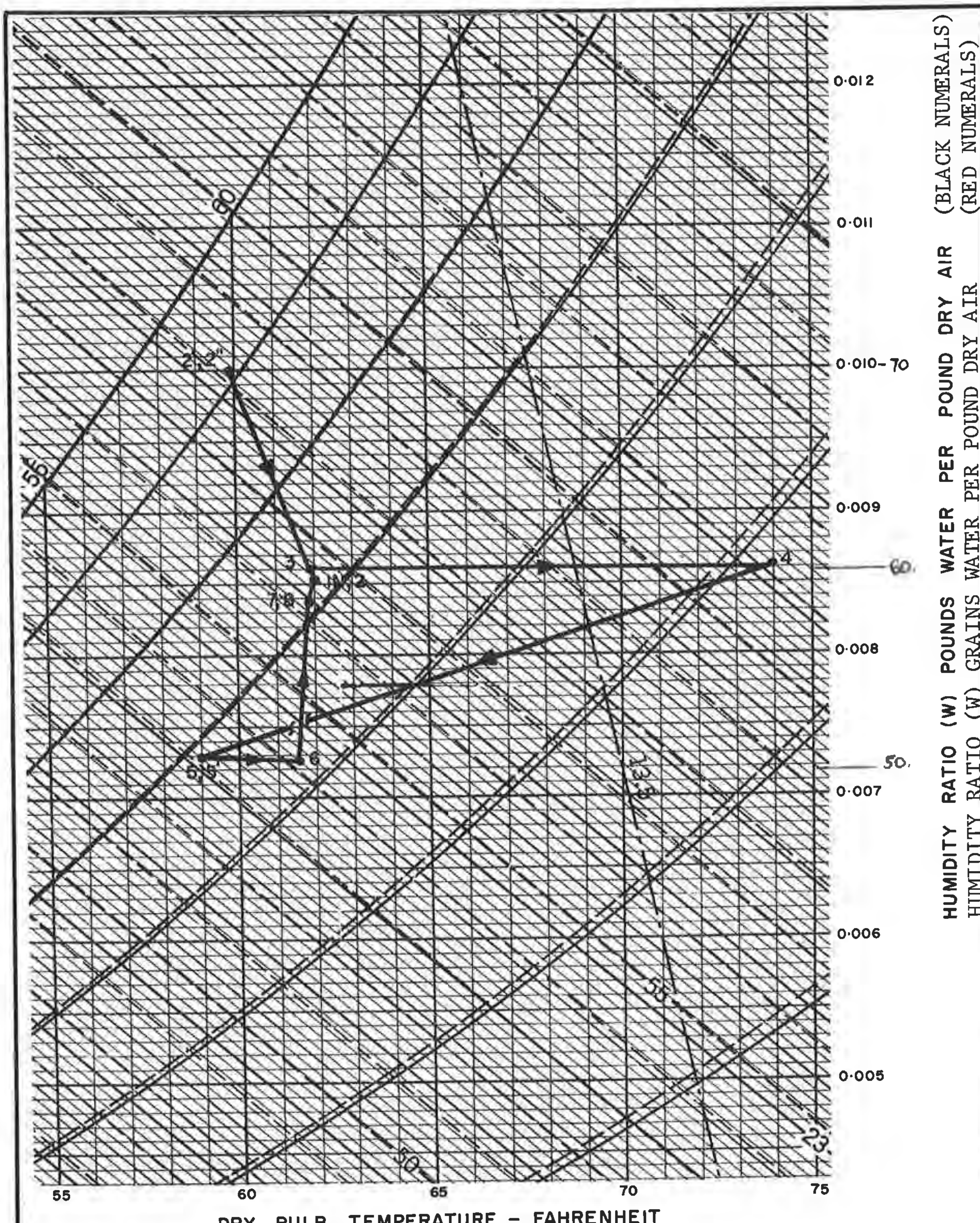
PSYCHROMETRIC CHART REPRESENTATION OF DIRECT EXPANSION COIL PERFORMANCE FOR VARIOUS ENTRY CONDITIONS.





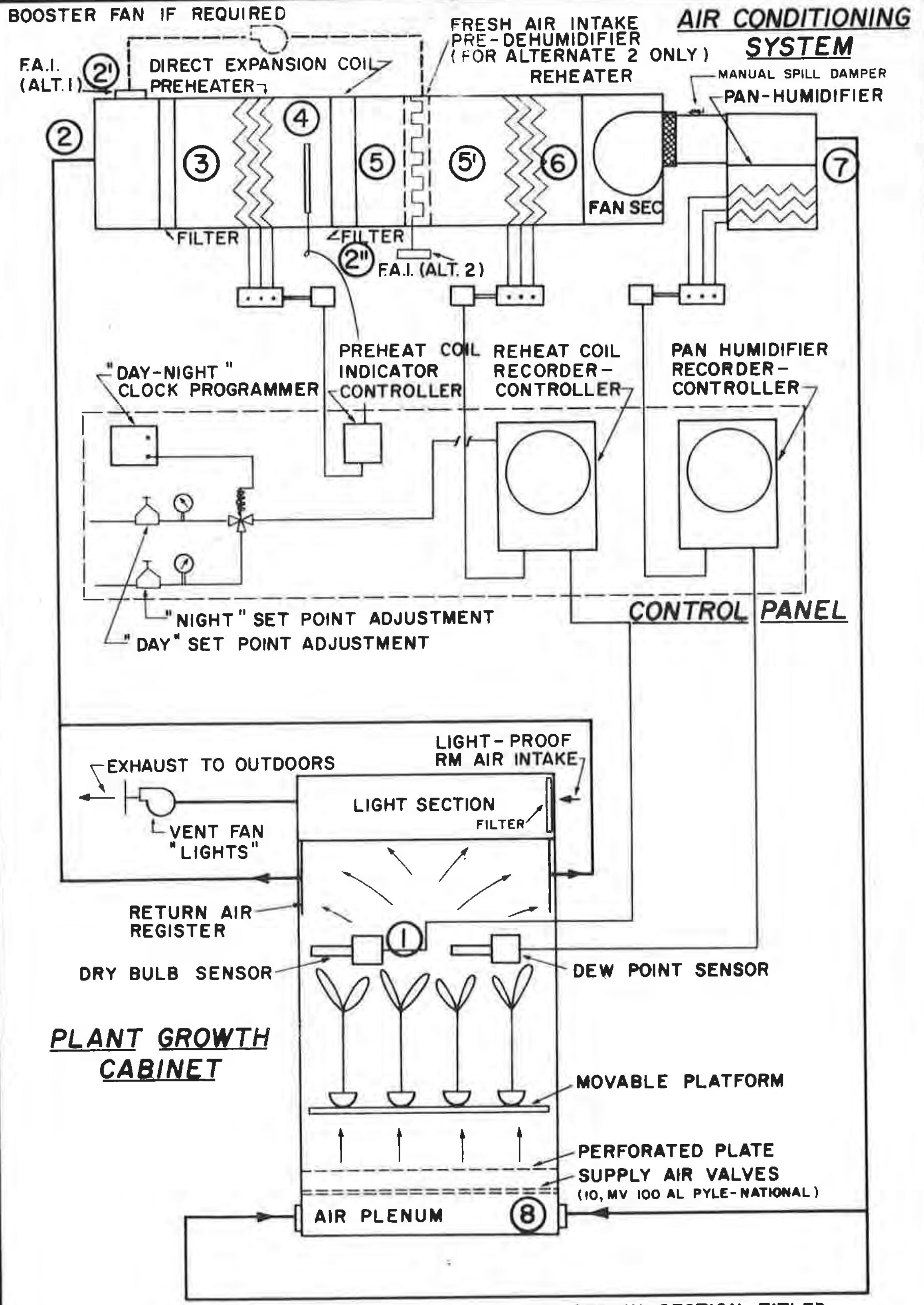
AIR SYSTEM PERFORMANCE FOR A DAY OPERATING SETTING APPENDIX VI

DRAWN ON ASHRAE PSYCHROMETRIC CHART



AIR SYSTEM PERFORMANCE FOR A NIGHT OPERATING SETTING.

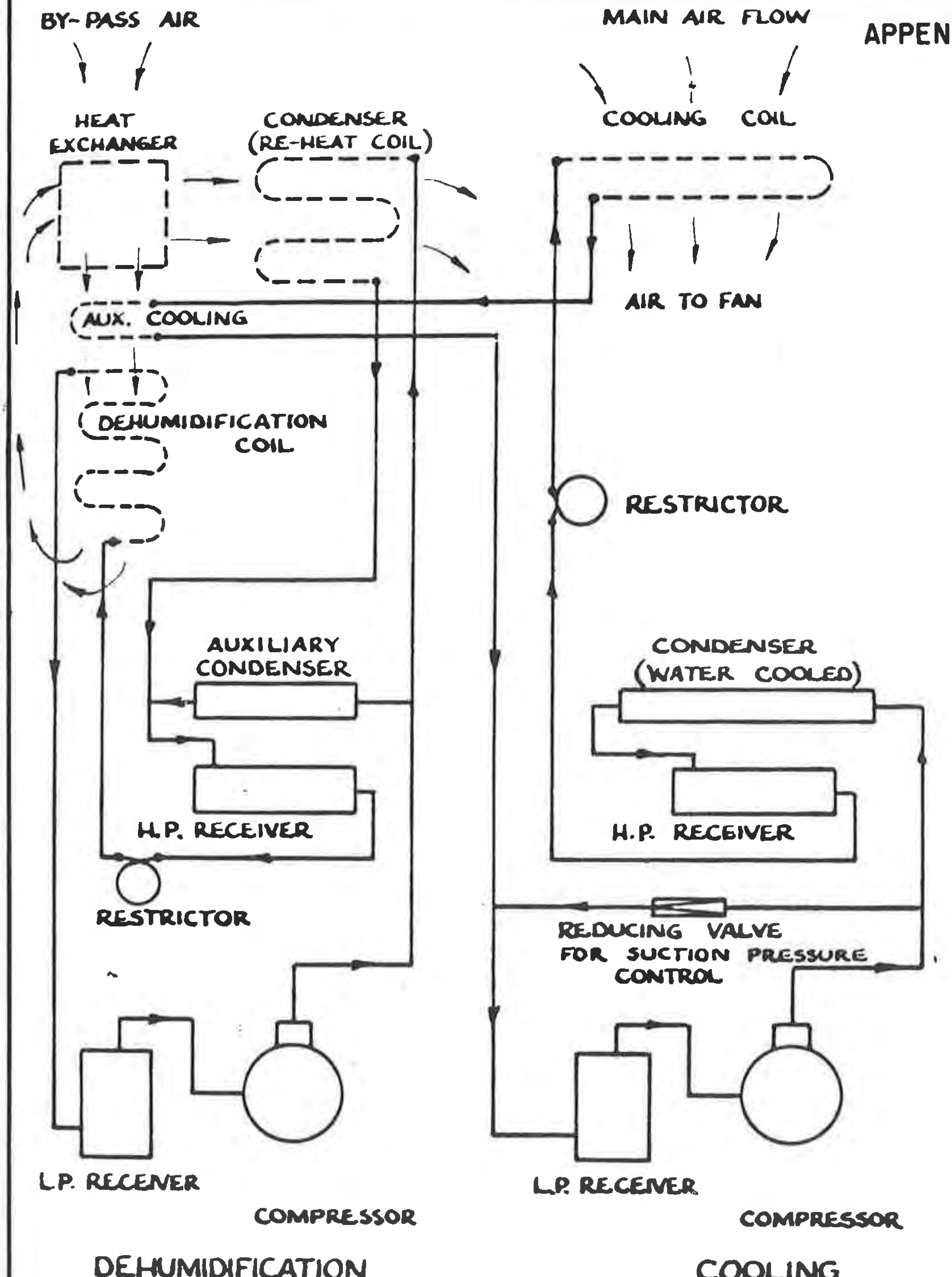
APPENDIX VII



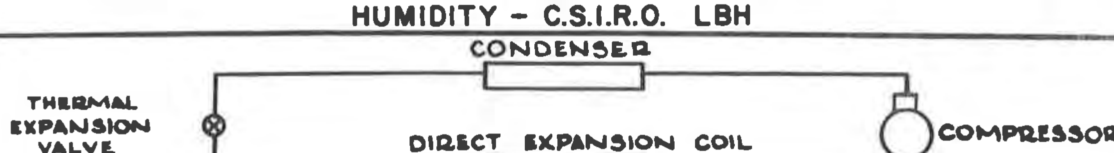
NOTE: NUMBERING CORRESPONDS TO SAME SYSTEM USED IN SECTION TITLED - PSYCHROMETRIC ANALYSIS. (ALSO SEE APPENDICES VI AND VII).

SCHEMATIC-AIR CYCLE AND CONTROLS APPENDIX VIII

APPENDIX IX



DIAGRAMMATIC ARRANGEMENT OF REFRIGERATION SYSTEM FOR HUMIDITY – C.S.I.R.O. LBH



DIAGRAMMATIC ARRANGEMENT OF REFRIGERATION SYSTEM FOR HUMIDITY – AT "WAITE"

APPENDIX X

KEY OF SYMBOLS

ADP	apparatus dew point
ALT	alternative
ASHRAE	American Society of Heating, Refrigerating and Air Conditioning Engineers
ASHVE	American Society of Heating and Ventilating Engineers
Btu	British Thermal Unit
cfm	cubic feet per minute, standard air
c_p	specific heat of dry air
CSIRO	Commonwealth Scientific and Industrial Research Organization
Δ	prefix indicating "change of"
da	dry air
dbt	dry bulb temperature
dep	wet bulb depression
dep r	wet bulb depression ratio
dpt	dew point temperature
dx coil	direct expansion coil
e	prefix indicating entering condition
fa	face area
fai	fresh air intake
fpi	fins per inch
fpm	feet per minute
h_a	enthalpy, per pound of dry air
h_{wv}	enthalpy equivalent to latent heat of vapourization, per pound of water vapour
hr	hour
in or "	inch
l	prefix indicating leaving condition
LBH	a temperature-humidity growth cabinet developed in Australia by the CSIRO
lb or lb_m	pound mass of dry air
lb_{wv}	pound mass of water vapour
$\dot{m}$	pounds of dry air per hour
min	minute
o d	outer diameter of tube
psia, psig	pounds per square inch absolute, gauge
Q_a	heat absorption capacity of the specific coil selected
Q_b	heat absorption capacity of manufacturer's basic test coil
Q_{LH}, Q_S, Q_{Tot}	latent, sensible and total heat absorption capacity of an airstream
rpm	revolutions per minute
SHR	sensible heat ratio
tcf	total correction factor
t_r	refrigerant temperature within direct expansion coil
$\bar{v}$	specific volume
w	grains of water vapour per pound of dry air

APPENDIX XI

DETAILED EXPLANATION OF THE DESIGN SOLUTION

This section is concerned with explaining in further detail the design solution. It will also attempt to place in proper perspective such factors as running cost and reliability.

The applicability of the design to naturally lit growth chambers and cost, as related to range requirements, rate of air flow and latent heat design loads will also be discussed in this Appendix.

1. THE FUNDAMENTAL PRINCIPLE IN DETERMINING THE DESIGN SOLUTION.

In the design of a phytotron, a clear distinction should be made between the cooling load variation for any one operating setting and the variation resulting when designing for a range of operating settings (Section 2.2). The former is minimal; and in fact is an advantage that is utilised in this design whereas the latter, as opposed to conventional air conditioning applications, is relatively large. It is difficult for systems using a direct expansion coil to cope with this range when both temperature and humidity control is specified. The process path across the direct expansion coil is associated with both sensible cooling and dehumidifying, which have a quantitatively fixed relationship to each other for any one entry condition. When wide range settings are specified the problem is intensified since, for different operating settings, there are different relationships between the heat and mass transfer taking place at the direct expansion coil. As a first step, the cooling and dehumidifying processes were studied over a wide range of entry conditions (Section 3.4.2.(d) and (e)) for the relationship between the mechanical engineering loads and the growth chamber loads. The selection of the system was based on obtaining the lowest possible expenditure of power that would meet the design requirements (Section 3.4.2.(d)). This minimal load condition was dependent on meeting the total design latent heat load by an equivalent dehumidification. A study of numerous entry conditions to the direct expansion coil (Section 3.4.2.(e)) led to the conclusion that for any one coil, dehumidification was at a minimum at a point combining high dry bulb temperature and low humidity ratio. This point was designated as critical point 1. Furthermore, this same study revealed that sensible cooling load and latent heat load was met or exceeded for all other possible entry conditions when the systems of refrigeration and air-conditioning were arranged as per section 3.3.2. The next step was to select a reliable direct expansion coil that would best suit every entry condition in the specified range. Since dehumidification was the limiting factor in coil selection, the goal was to use a coil having the lowest possible refrigerant temperature that would, nevertheless, be safely above the frost point. A further goal was to select a coil and establish its conditions of operation to minimize the excess dehumidification occurring at high dry bulb and high humidity ratio operating settings such as for critical condition number 3. Again, this approach reduced running costs both in dehumidifying and humidifying. (Pages 52 to 55 including figures 3 and 4 present this analysis). Thus both capital and running cost were kept at a minimum. Nevertheless, the total mechanical load was relatively large compared to the growth chamber load. To achieve a stable system, the next step was to eliminate variations in this relatively large load during any one operating setting. The aim was to devise a system in which the cooling, dehumidifying, heating and humidifying processes approach a constant load condition. The design solution seeks to develop the system with a minimal process rate, specifically, the insignificant change of load associated with the growth chamber load variations. These are the variations in the loads enumerated in sections 3.1.1., 3.1.2., 3.1.3. and 3.1.4. Thus the control system selected is concerned with the control of the "true" load, the agronomist's load, and not the mechanical engineering load associated with the range requirement and the use of a direct expansion coil.

Once the design aim was established it was possible to select the air-conditioning and refrigeration systems and to integrate them with a control system to carry out this aim. Section 3.3.2., "Factors upon which system performance depends" to section 3.6, "The Control System" presents the analysis of the method of selection to achieve the above aim.

2. APPLICATION TO NATURALLY LIT GROWTH CHAMBERS.

The system proposed for the "Waite" artificially lit cabinet is applicable to designs for naturally lit cabinets. Considerable difficulty has been encountered in maintaining conditions in these cabinets. With the elimination of the large instabilities due to the dehumidifying process, an addition of a "true load" variation would not affect the system adversely. The designer would have to calculate these load variations and limit them to narrow differentials for the controlled variables of equations 1 and 2, dry bulb temperature and humidity ratio. An increase of the air velocity, a limitation on the number and size of plants, the substitution of an instantaneous response humidifier and the use of 100 per cent recirculation plus CO_2 introduction to eliminate outside air latent heat load (this may not be necessary at all times) would result in the system proposed for "Waite" being applicable to naturally lit growth chambers.

3. RELIABILITY.

There are numerous factors that must be considered in selecting the systems to carry out the design aim. The fact that the design is for use in scientific research is a primary consideration. It is an important laboratory device and must be reliable, rugged, easy to service and maintain, capable of operating continuously over long periods of time and easy to operate and set up. It must be provided with ample adjustment to allow for effects of aging. It must incorporate safety measures to prevent accidents on failure or malfunction of a part. Listed below are some of the measures taken to assure the reliability of the system.

Reliability Considerations

(1) Component selection

Heavy duty, rugged, industrial components, designed for continuous operation and having certified engineering performance data, were selected. (Sections 3.3.1, 3.3.2(e), 3.4.2.(d)).

(2) Component spacing

Components were arranged to assure space for proper fluid flow for sensing and controlling, for larger rugged components, servicing and maintaining. The air conditioning system is separated from the growth chamber. The single self-contained unit is incompatible with the design requirements (Section 1). Compactness is considered secondary to the factors enumerated below.

(3) Open compressor

For the purpose of flexibility in adjustment and in obtaining optimum performance, an open type, variable speed, belt-driven compressor using Freon 12 as a refrigerant was selected (Section 3.4.2.(d)).

(4) Air distribution

The air distribution was engineered as described in Section 3.2 of the thesis. The components were selected on the basis of their performance data. The system was provided with volume and directional flow adjustments. Uniformity of temperature and humidity within the plant occupied area was the design aim.

(5) Low air velocities

Air velocities of 90 fpm were used through the growth chamber. This is below the flow rates commonly used in existing chambers and in accordance with agronomists' recommendations.

(6) Backward curved centrifugal supply fan

This larger and more expensive fan type was used to permit air flow rate adjustment by means of volume dampers.

(7) Avoidance of frosting

The system was selected to preclude the possibility of frosting. Provision was made for adjustment in the case of aging, i.e., clogged coils or lower air flow rates. (Sections 2.2.5.(g); 3.4.1.(a); 3.4.2.(d); 3.4.2.(e); figures 3 and 4; Appendix II.)

(8) Stable control system

The control system was selected to tie in closely with the design. It functions to control the slow process rates due to the growth chamber load variations.

(9) Humidity ratio - Sensing and Control

The pan-humidifier was selected to minimize interaction between humidity and dry bulb temperature control. The control system used a dew point temperature sensing element rather than a relative humidity sensing element. (Section 3.6.5).

(10) Safety protection

The automatic temperature and humidity control systems will be interlocked with the supply fan to prevent the heaters, humidifiers and light section from operating when the supply fan is off (Section 3.6.2). High-low pressure cut-off and low voltage protection are provided on refrigerant system. (Section 3.3.2.(f)).

(11) Conservative design loads

All estimates on loads were on the high or safe side.

4. RUNNING COSTS.

A further objective was to achieve the design aim at the lowest possible capital and running cost. However, reliability was not sacrificed in order to obtain this end. (Section 2.2.6). Thus most of the factors enumerated above under "Reliability" resulted in increasing the capital cost. The low air velocities through the growth chamber served to increase the running cost.

In the beginning of the thesis (Section 1), (Section 2.2), it was indicated that the aim was to produce an economically feasible unit. An examination of the literature and designs of narrow-range, commercial cabinets revealed that if their principle of operation had been extended to include the wide range of the "Waite" cabinet, they would have required an impractically large refrigeration plant having in some cases as much as a 17 ton refrigeration capacity and immense power needs for reheat and humidification. (Section 2.2.). On the other hand in a separate section below it will be demonstrated that, should the "Waite" requirements be changed to a narrower range of humidity (or to temperature-only cabinets) the capacity of the system would drop to a fraction of its present modest tonnage. Listed below are some of the measures taken to assure low running costs.

Running Cost Considerations

(1) Direct expansion coil selection was based on establishing minimum running cost - see analysis determining limiting condition for dehumidification critical point 1. (Section 3.4.2.(d)).

(2) Compressor capacity was established by the direct expansion coil capacity in (1) above. (Section 3.4.2.(d)).

(3) Preheat coil settings were established to operate the humidifier at minimum running cost. (Section 3.4.1.(c)).

(4) Direct expansion coil selection analysis was also concerned with minimum dehumidifying and humidifying running costs. The problem arises at high humidity ratio settings. (Section 3.4.2.(e); Figure 4).

(5) Variable speed compressor was selected to permit reduced running costs. (Section 3.4.2.(f)).

5. COST AS RELATED TO RANGE REQUIREMENTS and

Rate of Air Flow, Latent Heat Design Loads

It would be misleading to compare costs between systems without taking into consideration the range requirements. For example, if the "Waite" research requirements are altered by limiting the system to higher range humidities above 45% relative humidity, the cost of the system would be appreciably reduced. In such a case, due to this single change, that is, assuming identical light loads, heat gains through cabinet walls, transpiration loads and outside air intake latent heat loads as enumerated herein, a very different size condensing unit is required. The condition for critical point 1 would dictate the basis for selection of the condensing unit. In this example, critical point No. 6 (approximately) of Appendices IV and V takes over the role of critical point No.1. Note that the load ratio line has a greater slope and with the dehumidification of $3\frac{1}{2}$ grains of water vapour per pound of dry air, would be associated with sensible cooling of $4^{\circ}F$ and both the cooling and dehumidifying needs would be satisfied with a Δh_a of 1.5 rather than 5.0 Btu per hr. per lb. of dry air. Furthermore, on examination of Appendix IV, it can be seen that the <u>same</u> refrigeration system which had a capacity of 36000 Btu per hr. has increased 50% in capacity to 45,600 Btu per hr. The conclusion is obvious - a very much smaller condensing unit having a refrigeration capacity of about <u>one-fourth</u> the "Waite" cabinet would be required. It would be associated with equivalently smaller reheating and humidifying facilities. Similarly, it can be demonstrated that the capacity of the condensing unit and running cost are approximately inversely proportional to the air flow rate and proportional to the latent heat design load.

APPENDIX XII

COST OF BUILDING THE "WAITE" PHYTOTRON

The "Waite" phytotron is estimated to cost £A4300 as per Section 2.2.6.(a). These costs are detailed below.

Environmental Growth Chamber	£1450
(This estimate includes completely wired light section, diffusers, return air registers, light section vent fan).	
Condensing Unit	600
(This estimate includes supports and variable speed drive).	
Air Conditioning Cabinet	400
(This estimate includes preheater, reheater, direct expansion coil, filters, centrifugal fan).	
Control System	1100
(This estimate includes preheater, reheater, humidity, condenser and water controls, day-night programmer, provision for fresh air and total air metering, condenser water control valve, sensing devices, relays, safety devices).	

Ductwork	£250
(This estimate includes insulation).	
Electrical Equipment, Mechanical Piping	500
(This estimate includes contractors' services).	============

TOTAL COST £A4,300

APPENDIX XIII

OUTLINE OF THE METHOD OF CONSTRUCTION OF THE GRAPH FOR APPENDIX III

Step 1. The condition for critical point number 1 was determined. For the "Waite" design this was outlined in Section 2.2.5.(g) and the calculated results enumerated under the column labelled number 1 of Appendix IV. For the purposes of construction of the Appendix III curve the relevant data for critical point number 1 are as follows:

capacity of the compressor selected = 36,000 Btu per hour

refrigerant temperature in evaporator = 30.7°F

Step 2. The evaporator temperature and its corresponding pressure are plotted as the abscissa of Appendix III.

Step 3. The capacity of the compressor (for a constant condensing temperature) is plotted as the ordinate.

Step 4. The point representing condition number 1 :-

ordinate - 36,000 Btu per hour, abscissa 30.7°F

is entered on the graph.

Step 5. An allowance is made for pressure drop between the evaporator and the suction to the compressor. (Section 3.4.2.(c), page 49). This, too, is listed as the abscissa of Appendix III.

Step 6. An available open type compressor with known performance data was selected to meet this capacity, that is, its capacity versus suction pressure curve will pass through critical condition number 1 (see step 4 above). This curve represents a speed (of about 1000 RPM) which is in the middle of the compressor speed operating range in order to allow for variable speed drive adjustments in both directions. The performance curve of the compressor at the selected speed is entered in Appendix III.

Step 7. All other points: 2, 2a, 2b, 3, 4, 5, 6, 7, 8 and 9 are then determined by the trial and error method of solution or by the graphic simultaneous solution of the evaporator and condensing unit curves. (Section 3.4.2.(d), page 50). Condition number 4 was selected as a random point to illustrate this solution. (See note on Appendix III; also evaporator performance curve intersecting the compressor performance curve). This point of intersection represents for critical point number 4, the equilibrium condition of the continuously operating refrigeration system. It is at this condition that the evaporator and condensing unit performance are compatible.

Step 8. The method that is detailed below for condition number 4 is applied to all other entry conditions studied. The point of equilibrium between evaporator and condensing unit performance is then located on the compressor curve. This completes the graph of Appendix III.

APPENDIX XIV

DETAILS OF THE TRIAL AND ERROR SOLUTION OF EVAPORATOR AND COMPRESSOR CURVES FOR CONDITION 4 OF APPENDIX III

Step 1, trial 1. By observation select an evaporator capacity. In this case 52,000 Btu per hr was selected. The following steps to step 9 were taken to ascertain whether the capacity selection was compatible with compressor performance.

Step 2, trial 1. e dbt = 87°F (Similar to step 2 of section 3.4.2.(g)). (See condition 4 Appendix IV).

Step 3, trial 1. e wbt = 79.1°F (Similar to step 3 of section 3.4.2.(g)). (See condition 4 Appendix IV).

Step 4, trial 1. Q_a/fa (actual) = 52000/7 = 7429 Btu per hr.per sq.ft. (Similar to step 5 of section 3.4.2.(g)).

Step 5, trial 1. e dep = 87.0 - 79.1 = 7.9°F (Similar to step 6 of section 3.4.2.(g)). (See condition 4, Appendix IV).

Step 6, trial 1. t c f = 0.558 (Same as step 7 of section 3.4.2.(g)). (See condition 4, Appendix IV).

Step 7, trial 1. Q_b/fa (basic) = 7429/0.558 = 13,300 Btu per hr.per sq.ft. (Similar to step 8 of section 3.4.2.(g)).

Step 8, trial 1. t_r = 50.7°F (Similar to Step 9 of section 3.4.2.(g)).

Step 9, trial 1. The refrigerant temperature of Step 8 is entered on Appendix III at its intersection with the trial capacity of 52,000 Btu per hr. The first trial capacity is not compatible with the compressor performance curve on Appendix III. It is located below this curve.

Step 1, trial 2.	A second evaporator capacity, 55,000 Btu per hr. is selected. (This time a higher capacity, one that would fall above the compressor performance curve, has been assumed).
Step 2, trial 2.	e dbt = 87°F
Step 3, trial 2.	e wbt = 79.1°F
Step 4, trial 2.	Q_a/fa (actual) = 55000/7 = 7857 Btu per hr. per sq. ft.
Step 5, trial 2.	e dep = 7.9°F
Step 6, trial 2.	tcf = 0.558
Step 7, trial 2.	$Q_b/fa = \frac{7857}{0.558} = 14080$ Btu per hr. per sq. ft.
Step 8, trial 2.	t_r = 49°F
Step 9, trial 2.	Again the second trial is not compatible with the compressor performance curve of Appendix III. It is compatible only with the evaporator performance curve. This point is entered on Appendix III at the intersection of the trial capacity of 55,000 Btu per hr. and the refrigerant temperature of 49°F.
Step 10.	The 2 trial points entered on Appendix III for condition 4 are connected with a straight line. The solution lies at the intersection of the evaporator performance curve thus drawn and the compressor performance curve, t_r = 49.6°F

and Q_a = 54,000 Btu per hr.

Step 11. It is this value for capacity that is entered into Appendix IV for condition 4. All other points 2, 2a, 2b, 3, 5, 6, 7, 8 and 9 are obtained in a similar manner.

With the capacity of the refrigeration system established for a particular entry condition, the steps outlined for condition number 1 solution, (Section 3.4.2.(g)), are followed. These results are enumerated in Appendix IV and are used in the construction of Appendix V.

APPENDIX XV

DETERMINATION OF DESIGN LOAD - DETAILED

Sensible Heat Gains Associated With Temperature Gradient, (see definition of temperature gradient in section 3.2.9), is stated in section 3.1.1 to be 5,200 Btu per hr. for a temperature gradient of $3^{\circ}F$ and 1600 cfm.

The basis for the above figure is as follows:

light load based on data from agronomist	4100 Btu per hr.
cabinet wall, glass and ceiling losses (based on cabinet operating at its lowest dry bulb setting of $55^{\circ}F$)	1130 Btu per hr.
(there will be no losses due to infiltration through cracks since the cabinet is sealed and during 4% OA intake any losses due to leakage will be outward).	0
	5200 Btu per hr.

The summation of the loads listed above is equal to 5200 Btu per hr. as stated in section 3.1.1.

Solving for Δt in equation 1

$$Q = cfm \times 1.08 \times \Delta t \qquad 5200 = 1600 \times 1.08 \times \Delta t$$

$$\Delta t = 3^{\circ}F$$

(Section 3.1.1).

Sensible Heat Gains Not Associated With Temperature Gradient

(1) Loads due to outside air intake

The design "outside air" condition was assumed to be 90°F. (The "outside air" is actually interior building air. The laboratory is located below grade in a heavy building). The calculations to determine this load are based on the growth chamber operating at its lowest temperature setting of 55°F. Substituting in equation 1 for the 64 cfm, (4 per cent outside air volume flow rate):

$$Q_s = \text{cfm} \times 1.08 \times \Delta t$$

$$Q_s = (64)(1.08)(90 - 55) = 2420 \text{ Btu/hr.}$$

(2) Duct heat gains

The losses from an insulated duct length of 30 feet having an average surface area of 4 square feet per foot of length and having an overall duct wall coefficient of transmission of 0.25 Btu per (hour)(square foot)(°F).

$$Q = UA\Delta t$$

$$Q = (0.25)(120)(90 - 55) = \underline{1050} \text{ Btu/hr.}$$

(3) Fan heat and miscellaneous losses

A conservative estimate for this small fan and short run: 400 Btu/hr.

Total 3870 Btu/hr.

plus 10% factor to take account of variables in calculation: 4300 Btu/hr.

(Section 3.1.2)

The associated temperature difference is obtained from equation 1.

$$Q = \text{cfm} \times 1.08 \times \Delta t$$

$$4300 = (1600)(1.08)(\Delta t)$$

$$\Delta t = 2.5^{\circ}F \quad \text{(Section 3.1.2)}$$

Internal Latent Heat Gains

At Waite the agronomists estimated 40 ounces of water vapour per hour as a basis for the internal latent heat load due to transpiration and evaporation including plant watering. The term estimated to be about 2700 Btu per hour in section 3.1.3 is derived from the agronomists' above estimate.

40 ounces of water vapour = 2.5 lb_{wv}

Since 1 pound of water has its maximum latent heat capacity at the lowest saturation temperature, the value for latent heat capacity was taken at $55^{\circ}F$. This is 1062.7 Btu per lb. of water. Substituting in equation 2 $Q_L = \text{cfm} \times .67 \times \Delta w$

$$(2.5)(1062.7) = (1600)(.67)(\Delta w)$$

Δw = 2.5 grains of water vapour/lb_m

(Section 3.1.3)

Latent Heat Gains Due to Outside Air Intake

The 5000 Btu/hr. load (Section 3.1.4) is based on the assumption that the maximum humidity ratio of the room air intake at "Waite" is 134 grains of wv/lb_m. Substituting in equation 2 for 64 cfm air intake and for the minimum humidity ratio operating setting of 33 grains of wv/lb_m

$$Q_L = cfm\ (.67)(\Delta W)$$

$$Q_L = (64)(.67)(134 - 33)$$

$$Q_L = 5000 \text{ Btu/hr.} \qquad \text{(Section 3.1.4)}$$

This is equivalent to 4.6 grains wv per lb.da for 1600 cfm

Substituting in equation 2

$$Q_L = cfm \times .67 \times \Delta W$$

$$5000 = (1600)(.67)\Delta W \qquad \Delta W = 4.6 \quad \text{(Section 3.1.4)}$$

The method of meeting this load is explained in a later section.

In section 3.5.1 it is indicated that the "Waite" agronomists were willing to accept a 100 per cent recirculation system and therefore under that condition outside air latent load would be zero.

In section 3.5 it is indicated that the combination of factors which make for the outside air latent heat load being excessive and thus requiring changeover to 100 per cent recirculation (with metered CO_2 introduced) depend on the total latent heat load being high, a large number of fully grown plants, the outside latent heat load itself being high, the dry bulb being very high and the operating setting for the cabinet being a high dry bulb temperature and a very low humidity ratio.

Granted this rare occurrence, the simultaneous combination of factors at their maximum limits - in section 3.5 it is pointed out that change-over to CO_2 under this condition would not be required if the humidity ratio of the outside air does not exceed 65 grains of moisture per pound of dry air. The supporting calculations for this statement are as follows:

(1) Minimum dehumidification occurs at critical condition number 1. At this condition Appendix IV indicates that 3.7 grains of water vapour per lb. d.a. is removed. To calculate in terms of latent heat removal capacity for a 1600 cfm air system:

Substitute in equation 2

$$Q_L = \text{cfm}\,(.67)(\Delta W)$$

$$Q_L = (1600)(.67)(3.7) = 3970 \text{ Btu per hr.}$$

(2) The total internal latent heat load of the system is 2.5 grains of water vapour per lb. d.a. (section 3.1.3).

In terms of latent heat removal requirements, at 75°F the latent heat capacity per lb_{wv} = 1052.1 Btu per lb_{wv}.

40 ounces of water vapour per hr. = 2.5 lb_{wv} per hr.

$$= (2.5)(1052.1) = 2630 \text{ Btu per hr.}$$

The outside air latent heat load must not exceed the difference between the total latent heat capacity of the direct expansion coil and the internal latent heat load:-

3970 - 2630 = 1340 Btu per hr.

Substituting this value in equation 2 for a 4 per cent outside air intake of 64 cfm and solving for the maximum humidity ratio of outside air before changeover to the 100 per cent recirculation system is required:-

$$Q_L = \text{cfm}\ (.67)(W_{\text{outside}} - W_{\text{critical point 1 operating setting}})$$

$$1340 = (64)(.67)(W_{\text{outside}} - 38)$$

$$W_{\text{outside}} = 69.2$$

To take account of approximations, 65 grains wv/lb_m was established as the upper limit below which the system need not change over to 100 per cent recirculation (Section 3.5).

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