THE UNIVERSITY OF ADELAIDE

Department of Mechanical Engineering

A UNIFIED APPROACH TO THE DESIGN OF CLIMATE SIMULATORS

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Thesis for the Degree of Doctor of Philosophy

July 1979.

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STATEMENT

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

ALLAN SHAW

July, 1979

ACKNOWLEDGEMENTS

I wish to express my sincere thanks and gratitude to Professor R.E. Luxton for his supervision during the writing of this thesis. He has given much time to discussion and advice although he has had other very heavy commitments. Furthermore as Head of the Department he has helped obtain the equipment which was necessary for the project.

My thanks are also due to Professor H.H. Davis my former supervisor who provided the opportunity for building the basic prototype phytotron unit at the Waite Institute and for his support towards the new teaching and research system in the Department.

I also wish to thank Mr. D. Caldicott Senior Lecturer in Mechanical Engineering at the South Australian Institute of Technology for the time he so generously gave in discussion and his comments.

I wish to express my gratitude to members of the ancillary staff who worked untiringly as a team to help build test and maintain the equipment necessary to carry out the research, in particular to:

Mr. D. Kerr the laboratory manager for his technical advice and scheduling of the entire works programme.

Mr. P. Hall for his close cooperation in running and testing the new system.

Messrs. P. Walker, G. Osborne and P. Boult of the electronic workshop for their time and expertise.

I am grateful to Mr. M. Bethune for his excellent draftsmanship, to Ms. M. Lansley for her accurate typing and to Mr. J. Tomlinson who provided most of the photography.

I wish to thank Mr. R. Ellis of White Engineering for the high standard of workmanship in carrying out their part of the installation of the air and refrigeration systems of the Department of Mechanical Engineering System described in this thesis.

To Carrier Air Conditioning, Australia I wish to express my thanks for their donation of the dehumidifier.

My thanks to Ms. K. Leverett of the Barr Smith Library for her extensive assistance in obtaining publications.

Finally, I would like to thank my wife for her help with proofreading. Above all, I wish to thank her for her encouragement, assistance and continuous support.

LIST OF SYMBOLS

S1 Units are shown as a standard. Where there is an exception due to Imperial Units used in references this is stated.

A	Area	m^2
A_c	Exchanger minimum free flow area	m^2
A_f	Secondary external surface area (fins)	m^2
A_i	Inside pipe surface area	m^2
A_0	Total outside surface area sum of primary and fin surface areas	m^2
A_p	Primary surface area	m^2
ADP	Apparatus dew point average outside surface temperature of a dehumidifier	C
B	Ratio of A_0/A_i	dimensionless
BF	Bypass factor	dimensionless
BF_n	Bypass factor subletter 'n' refers to the number of rows in depth	dimensionless
c_p	Specific heat at constant pressure	kJ per (kg)(K)
c_{pm}	Specific heat at constant pressure of moist air	kJ per (kg)(K)
$c_{p\ell}$	Specific heat of liquid at constant pressure	kJ per (kg)(K)
c_{pv}	Specific heat of vapour at constant pressure	kJ per (kg)(K)
CSIRO	Commonwealth Scientific and Industrial Research Organization	-

D	Water vapour diffusivity	m^2 per s
D, d	Diameter	m or mm
D_e	Equivalent volumetric diameter	m
D_1	Wet bulb depression at entry condition	C
D_1	Outside pipe diameter (Figure 11.5)	mm
D_i	Inside pipe diameter (Figure 11.5)	mm
D_2	Wet bulb depression at leaving condition	C
D_2	Outside fin diameter (Figure 11.5)	mm
$\frac{D_2}{D_1}$	Wet bulb depression ratio	dimensionless
DBT, dbt	Dry bulb temperature	C
DPT, dpt	Dew point temperature	C
DX	Direct expansion coil	-
e	Entering condition (used as a prefix as for example edbt)	-
EPR	Constant pressure evaporator regulator	-
f_g	Coefficient of heat transmission through air surface film (Section 12.5)	Btu per (hr)(ft^2)(F)
f_R	Coefficient of heat transmission through refrigerant surface film (Section 12.5)	Btu per (hr)(ft^2)(F)
G_m	Mass velocity of moist air (V)(ρ)	kg per (s)(m^2)
G_r	Mass velocity of refrigerant (V)(ρ)	kg per (s)(m^2)

h	Specific enthalpy (Sections 4,5 and 6)	kJ per kg
h	Coefficient of heat transfer	W per $(m^2)(K)$
h_{cod}	Convection heat transfer coefficient for dry outside surface	W per $(m^2)(K)$
h_{cow}	Convection heat transfer coefficient for wetted outside surface	W per $(m^2)(K)$
h_{do}	Mass transfer coefficient for outside surface	kg per $(s)(m^2)$
h_i	Combined coefficient of heat transfer through water layer, metal and refrigerant film	W per $(m^2)(K)$
h_r	Enthalpy of refrigerant (Sections 4,5 and 6)	kJ per kg
h_r	Refrigerant heat transfer coefficient	W per kJ per kg
H	Distance between rows in direction of air flow (Figure 11.5)	mm
H	Specific enthalpy of air	kJ per kg
H_c	Specific enthalpy of air at surface of coil (Section 10)	kJ per kg
H_{fg}	Latent heat of vaporization	kJ per kg
ΔH_m	Log mean air enthalpy difference	kJ per kg
H_m	Log mean air enthalpy $(H_r + \Delta H_m)$	kJ per kg
H_r	Enthalpy of saturated air at refrigerant temperature	kJ per kg
H_s	Enthalpy of saturated air at water film temperature	kJ per kg
H_{sm}	Log mean enthalpy at wetted surface temperature	kJ per kg
ΔH_{sm}	Log mean enthalpy difference at wetted surface temperature	kJ per kg
HV	High velocity member of a pair of test runs where the velocity is twice that of the LV, low velocity member (Section 12)	-

J	Mechanical equivalent of heat = 778 (Equation 12)	ft lb_f per BTU
k	Unit thermal conductivity	W per (m)(K)
k_ℓ	Unit thermal conductivity for refrigerant	W per (m)(K)
ℓ	Leaving condition (used as a prefix as for example ℓdbt)	-
L	Heat exchanger flow length	m
L	Vertical distance between tubes (Figure 11.5)	mm
LV	Low velocity member of a pair of test runs where the velocity is half that of the HV, high velocity member (Section 12)	-
Le	Lewis number, $\left(\frac{\alpha}{D}\right)^{2/3}$	dimensionless
LSHE	Liquid suction heat exchanger	-
M	The Goodman (1936) symbol for Tie Line Slope (Section 11.5)	Btu per (lb)(F)
$\dot{m}$	mass flow rate	kg per s
$\dot{m}_a$	mass flow rate of air	kg per s
$\dot{m}_r$	mass flow rate of refrigerant	kg per s
N	Number of fins per inch (Section 11.7.2)	-
NATA	National Association of Testing Authority, Australia	-
P_r	Prandtl number for moist air $(\mu)(c_p)$ per k a fluid properties modulus	dimensionless
PH	Pressure enthalpy diagram	-

Q,q	Rate of heat transfer Refrigerant capacity	W
dq	Heat transferred through an incremental area	W
r_h	Hydraulic radius	m
$4r_h$	Flow passage hydraulic diameter (Figure 11.6)	feet
Re	Reynolds number for air $(D_e)(G_m)$ per μ	J per kg
RE	Refrigerating effect	kJ per kg
RH	Relative humidity	dimensionless
SEE	Society of Environmental Engineers	-
St	Stanton number, h_{cod} per $(G)(c_p)$ a heat transfer modulus	dimensionless
STDL	Surface Temperature Determination Line	-
t	temperature	C
t	fin thickness	mm
t^1	wet bulb temperature	C
t_i	temperature of air at surface of coil (Section 10)	C
t_r	Refrigerant temperature	C
t_s	Temperature of saturated moist air	C
t_{sm}	Log mean water film temperature obtained from psychrometric tables knowing H_{sm}	C
TLS	Tie Line Slope	kJ per (kg)(K)
TX	Thermostatic expansion valve	-

V	Velocity	m per s
V^1	Velocity	feet per minute
$V_{f,std}$	Face velocity of standard air approaching the coil (Section 11.7.5)	m per s
$V^1_{f,std}$	Face velocity of standard air approaching the coil (Section 11.7.5)	feet per minute
W,w	Humidity ratio	kg of water vapour per kg of dry air
W,w	Humidity ratio (Table 12.1)	g of water vapour per kg of dry air
WBT,wbt	Wet bulb temperature	C
X,x	vapour fraction or quality	-
α	Thermal diffusivity	m^2 per s
α	Heat transfer Area per Total volume (Figure 11.6)	dimensionless
μ	Dynamic viscosity	kg per (m)(s)
μ_ℓ	Dynamic viscosity of refrigerant	kg per (m)(s)
μ_ℓ	in Pierre equation 12, Section 11.7.7	lb_f per (ft)(hr)
ρ	Density	kg per m^3
σ	Free flow area per Frontal area (Figure 11.6)	dimensionless

1. SUMMARY

A climate simulator must have the capacity to produce not just a single set of controlled climatic conditions but must provide a whole range of climates.

The design approach takes two main directions. The first relates to those aspects of the problem which are peculiar to climate simulators. For example, there is a requirement to simulate *any* temperature and humidity combination within a climatic range and the requirement to change automatically from *one* combination of temperature and humidity to *any other* combination in the climatic range. Sections 3 to 6 inclusive present a unified system solution applicable to the total field of climate simulation with its wide variation of requirements within the physical and life sciences. There is a clear line drawn between this area and the design requirements of conventional air conditioning.

The climate simulator when viewed as a research system must offer the user a scientific tool of considerable sophistication. The user must be given full scope to apply the scientific method to his investigations. This necessarily calls for close tolerance performance and the maintenance or variation of climatic conditions to suite the purposes of the research project.

Facilities to study problems requiring control of climatic conditions are not easily obtainable. Many research workers requiring controlled thermal environments cannot proceed with their investigations due to the inability of existing systems to meet their standards and specifications. Recent years have seen a growing demand both in the physical and life sciences for two-variable, temperature-humidity controlled climate simulators, which will operate stably over a wide range. A search of the literature on the engineering aspects of climate simulation reveals discouraging recommendations. R.N. Morse

(1963) is of the opinion that one should not design for wide climatic range temperature-humidity systems. Another discouraging aspect is that, though there is a large demand, one user's specifications differ markedly from another, and depend on the area of interest and the degree of sophistication of the research program. The user may be a plant physiologist concerned with controlled experiments to study the relationship between the plant and its environment, or he may be an engineering scientist concerned with advanced studies in heat and mass transfer at evaporator surfaces. Through the unified approach that is presented in this thesis, one engineering system would be serving the broad spectrum of requirements for climatic simulation facilities in research and teaching programs.

The fundamentals of a basic engineering design (Shaw 1964) for climate simulation will be described. This design will be shown to be adaptable to the wide differences in the detailed specifications mentioned above. It can take the form of a low energy system, easily operable by non-engineers, and has the capacity to change over automatically between different temperature and humidity settings in accordance with some prescribed program. It is applicable to a wide range temperature-humidity system as well as to a narrow range temperature-only system. The system has the capacity to maintain its operating settings to close tolerances.

The second direction of the design approach relates to the selection of the dehumidifier. The approach used in the selection of the dehumidifier enables the operator to achieve a wider range including lower humidity ratios and reduces the energy requirements of the system. This very important component is concerned with the simultaneous role of offsetting both the sensible and latent heat loads.

A new approach is developed that affects not only climate simulation but also the field of air conditioning. Sections 10 and 11 study in depth the heat and mass transfer relationships at the dehumidifier and a new method of selection is described. The empirical work associated with Sections 10 and 11 was carried out at the Waite Institute phytotron unit designed by the writer. Section 12 of this thesis is concerned with the empirical confirmation of some of the findings described in Sections 10 and 11. This work was carried out in the new (February 1979) Heat and Mass Transfer Research and Teaching Laboratory located in the Department of Mechanical Engineering of the University of Adelaide.

2. THE NEED

2.1 In Agricultural Research

It is particularly in the area of agricultural research and the biological sciences that the need for climate simulators exists. In contrast to research in the physical sciences, where it has generally been possible to organize research environments within precisely known and controlled conditions, the life scientist has had to work largely with natural or imperfectly controlled (temperature-only) environmental conditions.

It has been very difficult to provide and control the proper environment for a plant. High light intensity is essential where mature green plants are investigated and this must be provided in a completely defined environment including control of temperature and humidity. In order to take in a wide variety of plant types over all seasons simulation of a wide variety of climatic conditions is important. In 1962 this need was reflected in the convening of an International Conference in Melbourne by the C.S.I.R.O. (Commonwealth Scientific and Industrial Research Organization) on the topic 'Engineering Aspects of Environment Control for Plant Growth'.

Following this conference the Waite Agricultural Research Institute of the University of Adelaide requested the writer to build a temperature-humidity phytotron unit to meet their specifications. This basic system (Shaw 1964) preceded the unified approach taken in this paper.

2.1.1 Controlled relative humidity

Relative humidity plays a very important role in determining plant growth and development. A study (Krizek et al 1971) on the seedling growth of three hybrid annuals grown at three levels of relative humidity, 40, 65 and 90 per cent reveals dramatic variations in height, fresh and dry weight of tops, and leaf area. This study also shows a very significant interaction between relative humidity and type of container used. Table 2.1 is reproduced below from this paper.

TABLE 2.1

Effects of Relative Humidity and Type of Container on the Growth of Ageratum Seedlings After 14 Days of Controlled Environment Treatment

Environmental Treatment	No. of Nodes	Height Main Shoot (mg)	Fresh weight Tops (mg)	% Dry Weight	Total Leaf Area (cm^2)
40% RH					
clay pot	3.7	11.5	121.0	11.0	4.3
plastic pot	3.8	12.8	149.3	10.7	5.5
65% RH					
clay pot	4.5	20.3	604.3	9.3	18.2
plastic pot	4.0	18.7	402.3	9.0	13.2
90% RH					
clay pot	5.0	22.7	785.3	9.0	25.5
plastic pot	4.7	26.2	486.0	9.0	18.8

Ageratum seedlings grown at 90 per cent relative humidity in a clay pot have 1.4 times more leaf area than if grown in a plastic pot. The same plant grown at 65 per cent relative humidity again has about 1.4 times more leaf area in the clay pot plants. However, a plant grown at 40 per cent relative humidity in a clay pot has 1.3 times less leaf area than if grown in a plastic pot.

Reversal of behaviour in studies with clay and plastic pots have also been found to occur because of a change of mean outside night temperatures from winter conditions of 1.5C to summer conditions of 10.7C.

The fresh weight and dry weight of petunias were significantly greater in plastic pots at every level of humidity, while marigold seedlings were unaffected by the type of containers used. Usually relative humidities below 65 per cent were associated with a drastic curtailment of growth.

In this experiment the greatest variation existed when ageratum seedlings grown at 90 per cent relative humidity in a clay pot had 5.9 more leaf area than when grown at 40 per cent relative humidity in a clay pot.

These findings point to the complexities arising from the interaction of numerous variables and the need for controlling them for a better understanding of plant growth.

The study concludes as follows:

"Although to date many investigators using plant growth chambers have considered the atmospheric moisture content to be of little importance, our findings suggest: (1) that more attention should be given to designing controlled environment systems with adequate humidity control; and (2) that greater care should be given in designing environments to assure adequate levels of atmospheric moisture."

2.1.2 Identification of critical climatic variables through climate simulation

Crop plants are not usually grown in habitats to which they are naturally suited. As a result there may be problems arising such as reduced yield, fruit drop and loss of fertility. Under controlled environment conditions, one can carry out research to identify and isolate the climatic variables responsible for the problem, determine the time in the plant's life when the problem arises, study the mechanism responsible and develop preventive treatment.

Under controlled conditions one can obtain reliable and reproducible information on the limits of tolerance of the plant to extremes of climate. Under controlled conditions those factors most conducive to optimal growth and productivity can more readily be determined.

2.1.3 Physiological factors and their relationship to the environment

Controlled experiments are the prime means of establishing the factors and mechanisms which determine the adaptation of the plant to its environment. The critical factors which govern the processes of germination, growth, reproduction, respiration, nutrition, fruit and seed formation, ripening, dormancy, death, etc., are fundamental physiological problems which, because of their inherent variability can best be studied in controlled environments. The more complex the research and the problems it aims to solve, the more important becomes the control of the environment and the separation and manipulation of its components.

2.1.4 Expansion of the research field through climate simulation

In uncontrolled green-houses it is not always possible to achieve conclusive results. There are numerous examples of opposite conclusions being reached in the repetition of a particular experiment.

A.I. Virtanen (1938) working at the Biochemical Institute, Helsinki, found that leguminous roots excreted nitrogenous compounds into the root medium when infected with root nodule bacteria during long day periods of the year. P.W. Wilson of the University of Wisconsin, was unable to reproduce these results. The reason for this was that the long day summer weather was not cold enough for this behaviour to occur in Wisconsin. Wilson (p.160, 1940) gives an account of this discrepancy in the work of two investigators.

F.W. Went (1961a) the director of one of the first Phytotron facilities in the U.S.A., the Earhart Plant Research Laboratory at the California Institute of Technology, Pasadena, California, enumerates long lists of important research projects that could only have been carried out successfully in a phytotron. Temperature, light, wind, length of day, length of night and thermoperiodicity, (the phenomena associated with plants requiring higher day and lower night temperatures for maximum growth and development), are variables which when selectively controlled extend the area of plant research.

2.1.5 Economic viability of phytotron units

Controlled environment chambers for plant growth are of course more expensive than greenhouses which are themselves not inexpensive and must therefore account for their additional cost.

One study at the Earhart Laboratories of the California Institute of Technology highlights the importance of artificially lighted climate simulators in reducing the space and time requirements for research into plants. The uniformity of the shape and size of plants can be correlated to both the plant's genetic constitution and the conditions of the environment in which it is raised. The more reproducible the genetic constitution and the environmental conditions under which growth takes place, the more uniform plants should be. The agronomist assigns a "coefficient of variability", (Went 1957), (Went 1961b), that is, a percentile of those plants that varied from a uniform standard. For plants raised in a conventional greenhouse the coefficient of variability was 20%, for those raised in an air conditioned greenhouse it was 10% and for those raised in an artificially lighted room at constant temperature is was 5%. In assessing the variation from a standard, a square relationship applies. Therefore, since the coefficient of variability of plants raised in an air conditioned greenhouse is one half of those raised in a conventional greenhouse, only one quarter of the number of plants is needed for the results to have the same significance. The coefficient of variability of plants raised in an artificially lighted chamber at constant temperature is one quarter of those raised in a conventional greenhouse. Therefore, only one sixteenth of the number of plants is needed for the results to have the same significance (F.W. Went 1961b). Thus, far fewer plants are necessary to establish the same facts. It would be of interest to know whether the number of plants needed would have been still further reduced had the room also been equipped to maintain humidity constant. The unified design approach that is presented in this thesis permits close control of this additional variable at a minimal additional cost.

Under artificial climates the worker is not constrained by the seasons. Four generations of tomatoes can be raised or tested four times faster when compared with an ordinary program.

The conclusion is obvious. A phytotron which can simulate climate over a wide range requires less space, less time, improves the efficiency of the research worker, increases the significance of the project and avoids much wasted work. In certain areas it is a necessity, no matter what the cost. When used properly it can be economically viable.

2.2 The Use of a Controlled Environment Laboratory as a Testing Facility

The nature of the design solution used in this unified approach enables the system to serve as a testing laboratory where control of temperature and/or humidity is required. These two properties can be maintained fixed or varied to some programmed pattern depending on the control system selected. Where the testing facility is used in the physical sciences, the system design requirements are less demanding than for the life sciences. Usually there is no need for a fresh air supply to support photosynthesis or respiration. The requirement for intense lighting to simulate sunlight is usually eliminated. As a consequence these systems cost less and require less power to operate.

2.3 The Use of a Controlled Environment Laboratory as a Teaching and Research System in the Physical Sciences

This unified approach includes the application of the design as a teaching aid for science courses. It leads to laboratory exercises in thermodynamics, air conditioning, refrigeration, psychrometrics, two phase flow, steady flow cycles, first and second law experiments, and heat and mass transfer. This use has not been an accidental side effect to the design but part of the original specifications to which the first prototype of the system was built.

When the Waite Agricultural Research Institute approached the Department of Mechanical Engineering for the design of a phytotron unit, it was found, as the design developed, that the same system could be valuable as a teaching laboratory for Mechanical Engineering. The specifications for both the phytotron unit and teaching laboratory are surprisingly similar. It was therefore decided to enter into a joint venture, where both the Waite Agricultural Research Institute and the Mechanical Engineering Department shared expenses remaining after grants and contributions. During the past six years the Waite system has been used extensively for laboratory experiments and research projects in thermodynamics. A detailed description of several of these experiments will be given in Sections 8, 11 and 12.

2.4 The Use of a Controlled Environment in Building Science Research

There has been a keen interest in the building and architectural areas for a greater understanding of the materials and systems used in building design. (Shaw pp.83-92, 1978). A controlled chamber can serve as an important research tool in reducing the capital and energy costs of heating and cooling buildings. Such a facility can be used to determine the thermal and moisture transfer properties of structural materials. With this system one could demonstrate heat and mass transfer through simple and compound walls. The system could be used to investigate temperature gradients, film coefficients, vapour barriers, the effectiveness of caulking compounds, the effect of building porosity and the moisture holding capacity of building materials.

If a lighting section is incorporated in the design to simulate sunlight many additional studies of interest can be included such as the effect of altitude and azimuth angles on the thermal properties of building structures, the effect of colour and the effect of haze conditions on solar transmission.

3. DETERMINATION OF THE FUNDAMENTAL CRITERIA

What is needed in the field of climate simulation is a single fundamental system of design which is readily adaptable to accommodate the wide variations associated with the needs of the different users and which is suitable for mass production. The alternative, custom built designs are necessarily expensive. In addition they often do not perform satisfactorily for reasons which are discussed in Sections 3.2, 10 and 11. It is against this background that a unified approach to the design of climate simulators is presented here.

An important factor in determining the applicability of a unified system of design is that there must be a lowering of cost and energy consumption commensurate with the reduction of the complexity of the users' specifications.

3.1 Use of a Phytotron Unit to Determine the Design Requirements for the Most Demanding Situation

In surveying the need for climate simulation in Section 2, the design requirements for systems serving plant research were highlighted.

This is not only because the plant scientists are the greatest users of this type of facility. It is also because their specifications are the most difficult to meet due to the large variations within this group. The life scientists' requirements vary in complexity from a temperature-only controlled system to a sophisticated temperature-humidity controlled system. Variations in range, size, control, lighting and air flow patterns highlight the need for a flexible system of design. This need plays an important part in forming the basis of the unified approach.

3.2 Design Requirements for Climate Simulation in Contrast with those for Conventional Air Conditioning

It is a mistake to attempt to design a climate simulator by applying existing air conditioning solutions. Such an approach is likely to lead to failure. Before presenting the fundamental criteria of the unified approach, the differences in requirements of a climate simulator with that of conventional air conditioning will be enumerated.

In developing this section more than simply a contrast with air conditioning objectives emerges. This analysis is also directed towards the design solution.

3.2.1 Range and tolerance

In the design of conventional air conditioning systems it is necessary to offset the sensible and latent heat additions for a particular desired operating condition having a fairly broad tolerance (see Section 3.2.5, Effective temperature). In a climate simulator the requirement is far more demanding because there is not one particular interior space condition, but many over a range, a range of temperatures and a range of humidities in any combination representing a broad spectrum of outside weather conditions. There is an inter-relationship between the heat loads and the dehumidifier performance due to the range. It is possible to have a particular operating point in the range where a high sensible heat load for a given dehumidifier is very easily offset in one section of the range, and a fairly low sensible heat load, which is impossible to offset in another section of the range. (See Section 5.2.2 and Figure 5.1).

This also holds true for the latent heat load. Fortunately the critical operating points at which the respective sensible and latent heat loads will be most difficult to offset occur at different conditions. Were this not the case the design that will be presented would include a larger energy penalty than it does.

The system must satisfactorily reach any selected operating settings from start-up. If the system is in operation, then it must permit changeover automatically from one set of operating conditions to any other set of operating conditions.

It is this range requirement which reveals the greatest difference when the system is compared with conventional air conditioning design specifications.

3.2.2 Space considerations

In conventional air-conditioning design practice, during the summer season air is introduced into the conditioned space at 11C to 14C below the design temperature. This is possible due to the relatively large volume of space, about 1 metre above the heads of the occupants, where the air can be introduced through ceiling diffusers, plenums, side registers etc., so as to reach the design temperature within an acceptable tolerance in the occupied section of the space.

In the case of phytotron units, the location of the heat load presents a varying problem that is dependent on the size of the plants and their foliage, and the design of the system of air distribution. With seedlings and small leaved plants the maximum source of sensible heat from the artificial lighting would be at the floor of the chamber; with fully grown plants having dense foliage it would be at the top of the leaves. In the former case, if air is introduced from below a large part of the heat load would be offset at the very entrance of the air to the chamber, while in the latter case the temperature would be below the design setting until the air reached the top of the leaves.

Thus it can be seen that in the design for climate simulation air should be introduced into the space at a temperature very nearly equal to the desired design setting to assure a small temperature gradient. (See Section 5.2.3).

3.2.3 Dehumidifier selection methods

The methods used by manufacturers to select dehumidifiers for conventional air conditioning are not applicable to the field of climate simulation. (Sections 10 and 11 treat in depth dehumidifier selection).

3.2.3(a) Enthalpy difference across dehumidifier

Closely related to space considerations and temperature gradient discussed in section 3.2.2 is the selection of the dehumidifier. It is desirable that the enthalpy difference across the dehumidifier be small, corresponding to the small temperature gradient.

In conventional air conditioning there is a large enthalpy change and the air is cooled close to saturation. The conventional methods used in dehumidifier selection adopt numerous simplifying approximations (see Section 10) which are applicable to the near saturated condition at the outlet of the dehumidifier. It is assumed that the average path of the air stream through the dehumidifier follows a straight line when drawn on a psychrometric chart. There is no provision for considering the true nature of the curved path of the air stream, which becomes relevant when small enthalpy differences across the dehumidifier occur. Thus, the very selection of the extended surface cannot rely on conventional practice.

3.2.3(b) Humidity operating settings near saturation

Very frequently it is desired to set the climate simulator at a high relative humidity. For example, in phytotron units it is a preferred condition for obtaining improved growth rate for many plants. (See section 2.1.1.) Those manufacturers of dehumidifiers using the wet bulb depression ratio in coil selection assume that the saturation curve of the conventional psychrometric chart is a straight line. This assumption is a fair approximation for entering conditions to the dehumidifier which are associated with relative humidities for human comfort. However a large error is introduced when the entering condition is at a high relative humidity.

3.2.4 Change in air volumes

The volumes required in conventional air conditioning will vary from about 6 to a maximum of 25 air changes an hour. In climate simulation it is a very different order of magnitude, 400 to 1000 air changes an hour. This is dictated mainly by the requirements of the plants for an air velocity similar to that of outside air movement. It is also compatible with the greater air flow rates necessary to obtain low temperature changes across the dehumidifier (see Section 3.2.2).

3.2.5 Effective temperature

In conventional air conditioning it is not usually important to maintain precisely the set point of a particular temperature and humidity as is the case in climate simulation. In comfort air conditioning the aim is to achieve an effective temperature. This indicates a range of temperature and humidity conditions for evaluating degrees of human comfort. Maximum comfort can be achieved at numerous combinations of temperature and humidity. Therefore the coil load ratio line slope is not as critical as when designing for climate simulation. For example, a comfortable room would be acceptable at a lower than design dry bulb temperature combined with a higher than design humidity ratio.

3.2.6 Sensible heat load

In conventional air conditioning practice the sensible heat load per square metre of floor area being treated is very low when compared with a phytotron unit, particularly if high intensity lighting to simulate sunlight is employed without a barrier between the lighting section and the plants. Under such conditions the sensible heat load per unit floor area may be as much as 50 times greater.

3.2.7 Continuity of operation

The capacity of a system to operate continuously with automatic changeover to any desired program is an essential requirement to meet any long range research or testing program. In a phytotron unit a particular project may call for weeks and even months of continuous operation, with automatic changeover according to a desired program of climate simulation.

A conventional air conditioning system would rarely be called on to operate under such demanding conditions.

This requirement carries with it several additional specifications. For example, where continuity of operation to close tolerances is imperative, a vapour compression cycle must avoid frosting and the associated defrost period. Furthermore the quality of the components must be excellent.

3.2.8 Operation, maintenance and service

The complexity of a sophisticated arrangement to meet all the design requirements will often be associated with a single reach-in cabinet. The operator of a phytotron unit is unlikely to understand the engineering system which serves him; he may not have a technical staff to set up and adjust a system that is built to meet complex design requirements. The system may be located in a country where up to date maintenance and service agencies are not available.

Apart from small residential air conditioning units which have a vast service organization behind them, a conventional central air conditioning system invariably is provided with skilled technical staff and the latest systems includes modern instrument and control consoles. In the environmental control field there are only a few centres such as the Dimock Laboratory at Cornell University in Ithaca, New York which can afford the employment of a separate division of skilled technicians to serve the community of research workers using these facilities.

Therefore it is necessary to have a system which is very simple to operate and using known components that are easily maintained and serviced. Preferably the operator should only be required to set his desired temperature and humidity conditions and his programme of changeover periods. Apart from minor corrections for drift occurring over long periods, no further adjustments should be necessary.

3.2.9 Load constancy

As indicated above in Section 3.2 to 3.2.6 usually a more difficult set of specifications must be met in climate simulation. However, in this section which considers load constancy the reverse is true. It should be noted that for artificially lighted climate simulators, phytotron units, in particular, (naturally lighted chambers would be an exception), the control problem is very much simplified. Though the heat loads (from lighting) and moisture loads may be very high, there is little change in these loads. On the other hand in air conditioning heat and moisture loads can vary considerably. Transmission and solar loads in particular can rise and fall very rapidly. This load constancy is one characteristic that the climate simulator design should use to advantage to obtain stable operating conditions.

3.3 The Fundamental Criteria Expressed as a Set of Specifications

It is the purpose of this thesis to present a simple unified system which can competitively satisfy a portion of the broad research testing and teaching requirements for controlled thermal environment facilities in its many varied forms.

In Summary - the aims of the unified system may be expressed as a set of specifications as follows:-

(1) *System must be designed to maintain its temperature and humidity operating conditions to close tolerances and with low temperature gradients.*

(2) *System shall have the capacity to automatically changeover temperature and humidity operating settings according to some prescribed program.*

(3) *System must have the capacity to offset all sensible and latent heat loads.*

(4) *System must have the capability to operate over a wide range of temperature and humidity conditions, above and below ambient conditions.*

(5) *System shall have the capacity to function continuously over long periods of usage. Momentary interruptions such as are caused by defrost cycles are not admissible.*

(6) *System must be easily operable by non-engineers.*

(7) *System must be easily maintained and serviced by commercial refrigeration servicing agencies.*

(8) *System engineering components to occupy minimal space.*

(9) *System must be amenable to modification and adjustment so that it could competitively satisfy the widely different specifications of the research community using these facilities while maintaining a uniformity that would permit a mass production approach by the manufacturer.*

(10) *System must be designed to fail safe.*

(11) *System to be inexpensive in capital costs.*

(12) *System to have low energy requirements.*

4. THE DESIGN SOLUTION

4.1 The Basic System - Introduction

The basic design solution was reached after considering many factors, some of which have been mentioned in Section 3 above. It followed from:

1. the specifications of the various users of climate simulators,
2. the wide variations that existed among these users' specifications,
3. the information gained by contrasting the design problems of climate simulation and conventional air conditioning specifications,
4. the performance of existing climate simulator systems,
5. a study of the existing literature on the engineering aspects of climate control,
6. a study of the patent literature,
7. a study of heat and mass transfer performance of direct expansion evaporators over small temperature differences. (See Section 11).

4.2 The Basic System - The Thermodynamic Design Aspects

4.2.1 Arrangement of the basic system

Consider a conventional vapour compression refrigeration cycle operating continuously, using a thermostatic expansion valve and having a fixed condensing temperature and a constant air flow rate. The cycle employs a single direct expansion coil with a selected surface whose performance can be determined. A schematic diagram of the system is as shown in Figure 4.1.

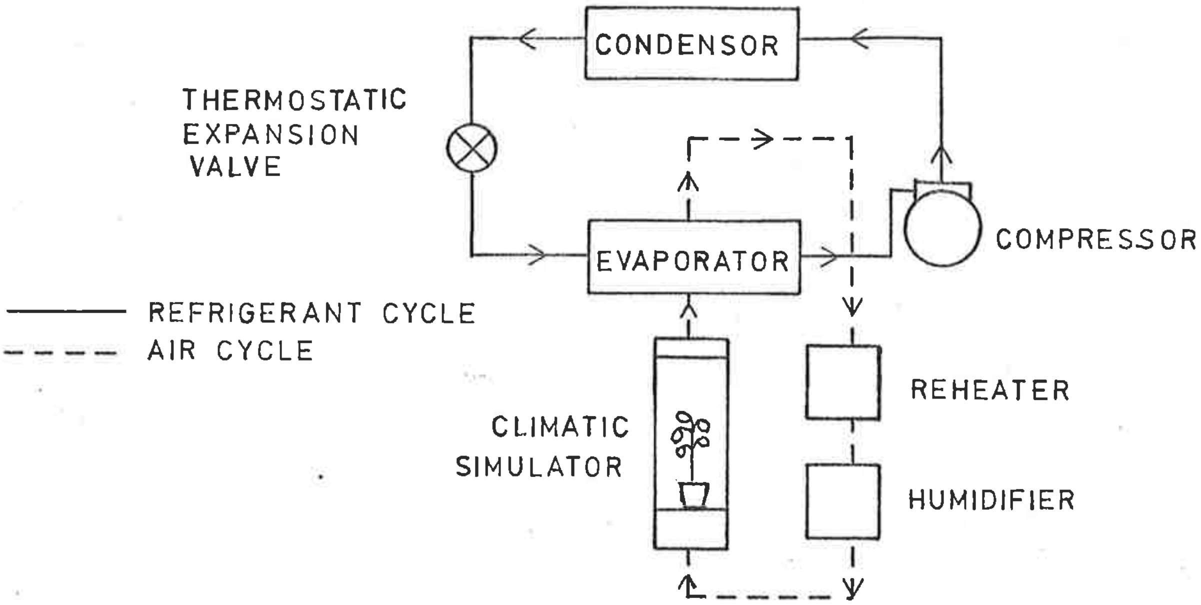


Fig. 4.1 Schematic Diagram of Basic System

Assume some specified climatic range to be the shaded area indicated on a psychrometric chart. This range may cover a dry bulb temperature from about 4 C to 43 C and a humidity ratio of from 0.005 to 0.025. Assume this area is divided by five equally spaced constant enthalpy lines as indicated in Figure 4.2 on a psychrometric chart.

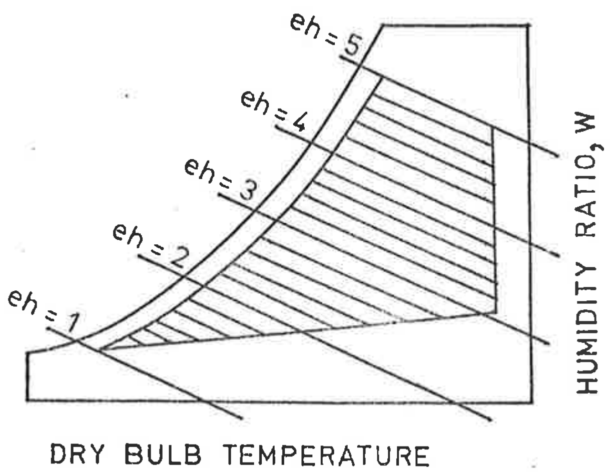


Fig. 4.2 Climatic Range

4.2.2 Inter-relationship between evaporator and condensing unit

Now, in the context of the constraints set forth above assume that air enters the direct expansion coil at an enthalpy of "3". Figure 4.3 shows the condensing unit and the direct expansion coil performance curves on a capacity versus suction temperature, pressure diagram.

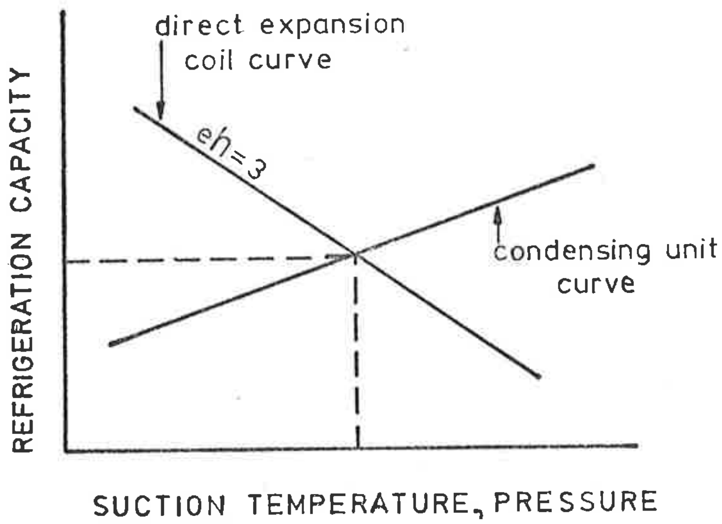


Fig. 4.3 Balance Point for Two Steady Flow Streams

It may be of interest to examine the relationship between these two curves. The condensing unit curve slopes up to the right and its left extremity has a very reduced refrigeration capacity, attributable mainly to the low evaporator pressures. The density of the vapour that is drawn into the cylinders of the compressor is at a minimum and this is associated with a minimum mass flow rate of refrigerant. On the other hand, the highest refrigerant capacity occurs at the upper right end of the condensing unit curve where the refrigerant density is at a maximum.

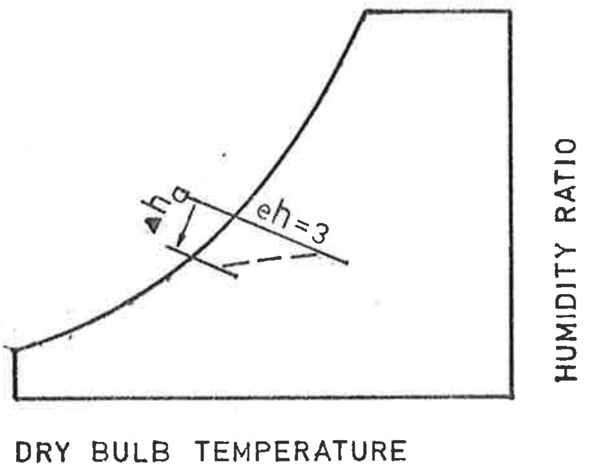
While there is a decrease in refrigerant capacity of a condensing unit with reduced suction temperatures, the opposite is true for the refrigeration capacity of an evaporator. In this case the refrigeration capacity increases with falling suction temperatures. This is to be expected. For a given entering energy level of the air stream to a cooling coil per unit flow rate, the lower the temperature of the refrigerant, the greater will be the heat and mass transferred from the air stream to the evaporator. On the other hand, for the same entering energy level to the cooling coil at maximum refrigerant temperature, the mean enthalpy potential difference will be at its minimum. Thus the evaporator performance curve moves down to the right.

By drawing the condenser unit curve and the evaporator performance curve on the same refrigeration versus evaporator temperature and pressure diagram it is apparent that both curves are satisfied at their point of intersection. This determines the operating point.

The intersection of the condensing unit curve with the curve at the direct expansion coil would result in a simultaneous solution which would represent the balance point for the air and refrigerant steady flow streams. The heat and mass transfer from the air stream at entering enthalpy "3" can be equated to the heat transferred to the refrigerant.

With reference to Figures 4.4 and 4.5,

$$\dot{m}_a \Delta h_a = \dot{m}_r \Delta h_r$$



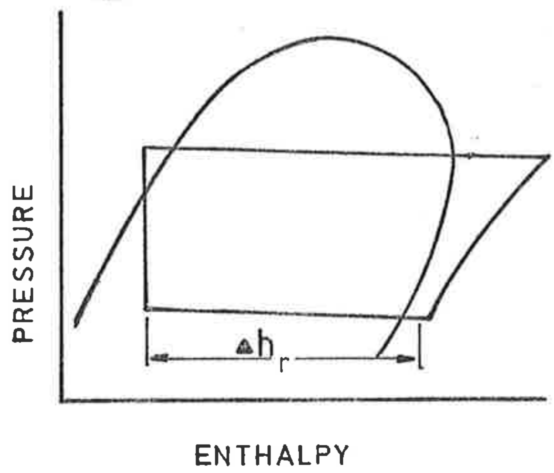


Fig. 4.4 Entry and Leaving Conditions of Air at Direct Expansion Coil.

Fig. 4.5 Refrigerant Cycle with Refrigerating Effect $= \Delta h_r$

Now, as indicated in Figure 4.6, this same process will be repeated for all of the five entering enthalpy conditions, still maintaining the same condensing temperature, the same air flow rate and the same components in the refrigeration cycle.

The results on the refrigeration capacity vs suction temperature, pressure diagram would indicate five fixed refrigeration capacities each related to five fixed suction temperatures and pressures and five fixed balance points and the dry bulb and dew point temperatures of the operating settings which combined, represent these enthalpies.

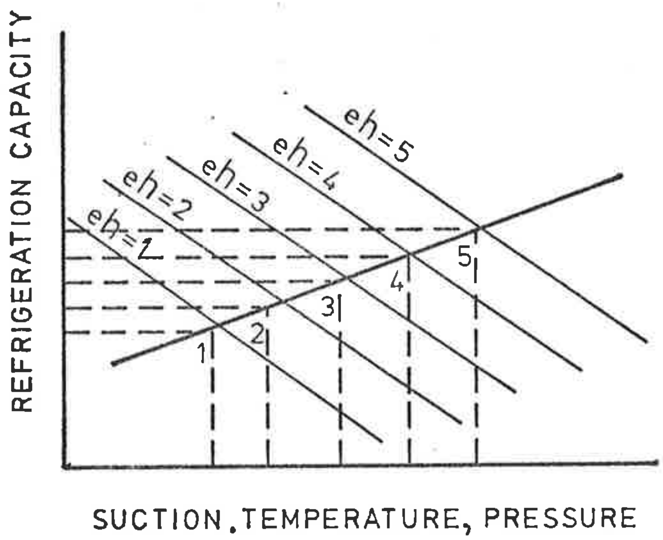


Fig. 4.6 Balance Points for Five Different Entry Air Enthalpy Conditions

4.2.3 Completing the basic system - addition of controls

Up to this point only one component having a control action has been indicated, the totally self-contained performance of a thermostatic expansion value.

Now consider that the system is required to cover a range as depicted on the psychrometric chart of Figure 4.1 with a specific enthalpy range from h'1' to h'5'. To this system is added a heater and a humidifier in the air stream downstream of the direct expansion coil.

Two proportional controllers with sensing elements within the chamber of the climate simulator act to maintain a fixed dry bulb temperature and a fixed dew point temperature by the addition of heat to a reheater and to a humidifier respectively. Consider five different chamber operating settings of temperature and humidity.

Each combination has a different corresponding enthalpy of '1', '2', '3', '4' and '5' at the exit condition of the test chamber. In each case precise control would be obtained, provided the direct expansion coil was selected so that in all cases the dry bulb and the dew point temperatures of the air leaving the coil are found to be sufficiently below the operating set points to allow for the offsetting of all sensible and latent heat loads. In each case the controllers add heat and moisture to the air stream so that the dry bulb and dew point temperature operating settings are reached. When this occurs the respective entering enthalpy condition would also be reached.

This system is a self realizing one. On start up or change-over, the dry bulb and dew point temperature controllers begin to act when the refrigeration cycle has cooled and dehumidified the air stream to a point below the respective operating settings. When this occurs, the combination of the sensible heat load (provided there is one) plus the sensible heat added by the dry bulb temperature controller in the reheater, plus the latent heat load and the water vapour introduced into the air stream by the dew point temperature controller together reach the fixed entering enthalpy corresponding to the selected combination of operating settings ('1' to '5').

Thus the refrigeration cycle is driven to a steady flow state having a fixed suction temperature and a fixed capacity as the controllers add heat and moisture to satisfy the desired operating settings.

Now assume a load change takes place within the controlled air section so as to increase the sensible heat load. This occurrence would have no effect on the steady flow nature of the refrigeration cycle since the dry bulb temperature controller would act to reduce the sensible heat addition from the reheater an amount equivalent to this increase. Thus both the refrigeration capacity and suction temperature would remain fixed.

4.3 Basic System Design Objectives

Described above is the system of wide range temperature - humidity control that is basic to the unified approach of this thesis. An assessment of the system in achieving the design goals is given below.

4.3.1 Simplicity

The system possesses a unique simplicity.

1. It requires only one evaporator serving both the sensible cooling and dehumidifying functions.
2. There is only one air circuit.
3. There is only one thermostatic expansion valve in the refrigeration cycle and its self contained action is automatic.
4. The basic refrigeration cycle is the simplest possible example of a vapour compression cycle.
5. The temperature and humidity controllers act only in one direction. Control is brought about by the addition of sensible and latent heat to an air stream.
6. There are no controls (apart from conventional thermostatic expansion valve performance) on the cooling or dehumidifying processes.

4.3.2 Energy level and the design solution

Stable performance, range, steady state and a fixed energy datum level established at the evaporator are all inter-related.

In the design described above for a simple basic system, the focus was on range. The aim was to find an engineering system utilizing a direct expansion coil and a commercially available condensing unit which at any dry bulb temperature and at any humidity (within a climatic range) would reach steady state or near steady state conditions for both the refrigeration cycle and the air stream. The system described above does this very thing. Stable, close tolerance temperature and humidity is obtained with dry bulb and dew point temperature controllers adding heat and humidity to a fixed energy datum established by the refrigeration system. Though the control system can be selected to handle load change, as would be the case for a naturally lighted climate simulator, when there is little load change, particularly stable operating conditions are obtained. The control system in the phytotron unit functions mainly during start-up and changeover. It plays a minor role once steady state and the desired operating settings are reached in a system with little load change. For example in the case of the Waite Phytotron Unit, if the system after reaching the desired set points were switched from automatic to manual control, the desired conditions would hold for hours with only minor drift associated with small load changes. This highlights the importance of range in the design.

The stable performance of this system is dependent on the establishment of a fixed energy level at the evaporator surfaces.

4.3.3 Criteria to establish a fixed energy datum

4.3.3(a) Entering enthalpy of moist air and mass flow of air stream

These properties are kept constant in the simple, single, circuit air stream of the basic system. Once steady flow conditions for a particular operating set of temperature and humidity are reached they do not vary.

4.3.3(b) Constant condensing temperature/pressure

Variation in condensing temperature is not as critical in affecting refrigeration capacity as variation in evaporator temperature. When the change in condensing temperature is very gradual, close tolerance performance would not be affected and it is possible to omit condensing temperature or pressure control. When air condensing is employed in wide range systems some form of control is advisable. (See Section 4.3.3(c)).

4.3.3(c) No intermission in system performance permissible

The design centers on establishing steady state conditions. The use of proportional rather than on-off control is therefore preferable. The occurrence of frosting would be counter productive to the design aims. Frost would change the performance of the evaporator surfaces and the required corrective action using a defrost cycle in any form would further upset the system performance. In the basic system design it may be very difficult to prevent frosting where air condensers are employed without control of condensing temperature. This could be responsible for frosting during low load operating conditions and ambient conditions improving condensing unit performance. It can be demonstrated on a refrigeration capacity versus evaporator temperature plot showing the condensing unit and the evaporator performance for a summer night condition operating at a point just above frosting. (See solid line curves in Figure 4.7 below).

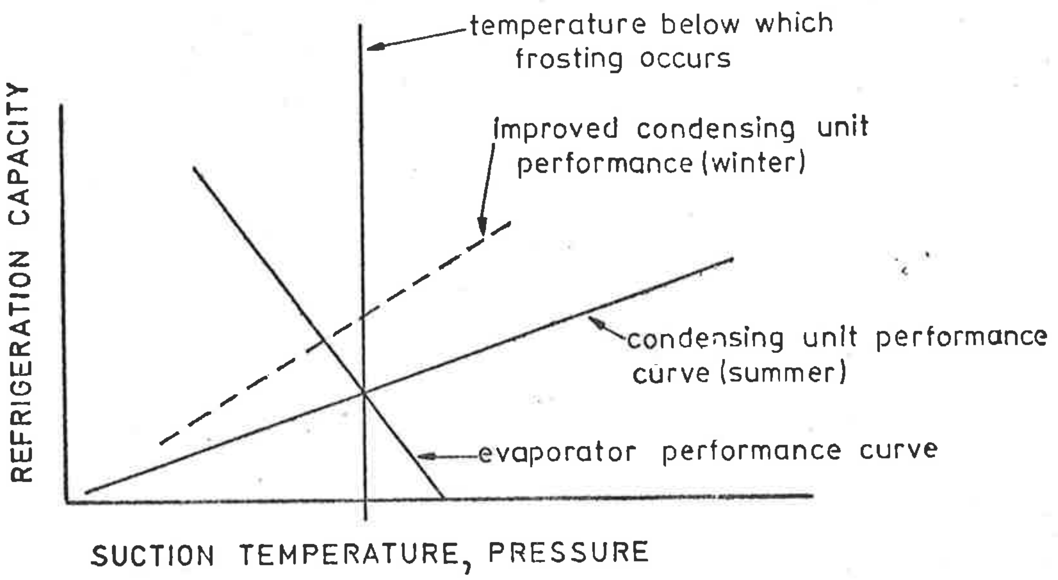


Fig. 4.7 Effect of Improved Condensing Unit Performance

A cool winter night would affect the condensing unit curve so as to improve its performance as per the dashed condensing unit curve drawn above. As a result the new operating point would be a lower temperature than the minimum permissible and frosting would occur. Several systems to maintain a constant condensing temperature will be discussed for both air and water condensing.

One characteristic of the basic system is that the evaporator has less capacity to dehumidify for low humidity ratio entering air conditions. For any given entering dry bulb temperature, as for example conditions 1, 2 and 3 of Figure 4.8, there is a progressively shallower slope to the coil condition curve as the humidity ratio is lowered.

For a given humidity ratio as for example conditions 4, 5 and 6 of Figure 4.8 there is also a progressively shallower slope to the coil condition curve as the dry bulb temperature is increased. During the lowest humidity ratios of a range, dry coil performance is approached.

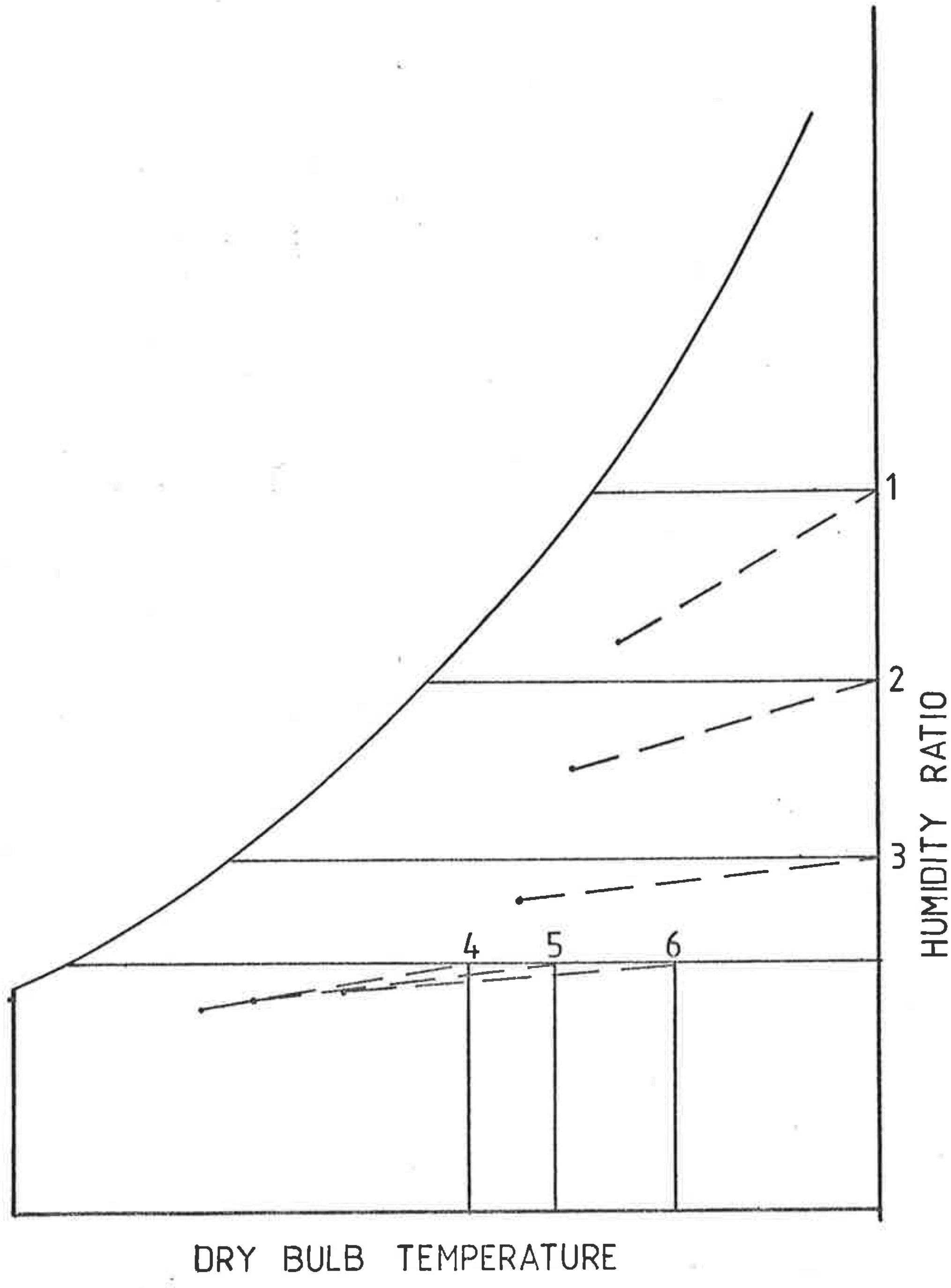


Fig. 4.8 Comparison of Various Entering Conditions to Evaporator

When low humidity ratios are included in the range it is necessary to sensibly cool the air more in order to offset the sensible loads. This too could cause frosting on the evaporator surfaces.

4.3.3(c)1 The preheater

Since there is excess sensible cooling, one of a number of ways of avoiding frosting is to preheat the air before it enters the evaporator to a point which would prevent the occurrence of frosting. This preheated air does not serve as an additional energy penalty. The penalty is already there in the performance of the reheater. Any heat that the preheater introduces will relieve the reheater of an equivalent amount. The preheater (or some equivalent alternative) is an important part of the basic system, since by preventing frosting it preserves the steady flow state which is essential to this design solution. (For alternative solutions to prevent frosting and still preserve the steady flow state, see Sections 6.2 to 6.8).

4.3.3(c)2 Constant pressure evaporator regulator

A refrigerant constant pressure evaporator regulator, E.P.R., can be used with the basic and modified systems to prevent frosting. It dispenses with the use of a preheater in a manner which does not disturb the constancy of the energy datum. The valve action can best be described with the aid of the refrigeration capacity versus saturated suction pressure diagram of Figure 4.9. Assume that the intersection of the evaporator performance curve, marked $\dot{m}_a eh_a = 1$, with the condensing unit curve at design conditions is located at a minimum evaporator temperature, below which, frosting would begin.

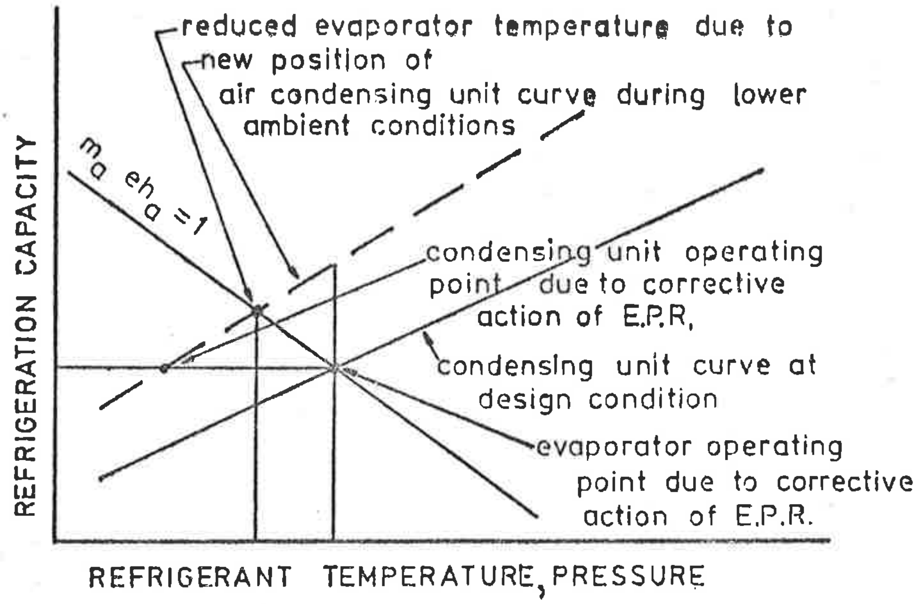


Fig. 4.9 Performance of Constant Pressure Evaporator Regulator, E.P.R.

As indicated under Section 4.3.3, for the basic system with the absence of a constant pressure evaporator regulator on a cool night the air condenser performance improves to a condition that is represented by the dashed line. Thus without the regulator the new intersection of the condensing unit curve with the evaporator performance curve would have been located at a lower refrigerant temperature and a higher refrigerant capacity. Frosting would result if some corrective action were not taken.

The evaporator pressure regulator valve seat moves towards closing the inlet refrigerant flow to the regulator when the evaporator pressure drops below its setting, and thus the minimum evaporator pressure is maintained. At the same time, the compressor suction pressure is reduced. The diagram of Figure 4.9 is labelled to show how the new operating condition is increased to the minimum evaporator temperature and the suction pressure to the compressor is reduced along with the refrigeration capacity.

The evaporator pressure regulator can be used in conjunction with face and bypass dampers in order to reduce the energy requirements during part load conditions. (See Section 6.6).

4.4 Basic System Design Viewed Against Background of Criticism of Existing Two Variable Controlled Range Systems

4.4.1 Interaction between temperature and humidity controllers

A design solution has been offered which meets the numerous criticisms that have been made in the literature of two variable controlled climate simulators. This system is free of hunting and cycling that arises due to interaction between the temperature and humidity controllers. In this solution there is some interaction between the controllers particularly if a pan humidifier is used to introduce moisture to the air. There is a relatively small amount of sensible heat addition which occurs at the water surface. However the effect is not that of cycling but simply of relieving the reheater of an equivalent addition of heat. The dry bulb temperature sensor within the climatic chamber does not differentiate between sensible heat introduced by the reheater or the humidifier. The humidifying process drawn on a psychrometric chart indicates a very nearly vertical line upwards but slightly to the right. In this design solution one reason the temperature and humidity sensors do not cause interaction is that the dry bulb and the dew point temperature sensors are measuring independent variables. Systems controlling relative humidity and dry bulb temperature simultaneously often move into a cycling relationship between variables. This was avoided.

4.4.2 Interaction between the cooling and dehumidifying processes

In this design the cooling and dehumidifying processes occur together at the single evaporator. Since the evaporator surface serves to establish a fixed energy datum level no interaction occurs.

The relationship between the amount of cooling and dehumidifying may vary for different set points. However for any operating pair of temperature and humidity settings they have a fixed relation to each other which is compatible with the steady state of the refrigeration and air cycles. See Section 7.1.2 and Appendix I, indicating on a recorder chart, performance of temperature and humidity over a several day period with changeover between 'day' and 'night' conditions at the Waite Institute Phytotron Unit.

4.4.3 Interaction between cooling and heating, dehumidifying and humidifying

In this design controllers are not needed for either cooling or dehumidifying. By virtue of the design these processes have been brought to a condition establishing the fixed energy datum by way of the temperature of the wetted surface of the evaporator. In lieu of controls for cooling and dehumidifying these processes have been allowed to freely establish an inter-relationship between each other and reach a balance point with the refrigerant flowing through the evaporator.

Since the design establishes a fixed energy datum by way of the surface temperature of the evaporator, there can be no interaction with the heating and humidifying processes. The reheater and the humidifier transfer heat and moisture to an air stream which has a fixed dry bulb and dew point temperature leaving the evaporator.

4.5 Summary. The Essence of the Design Solution

The essence of the design solution is to select a continuously operating refrigeration cycle that has the capacity to offset in an air stream both the sensible and latent heat loads for every combination of temperature and humidity within a climatic range. The control system becomes operative in one direction only - to add heat, sensible and latent when the temperature and/or humidity fall(s) below the operating settings, thus establishing the desired conditions. When this is achieved the continuously operating refrigeration cycle reaches steady state. This occurs when the balance point between the condensing unit curve and the evaporator performance curve on a refrigeration capacity versus evaporator temperature, pressure plot is established. This balance point will vary depending on the operating settings of the system. The system described has the ability to automatically move from one set of temperature and humidity conditions to another set. This will occur even though there may be very different sensible and latent heat loads that are associated with the settings from which and to which changeover is taking place. An example would be the case of moving from a 'night' temperature and humidity to a 'day' temperature and humidity in a phytotron unit.

The unified approach therefore has resulted in a system which for all operating settings drives to steady state conditions and which has the capacity to control temperature and humidity by means of the addition of heat alone. This is common to all applications of the design. However the system is amenable to numerous arrangements, in order to comply with variations in the users' specifications and to conserve on energy requirements.

It is the ability of the system to adjust to differing specifications which permits the application of the unified approach described in this thesis. A number of these arrangements are described in Section 6.

5. ENERGY CONSERVATION

5.1 Background

One of the most difficult problems that is associated with the engineering of climate simulation systems is to find a basis wherein a particular design can be rated with regard to energy consumption. There are so many factors involved which directly or indirectly interplay with each other that only a total evaluation can avoid misleading conclusions. If a comparison of energy consumption is made of several different conventional systems for air conditioning, these being relatively simple systems since range is not a factor, it would be quite difficult to come to a decisive conclusion.

For example, a single duct reheat system at first glance may appear more wasteful of energy than a dual duct system of air conditioning. In the reheat system, air that has been overcooled is reheated to reach the design temperature - if waste heat is unavailable this could be a double or even a quadruple energy penalty depending on the COP and whether reheat is obtained from thermal or electrical source. (See Section 11.14, including footnote.) In the dual duct system relatively hot and cold air is made available at the entrance to a mixing box. These air streams are so controlled as to maintain a constant flow after combining in correct proportions to meet the needs of an air conditioning space so that no waste of cooling or heating occurs. During summer cycle the so-called hot air may be at ambient temperature, only relatively hot.

The reheat system if applied to variations within a zone, (a collection of areas having very similar energy load conditions), would involve only minor reheat requirements, would cost less initially, would take up less space and may even involve less total energy usage.

The dual duct system because of the space requirements of its ducts is most applicable to those buildings which have a maximum difference between the simultaneous sensible heat load and the summation of the individual room or zone sensible heat peak loads that are served by a particular air conditioning unit. (Shaw 1967). Oddly enough this very criterion may actually involve the dual duct system into "reheat" of the "bypass, hot" air in order to prevent rooms that have negligible loads from being over-cooled.

A look in depth considering all factors including space utilization, building characteristics, thermal loads, manufacturing costs, maintenance costs, as well as running costs would reveal that in most cases for conventional usage the reheat system would far exceed the dual duct system in overall economy. In a few exceptional cases such as museums and department store air conditioning, the dual duct system would prove feasible though not necessarily economical. To determine which of the two systems are preferred would involve extensive and costly calculations.

Now when *range* is also a factor and the system to be analysed can be applied to narrow as well as wide range, to single variable control (temperature-only) or to two variable control (temperature and humidity) and the design criteria includes numerous aims, (see Section 3.3), it must become apparent that a comparison of different systems of climate simulation is exceedingly difficult. It will not be attempted here since it would involve a diversion in depth requiring detailed knowledge of other systems of climate simulation much of which is unavailable. Also it would introduce an additional complexity. How do the systems being compared perform? An economic rating without also assigning a value judgement on performance is of little significance, particularly in this field. There have been many systems designed with energy conservation as a first priority and unfortunately performance suffered to the extent that these systems were of no value.

In this section, the basic unified system will be examined from the point of view of energy expenditure occurring in excess of what is essential to offset the thermal and humid loads.

Section 6 describes the various steps taken to reduce energy expenditure to a minimum. This will take the form of different arrangements modifying the basic system. Only stable performing arrangements will be examined. All of the modifications suggested will continue to maintain the major objectives of the unified system. In every case the refrigeration system on reaching the desired temperature and humidity setting within the climate chamber will reach a steady flow state having a fixed evaporator temperature and pressure. This will establish a datum temperature for a fixed air flow rate which will maintain its setting over a range of temperature and humidity conditions by means of the addition of sensible and latent heat.

The basic system itself gives rise to considerable energy losses. However, in Section 6 the system is shown to have an unusual flexibility which permits variations and modifications. Some of these will be employed to reduce energy expenditure to a minimum. In addition Sections 10 and 11 treat in depth with energy conservation in dehumidifier selection.

5.2 Basic System Energy Expenditure

A basic design has been presented in Section 4 with little reference to energy requirements. Up to this point, the main priority was to design a simple, stable, close tolerance, two variable controlled range system. Both the condensing unit and the evaporator surface had to be selected to have a minimum refrigeration capacity sufficient to offset both the sensible and latent heat loads for all operating conditions in the range.

5.2.1 High entering enthalpy settings; range

For a given mass flow of air an energy penalty occurs in the basic unified system for operating settings in the range having a high entering enthalpy. This is clearly shown on the refrigeration capacity versus suction temperature pressure plot of Figure 4.6. The penalty is associated with the balance point location. The evaporator performance curve intersects the condensing unit curve at higher and higher refrigeration capacities as the entering enthalpy of the operating settings increase. The larger the span of enthalpy difference between the operating settings within the range, the greater the energy penalty.

Though this energy penalty can partially be eliminated by modifying the basic system in several ways, (see Section 6.4), it may be found quite acceptable in certain applications where small chambers are concerned particularly if the climatic range is not too wide. This is justifiable because of the simplicity of the basic system.

5.2.2 Dehumidification by cooling

A range system using a single direct expansion coil to offset both the sensible and latent heat loads will of necessity include some energy penalties.

There will be occasions when dehumidification will be more than is actually required because of the need to offset the sensible heat load. This will occur during operating settings having high relative humidities. (See Figure 5.1). Obviously if ΔW_1 of curve 1 has the capacity to offset all latent heat loads for operating condition 1 then ΔW_2 of curve 2 indicates excessive dehumidification for operating condition 2.

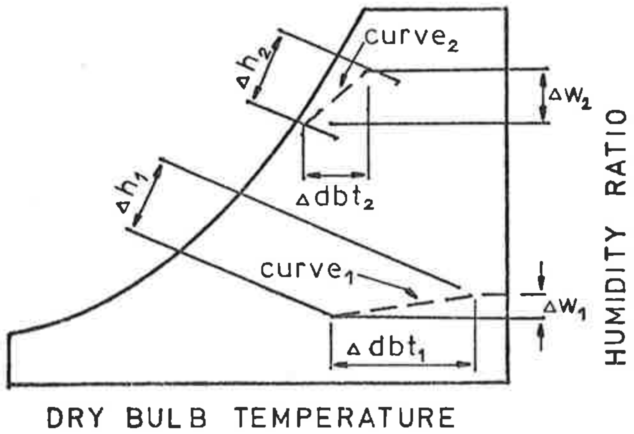


Fig. 5.1 Energy Expense where Both the Cooling and Dehumidifying Function is Carried Out by a Single Direct Expansion Coil

On the other hand, there will be occasions when the sensible cooling will have to be more than is actually required because of the need to offset the latent heat load with sufficient dehumidification. This will occur during operating settings which simulate high dry bulb temperatures combined with arid conditions. At these times the direct expansion coil will be approaching dry performance. See curve 1 in Figure 5.1, also curves 4, 5 and 6 of Figure 4.8.

If Δdbt_2 of curve 2 has the capacity to offset all the sensible heat loads for operating condition 2 then Δdbt_1 of curve 1 indicates excessive sensible cooling for operating condition 1.

The energy penalty associated with dehumidification by cooling is still further aggravated due to the need in this system of establishing steady state conditions. The surface temperature of the evaporator serves as the datum for both the cooling and the dehumidification process. This temperature has a minimum limit. The surface of the evaporator must never drop to a level which would result in frosting. (See Section 4.3.3(c)). The build up of ice on the evaporator would be accompanied by a constantly reducing refrigeration capacity. As a consequence of this limitation there is a limit to the maximum available driving potential for dehumidification. Therefore, to increase mass transfer it may be necessary to increase the refrigeration capacity and the mass flow of air beyond the amount necessary to offset the sensible heat loads. This penalty arises in the high dry bulb temperature, low humidity, ratio area of the range as has been demonstrated by curve 1 of Figure 5.1.

It is due to the energy penalties in this area of the psychrometric chart range, that a point is reached which defines the limits of economic feasibility. Limitations are imposed on the range of all systems using dehumidification by means of evaporators in coolers or scrubbers. In Figures 5.2, 5.3, 5.4 and 5.5 the temperature-humidity ranges of an Australian, Canadian, English and Swedish system are shown. The systems have different designs but in all cases cooling is used to dehumidify. Note that the lower right hand corner of the range as shown on American, Canadian and English psychrometric charts is restricted to avoid the 'dry' coil energy penalty becoming excessive. The corresponding area on the Swedish psychrometric chart which is plotted, turned ninety degrees and mirror image to the American, Canadian and English charts, (i.e. dry bulb temperature is the ordinate and humidity ratio the abscissa), is in the upper left hand corner.

There is a large energy savings if the specified range excludes this corner of the psychrometric chart.

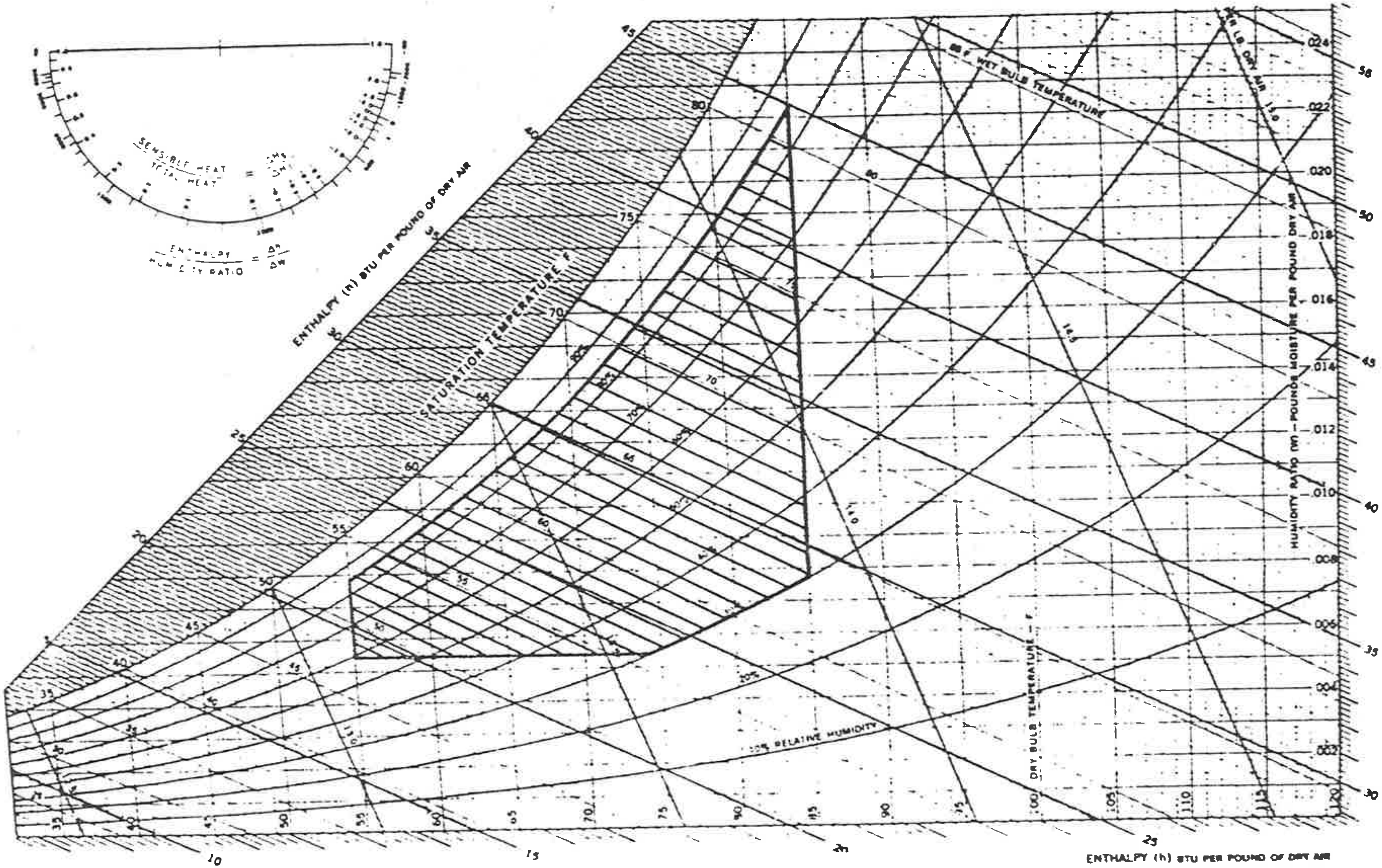


Fig. 5.2 Range of an Australian System

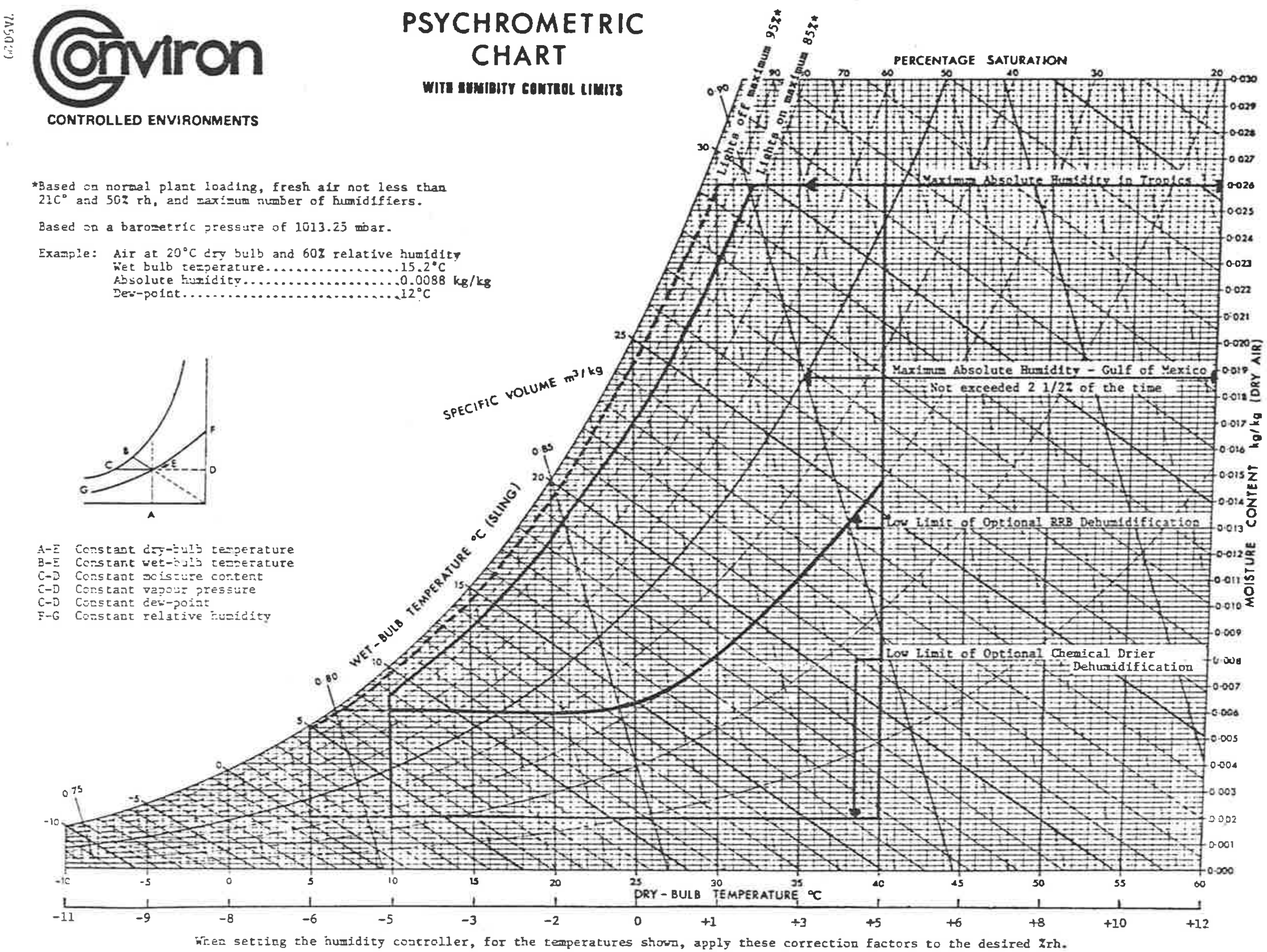
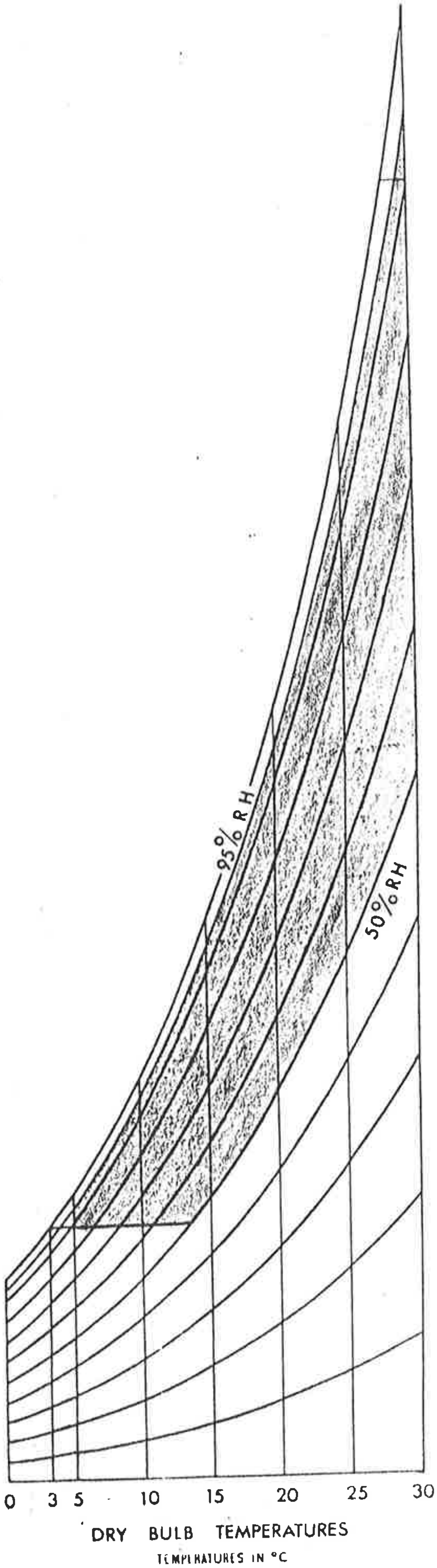


Fig. 5.3 Range of a Canadian System



PSYCROMETRIC CHART

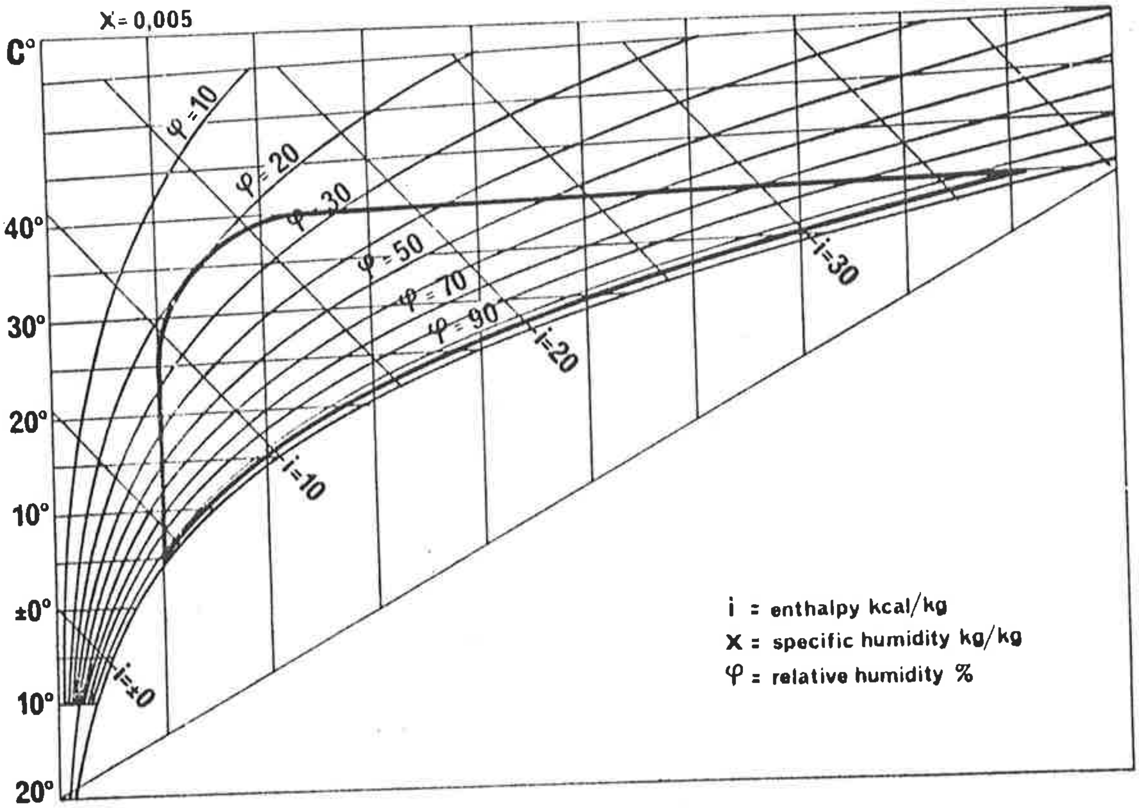
Limiting factors of performance are:

1. Maximum dry bulb temperature 30°C
2. Minimum dry bulb temperature 5°C
3. Relative humidity range 95/50%
4. Minimum dewpoint 3°C

The Prestcold Plant Growth Cabinet will supply conditioned air to the plant bed at any predetermined point within the area shown by the graph. Control of dry bulb temperature will be ±0.5°C, and of relative humidity ±2%.

Fig. 5.4

Range of an English System



Operating range

is determined by the specific humidity line for 0.005 kg/kg and the dry-bulb temperature lines for +5°C and +40°C. Air humidities up to 95—98% are obtainable. Temperature and humidity sensing elements are located in the unit room. The sensitivity of these controllers determines the accuracy with which temperature and humidity are maintained.
In cases where temperatures below +5°C are considered necessary, the required conditions can be secured by means of special equipment.

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Fig. 5.5 Range of a Swedish System

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There are alternative approaches that may be taken where it is necessary to work in lower humidity ratios than 0.005. For example the use of absorbents may become necessary. These too have their drawbacks in more controls and increased system costs, space requirements and operating complexity. Here too, there are energy penalties such as the heat required for regeneration.

Several modifications to reduce the energy penalties are described in Section 6. They will be compatible with the unified approach.

Relevant to the energy penalty, the selection of the direct expansion coil is a very important aspect of the design and is discussed in detail in Sections 10 and 11.

5.2.3 Flow pattern and percent load considerations

The energy expenditure is affected by the method of introduction of the air stream and its relationship to the heat and moisture loads, the specified temperature and humidity gradient through the controlled space of the chamber, the part load and full load operating conditions, and in the case of phytotron units, the number, size and foliage areas of the plants within the chamber.

The basic system does not allow appreciable energy savings for part load operation. In fact, it gives rise to an additional energy penalty since the same operating setting would require more reheat during part load conditions. This might be acceptable for many applications involving small reach-in chambers. However, in some cases, as for example a large day-night phytotron unit, particularly if it is designed with non-barrier* lighting, the part load penalty during the

---

* A non-barrier system is one in which the lighting section is not separately partitioned from the plant growth chamber. Thus all the heat from lights must be offset by the refrigeration system. In a barrier system, the lights are separated from the chamber proper and the light section usually ventilated. Thus the greater portion of the heat from lights is dissipated to ambient.

night cycle would be excessive. In such cases the basic system would be used in one of the modified forms, for example the one described in Section 6.6. The refrigeration capacity during the full load operating day cycle is higher for the basic system when the air stream passing through the plant growth chamber has an upward flow pattern. With this arrangement as the air stream passes up or across the top of the phytotron chamber there is a convective heat transfer from the lighting section. (This will affect the refrigeration capacity but not the temperature gradient across the plant occupied section of the chamber). Thus during a 'day' (full load) operating cycle the enthalpy of the air entering the direct expansion coil will be more than the enthalpy of the air at the operating setting of the system. (These enthalpies would be equal for the downward flow full load operating day cycle). During the 'night' (no load) operating cycle the enthalpy of the air at the operating setting of the basic upward flow system and the enthalpy of the air entering the direct expansion coil are the same. See Figures 5.6 and 5.7.

Though the refrigeration capacity during the part load (night) cycle is lower than the full load (day) cycle for the upward flow pattern, (due to the higher entering enthalpy of the day cycle, see Section 5.21) this is not to be misunderstood as the reason for recommending upward flow. Downward flow has a slightly lower refrigeration capacity requirement and a slightly lower reheat requirement during full load (day) operating conditions. This is demonstrated by Figures 5.8 and 5.9 for the downward night and day flow patterns.

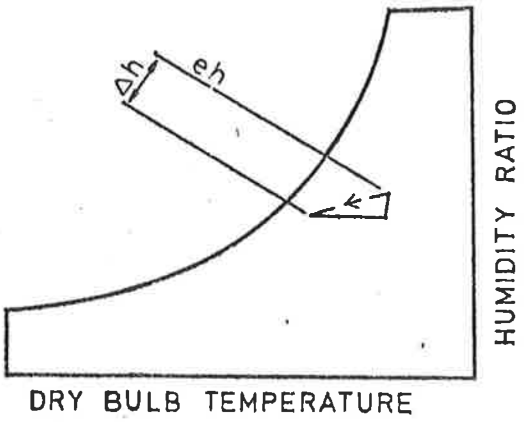

FIG 5.6 NIGHT CYCLE

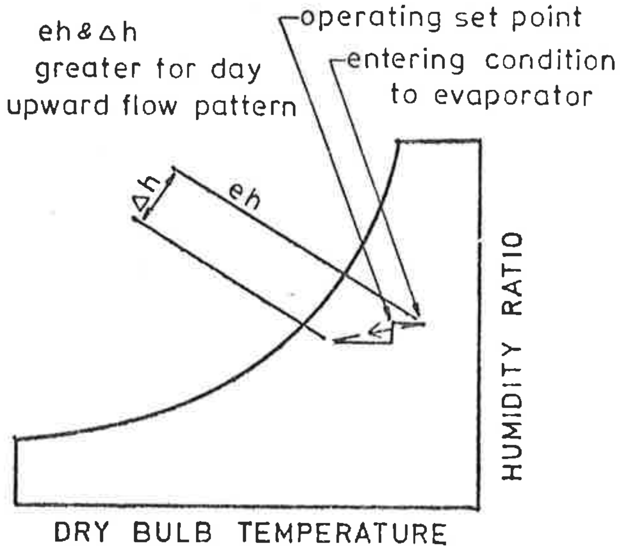

FIG 5.7 DAY CYCLE

UPWARD FLOW PATTERN

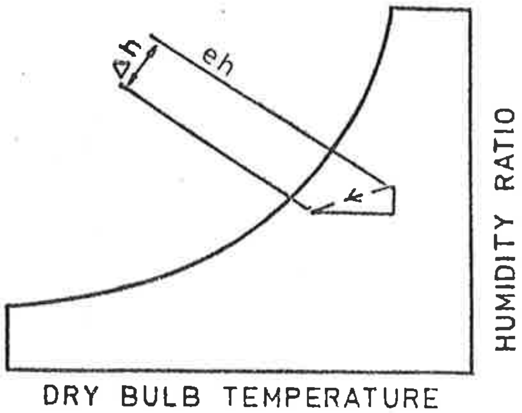

FIG 5.8 NIGHT CYCLE

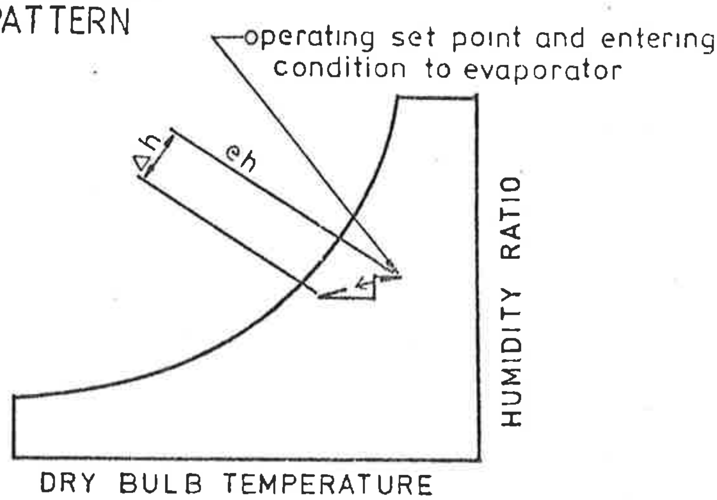

FIG 5.9 DAY CYCLE

DOWNWARD FLOW PATTERN

Upward flow results in the least temperature gradient and the most uniform air stream. On the other hand, downward flow for a phytotron unit is very poor when a barrier is employed between the lighting section and the plant occupied chamber. Due to the 'ceiling' of the chamber being occupied by lights air introduction is usually limited to top grilles located at the sides of the chamber. A uniform air stream from this type of supply source is not obtainable within the narrow confines of a phytotron unit, vortices and eddies form, and the temperature gradient can be very high.

It is for this reason that the upward flow pattern despite its (slightly) larger full load penalties is often preferred. With upward flow the air can be introduced through a perforated 'floor' of the plant growth chamber and be positively displaced upwards towards top grilles located high along the sidewalls. (See Appendix IV). The convective heat transferred from the barrier of the lighting section does not affect the temperature gradient across the chamber. It is therefore particularly low when the plants are seedlings. The major load which is from the radiant energy of the lights is associated with a sudden temperature rise at the floor of the chamber before it enters the plant occupied space. With crowded plants and large foliage the gradient increases. However it would be fairly uniform at each horizontal cross-section of the plant growth chamber.

6. ARRANGEMENTS AND MODIFICATIONS TO THE SIMPLE BASIC SYSTEM IN ORDER TO SATISFY WIDE DIFFERENCES IN USERS' SPECIFICATIONS AND TO EFFECT ENERGY SAVINGS

The basic system has been shown to possess energy penalties associated with range and air flow pattern during part load and full load operation. The design approach to establish steady flow air and refrigeration cycles has further aggravated the excess refrigeration capacity by the need for reheating and the need to avoid frosting. These energy expenditures can often be tolerated for small chambers having refrigeration capacities of about 3kW.

Without departure from the fundamental criteria it is possible to adapt the system to wide differences in users' specifications and energy performance characteristics.

6.1 Open Compressor and Fan with Variable Speed Control

One arrangement to counter the increased energy requirements for increased entering enthalpy settings is to employ an open type compressor with a manual or automatic control over the compressor speed. It is then possible to maintain a fixed evaporator temperature (pressure) and therefore a constant minimum refrigeration capacity. Thus Figure 6.1 below demonstrates the refrigeration capacities at the set points which result in the maximum entering enthalpy of the air streams to the evaporator, eh = 5 in this particular example. With a reduced speed of the compressor, the same refrigeration capacity as for the minimum entering enthalpy of the air stream, eh = 1 is obtainable. However this method of reduction of refrigeration capacity would only be applicable to very limited range systems. It should be noted that though by means of a lower compressor speed the refrigeration capacity has been reduced to that of the evaporator performance curve for condition eh = 1 in Figure 6.1, the suction temperature has increased. This would reduce the extent of dehumidification. To maintain the original suction temperature the fan speed would also have to be reduced.

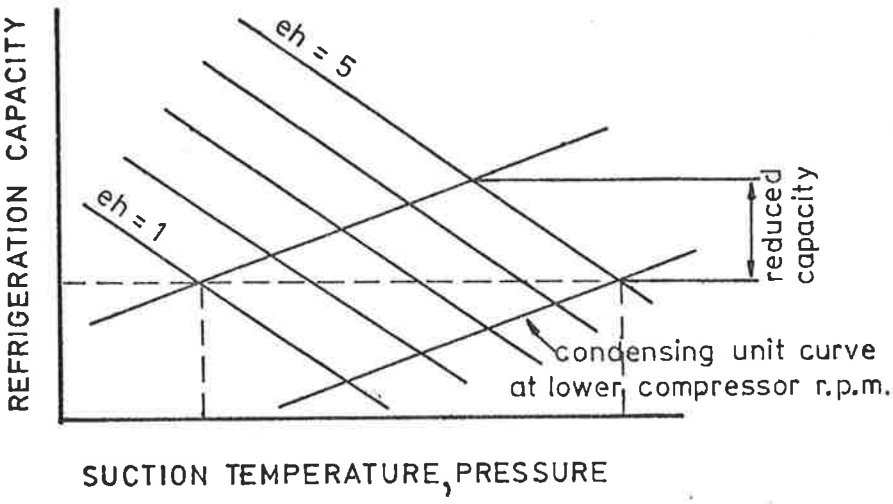

Fig. 6.1 Reduction of Refrigeration Capacity with Reduced Compressor Speed

A variable speed controlled compressor has much to commend it in other respects. It would give research workers a more flexible condensing unit. Performance would not be restricted by a particular condensing unit curve but by many curves covering a broad area. Where such an open type compressor system is being used with a phytotron operating on a day/night basis, automatic changeover of compressor speed and fan speed can be used to maintain the desired refrigerant temperature with minimum energy expenditure.

However the use of open type compressors is not likely to suit commercial applications. The open type compressor requires more servicing and it has been largely replaced by hermetic units. Furthermore since air flow rate would also have to be controlled (see Section

6.2 below) to keep the air refrigerant relationship compatible it would not be suitable for mass production purposes.

6.2 Variation in Flow Rate of Air Stream

One very simple modification to save energy is to decrease the air flow rate by controlling an air damper downstream from the fan either manually or automatically. In the case of a phytotron unit, which operates either full load during a day setting and close to no load during a night setting, two positions of the damper may suffice. Figure 6.2 below indicates the drop in refrigeration capacity when the air damper moves to partially closed position.

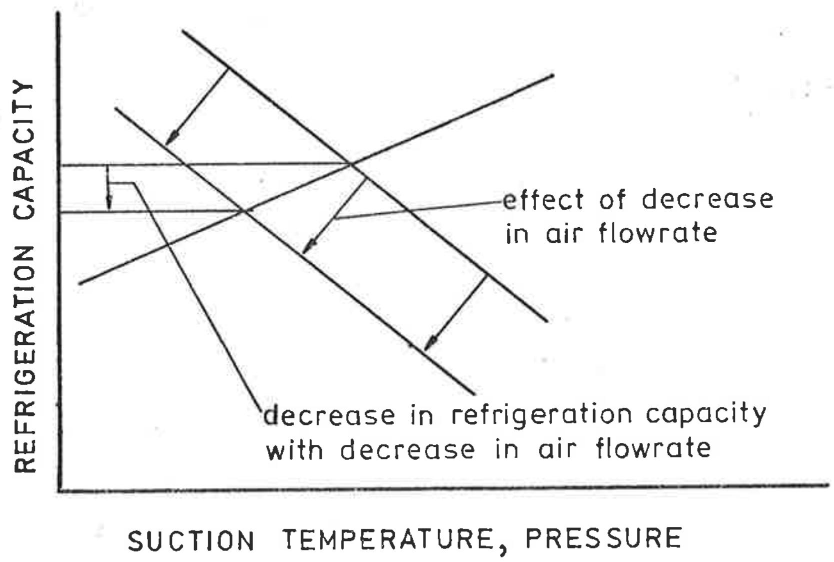

Fig. 6.2 Effect of Decrease of Air Flow Rate

Though this modification will obviously reduce the energy costs, it involves varying the air flow rate to the test chamber. This would not satisfy the requirements of many users.

Further energy savings could be made with this system during reduced load operation by the addition of a constant pressure evaporator regulator. (See Section 6.6).

6.3 Two Direct Expansion Coils and Two Hermetic Condensing Units

This modification permits the system during one part of the range and also under part load conditions to use half the total system capacity. It also provides the assurance that during a breakdown of one of the compressors the system would not be completely inoperative and might serve as a stand-by until repairs are made. This system uses two independent condensing units; two evaporators which may be piped in a manner such that every other tube in each evaporator is exclusively associated with one of the two condensing units; a single preheater, reheater and humidifier, each split between two air paths; two thermostatic expansion valves, solenoid valves, driers, liquid-suction heat exchangers. The number of condensing units operating are associated with the particular operating settings desired. Though there is no actual bypass built around the evaporator, when only one condensing unit is on, every other tube in the evaporator is not operating and thus there is the equivalent of an additional 50 per cent bypass of the evaporator. The system performance was of the same order as that of the Waite Institute Phytotron Unit. See Appendix V for a photograph of this unit. Note the two hermetic condensing units are located to the left.

6.4 Face and Bypass Dampers

Another more practical way of reducing the energy requirements for high entering enthalpy settings is to include a face and bypass damper in the system which can be made to act completely automatically to maintain a fixed refrigeration capacity. In this case an open compressor would not be required.

The use of a face and bypass damper has a somewhat similar effect to the damper at fan discharge arrangement of Section 6.2, where the total air stream has its mass flow rate reduced. In the arrangement described in this section as the bypass damper opens and the face damper closes the evaporator performance curve moves down to the left on the refrigeration capacity versus suction temperature, pressure plot. When the reverse action takes place, the closing of the bypass damper and opening of the face damper, the evaporator performance curve moves up to the right. Unlike the case of Section 6.2 above, the total mass flow rate of air entering the climate simulator chamber remains approximately constant.

Reference is made to a psychrometric chart outlining a climate range that may be served by a climatic chamber. See Figure 4.2 repeated below.

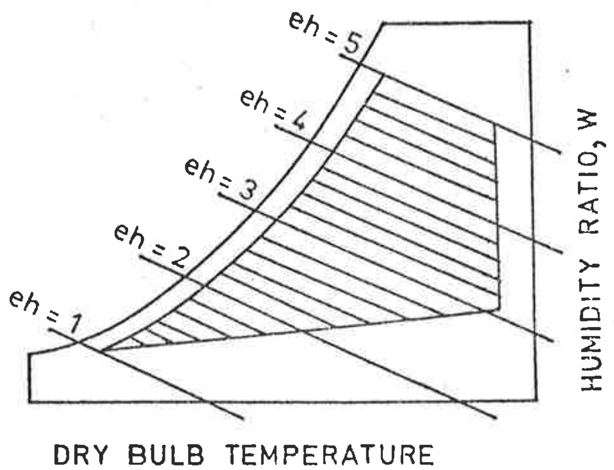

Fig. 4.2

Climatic Range
(Repeated)

Corresponding to this psychrometric range, for the basic system design, there is a fixed mass flow of air through the evaporator at all operating settings. Figure 6.3 below represents the entire range performance on a refrigeration capacity versus suction temperature plot similar to Figure 4.5. Points 1 to 5 inclusive represent in five steps the various entering enthalpy conditions corresponding to the total range of operating settings.

Consider a simple case where for all operating settings having an entering enthalpy value upstream of the evaporator of eh = 1, the bypass damper in a face and bypass arrangement is fully closed and the face damper fully open. Now consider all the other operating settings having higher entering enthalpy values of eh = 2 to eh = 5 respectively. Figure 6.3 indicates diagrammatically the effect of varying degrees of small, medium, large and maximum bypass associated with maximum, large medium and small face velocities. Since position of the evaporator performance curve on this diagram is a function of the product of the entering enthalpy and the mass flow of air passing through the evaporator, the basic system evaporator performance curves of eh = 2, 3, 4 and 5 can all be relocated to coincide with that of eh = 1 by simply reducing the mass flow of air through the evaporator by an amount that is compatible with the respective operating setting enthalpy until finally the highest enthalpy operating condition in the range, eh = 5, has the least air flow through the evaporator coil. (This corresponds to the maximum bypass shown in Figure 6.3). (Shaw p.90, 1978).

Thus

$$\dot{m}_1 eh_1 = \dot{m}_2 eh_2 = \dot{m}_3 eh_3 = \dot{m}_4 eh_4 = \dot{m}_5 eh_5 \ .$$

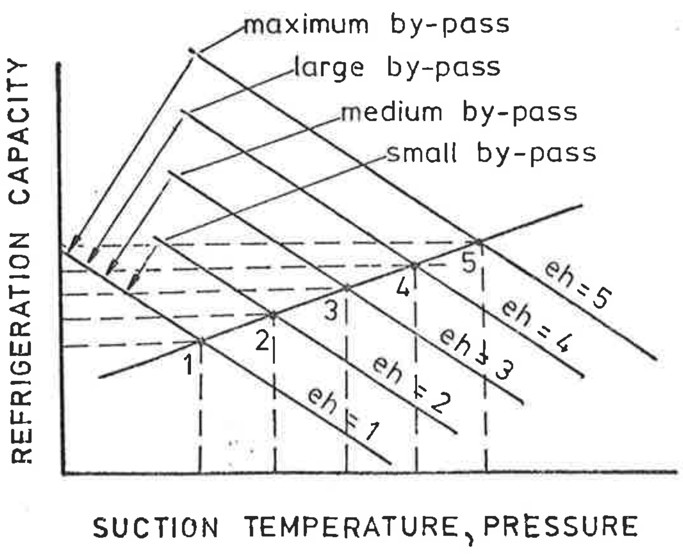

Fig. 6.3

Face and Bypass Dampers to Obtain Constant Capacity for all Operating Conditions

The significance of the energy savings realized by the reduction of refrigeration capacity for all entering enthalpy conditions above eh = 1 to the fixed refrigeration capacity occurring at eh = 1 can also be seen by inspecting the nature of the line connecting entering and leaving conditions at the test chamber. Figure 6.4 below shows the enthalpy change associated with the fixed refrigeration capacity and a constant <u>total</u> air flow stream of combined face and bypass air that enters the test chamber (NB: not the varying mass flow of air passing through the evaporator). The "average" paths of 1 through 10 all have the same enthalpy change to the air stream. This change represents for all operating settings the refrigeration capacity corresponding to point 1 on Figure 6.3 above.

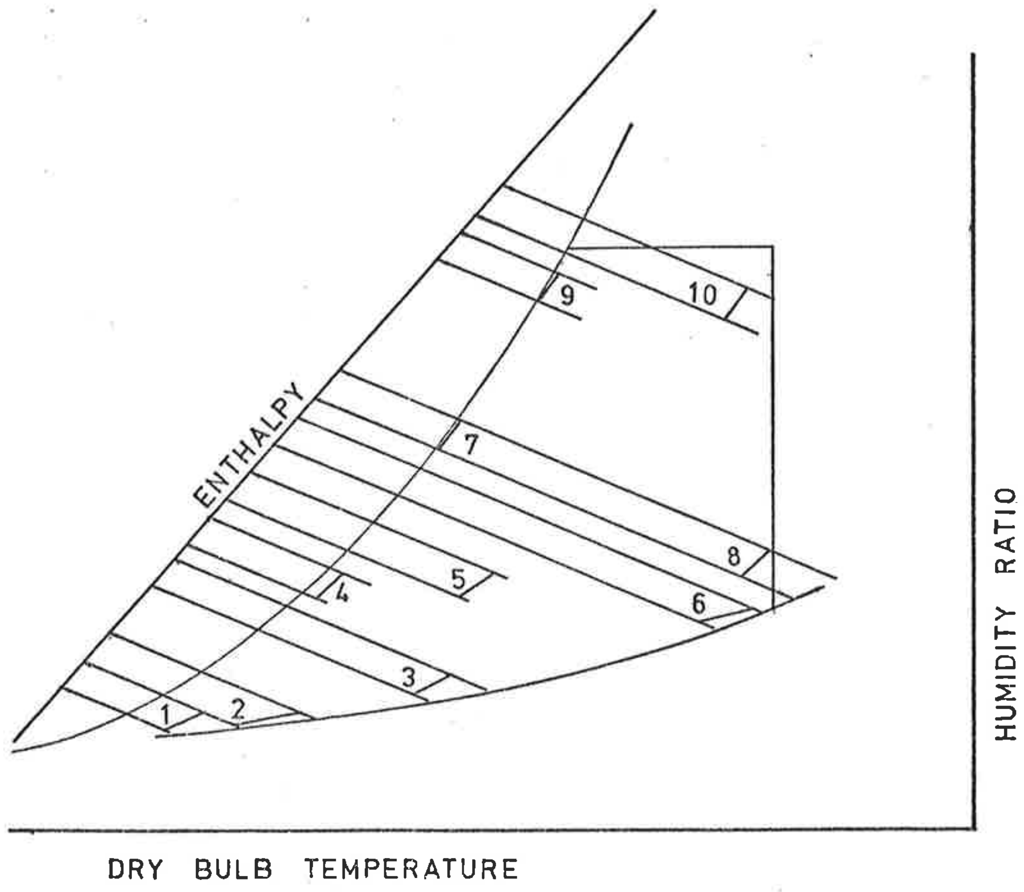

Fig. 6.4 Face and Bypass System: All Operating Settings Reduced to Same Refrigeration Capacity

The energy savings resulting from this modification must be balanced against the costs arising from the use of face and bypass damper motors, relays and in some applications an automatic control system to adjust the damper settings. (This is necessary to maintain both a day and night wet bulb temperature when changeover is required or to follow some program associated with wet bulb temperature operating settings). In the former case, when it is required to operate at a particular setting of day and night conditions it is not necessary to employ a controller. All that is necessary is to arrange that the time clock operating a day-night relay will actuate a damper motor to establish a particular fixed face and bypass damper setting. The desired fixed damper setting is a function of the wet bulb temperature setting (used in lieu of enthalpy) and therefore the operator need only set his control point adjustment for the wet bulb temperature desired, and by doing this the damper motor moves to the corresponding setting. For system performance of a commercial prototype built with the face and damper modification see Appendix VI. Performance includes the changeover between ten test points.

The control panel would be very simple, as indicated in Figure 6.5 below.

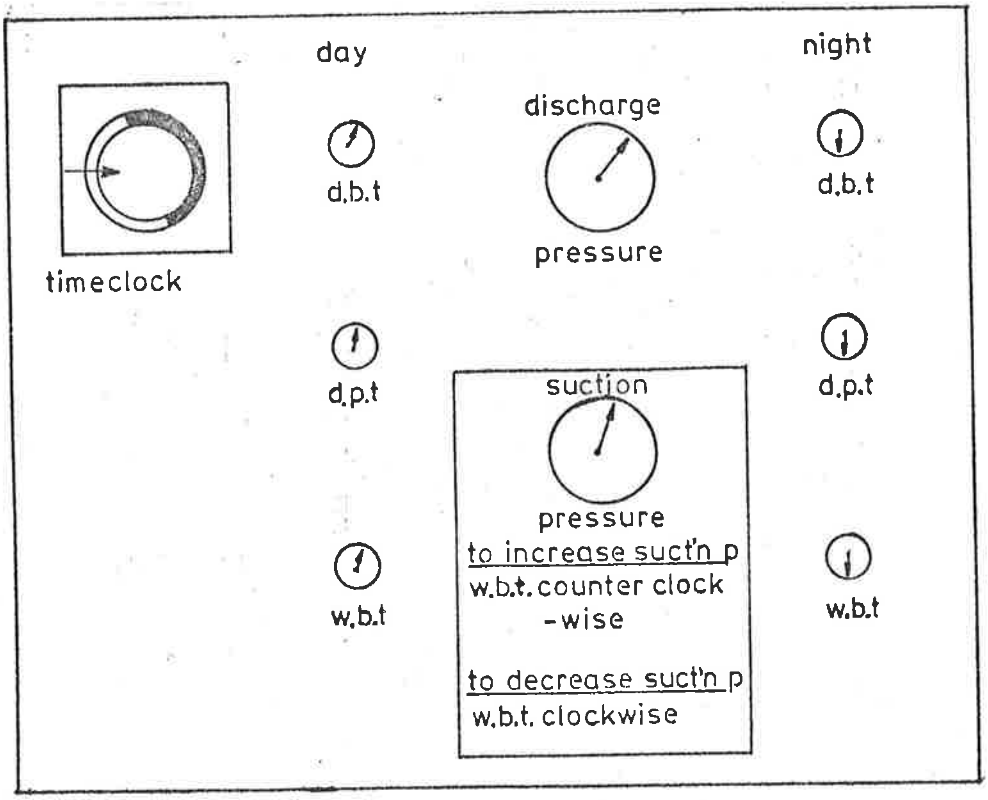

Fig. 6.5   Control Panel

In a day-night commercial system it would not be necessary to calibrate the wet bulb temperature control point adjustment potentiometer. In lieu of this, instructions below the suction pressure gauge would indicate the direction of movement of the adjustment dial to obtain the one particular suction pressure that is common for all operating settings for this face and bypass damper arrangement. See Figure 6.3 above.

There is no energy saved during part load operation for the face and bypass modification. For this arrangement all entering air energy levels at inlet to the direct expansion coil are identical. They have been reduced by the face and bypass damper positions to an air flow rate associated with a single minimum energy level. Thus if a comparison were to be made, between a full load day setting and a no load night setting, during the day cycle the face damper would close more (the bypass damper would open more) than during the night cycle. The mass flow rate of air to the evaporator for the 'day' condition would be lower but the entering enthalpy to the evaporator would be higher. Both total energy conditions would be the same. Thus for the face and bypass damper modification there would be no energy savings for the no load, night cycle. Instead, there would be a reheater energy penalty for the night cycle. In order to obtain energy savings during reduced load operation further modification is necessary. Section 6.6 below describes how this can be achieved with the face and bypass system.

6.5 Face and Bypass Dampers with a Minimum Stop

It may not be possible to obtain operating settings in the range having high entering air enthalpies if the face and bypass dampers operate to reduce the refrigeration capacity to the minimum value of eh_1 as indicated in Section 6.4 and Figure 6.3 above. At these conditions, a greater refrigeration capacity may be required.

One solution which would not impose any additional action by the operator of the system would be to employ a 'stop' to prevent the face damper from closing beyond a minimum setting. When a stop is employed a large portion of the range operates exactly as described in Section 6.4 above. The control system senses that the entering enthalpy (or wet bulb temperature) is reached corresponding to the minimum face damper setting, then for that entering enthalpy, and all entering enthalpies greater than it, there will be a gradual increase in the suction temperature. Thus a stop performs as described for the basic system in Section 4.2.1 to 4.2.3 and Figure 4.6.

The combination of a face and bypass damper and the use of a stop therefore represent a dual arrangement. Performance at the lower section of the range is such as to maintain a constant reduced refrigeration capacity and suction temperature/pressure. At the upper section of the range, the refrigeration capacity and suction temperature/pressure increases in association with rising entering air enthalpy.

To illustrate what occurs in this dual arrangement, consider Figure 6.6.

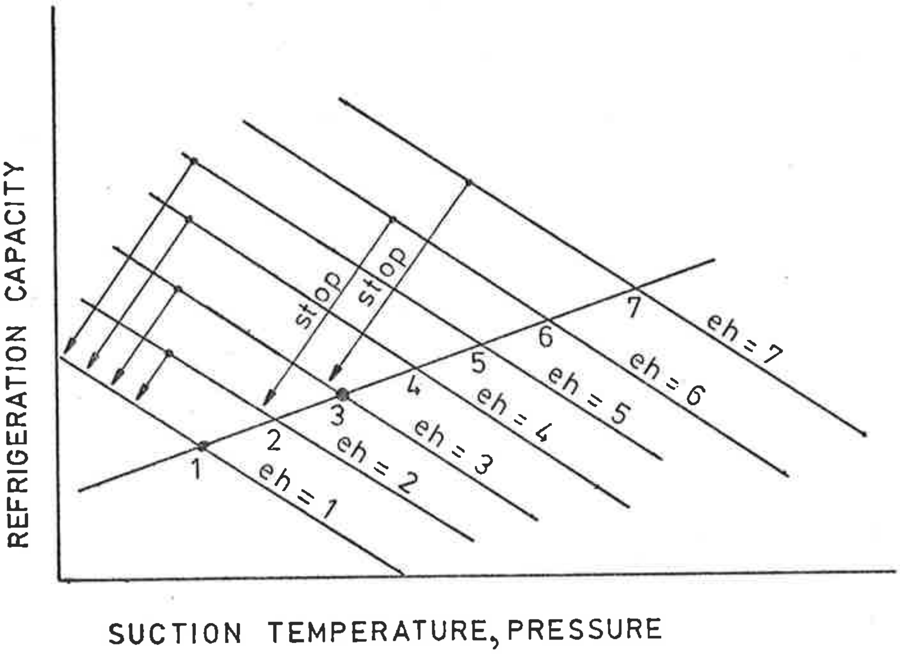

Fig. 6.6 Dual Arrangement, Face & Bypass Damper with a Minimum Stop.

Here, for every entering enthalpy up to the value of eh_5 the face and bypass dampers would operate to reduce the mass flow of the air through the direct expansion coil so that the performance curve for all conditions between $\dot{m}_a eh_2$ and $\dot{m}_a eh_5$ is equal to that of $\dot{m}_a eh_1$.

Now for all higher enthalpy entering conditions in the range from eh_5 to eh_7 the dampers would no longer further restrict the mass flow of air through the direct expansion coil. As a consequence, between the entering enthalpy condition of eh_5 and eh_7 there would be a gradual increase in the refrigeration capacity from $\dot{m}_a eh_1$ to $\dot{m}_a eh_3$. Thus this section of the range would perform very much as shown in Figure 4.6 except that it would benefit from the minimum opening setting of the face damper and therefore eh_6 would perform as if it were at curve eh_2 and eh_7 would perform as if it were at curve eh_3.

6.6 Refrigerant Constant Pressure Evaporator Regulator Used in Conjunction with Face and Bypass Dampers

The operation of a constant pressure evaporator regulator was described under Section 4.3.3(c)2 and Figure 4.9 in connection with the prevention of frosting in the unified system. A refrigerant constant pressure evaporator regulator used in conjunction with the face and bypass damper modification to prevent frosting can, in addition, play an important role in energy savings during reduced load conditions.

The face and bypass dampers described in Section 6.4 modify the basic system by reducing the energy penalty for all operating settings in the range to a single minimum value. As explained in Section 6.4 there is no appreciable reduction in refrigeration capacity during part load conditions. Furthermore, in order to prevent frosting eh_1 of Figure 6.2 may have to be kept higher than otherwise necessary. As with the basic system, a preheater or some other means to prevent frosting would be required for systems having ranges approaching near freezing temperature settings.

The addition of a refrigerant constant pressure evaporator regulator to the face and bypass system described in Section 6.4 serves to reduce the energy penalty during reduced load conditions and eliminates the need for the preheater or other equivalent frost preventing device. It is designed to maintain a minimum constant evaporator pressure (and temperature). However it will not control the evaporator pressure when this is above its minimum operating setting.

The valve is particularly suited to energy savings during part load conditions because the valve action in maintaining the evaporator pressure minimum setting acts to reduce the compressor suction pressure. Consequently the refrigeration capacity of the system as well as reheat requirements are also reduced. In its performance there is no interference with the constancy of the energy datum established by the evaporator pressure. The valve action can best be described with the aid of the refrigeration capacity versus evaporator pressure diagram of Figure 6.7 which is essentially a composite of Figure 6.3 explaining the face and bypass arrangement and Figure 4.9, explaining the action of the constant pressure evaporator regulator.

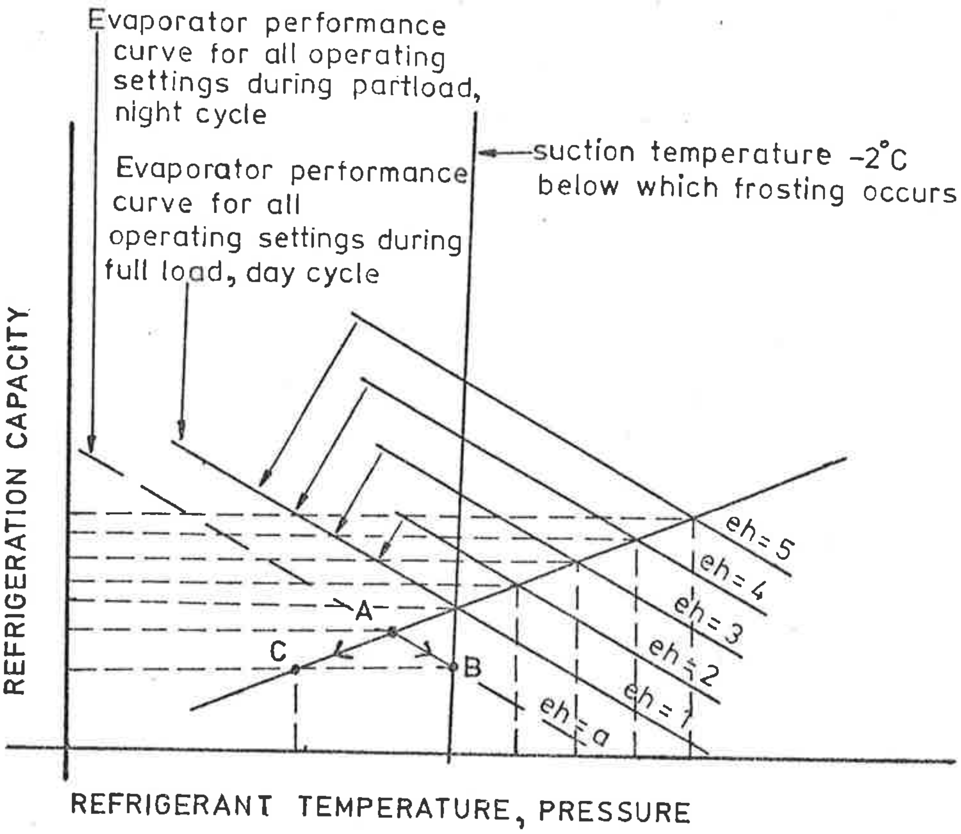

Fig. 6.7 Energy Savings During Reduced Load Operating Conditions Using A Constant Pressure Evaporator Regulator.

The following conditions are assumed to exist: The evaporator performance curve, eh = 1, intersects the condensing unit performance curve to result in a desired suction temperature compatible with the temperature range of the system. In order to achieve a wide range system eh = 1 would be at the minimum permissible evaporator pressure before frost would form on the evaporator surface. The evaporator performance curve at eh = 1 represents a 'day' operating condition of a phytotron unit. For all entering enthalpy conditions in the range, the air flow rate resulting from the positions of the face and bypass dampers is such as to reach the minimum evaporator temperature that occurs at the intersection of the condensing unit curve with the evaporator performance curve of eh = 1 as illustrated in Figure 6.7.

A refrigerant constant pressure evaporator regulator would have been set to maintain the desired evaporator pressure corresponding to the full load entering total enthalpy condition, $\dot{m}eh = 1$, as a minimum. This condition would result in a fixed evaporator pressure for all operating settings as described in Section 6.4

A face and bypass damper motor position would have been set by means of the night (part load) wet bulb temperature control point adjustment potentiometer of Figure 6.5, to operate manually or automatically during part load conditions at the lower value of total enthalpy, $\dot{m}_A eh = A$. This condition would have an evaporator performance curve located below the full load operating position of $\dot{m}_1 eh = 1$, passing through condition A. (See the refrigeration capacity versus evaporator pressure diagram of Figure 6.7). For every operating setting within the range there would be more air bypassing and less air through the dehumidifier than under corresponding full load face and bypass conditions. Thus during part load operation the reduced face (and increased bypass) damper setting would change the valve position. The valve seat would move towards a more closed position to reach a new balance point. The effect would be to:

1. increase the evaporator pressure to the desired minimum, preventing frosting, position B of Figure 6.7;
2. decrease the suction pressure of the compressor, resulting in a reduced refrigeration capacity, and a reduced reheat requirement, position C of Figure 6.7;
3. this dual action would maintain the steady flow nature of the basic unified system while resulting in reduced part load energy consumption.

6.7 Refrigerant Constant Pressure Suction Regulator

The modern small hermetic air condensing unit is limited by the maximum suction pressure permissible at the inlet to the compressor. Above this limit it is possible to overload the motor. Yet at high entering enthalpy set points the basic system balance point may occur at a suction pressure that is above the manufacturer's maximum limit.

The installation of a compressor suction regulator would resolve the problem. When the suction pressure seeks to rise above the maximum limit, to a point such as X, as shown on Figure 6.8, then the regulator valve begins to close its outlet port and thus reduces the compressor suction pressure to a safe level as indicated by point Y and at the same time increases the evaporator pressure to a point Z having the same reduced refrigeration capacity as Y.

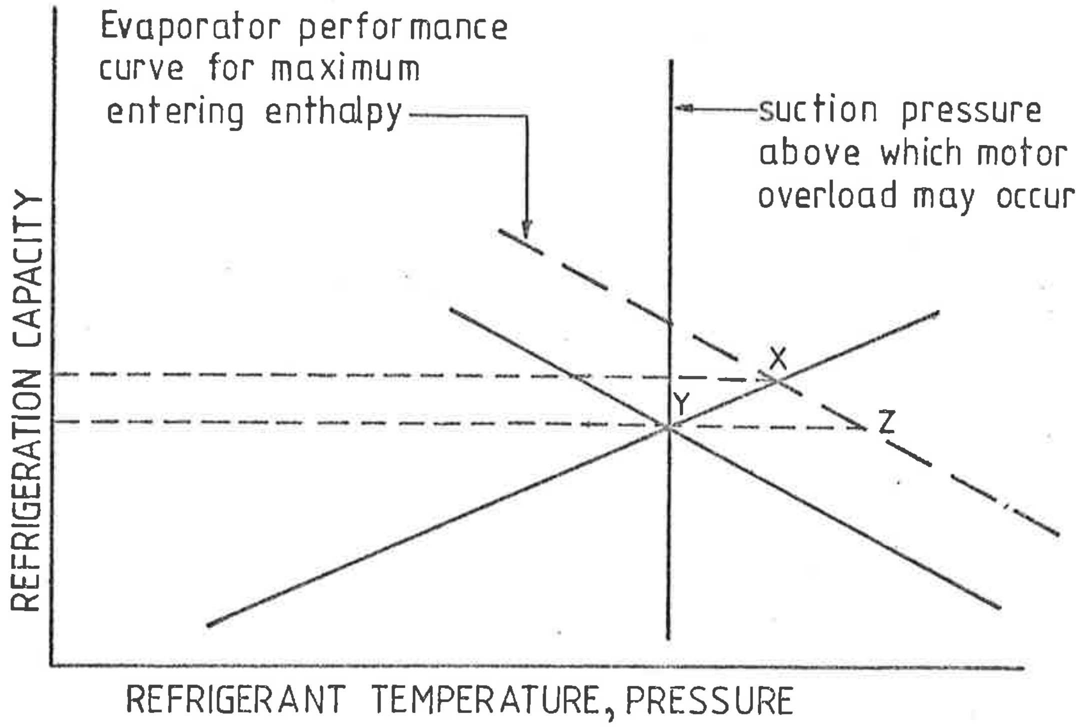

Fig. 6.8 Increased Range Using a Constant Pressure Suction Regulator.

Two problems are resolved by the presence of this valve, the compressor motor is not overloaded and the range of the system is enlarged to include higher enthalpy settings at reduced energy expenditure. This refrigerant constant pressure suction regulator can be installed as an addition to the basic system or with any of its modifications. As long as the minimum evaporator pressure regulator setting is far enough removed from the maximum suction pressure regulator setting, as in the case in this application (where for Freon 12 the evaporator pressure setting is 184 kPa gauge and the suction pressure setting is 285 kPa gauge), the two self contained regulators are completely compatible with each other. They would never be operating simultaneously. When one is called on to move towards closed position, the other is in its open position.

6.8 A Practical Low Cost, Wide Range, Temperature Humidity Climate Simulator

A simple low cost system utilizing the unified approach could satisfy a large number of specifications using inexpensive commercial components that are readily available:

1. An air cooled condensing unit

   Air condensing was selected in order not to limit the market. Frequently a condenser water supply would not be available. The expense of a cooling tower or pump would best be avoided. On the other hand, where it is available the option to supply a water condenser may be offered.

2. Fan

   The fan would require a very low horsepower since the wide range system would be designed in keeping with the findings of Sections 10 and 11.

3. <u>A single direct expansion coil</u>

   The coil selected would be shallow, with a low ratio of secondary fin to primary tube surfaces. See Section 11.

4. <u>Thermostatic expansion valve</u>

   Selected as for basic system. See Section 4.2.3.

5. <u>The reheater and proportional controller</u>

   Selected as for basic system. See Section 4.2.3.

6. <u>The pan humidifier and proportional controller</u>

   Selected as for basic system. See Section 4.2.3.

7. <u>A manually adjustable damper for bypass of air</u>

   A low cost system would avoid the expense of face and bypass dampers and their automatic control. However, provision of some simple system, such as a movable plate bypass damper adjustable in the factory or field would satisfy the variations in users' specifications.

8. <u>An evaporator pressure regulator</u>

   This inexpensive self-contained regulator would serve to prevent frosting. (See Section 4.3.3(c)2). It would be set at the lowest evaporator pressure possible when it is desired to maximize the range. This would probably be approximately -1.7C. A preheater and associated control would not be required. Furthermore, operating settings having low air enthalpies could be included which would normally result in evaporator temperatures below frosting and therefore be out of the permissible range. However, the loads would have to be low enough to be satisfied by the reduced cooling capacity associated with the drop in suction pressure due to the regulator action.

9. A suction pressure regulator

The small modern mass produced hermetic air condensing unit is usually limited to a suction pressure corresponding to an evaporator temperature of 7.2C. The use of this regulator satisfies the dual requirement of limiting the suction pressure to avoid overloading the compressor motor and also obtain air operating settings in the range which otherwise would have resulted in higher suction operating pressures. Not only is the steady flow nature of the unified system retained, but also there is a reduction of the refrigeration capacity for the high entering air enthalpy conditions of the range. See Section 6.7.

10. A reach-in cabinet

This would have to be modified to the user's requirements. If to be used as a plant growth environmental chamber then a barrier lighting system would be recommended with its own fan extraction system.

11. Miscellaneous components and instruments

It would be advisable to include a liquid suction heat exchanger to protect the compressor; a solenoid valve used in connection with a commercial low suction pressure cut-off safety device to evacuate the system on shut down; a simple control panel as in Figure 6.5. However, the wet bulb temperature adjustment potentiometers along with the instruction for their use would be eliminated; a time clock to permit automatic changeover between two operating conditions such as day and night settings, if this is specified. By constraining the refrigeration capacity of this type of hermetic condensing unit to operate between a minimum refrigerant temperature of -1.7C and a maximum temperature of 7.2C only a small energy penalty is imposed during the higher evaporator temperatures. The refrigeration capacity involved would be about that of a large domestic refrigerator. However the coefficient of performance would be considerable better due to the use of a thermostatic expansion valve rather than a capillary tube. The system functions as constrained by the condensing unit curve, the evaporator performance curves and their inter-action with the evaporator pressure regulator and the suction pressure regulator. As indicated above the two regulators are completely compatible with each other. They do not operate simultaneously. When one is operating the other is in it's open position since their respective pressure settings are far enough apart to prevent any overlap of control action. Figure 6.9 may be viewed as representing a combination of Figures 6.7 and 6.8 in its effect on the system.

6.9 Accommodation of Differences in User's Specifications

A manufacturer would be able to offer the unified system in a variety of forms. A typical line of chambers is listed in Table 6.1 giving a wide choice in size, range, controls, lighting and air flow pattern.

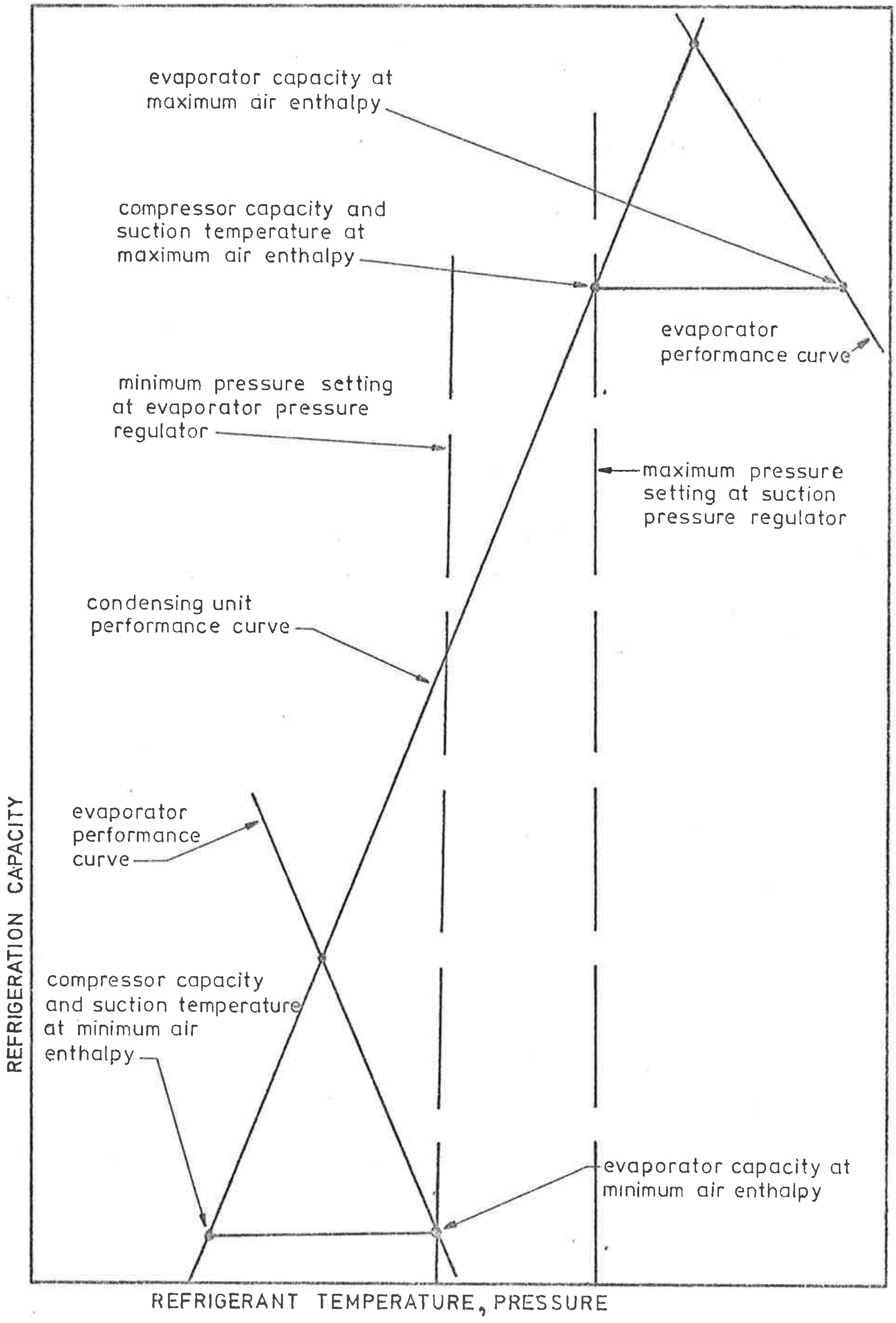

Fig. 6.9 Performance of Low Cost, Wide Range Temperature Humidity Climate Simulator.

TABLE 6.1 A TYPICAL LINE OF CHAMBERS PER UNIFIED SYSTEM

Catalogue Number	Size Per Type	Bench Area (m^2)	Inside Dimensions W. ×	H. ×	D. (m)
S-1*	Reach-In	0.84	1.37	0.91	0.61
S-2**	Reach-In	1.49	1.98	1.22	0.76
S-3	Reach-In	2.97	1.98	1.22	1.52
S-4	Walk-In	2.42			
S-5	Walk-In	3.44			
S-6	Walk-In	5.57			

Height may be varied to user's requirements;
*Will fit through a 0.76m. wide door;
**Will fit through a 0.91m. wide door.

Catalogue Number	Range	Temperature Range	Humidity Range
R-1	Temperature-Only	4.4C to 43C	
R-2	Standard Range	4.4C to 43C	18% to 95% Relative Humidity
R-3	Wide Range	4.4C to 43C	12% to 95% Relative Humidity

Temperature and Humidity Range may be enlarged on application.

Catalogue Number	Provision for Changeover of Set Points
C-1	Manual Changeover
C-2	Automatic Changeover via Time Clock (Day-Night)
C-3	Automatic Changeover via Cammed Program

Catalogue Number	Lighting
L1	Interior Lights - operated by on-off switch
L2	Artificially Lighted to Simulate Sunlight
L3	Naturally Lighted without provision for Automatic Night "Blackout"
L4	Naturally Lighted with provision for Automatic Night "Blackout"

Catalogue Number	Air Flow Pattern
A1	Cross Flow
A2	Upward Flow

7. DESCRIPTION OF SYSTEMS BUILT TO THE UNIFIED APPROACH

7.1 The Waite Institute Phytotron Unit

7.1.1 Description

The Waite Institute System represents the first prototype built to the unified approach. It therefore most closely represents the basic system described in Sections 4.2 to 4.2.3 inclusive.

In addition to the basic system it possesses the flexibility afforded by an open belt-driven compressor. It is equipped with several sizes of sheaves which can be quickly rearranged to permit compressor speeds varying from 460 to 1440 r.p.m. in increments of about 100 r.p.m. There is also a manual damper located at the outlet of the fan which permits adjustment to the flow rate of the air.

The condenser is water cooled. This gives some latitude for adjusting the condensing pressure by means of controlling the condenser water flow rate.

The system is equipped with a time clock and relays which are interlocked with the lighting section simulating sunlight. The system has the capability of automatically changing over between any combination of 'day' temperature and humidity to any other combination of 'night' temperature and humidity. The proportional controller as described in Section 4.2.3 acts to establish a steady flow refrigeration cycle. The desired dry bulb and dew point temperatures are obtained by the addition of sensible and latent heat to a fixed energy datum established by the evaporator surfaces.

7.1.2 Performance

The performance of the Waite Institute Phytotron Unit was reported in the Journal of the Society of Environmental Engineers, (Shaw 1975). A period of continuous operation over several days was recorded and at the end of this period a meeting of the Society of Environmental Engineers was held at the site to observe the system in operation.

The following is a description of the report. The associated recorder chart performance data is presented in Figure 7.1.

A twenty-four point temperature recorder is located in the upper right hand corner of the instrument and control panel, (Appendix III). The chart is moving at about 10 cm per hour. The pressure gauges can be seen on the upper left hand panel. A time clock is located to the left of the centre panel. Two wattmeters are located to the right of the time clock. The wattmeters measure the energy input to all heaters including a pan humidifier.

In reporting system performance there are five temperature readings that are most important. In order to facilitate reading these points a continuous line connecting the print-out points was drawn onto the recorder chart. These points are numbered 2, 8, 10, 19 and 21. (See Figure 7.1). Point 19 represents the air dry bulb temperature before entering the chamber from a plenum below.

Point 21 represents the controlled dry bulb temperature within the climate simulator.

Point 8 represents the controlled humidity by way of a cavity temperature within the bobbin of a lithium chloride dew point temperature sensing element. The cavity temperature recorded, Point 8, is very nearly directly proportional to the dew point temperature.

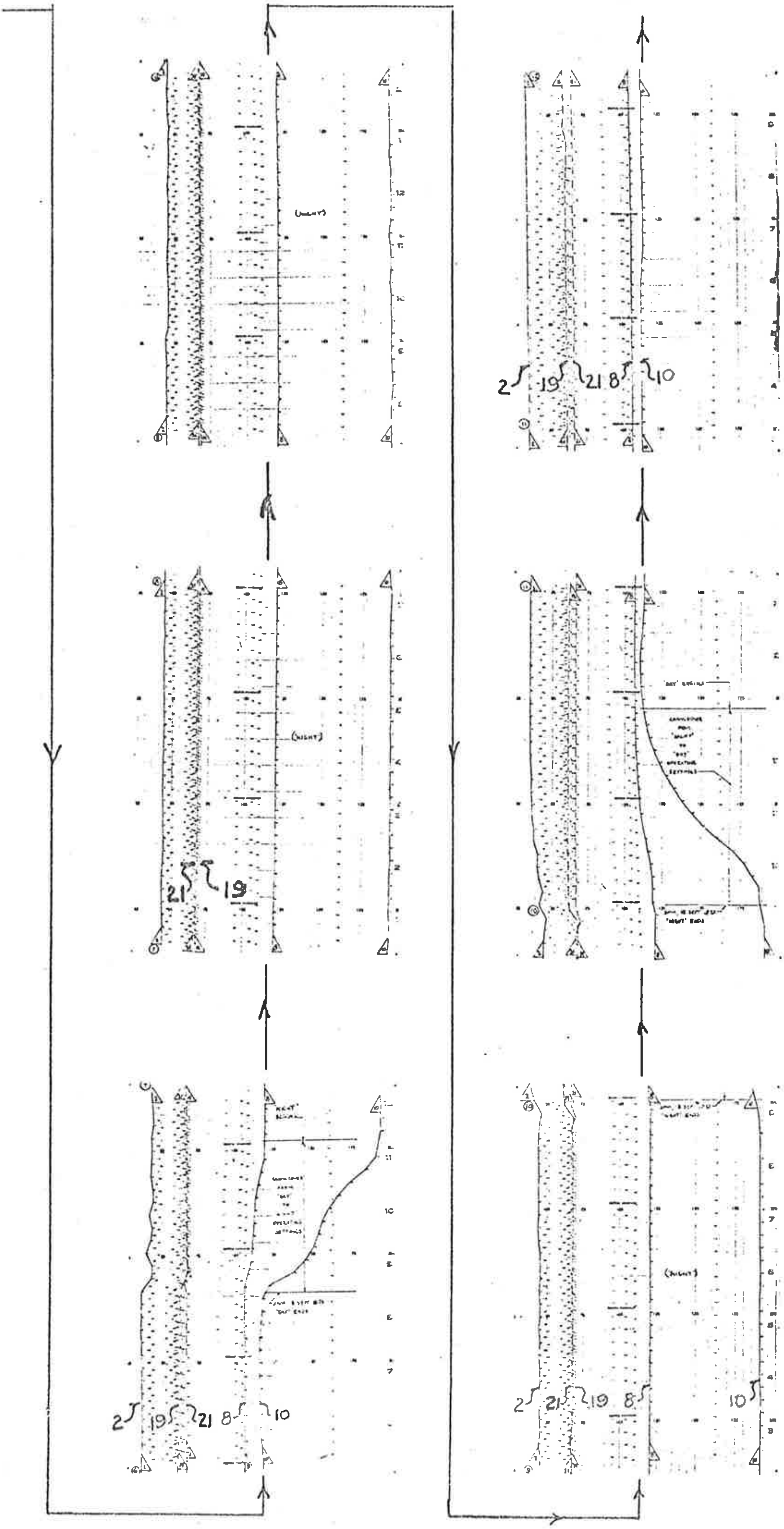

FIG. 7.1 PERFORMANCE RECORD
WAITE INSTITUTE PHYTOTRON UNIT

Point 2 represents the temperature of the refrigerant leaving the evaporator.

Point 10 represents the temperature of the water in the pan-humidifier. This water surface temperature is controlled to maintain the humidity setting. The relationship between these two points, 8 and 10, is apparent when one examines the recorder chart.

A precision Assmann Psychrometer is located within the climate simulator. This instrument can be seen in the photograph of the chamber, (Appendix IV).

During the S.E.E. visit the Waite Institute Phytotron Unit was being used in a research program by the Agronomists. Control points of their choice were selected to vary automatically from one set of operating conditions, (temperature and humidity), under simulated 'day' to another set of 'night' conditions. Thus the dynamics of the control action over this period is demonstrated on the recorder chart.

The twelve sections of the chart cover the period mentioned above with changeover occurring automatically between day and night settings. Each of the twelve sections overlaps in order to maintain this continuity. The triangular flags help identify the five print-out points described above. During the day cycle the lighting section is on. There are fourteen 400 watt mercury vapour lamps occupying the 1.9 m^2 ceiling or 'sky' area of the chamber. The radiant effect can clearly be seen when comparing the temperature difference between Points 21 and 19 during the day and during the night periods.

The minor fluctuation of Point 2 is to be expected when a commercial thermostatic expansion valve is used to maintain a fixed superheat condition.

In addition to the five points on the recorder chart described above, which have had a continuous line drawn in connecting their print-out points, there are several other points worthy of mention.

Point 4 is recorded approximately 11C above Point 2. It represents the temperature of the superheated refrigerant vapour at the exit of a liquid suction heat exchanger. This exchanger is used in the system as a further safeguard against liquid refrigerant entering the compressor inlet.

Point 3 represents the compressor discharge temperature.

Point 5 represents the dry bulb temperature upstream of the pan humidifier.

In all, there are thirty six different temperature readings that can be measured on this twenty-four point recorder. Twelve points have two readings each. The position of a sliding switch determines which of the two readings is being recorded.

The recorder chart demonstrates the close tolerance control of the dry bulb temperature Point 21 and the humidity, dew point cavity temperature, Point 8. Automatic changeover is identified on the recorder chart. From this recording it can be observed that no cycling occurs between the two controlled variables either during the change-over (or start-up) period or after the operating settings are reached. Furthermore repeatability, the automatic recurrence of the desired settings, is also demonstrated.

Thermohydrographs for several random operating conditions are also included in Appendix I, A to E.

The shallow, 3 row deep coil installed at the Waite Institute has a face area of 0.66 m^2. The average face velocity is well below conventional practice, ranging from 0.5 to 1.4 m/s. See drawing of this dehumidifier in Appendix II.

7.1.3 Patents

With the support of the Rural Development Corporation of London, patent applications were filed in four countries, Australia (Shaw 1969a), United Kingdom (Shaw 1969b), The United States of America (Shaw 1969c) and the Federal Republic of Germany (Shaw 1973). In order to limit the size of this thesis, the first page and only the main claims of the patent specifications are presented in Appendix VII A to D.

7.2 Commercial Systems

Several prototype commercial systems were built under licence.

A two direct expansion coil and two hermetic air cooled condensing unit system was built as described in Section 6.3. A photograph of this system is shown in Appendix V** (The two condensing units are located to the left of the photograph).

A face and bypass system using a single direct expansion coil and an air cooled condensing unit was built as described in Section 6.4. Performance of both systems was similar to that of the Waite Institute System. Performance data for the face and bypass system at ten test points in the range are indicated on a thermohydrograph (see Appendix VI).

7.3 The Mechanical Engineering Department Unified System

7.3.1 Description

The design is a modified version of the basic system described in Section 4. The modifications and spatial features of the system include:

an open compressor with manual variable speed control to compressor motor, (Figure 7.8)*,

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* Figures 7.2 to 7.12 are located at the end of Section 7.

** Courtesy of Paton Industries Pty. Ltd.

fan with manual variable speed control to fan motor, (Figure 7.3),

evaporator pressure regulator with bypass,

liquid suction heat exchanger with bypass, (Figure 7.8),

condenser pressure controller, (Figure 7.9),

timer and cascade controllers to permit automatic changeover between pairs of temperature-humidity operating settings, (Figure 7.9),

a condenser water system with cooling tower, (Figure 7.11), and controlled cooling tower fan.

The system is extensively instrumented. There are temperature and pressure measurements taken along the paths of air, water and refrigerant flow.

Rotameters are installed in the refrigerant and condenser water systems. (See Figure 7.8).

Wattmeters are installed to measure rate of preheat reheat and humidification. (See Figure 7.9).

Assmann psychrometers with electrically driven fans are installed within the air cycle upstream and downstream of the dehumidifier, (Figure 7.6) and

Provision is made for measuring the condensed water flow rate at the dehumidifier.

The pan-humidifier section is completely insulated. (See Figure 7.10).

One major modification to the basic Waite Institute prototype unit was to upgrade the air system for use in teaching and research. In the new research system air flow rate can be determined through a Venturi tube employing an inclined manometer. (See Figure 7.4). The accurate measurement of air flow is very difficult and secondary methods were considered inadequate. The Venturi tube adheres closely to the British Standards Institution (1964).

The air stream entering the dehumidifier has a very good flow pattern. There is a contraction having an area ratio of 8:1 immediately before the dehumidifier. Directly upstream the contraction is a grid section packed with 114,000 straws. Each straw is 212 mm long by 5 mm in diameter. (See Figure 7.5).

At the Waite Phytotron Unit space requirements made it very difficult to measure the dry bulb and the wet bulb temperatures downstream from the direct expansion coil and the humidifier. There was less than 0.5 metres between some sections of the downstream side of the dehumidifier and the upstream side of the reheater.

In the Department of Mechanical Engineering system the duct work extends almost the complete length of the laboratory plus part of its width. This ensures uniformly mixed air at all critical measuring stations located between the dehumidifier, reheater, pan humidifier, fan and Venturi meter. See Appendix VIII A for the duct-work drawings. See Figure 7.2 for photographs.

Two twenty-four point temperature recorders are mounted on the instrument panel. (See Figure 7.9). Recorder 1 employs resistance bulb-sensors, Recorder 2, thermocouples. The calibrated correction chart, Table 7.1, identifies each print out point.

	NO.	CALIBRATION	POSITION	NO.	CALIBRATION	POSITION
RECORDER NO 1	1	-1°C	FREON UPSTREAM TX VALVE	13	-3.2	CWS AT MAKE UP TANK
	2	-2.5°C	CWS SUCTION TO PUMP	14	-2.7	AIR DBT CHAMBER
	3	-1.2	FREON DOWNSTREAM DX COIL	15	-	OPEN CIRCUIT
	4	-1	FREON SUCTION TO COMPRESSOR DOWNSTREAM LSHE	16	-	OPEN CIRCUIT
				17	NOT CALIBRATED	DX COIL RETURN BEND
	5	-1.7	FREON CONDENSER	18	-1	AIR DOWNSTREAM DX COIL
	6	-1.5	FREON LIQUID DOWNSTREAM CON-DENSER UPSTREAM FLOW RATOR	19	-1	FREON LIQUID UPSTREAM LSHE
	7	-2.5	AIR DBT DOWNSTREAM REHEATER	20	NOT CALIBRATED	AMBIENT
	8	-1	FREON LIQUID DOWNSTREAM LSHE	21	-1	AIR DBT UPSTREAM DX COIL
	9	-1	CWS AT CONDENSER INLET	22	-	OPEN CIRCUIT
	10	-1.5	CWR AT CONDENSER OUTLET	23	-1.5	AIR DBT UPSTREAM PREHEATER
	11	+ .5	CWR COOLING TOWER	24	+5.5	FREON DOWNSTREAM DX COIL & UP-STREAM EVAP. PRESS. REG.
	12	-2.5	DBT FAN DISCHARGE			
RECORDER NO 2	1-12	-	CONNECTED TO VARIOUS POINTS ON DX COIL, (NOT IN USE AT PRESENT)			
	13		PAN HUMIDIFIER SURFACE WATER			
	14-23	-	OPEN CIRCUIT			
	24		DEW PROBE CAVITY			

TABLE 7.1 RECORDER CHARTS
MECHANICAL ENGINEERING DEPARTMENT UNIFIED SYSTEM

Recorder chart sections for several operating conditions are shown in Figures 7.12 A to D*. Point 14 on Recorder 1 and Point 24 on Recorder 2 are two important readings. They indicate the ability of the control system to maintain constant dry bulb and dew point temperatures.

Point 1 of Recorder 1 reflects the dryness fraction of the refrigerant. See Section 12.2.3 paragraph numbered 5 regarding the range of adjustment available.

When steady flow is established Point 3 on Recorder 1 remains fairly constant reflecting only the minor action of the thermostatic expansion valve to maintain a fixed superheat setting. See Section 12.2.3 paragraph numbered 6 regarding the range of adjustment that is possible to the superheat setting.

The use of the system as described in Sections 8, 9 and 12 attests to its applicability for teaching and research in the air conditioning and heat and mass transfer fields.

Full manufacturer's data on all system components are available.

The system is further described in Sections 12.2 to 12.2.4 in association with the research program confirming the findings developed in Sections 10 and 11. Drawings of the major components are presented in Appendix VIII, A to E. These are located in a pocket inside the back cover.

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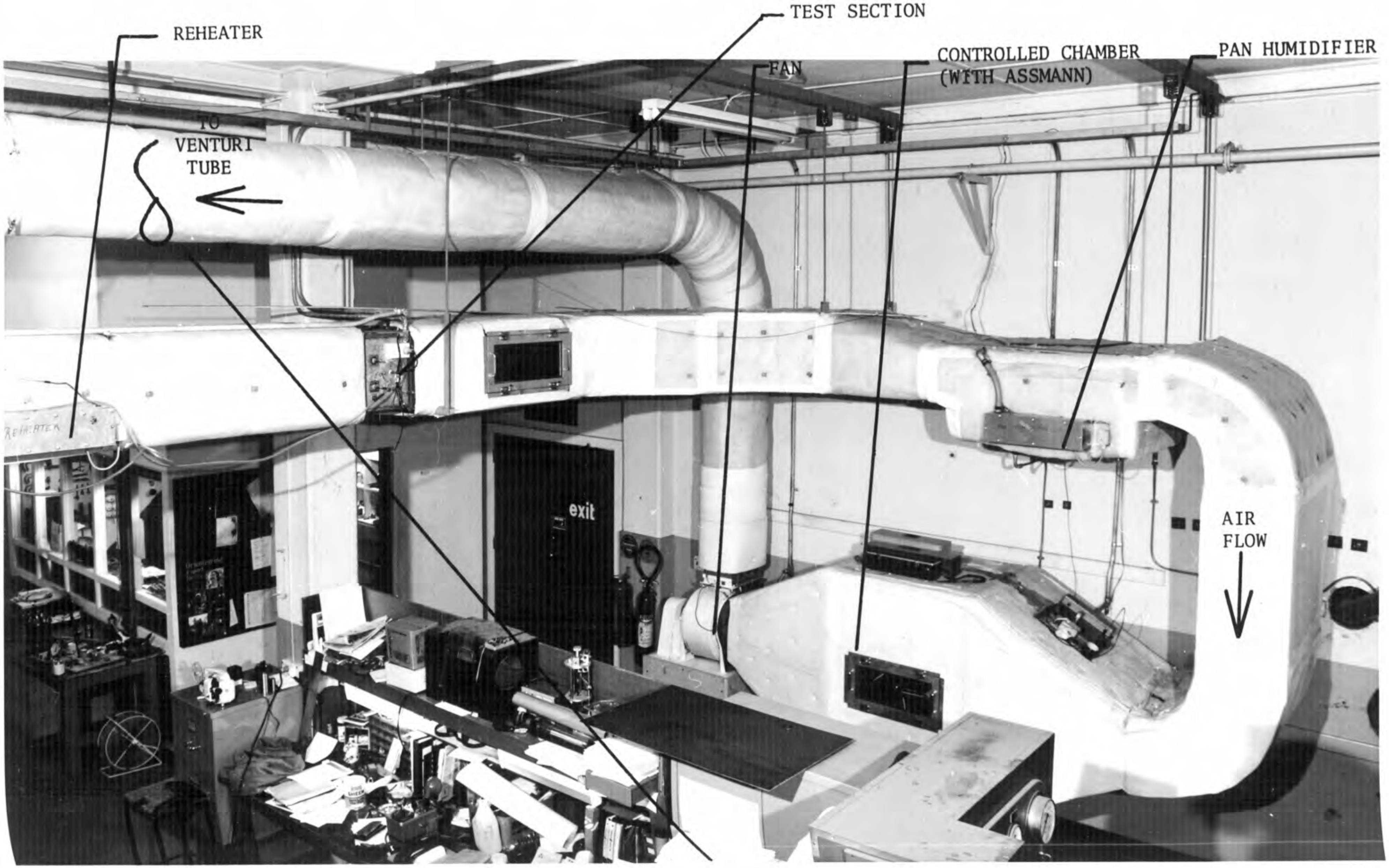
* The recorder charts of Figures 7.12A and B represent operating conditions of the high velocity member of Run 7. The recorder charts of Figures 7.12C and D represent the low velocity member of Run 7. These runs are part of a series that were used in Projects I and II as described in Sections 12.5.1 and 12.5.2.

7.3.2 Performance

In most respects the performance of the system meets the design expectations. There is no voltage regulator and this causes minor variations that are observable during peak periods and change-over periods at the power station. However in spite of these effects, the dry bulb-temperature setting is maintained within a tolerance of $\pm$ 0.1C and the dew point temperature setting within a tolerance of $\pm$ 0.2C. This system has advantages over the Waite Institute Phytotron Unit which give increased scope to teaching and research projects.

7.3.3 Patent

The confirmation of the findings of Section 11 by use of the Mechanical Engineering Department Teaching and Research System has led to the filing of a provisional patent in Australia, Shaw and University of Adelaide (1979). The first two pages and claims are presented in Appendix IX.


REHEATER
TEST SECTION
FAN
CONTROLLED CHAMBER
(WITH ASSMANN)
PAN HUMIDIFIER
TO
VENTURI
TUBE
REHEATER
exit
AIR
FLOW

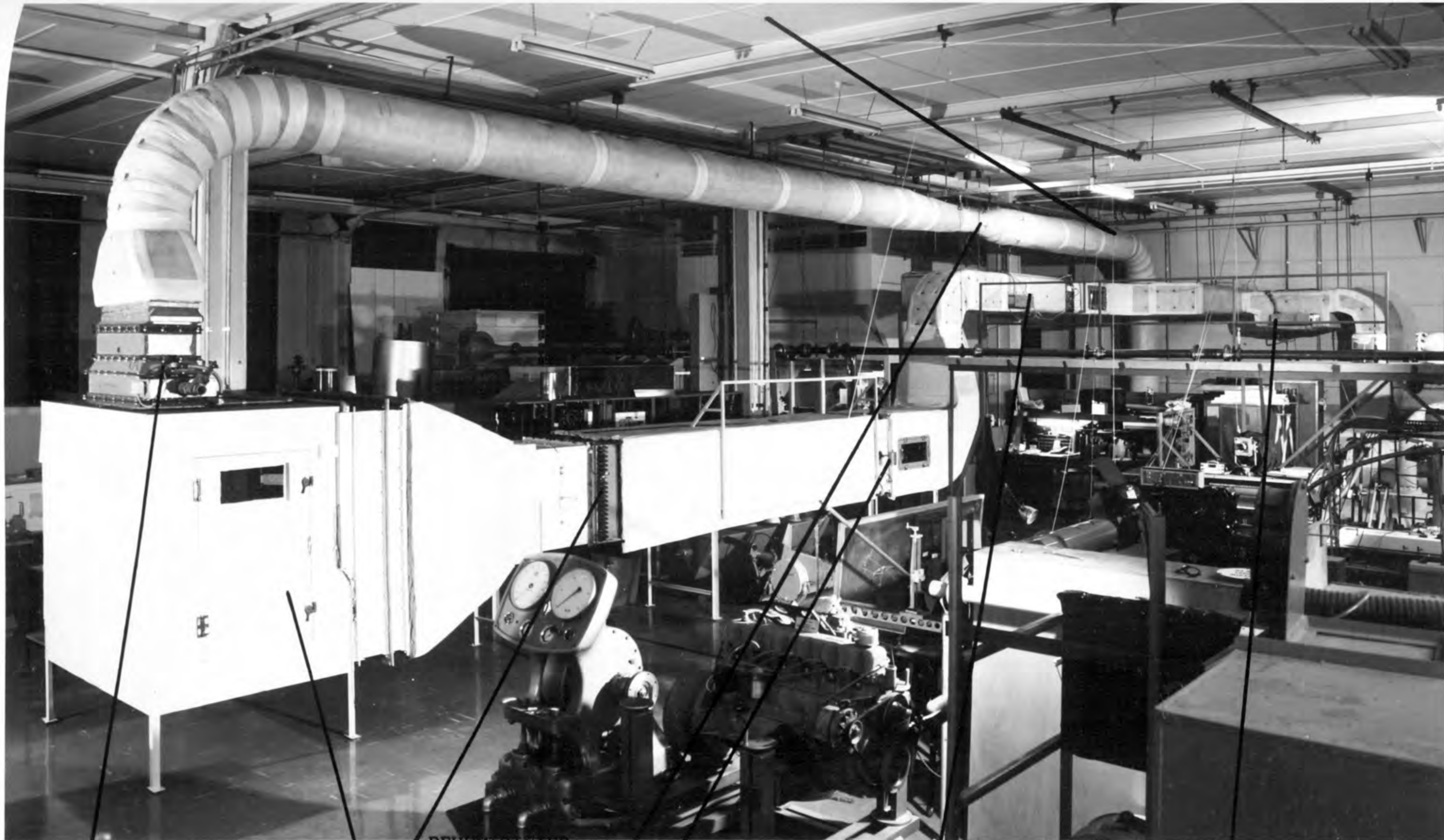


FIG. 7.2 AIR CYCLE

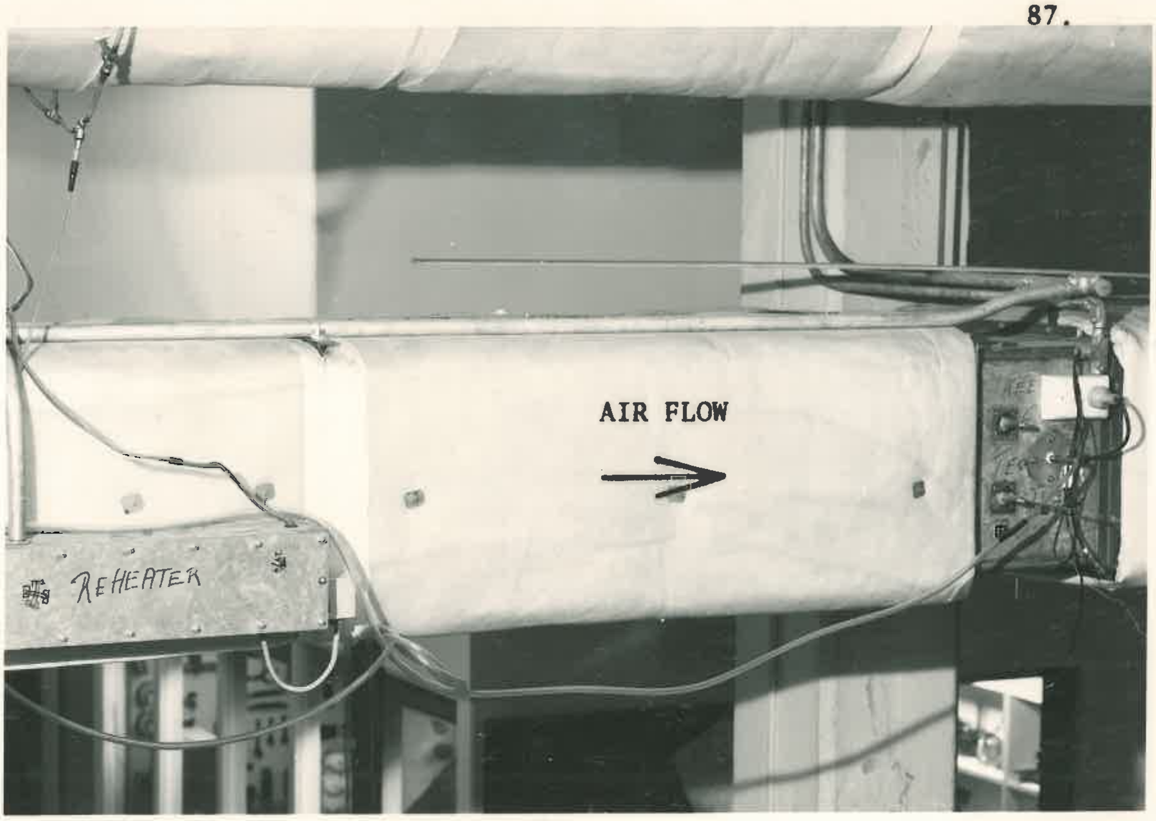
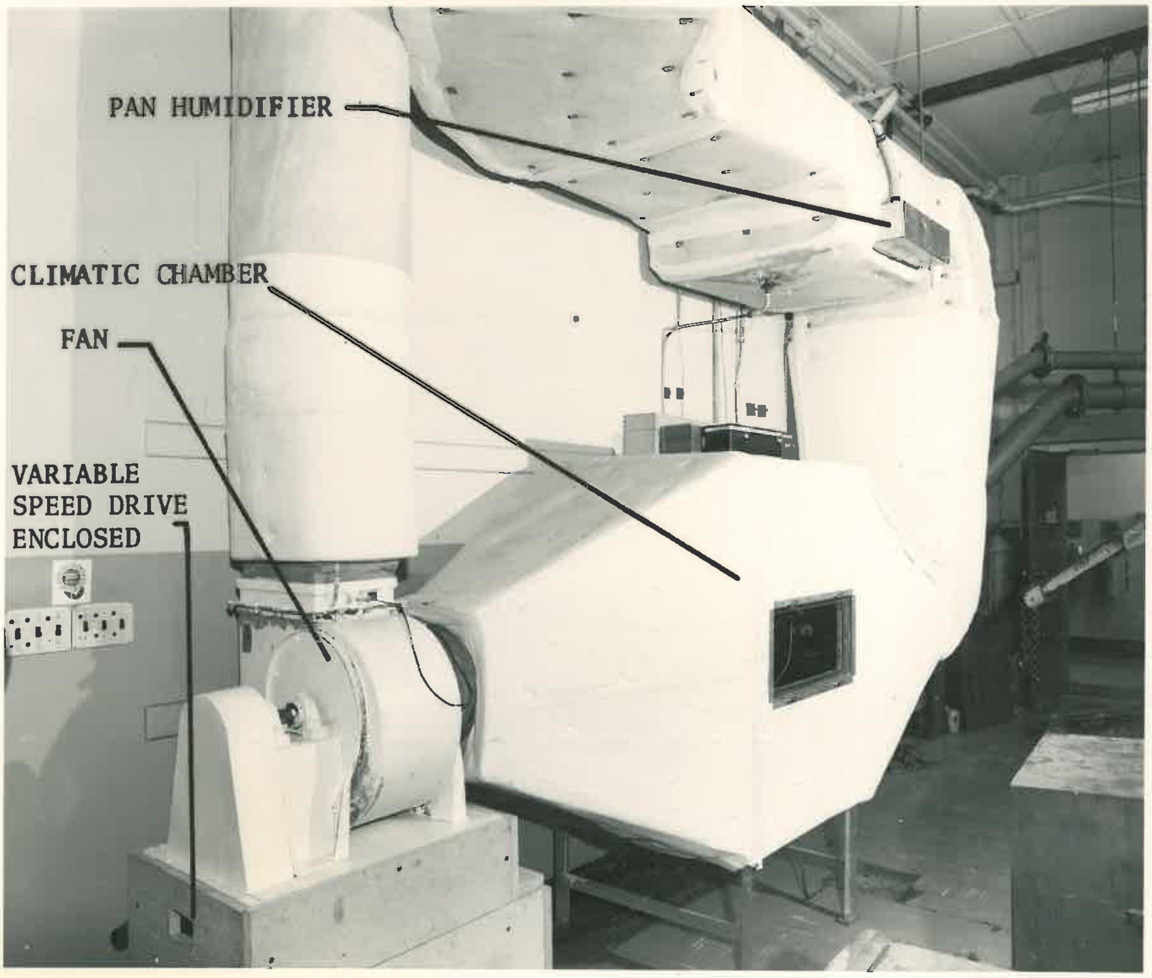


FIG. 7.3 REHEATER AND TEST SECTION (Above)

PAN-HUMIDIFIER AND CLIMATE SIMULATOR (Below)



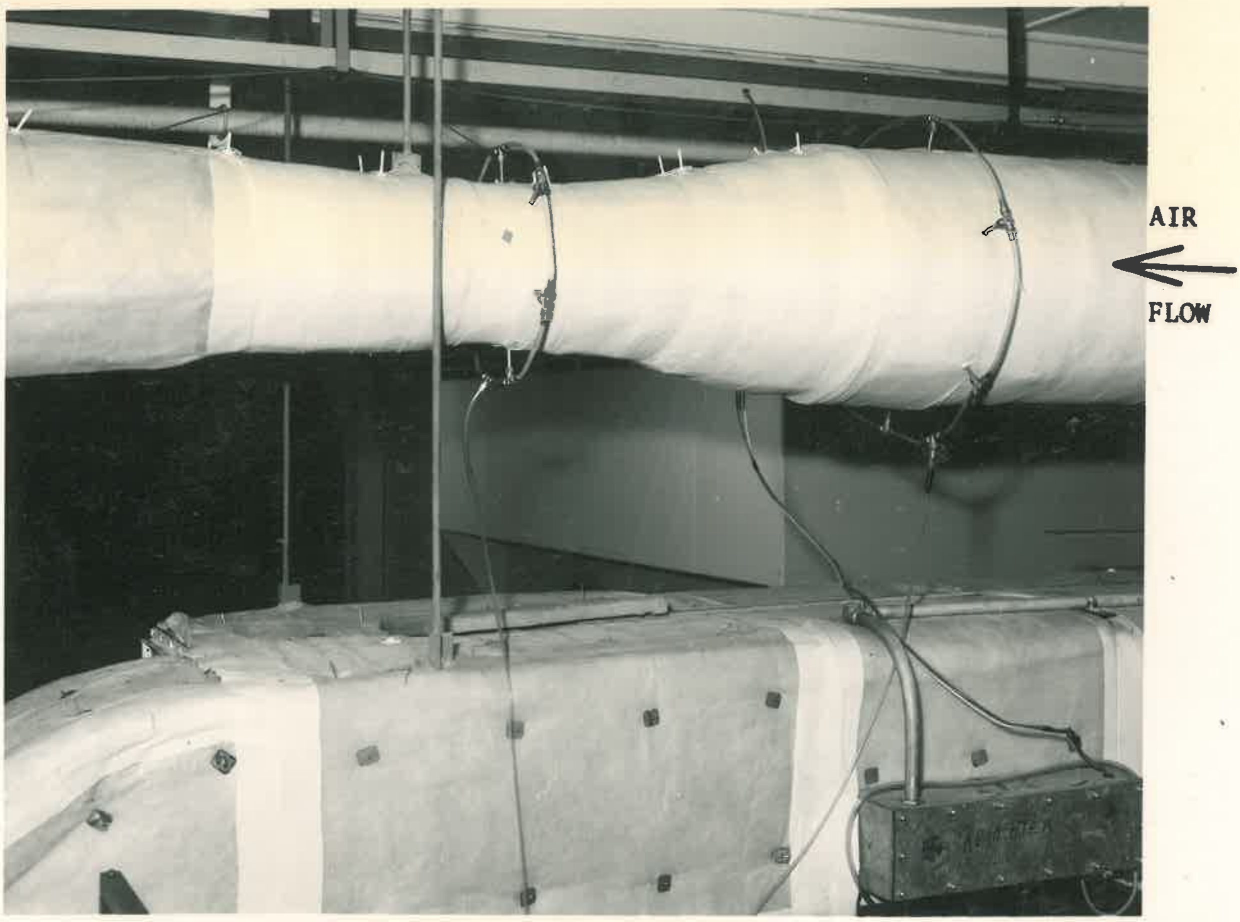
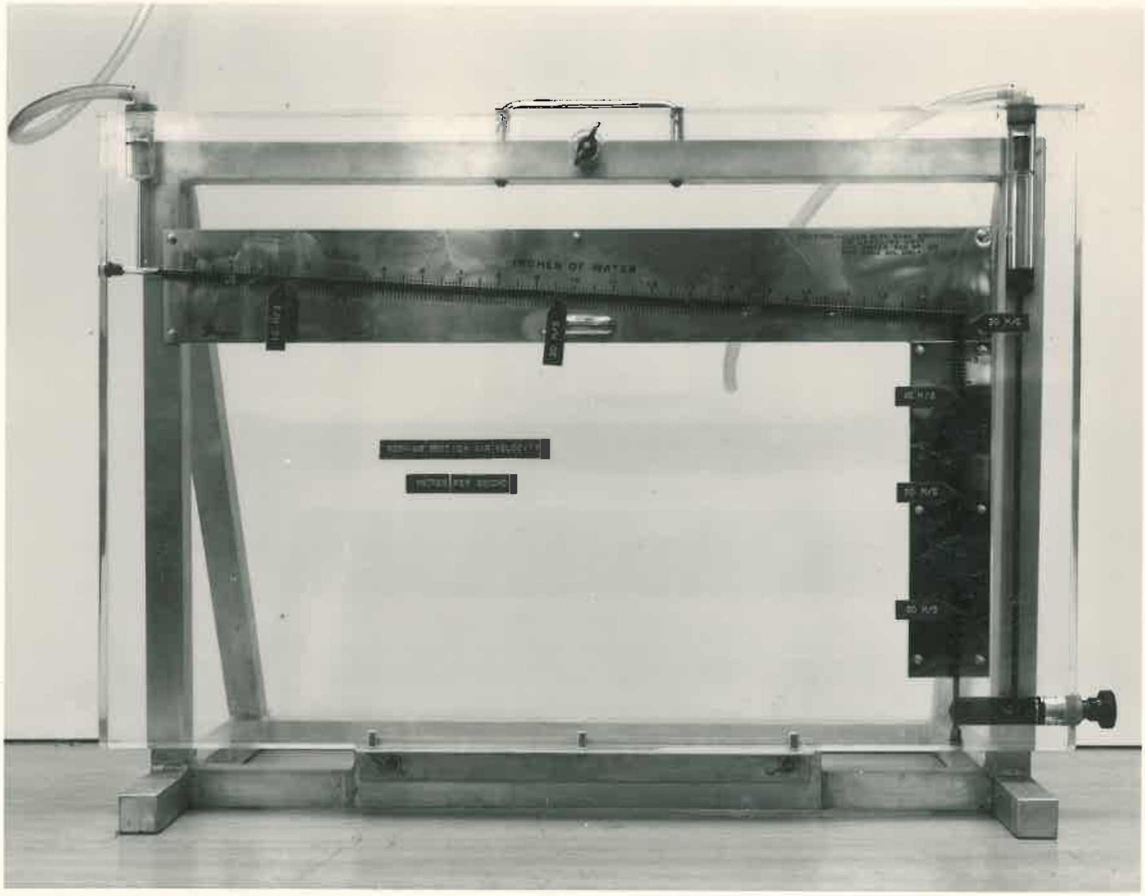


FIG. 7.4 VENTURI TUBE (Above)

INCLINED MANOMETER (Below)



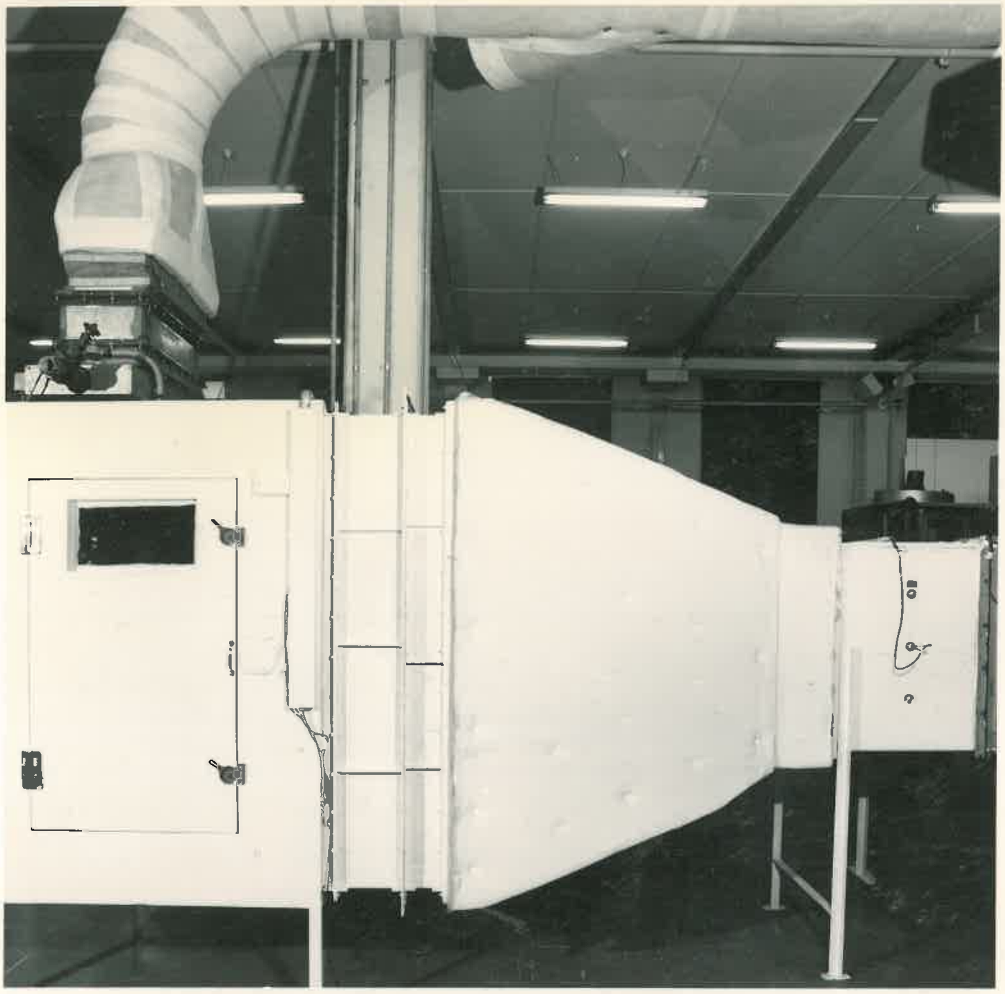


FIG. 7.5 AIR PLENUM AND CONTRACTION (Above)

GRID OF STRAWS UPSTREAM OF CONTRACTION (Below)

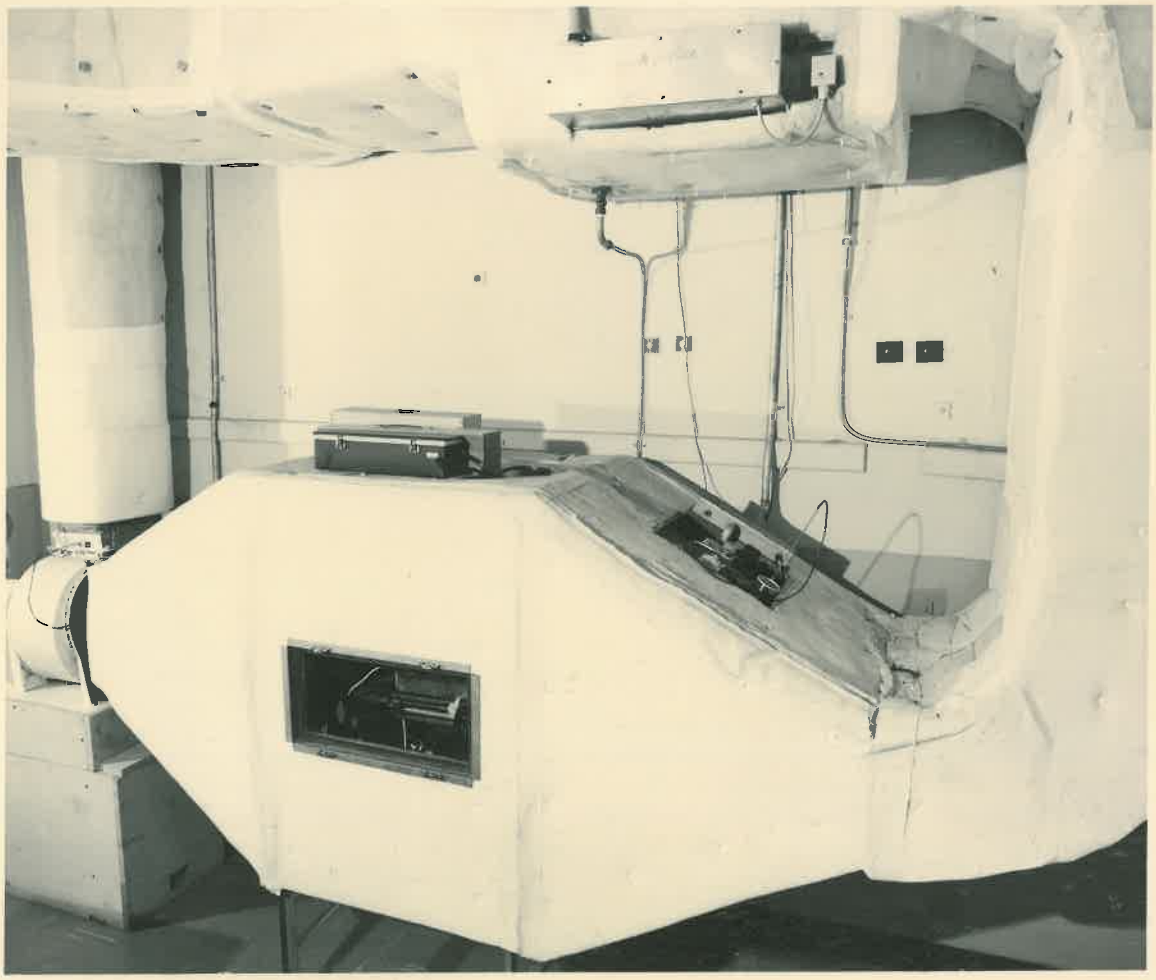
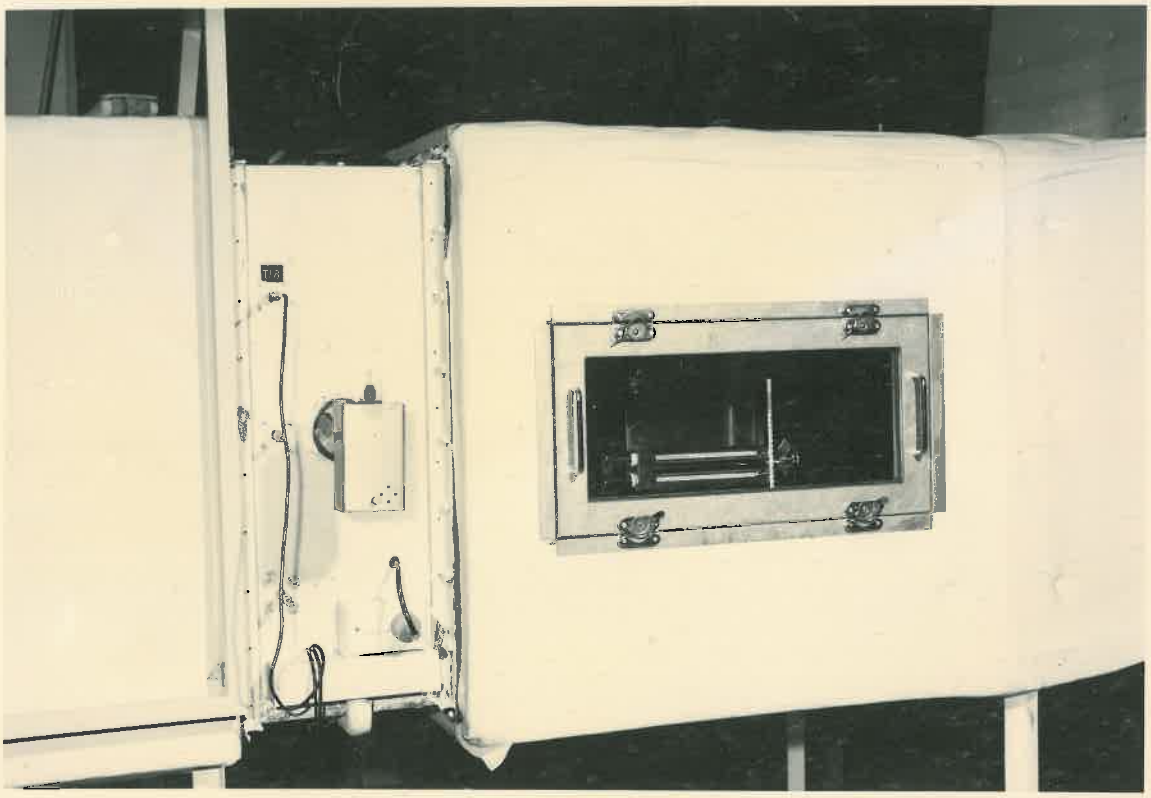


FIG. 7.6 ASSMANN PSYCHROMETER LOCATIONS

UPSTREAM DEHUMIDIFIER (Above)

DOWNSTREAM DEHUMIDIFIER (Below)



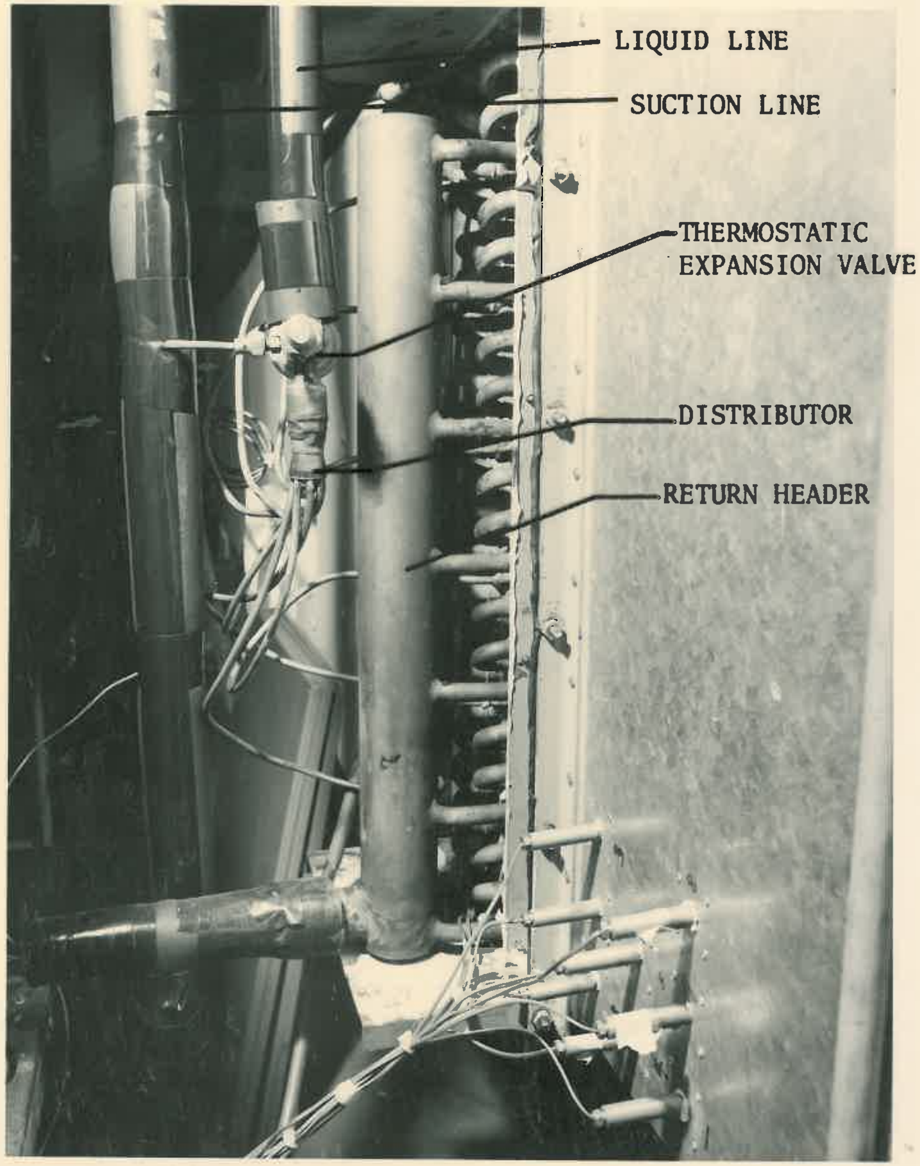
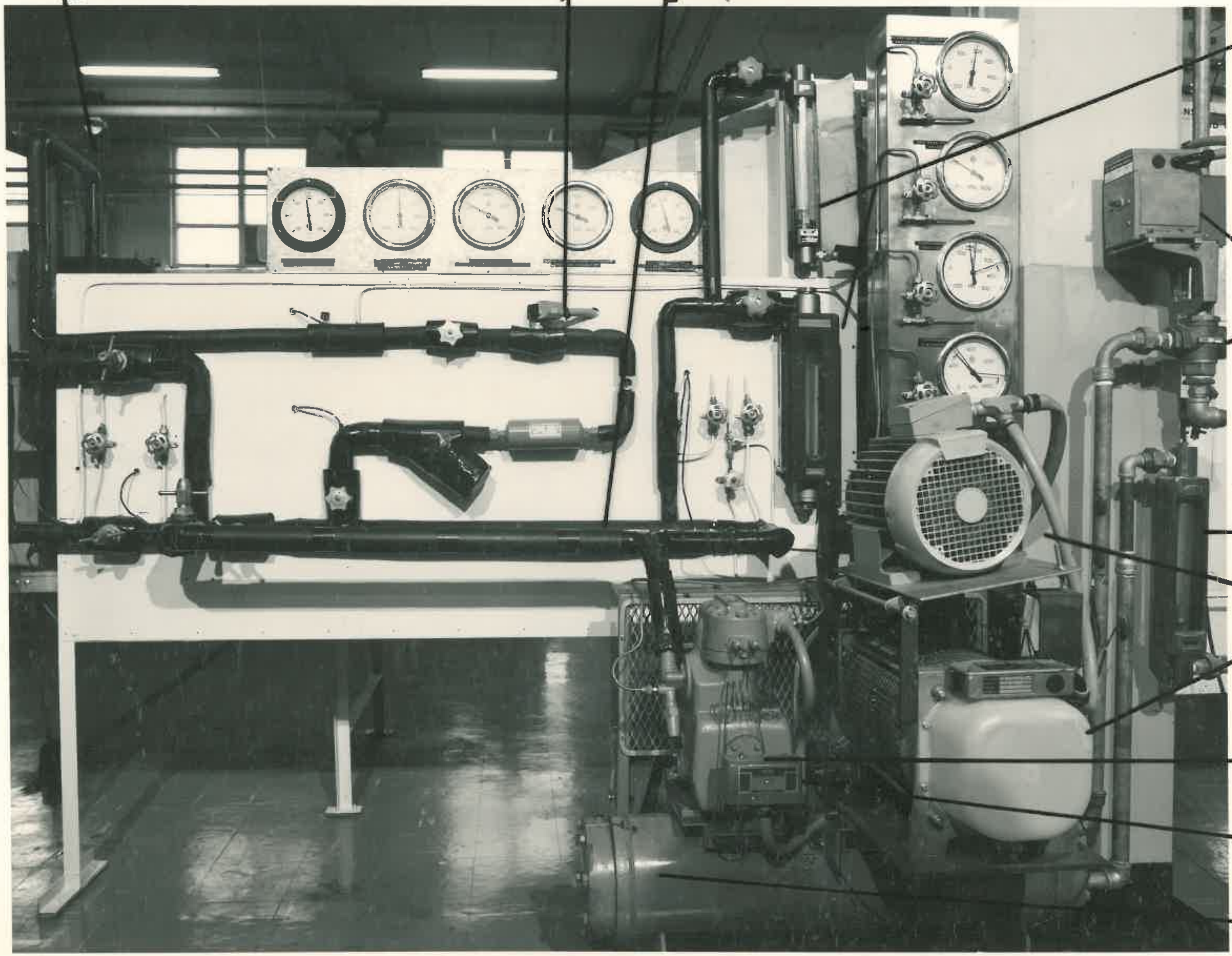


FIG. 7.7 DEHUMIDIFIER AND CONNECTIONS

(DEHUMIDIFIER CONTRIBUTED BY CARRIER AUSTRALIA)



<u>FIG. 7.8</u> REFRIGERATION SYSTEM

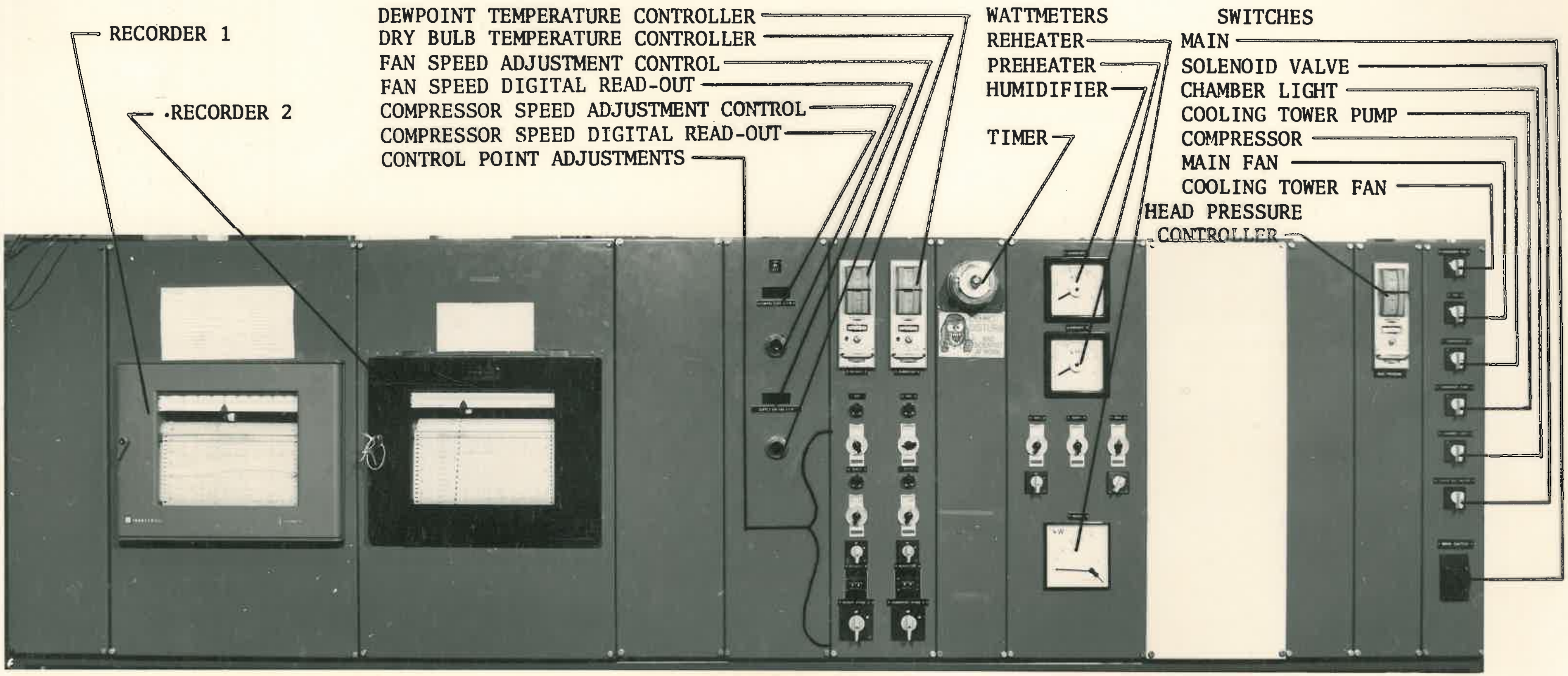


FIG. 7.9 CONTROL AND INSTRUMENT PANEL

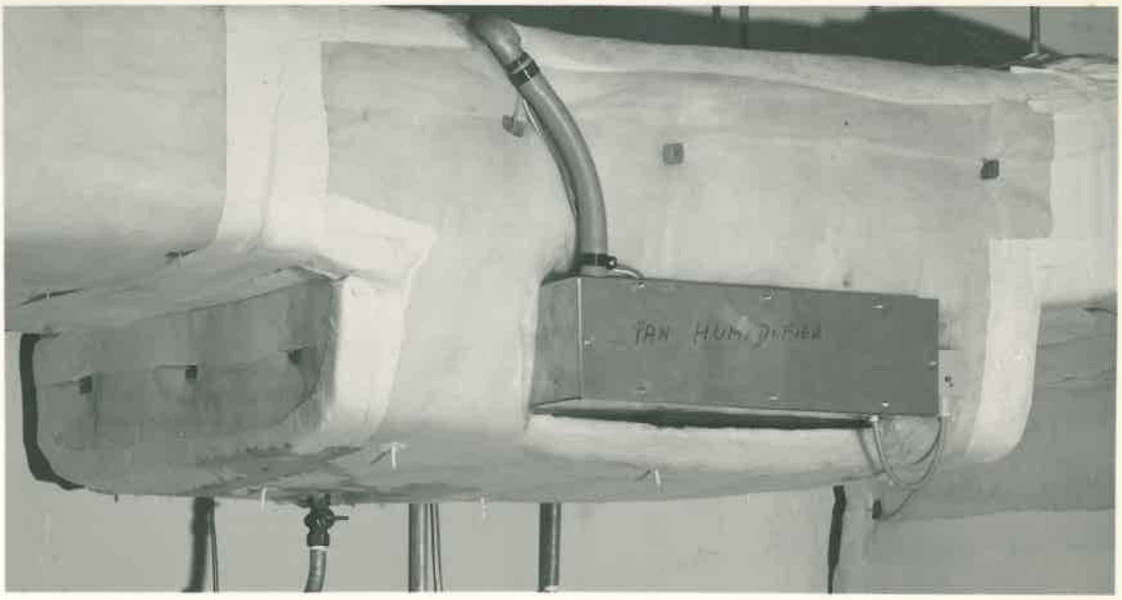
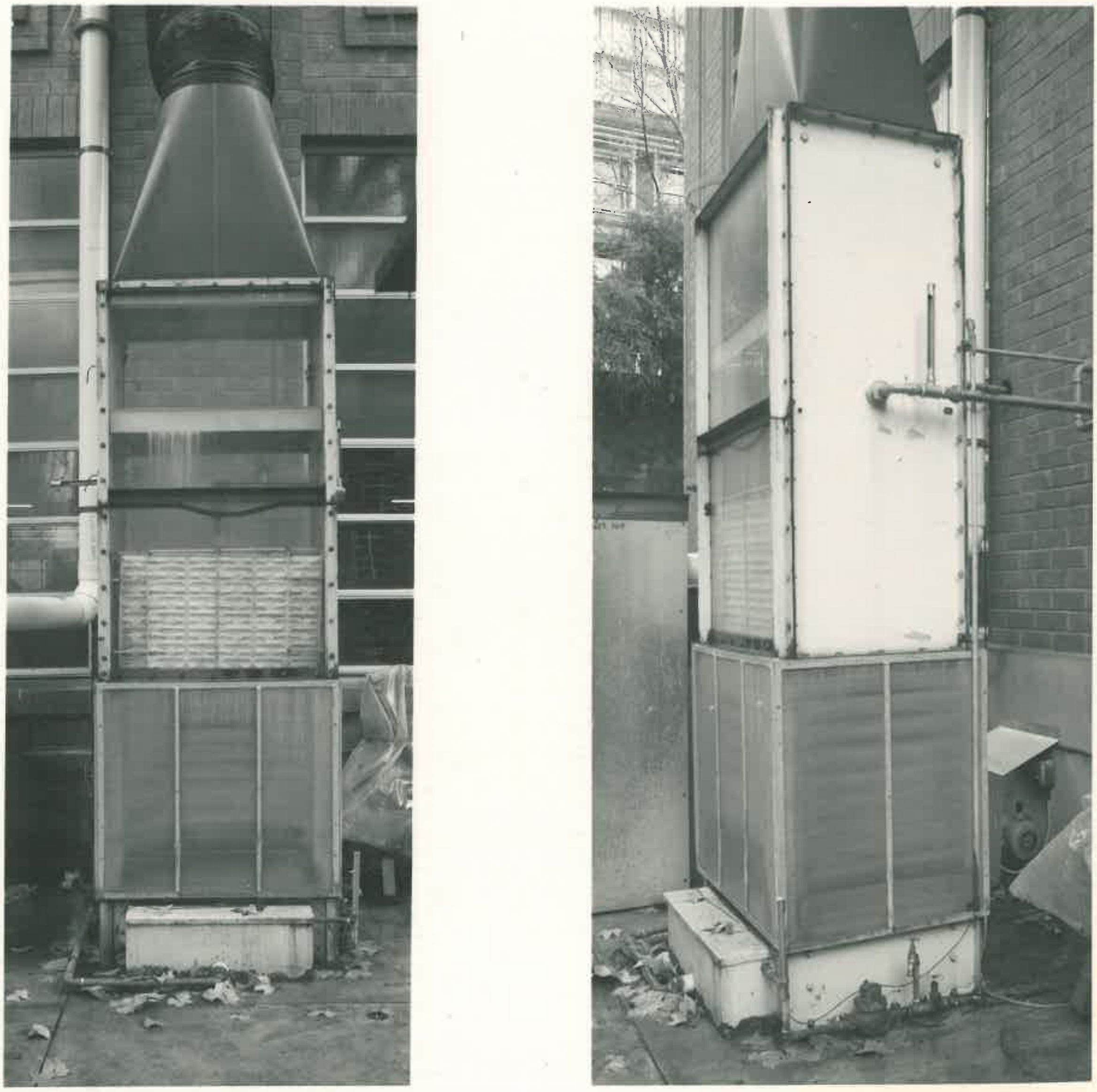


FIG. 7.10 PAN HUMIDIFIER (Above)

FIG. 7.11 COOLING TOWER (Below)



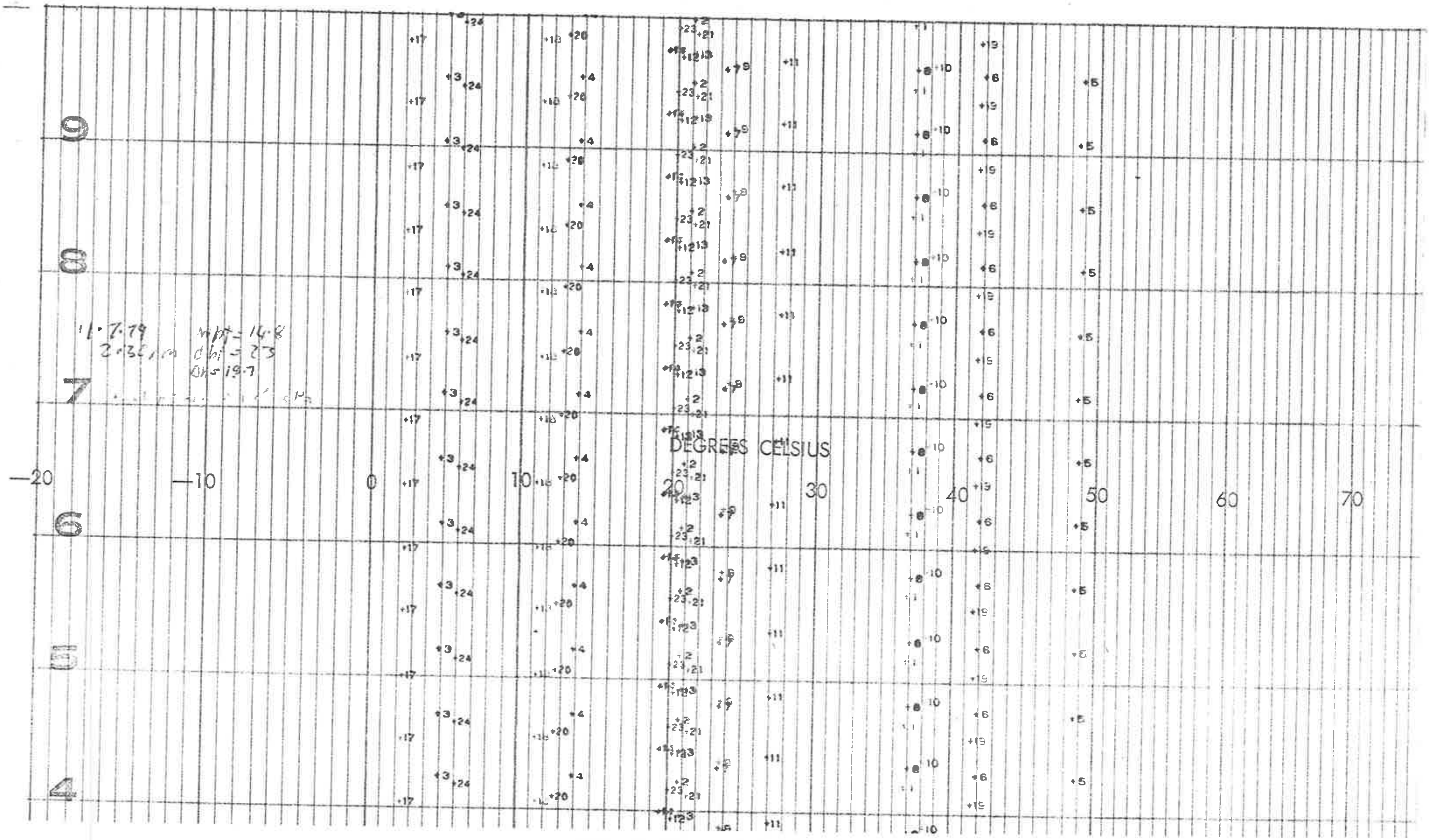


FIG. 7.12A RECORDER CHART 1, RUN 7HV

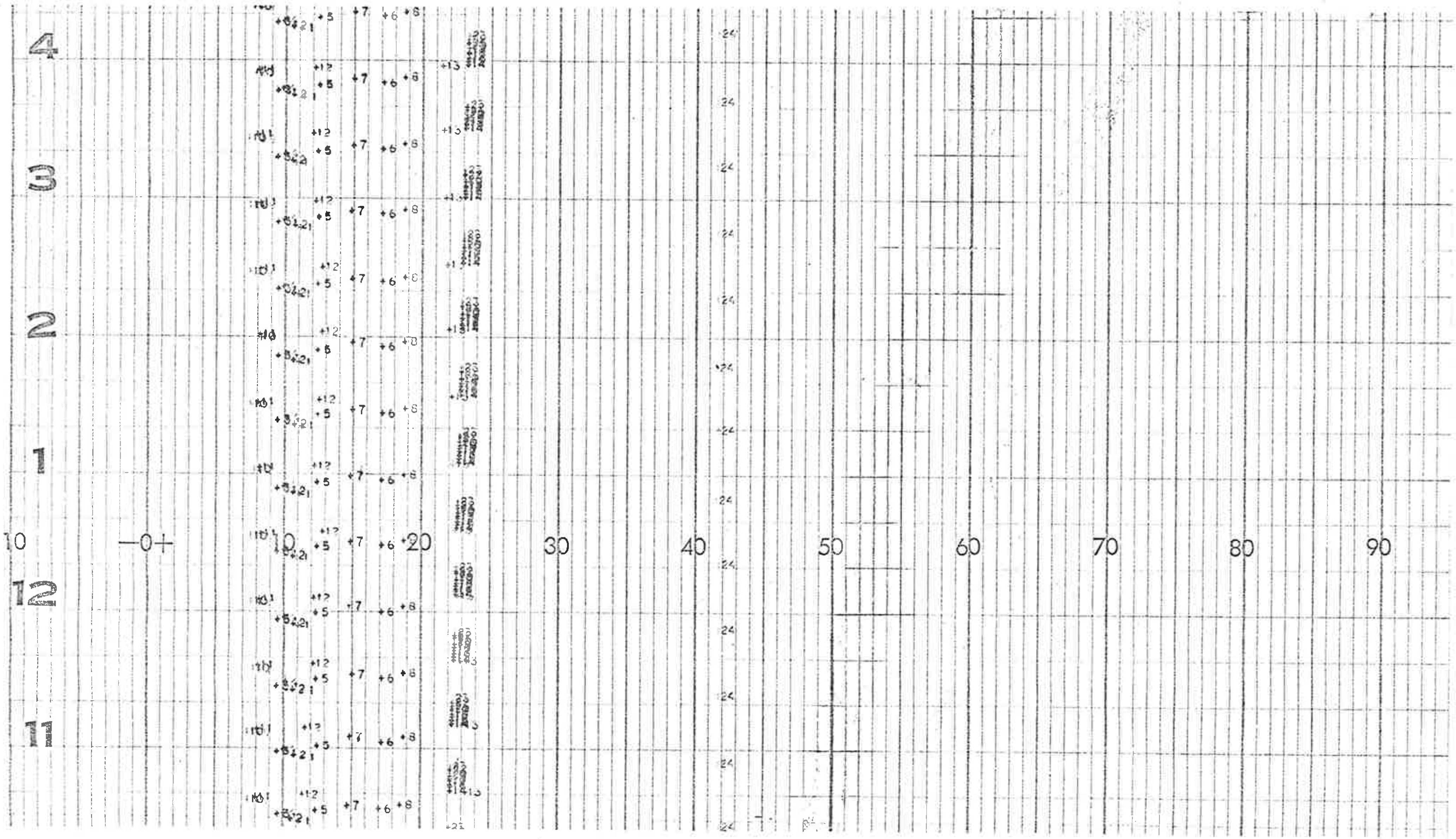


FIG. 7.12B RECORDER CHART 2, RUN 7HV

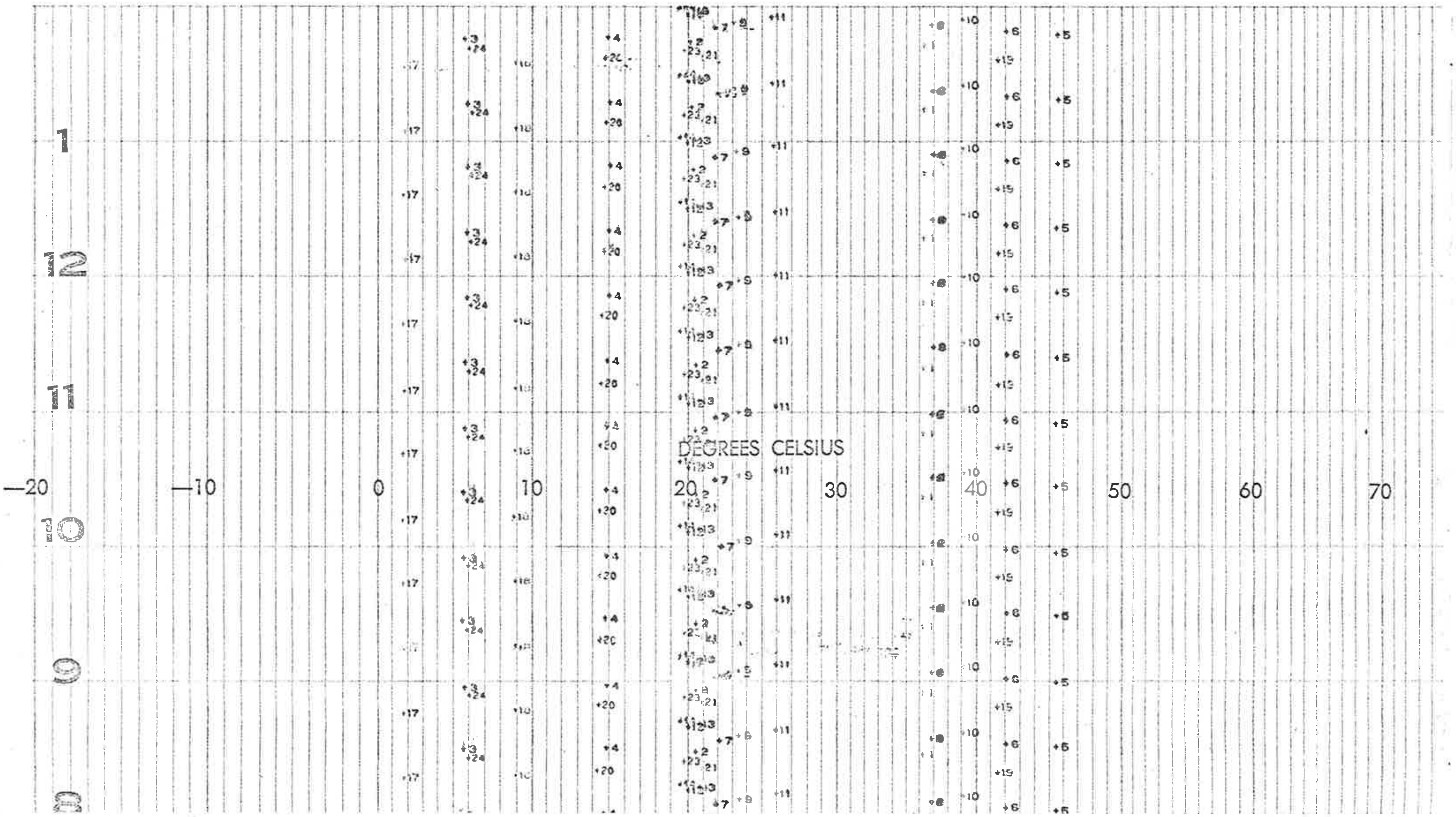


FIG. 7.12C RECORDER CHART 1, RUN 7LV

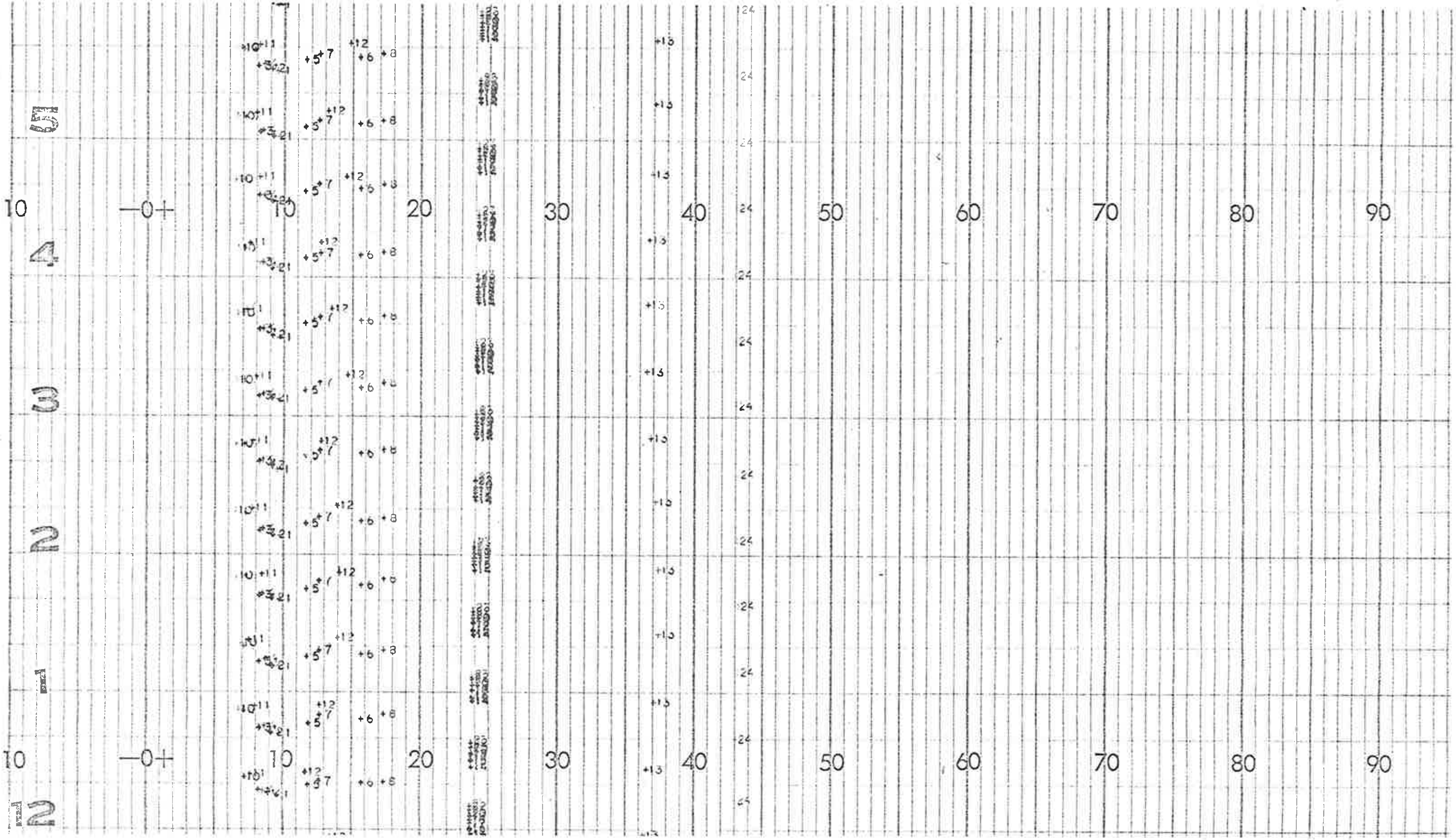


FIG. 7.12D RECORDER CHART 2, RUN 7LV

8. THE USE OF THE UNIFIED SYSTEM FOR TEACHING

8.1 Description of an Experiment Performed on the Waite Institute Unit and the Mechanical Engineering Department Teaching and Research Unified System

TITLE: Experiment on First Law Applications to Steady Flow Systems.

TIME: The experiment is conducted over a three week period with one three hour session per week. In addition, it includes preliminary lectures and a write up period.

DESCRIPTION: The experiment involves three steady flow cycles, interacting adiabatically at an evaporator and at a condenser. In addition, there are open steady flow paths involving

(a) make-up water to condenser;

(b) air path through cooling tower;

(c) make-up water to pan humidifier;

(d) water condensed at direct expansion coil.

OBJECT OF THE EXPERIMENT:

General:

The experiment is planned to supplement the lecture material on first law of thermodynamics, cycles, gas mixtures, two phase flow, psychrometrics and refrigeration.

Special Requirements:

It is the purpose of this experiment to:

1. Demonstrate the students' ability to record and organize relevant data.
2. To demonstrate the students' use of relevant property charts:
   (a) Draw refrigerant cycle on P.H. diagram.
   (b) Draw air cycle on Psychrometric Chart.

3. To determine air flow rate from Venturi tube upstream pressure and pressure differential readings. (Drawings giving Venturi tube design distributed).

4. To indicate the many first law of thermodynamics relationships that can be established based on the system performance data, some of the more important relationships sought are as follows:

   (a) Determine mass flow rate of refrigerant, $\dot{m}_r$, from the condensing process involving transfer of heat to the condenser water system, the relationship being

$$\left.\dot{m}c_p\Delta t\right]_{\text{condensing water}} = \left.\dot{m}_r\Delta H_r\right]_{\text{refrigerant}} .$$

   (b) Determine the heat transferred to evaporator knowing $\dot{m}_r$ and Refrigerating Effect, RE, according to the following equation

$$\left.Q\right]_{\text{Refrigerant Capacity}} = \dot{m}_r\ (\text{RE}) .$$

   (c) Determine the mass flow of dry air $\dot{m}_a$ from $\dot{m}_r$, RE and the enthalpy drop across the dehumidifier, ΔH_a

$$\dot{m}_a = \dot{m}_r\ \text{RE}/\Delta H_a .$$

   The student is expected to inter-relate ΔH_a with the enthalpy change of the refrigerant, ΔH_r across the evaporator according to Figures 4.4 and 4.5 of Section 4.2.2.

(d) Determine $\dot{m}_a$ from measuring the rate of water condensed at the dehumidifier and Assmann readings upstream and downstream from the dehumidifier, with the humidity ratio drop across the dehumidifier, ΔW, determined from Assmann readings, thus

$$\dot{m}_a = \frac{\dot{m}_{H_2O}}{\Delta W} .$$

(e) Compare ΔW as determined from Assmann readings upstream and downstream from the dehumidifier with $\dot{m}_a$ determined from upstream and differential pressures across Venturi tube and the rate of flow of condensate from drain pan of dehumidifier.

(f) Determine $\dot{m}_a$ from the measured heat input to a reheater and the temperature difference across the reheater.

(g) Determine the Refrigerating Capacity from the heat and mass transferred in the reheater and the pan-humidifier assuming no interchange with ambient, thus

$$\dot{m}_r \, RE = Q\big]_{Reheater} + Q\big]_{Humidifier} .$$

(h) Determine the specific heat at constant pressure of the refrigerant vapour, $C_{p,v}$, knowing $\dot{m}_r$, the refrigerating capacity, and the superheat rise across the vapour path of a liquid suction heat exchanger.

(i) Determine from a heat balance around a liquid-suction heat exchanger the specific heat of the refrigerant liquid $C_{p\ell}$ knowing the specific heat of the refrigerant vapour. Check the answer with property tables.

(j) Determine the COP of the refrigeration cycle.

(k) Compare the measured refrigeration capacity with the manufacturer's data.

8.2 Exercise in Inter-relationships Between Evaporator Performance and Condensing Unit Performance

The first law experiment was described for a system running at one steady flow state condition. In this suggested experiment the effects of changing of compressor speed, fan speed, and specific enthalpy settings can be explored. Essentially, the relationships established in Section 4.2.2, Figures 4.3 and 4.6; in Section 4.3.3. and Figure 4.7; in Section 6.1 and Figures 6.1 and 6.2; in Section 6.4 and Figure 6.3 would be examined.

8.3 Other Experiments

A complete manual of experiments could be drawn up similar to the ones described which could be carried out in the Mechanical Engineering Department Teaching and Research Unified System. A few of these will briefly be enumerated in order to indicate the scope of these facilities.

8.3.1 Experiment in cooling tower performance

Both the air and water sides are completely instrumented. Any desired load can be imposed on the condensed water system. The air flow rate can be controlled by way of a manual damper at the outlet of the pull-through centrifugal fan.

8.3.2 Experiment in second law applications

The effect of varying the high and low side pressures of a vapour compression cycle can be analysed. Both the supply fan and the compressor motor speeds can be varied through complete working ranges. This experiment may be conducted under constraints where the same subcooled condition to the liquid entering the thermostatic expansion valve and the same super heat condition to vapour leaving the evaporator is maintained.

8.3.3 Experiment in enthalpy potential

This experiment is described as a research project in Section 9.1 and in Section 12.5 Project II. It is very suitable for a final year laboratory exercise.

8.3.4 Performance of expansion valves and evaporator pressure valves in the vapour compression refrigeration cycle

The importance of control valves to the vapour compression refrigeration cycle merits their study as a laboratory experiment. The inter-relations between the valve action and the system performance is closely related to basic lecture material on refrigerants. The effect of these valves on the system would be analysed.

The importance of the self-contained automatic control action of the thermostatic expansion valve to maintain refrigeration system performance compatible with load changes can be studied. The teaching facility is provided with means for changing the loads, changing the extent of subcooling and superheating. The effect of different operating settings on the superheat settings can also be studied including changes to the valve spring settings.

An evaporator pressure regulator is installed in the cycle with a bypass around it. This valve can therefore be included or excluded from the cycle. Pressure gauges fully reveal valve performance. As the evaporator pressure drops below the valve minimum setting the valve acts to simultaneously increase this pressure and reduce the suction pressure. The effect of this valve action in reducing the refrigeration capacity can be fully studied.

<u>General Comment</u>: Many of the experiments suggested above may be carried out at several levels of student involvement. Though exercises lasting over several hours are envisioned in certain areas it is possible to demonstrate system characteristics to a large group of students in a brief period. For example the effect of changing the condensing pressure while maintaining the evaporator pressure constant and the effect of changing the evaporator pressure while maintaining the condensing pressure constant. The system responds instantaneously to changes in air flow rate and refrigerant flow rate. Thus the performance characteristics indicated in Figures 6.1 and 6.2 can readily be demonstrated.

9. THE USE OF THE UNIFIED SYSTEM FOR RESEARCH

9.1 A Research Project in Enthalpy Potential

The objective of this project is to investigate the enthalpy as the driving potential for the simultaneous transfer of heat and mass in an air water vapour mixture.

In this experiment the research system is operated under a number of fixed constraints:

air flow rate

condensing temperature

evaporator temperature

superheat condition leaving evaporator

dryness fraction entering evaporator

entering air enthalpy.

Several pairs of dry bulb temperature and wet bulb temperature conditions are investigated, each pair having a different dry bulb temperature. However each pair have the same specific enthalpy at entry to the dehumidifier.

The system is run for each pair of conditions maintaining the constraints listed above. When steady flow conditions are reached, the final leaving dry and wet bulb temperatures would be determined. For enthalpy potential theory to be valid, it would be expected that the leaving specific enthalpy for each pair would be the same.

The experiment is run at various air velocities and entry specific enthalpies to determine the conditions under which deviation may occur. One of the test runs would be selected so that the direct expansion coil performs dry.

9.2 A Research Project in Tie Line Slope Relationships

9.2.1 The datum coil Tie Line Slope values determined at the Waite Phytotron Unit

Prior to the completion of the Mechanical Engineering Department Unified System a number of Tie Line Slope values were determined empirically for various operating conditions using the Waite Phytotron Unit. The data thus gathered was used to determine the Tie Line Slope non-empirically for a problem which served to establish the findings of Section 11 of this thesis. It is therefore entered here as an example of a research project carried out at the Waite Phytotron Unit Unified System.

9.2.2 Confirmation of the findings of Section 11 using the Mechanical Engineering Department Unified System

Section 12.3 describes a research project carried out at the newly completed Mechanical Engineering Department Unified System. With this system Tie Line Slope relationships can be obtained empirically. The project logically follows Sections 10 and 11 and will therefore be located in Section 12.

9.2.3 Other research projects

A number of research projects can be carried out with the use of the Mechanical Engineering Department Unified System. The Commonwealth Scientific and Industrial Research Organization, Division of Mechanical Engineering, Highett, Victoria is negotiating the use of the system in connection with the measurement of the moisture diffusivity of materials.

The South Australian Institute of Technology will be conducting laboratory sessions using this system. Also, an Adelaide refrigeration equipment manufacturer is interested in using the system for a research project.

10. CRITERIA FOR SELECTION OF COOLING SURFACES TO SIMULTANEOUSLY MEET SENSIBLE AND LATENT HEAT LOADS - SECTION BASED ON COMMERCIAL MANUFACTURERS' COIL SELECTION DATA

10.1 Introduction

In this section some of the existing commercial practice employed in dehumidifier selection for climate simulation and air conditioning is examined. It will be shown that by applying principles developed here and in Section 11, energy requirements associated with dehumidifier performance can be reduced.

10.2 Background

A cooling coil does least dehumidifying when the entering condition of the air is at a combination of a high dry bulb temperature and a low humidity ratio.

Figure 4.8 repeated below indicates that for any given entering dry bulb temperature, as the entering humidity ratio drops, there is a progressively shallower slope to the coil condition curve, as shown when moving from entering condition 1 to 2 to 3. This figure also indicates that for any given humidity ratio, there is a progressively shallower slope to the coil condition curve as the dry bulb temperature is increased, for example, moving from entering condition 4 to 5 to 6. Thus in climate simulation for a wide range of temperature and humidity, it is important to check that the coil selected will offset the latent heat loads during operating conditions with a combination of low humidity ratio and high dry bulb temperature when minimum dehumidification performance occurs. Furthermore, the coil selection must consider peak design loads. Therefore associated with this combination would be a fresh air intake requirement during a high humidity ratio ambient condition, and the highest condensing

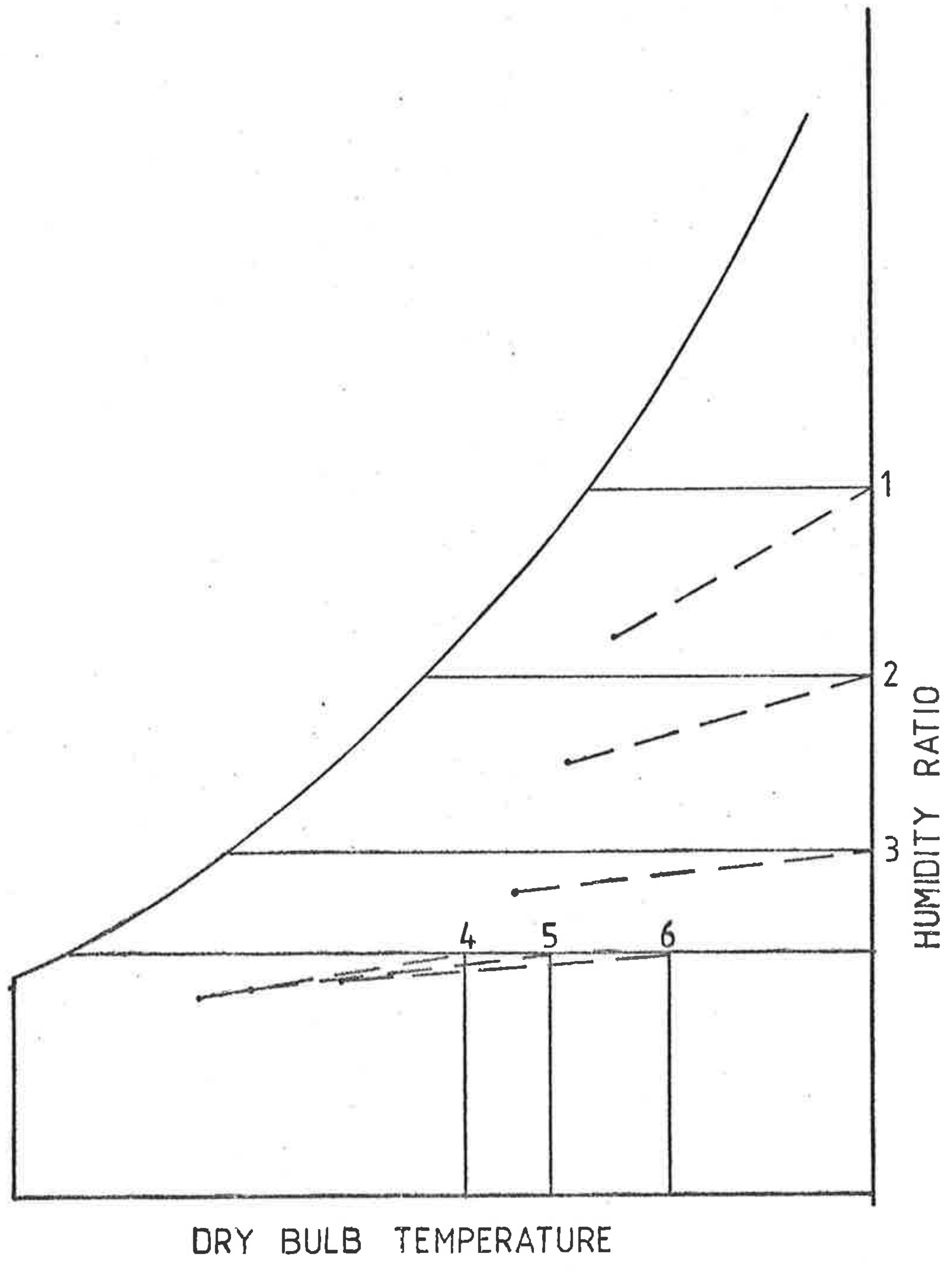


Fig. 4.8 Comparison of Various Entering Conditions to Evaporator

[Repeated]

temperature that may occur. The evaporator temperature would be at the minimum permissible.

In view of the above combined requirements the selection of a direct expansion coil can be very critical in meeting the specifications. (See Section 5.2.2). It is believed that some of the principles that will be presented here are applicable to the air conditioning field. Little attention has been paid to coil selection in conventional air conditioning design. There is almost unanimous agreement within the air conditioning design community in the approach that coils should have increased rows of depth whenever the problem calls for dehumidification to offset high latent heat loads. The literature and manufacturer's technical data support this approach. Thus, cooling coil characteristics are shown by performance curves drawn on a psychrometric chart such as in Figure 10.1, (Carpenter, 1950). The curve is labelled with numbers indicating how many rows deep would be required to reach an outlet condition so marked. Deeper and deeper dehumidification is obtained as the rows of depth are increased from two to six.

The above coil performance characteristics are not disputed. However, in respect to the use of dehumidifiers for climate simulation and air conditioning, it alone is insufficient to establish dehumidifier selection. It is important to consider another relationship that will be developed in this paper. When the face area is adjusted to maintain the same mass flow rate of air, the shallower the coil, the greater the rate of dehumidification with respect to sensible cooling in the simultaneous heat mass transfer process. It is this relationship that permits adjustment of the face area (or the face velocity) to best satisfy a design problem. Adhering simply to the rule, the deeper the coil the greater the dehumidification can carry with it an excessive energy penalty and result in poor performance. Arbitrarily limiting

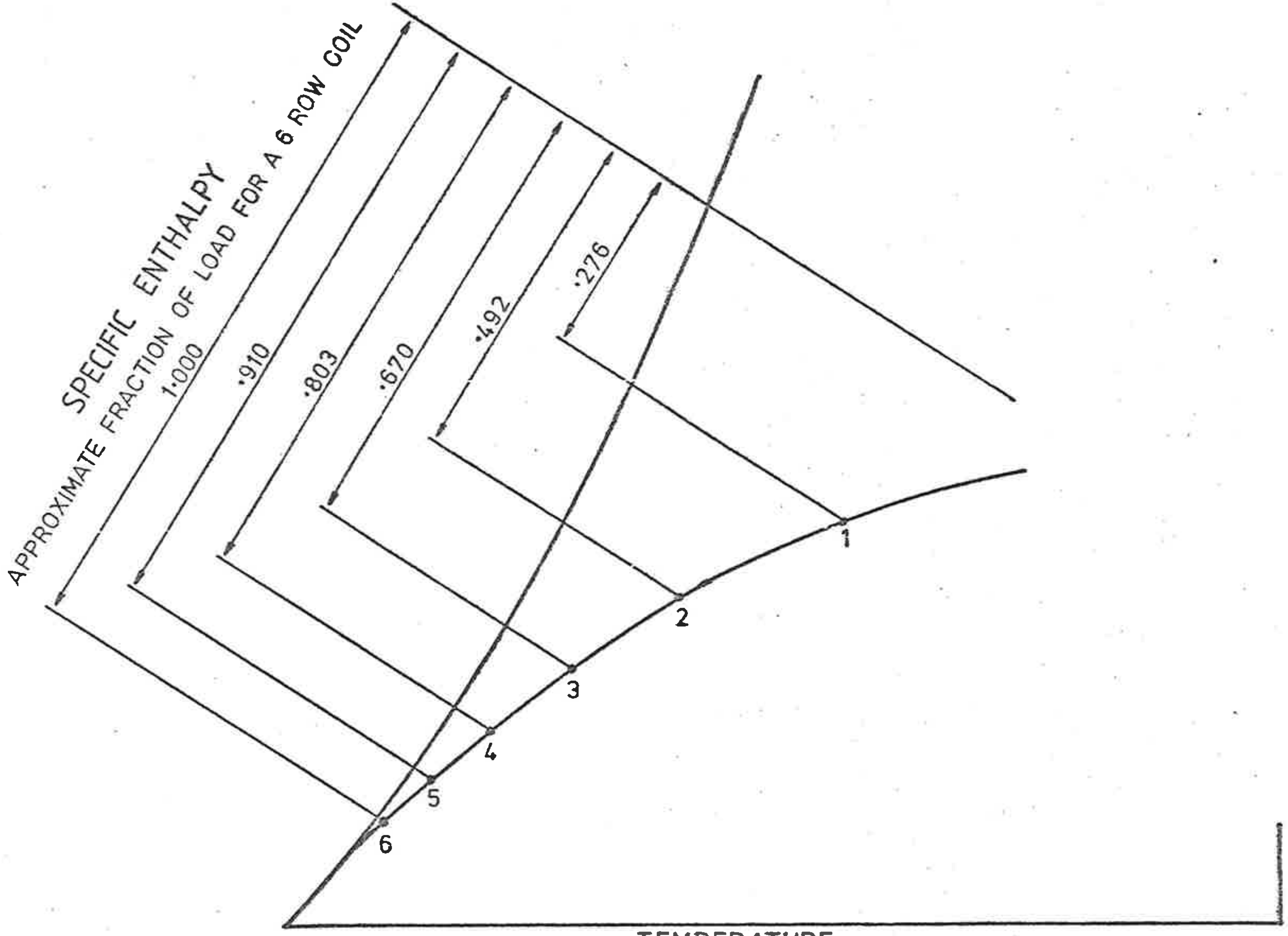


FIG 10·1 AIR SIDE COIL PERFORMANCE

the face velocity of a dehumidifier to a minimum of 2 m/s may also be a source of energy loss.

In environmental chambers having ranges which extend to high dry bulb temperatures combined with low humidity ratio settings, (a very common climatic condition), the use of conventional methods of industry would be very wasteful of energy particularly when the temperature gradient in the conditioned space is small. See problem number 1 in Section 11.14.1 on operating costs and dehumidifier selection. As will be demonstrated, low velocity shallower coils with less fins per unit length of primary tubing are more suitable for problems involving low sensible heat ratios. For these, a coil having a relatively low secondary to primary surface area and a very high fin efficiency is preferred. In the case of climate simulation design, deep coils at conventional air velocities can not only introduce prohibitive energy penalties but can prevent attaining the design range. Unfortunately, more space may be required and higher initial costs may be involved in accommodating the dehumidifier. However when viewed against the total complex and particularly with reference to energy savings, the proposed approach should be given serious consideration.

10.3 Reason for the Existing Methods Used in the Air Conditioning Field

In air conditioning design for comfort there are a number of reasons for the use of deeper coils at conventional face velocities to meet high dehumidification needs.

10.3.1 Spatial reasons

The cross-sectional areas of air conditioning units are often restricted in space. This may be the major reason for limiting the minimum air velocities permissible through cooling surfaces. Air flow rates of 2m/s (400 fpm) are considered low and commercial units frequently will employ air rates exceeding 3m/s (600 fpm) even though they have to add eliminator sections to remove water droplets which are carried over by the high velocity air stream. Recent innovations in high velocity air filters (e.g. bag filters) further encourage higher air flow rates through air conditioning units. On the other hand, in many factory built air conditioning units the cooling surface is located directly downstream from the filter bank. Many of these filters require considerably lower air stream velocities than is common practice for cooling coils. Therefore, certainly in these applications lower face velocity coils could be employed.

10.3.2 Temperature gradients

In air conditioning application higher temperature gradients cause the leaving conditions to approach saturation. Under these conditions it may be acceptable to select the deeper coils that are in present usage. (See Section 3.2.2 and 3.2.3.(a)).

10.3.3 Effective temperature

In conventional air conditioning applications it is not usually important to maintain precisely the set point of a particular temperature and humidity. Therefore higher humidities within the space may be acceptable. (See Section 3.2.5). (However, see Section 11.12 regarding part load performance).

10.3.4 Minimum humidity ratios

In air conditioning applications there is usually no requirement to obtain humidity ratios much below 0.009, whereas in the design of climate simulators humidity ratios of 0.005 are often a desired minimum. Thus the problem of dehumidification during settings when the coil condition curve has a shallow slope would not arise as frequently as it does in air conditioning applications if some of the conclusions from this study were put into practice.

10.3.5 High sensible heat ratio characteristic

In many conventional air conditioning applications high sensible heat ratios exist and dehumidification is of minor importance. A sensible heat ratio of 0.9 is quite common. In these cases, unless part load periods are adversely affected, existing air velocities may be quite acceptable.

10.4 Coil Selection Methods Holding to Fixed Face Velocity Can Result in Needless Energy Penalty

Consider a system which has sufficient sensible cooling capacity being enlarged with a deeper coil in order to accommodate a larger latent heat load. A 4-row coil will have twice as many passes as a 2-row coil. For the same air flow rate the 4-row coil has doubled the heat exchange surfaces and an 8-row coil has quadrupled it. This can result in an enormous, wasteful energy penalty to achieve some additional dehumidification. Not only would it be associated with larger refrigeration capacity but it may also require excessive reheating. If the mechanism to effect heat and mass transfer called for this approach to cooling and dehumidification there would be no choice but to accept this solution. However, fluid flow characteristics and the laws governing simultaneous heat and mass transfer point to a better approach.

10.5 A New Approach to Dehumidification Using Commercial Manufacturers' Data

Rather than considering dehumidification by way of addition of rows of depth to a fixed face area coil as in Figure 10.1, in this section by means of a comparative study a new basis will be developed. The performance of two coils having the same fin and tube total air side and coolant side surface areas will be studied.

10.5.1 Comments on commercial data for coil selection

Commercial methods for rating coils are not precise. They are based on the assumption that the coil condition curve is that of a straight line when depicted on a psychrometric chart. This relates back to the Contact Mixture Principle developed by Carrier (1937). Apart from the approximation that the process path through a dehumidifier follows a straight line when drawn on a psychrometric chart, other errors are introduced into the commercial methods. It is assumed, when wet bulb depression ratios are used, that the saturation curve of the psychrometric chart is a straight line over the section at which the coil performance is being considered. Further error is introduced when the cooling coil is assumed to have a single surface temperature whereas in fact this drops in the direction of the air flow.

In Section 11 of this paper a more accurate analysis will be presented. The commercial methods that will be used in this section will be sufficiently accurate to demonstrate qualitatively the relative slopes of the coil condition curves. They represent methods used by major reliable manufacturers.

10.5.2 Wet bulb depression ratio (and bypass factor)

In this section a number of terms that are common to commercial usage will be employed. Reference will be made to the wet bulb depression ratio in using manufacturer's data.

Firstly to explain the significance of the term it is best to associate it with the basic relationship to bypass factor, the term introduced by Carrier (1937), as part of the contact mixture principle. Reference is made to Figure 10.2.

Between entering and leaving conditions 1 and 2 the bypass factor, BF, is defined as:

$$BF = \frac{t_2-t_c}{t_1-t_c} = \frac{H_2-H_c}{H_1-H_c} \qquad \ldots(1)$$

and hence

$$t_2 = (BF)t_1 + t_c(1-(BF)) \qquad \ldots(2)$$

Thus knowing t_1, t_c and the bypass factor, BF, (or as is used by some coil manufacturers, wet bulb depression ratio), the leaving dry bulb temperature can be determined for dehumidifiers.

The terms $t_1 - t_1'$ or D_1 and $t_2 - t_2'$ or D_2 are known as wet bulb depression. It is defined as the dry and wet bulb temperature difference for a particular condition. The term t_c represents the coil surface temperature as determined by the intersection of the assumed straight line coil condition curve with the saturation line of the psychrometric chart. If the curve of the saturation line is considered to be straight and if the lines of constant enthalpy are assumed to be parallel to the lines of constant wet bulb temperature, then as shown in Figure 10.2, based on similar triangle relationships, the wet bulb depression ratio, D_2/D_1 and the bypass factor, BF, are

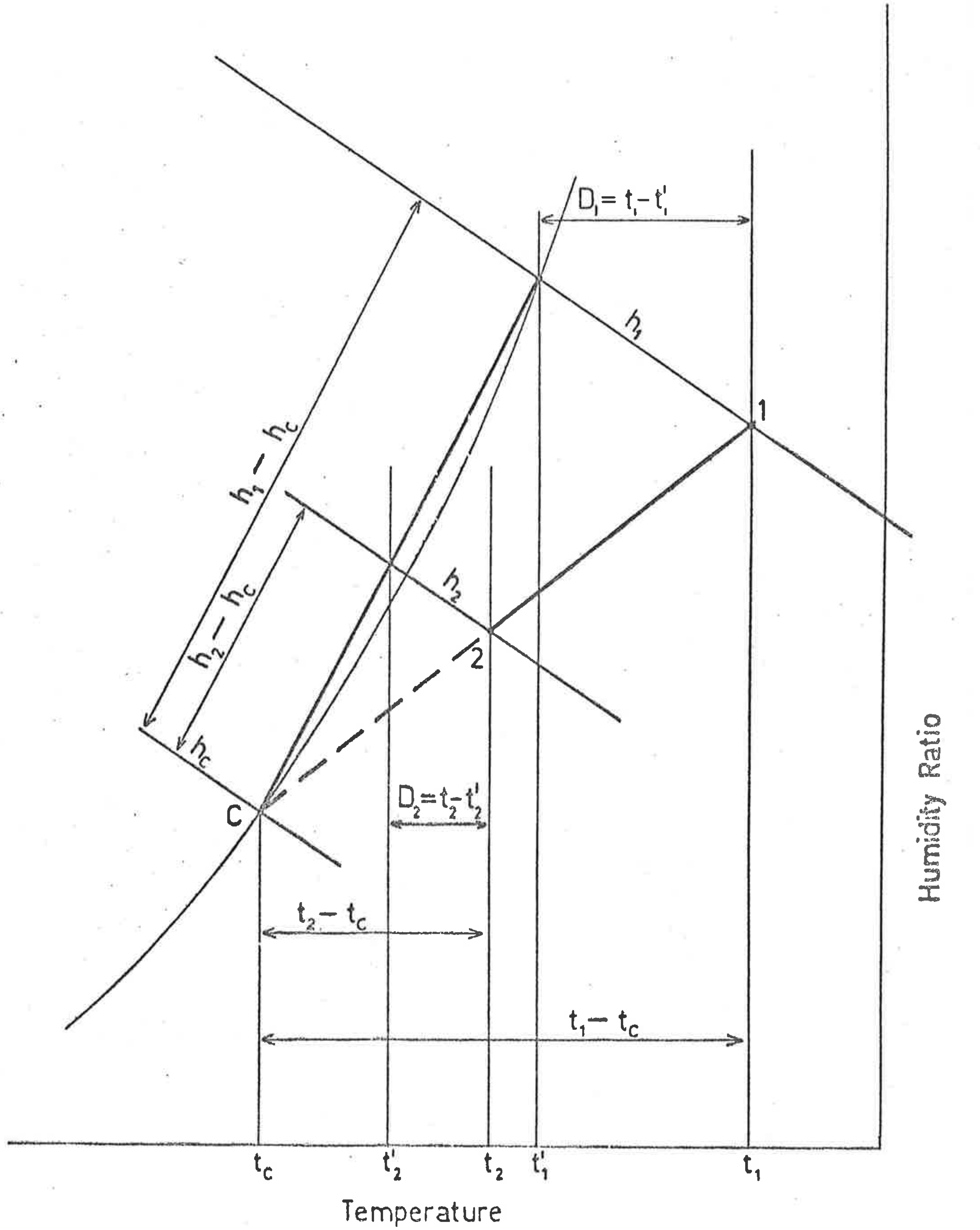


FIG 10.2 RELATIONSHIP BETWEEN BYPASS FACTOR AND WET BULB DEPRESSION RATIO

equal, since

$$\frac{H_2 - H_c}{H_1 - H_c} = BF \qquad \ldots(3)$$

and

$$\frac{H_2 - H_c}{H_1 - H_c} = \frac{t_2 - t_2'}{t_1 - t_1'} = \frac{D_2}{D_1} \qquad (\ldots\text{See Fig. 10.2})$$

then on the basis of the above approximations

$$BF = \frac{D_2}{D_1} \,. \qquad \ldots(4)$$

For the *same* coil having the same refrigerant temperature and face velocity and having the same entering wet bulb temperature but different entering dry bulb temperatures, and the same leaving wet bulb temperatures but different leaving dry bulb temperatures, the wet bulb depression ratios will be (approximately) the same for both conditions. This assumption can readily be seen by comparing the ratios of lines as drawn on the psychrometric chart for two conditions a and b. (See Figure 10.3). The depression ratio is used by numerous manufacturers in determining the performance of wet cooling surfaces.

The bypass factor and the wet bulb depression ratio have been shown to be very nearly equal. The bypass factor for one row and for several rows can be shown to be related as in the following equation

$$(BF)_n = (BF)_1^n \,. \qquad \ldots(5)$$

This states that the bypass factor for a number of rows, n, will be equal to the bypass factor for a single row raised to the nth power. This relationship can be checked with reference to the manufacturer's table (see Tables 10.1 (Trane 1956), and 10.2 (Carrier et al p.319, 1959a) below). It follows that by taking the square root of

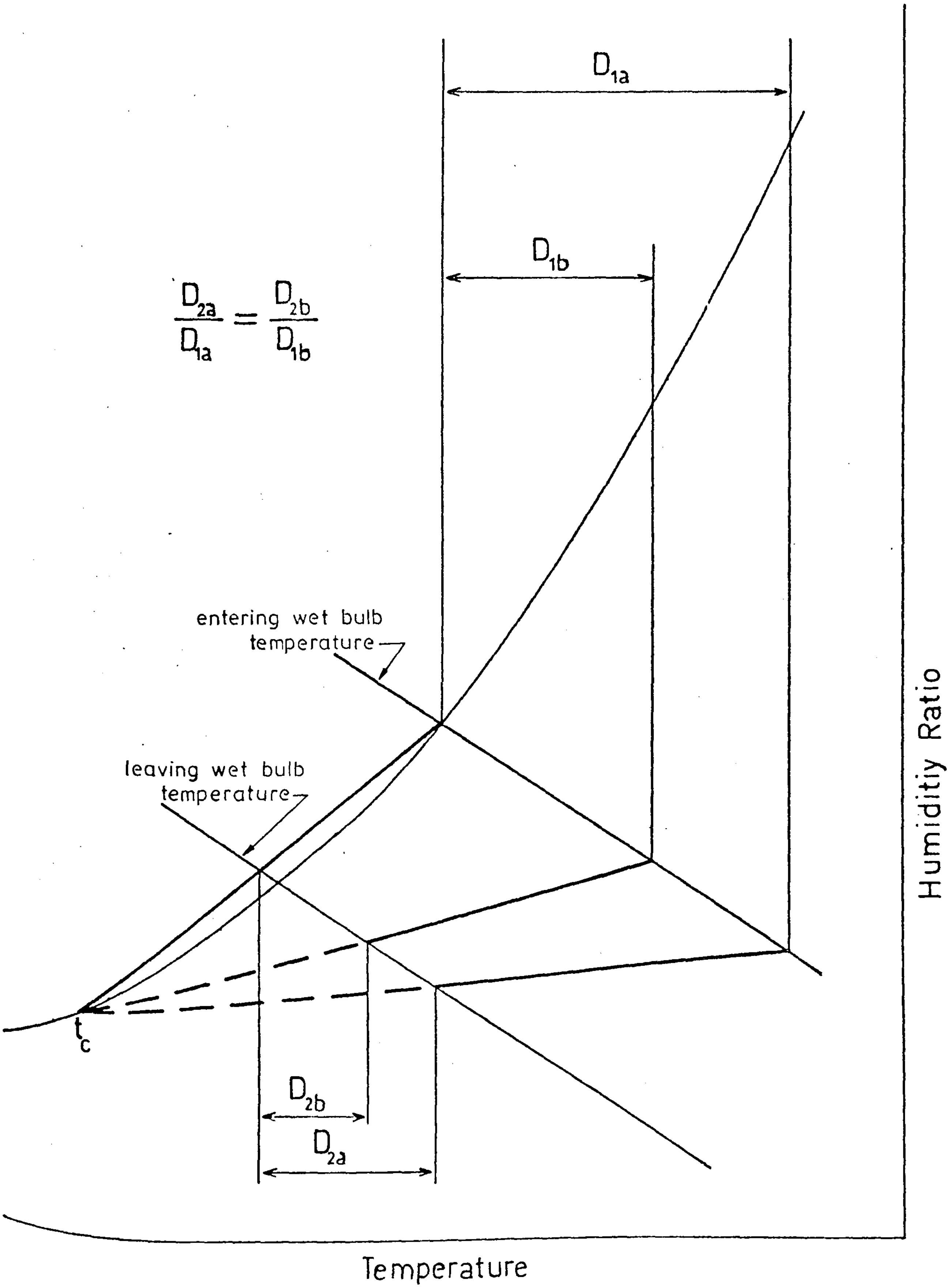


FIG 10.3 WET BULB DEPRESSION RATIOS ARE EQUAL WHEN THE ENTERING AND LEAVING WET BULB TEMPERATURES ARE THE SAME

the wet bulb depression ratio for two rows, we will find the depression ratio for a single row coil. Raising this BF value for one row to a power equal to the number of rows should yield the same value as shown in Table 10.1.

For example at 1.5m/s (300 fpm) air velocity and two rows deep $(BF)_2 = 0.440$. Applying the bypass equation above

1 row	$(0.440)^{\frac{1}{2}}$	= 0.663	
2 row	(0.440)	= 0.440	From Table 1
3 row	$(0.663)^3$	= 0.292	
4 row	$(0.663)^4$	= 0.193	
5 row	$(0.665)^5$	= 0.128 .	

It can be seen from this example that these values obtained through the equation agree with the values listed in Table 10.1 below for the column representing 1.5m/s.

Ratio of Final to Initial Wet Bulb Depression
For Direct Expansion Coils

Rows of Wet Tubes	Air Velocity									
	m/s	1.5	2.0	2.5	3.0	3.6	4.1	4.6	5.1	6.1
	fpm	300	400	500	600	700	800	900	1000	1200
2		0.440	0.472	0.497	0.512	0.527	0.543	0.554	0.566	0.583
3		0.292	0.323	0.346	0.368	0.387	0.399	0.411	0.423	0.445
4		0.194	0.221	0.244	0.264	0.281	0.295	0.307	0.320	0.340
5		0.129	0.153	0.172	0.188	0.204	0.217	0.228	0.239	0.259
6		0.085	0.104	0.121	0.135	0.148	0.160	0.170	0.179	0.198
7		0.057	0.071	0.085	0.097	0.108	0.118	0.126	0.135	0.151
8		0.038	0.049	0.060	0.069	0.078	0.086	0.094	0.101	0.114
10		0.016	0.023	0.030	0.036	0.042	0.047	0.052	0.057	0.067
12		0.007	0.011	0.015	0.018	0.022	0.025	0.029	0.032	0.039
14		0.003	0.005	0.007	0.009	0.012	0.014	0.016	0.018	0.023

TABLE 10.1 (Reproduced from Trane (1956)
With Metric Units Added)

Typical Bypass Factors for Cooling Surface

(5/8 in. OD tubes, 8 crimped helical fins per inch, 0.008 in. thick, 13/32 in. fin height, surface ratio 12.3)

Rows Deep	Face Velocity m/s (fpm)			
	1.5 (300)	2.0 (400)	2.5 (500)	3.0 (600)
	Bypass Factor			
1	0.61	0.63	0.65	0.67
2	0.38	0.40	0.42	0.43
3	0.23	0.25	0.27	0.29
4	0.14	0.16	0.18	0.20
5	0.09	0.10	0.11	0.12
6	0.05	0.06	0.07	0.08
7	0.03	0.04	0.05	0.06
8	0.02	0.02	0.03	0.04

(5/8 in. OD tube, 14.4 smooth helical fins per inch, 0.012 in. thick at base, 13/32 in. fin height, surface ratio 21.5)

1	0.48	0.52	0.56	0.59
2	0.23	0.27	0.31	0.35
3	0.11	0.14	0.18	0.20
4	0.05	0.07	0.10	0.12
5	0.03	0.04	0.06	0.07
6	0.01	0.02	0.03	0.04

TABLE 10.2 (Reproduced from Carrier et al (1959a) With Metric Units Added)

10.6 Comparison of Two Coils with the Same Primary and Secondary Surface Areas

Consider two coils:

Both coils have the same

1. total length of primary tubing
2. secondary fin to primary surface configuration
3. quantity of heat exchange surfaces
4. mass flow of air.

Both coils operate

5. with the same specific enthalpy, dry bulb temperature and humidity ratio at inlet to the coil.

However, one coil has

6. twice the face area but half the number of rows in depth of the other coil.

Two coil condition curves having the same entering condition will be drawn on a psychrometric chart. The object will be to determine the relationship between coil condition curve slope and wet bulb depression ratio. (See Figure 10.4.)

The relevant equation for wet bulb depression ratio for the steeper coil condition curve is

$$\frac{t_{2a} - t'_{2a}}{t_1 - t'_1}$$

and this is greater than the wet bulb depression ratio for the shallower coil condition curve which is

$$\frac{t_{2b} - t'_{2b}}{t_1 - t'_1} .$$

$$\frac{t_{2a} - t'_{2a}}{t_1 - t'_1} > \frac{t_{2b} - t'_{2b}}{t_1 - t'_1}$$

$$\frac{D_{2a}}{D_1} > \frac{D_{2b}}{D_1}$$

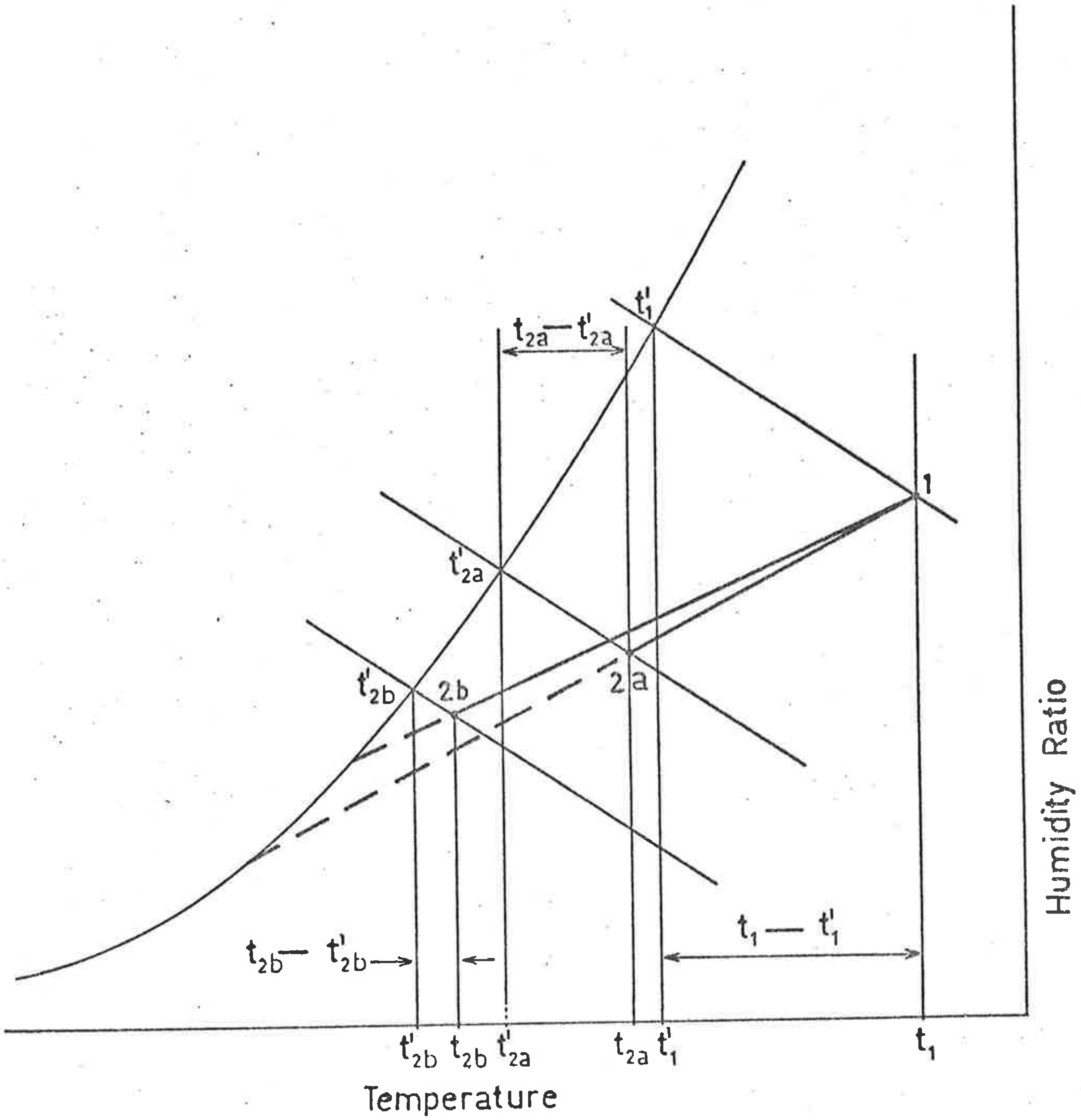


FIG 10.4 THE COIL WITH THE STEEPER COIL CONDITION CURVE HAS THE LARGER WET BULB DEPRESSION RATIO

It can be seen that the denominators, entering wet bulb depression, in the expressions for the two coil load ratio lines are equal. The numerators of the leaving wet bulb depression are greater for the coil load ratio line with the steeper slope. Thus it has been demonstrated the coil with the steeper slope will have the larger wet bulb depression ratio (and the larger bypass factor) under the six conditions of comparison enumerated above.

It is therefore possible to determine from wet bulb depression ratio data on coils the geometry which will be preferred to offset latent heat loads. It should be noted that the outlet condition, 2a, for the coil with the steeper coil condition curve is at a higher wet bulb temperature than the outlet for the shallower coil condition curve 2b. This was done to anticipate what actually does occur in the comparison of Section 10.6. However, it should be noted that had the outlet condition of the steeper coil condition curve been at the same wet bulb temperature as for the shallower coil condition curve, the same conclusion would have been drawn. The coil with the steeper coil condition curve would continue to have the larger wet bulb depression ratio. (This, as will be developed later, is because the steeper coil condition curve is constrained to have the lower face velocity).

In this comparison it is desired to identify the coil configuration associated with each of the coil condition curves. Is it to be expected that a deep, 6-row coil with a relatively high face velocity of 3.0m/s (600 fpm) would have the steeper slope to its coil condition curve than a shallow 3-row coil having a lower face velocity of 1.5m/s (300 fpm)?

It is to be noted that whereas in Figure 10.1 the addition of rows of depth to a fixed face area coil to accomplish greater dehumidification meant additional primary and secondary surface area to the coil, in this comparison the constraints set up operate to maintain the same total area of coil surfaces both for the air and the coolant.

Table 10.1, (Trane 1956), repeated below, relates wet bulb depression ratio with face velocity and depth of coil for a particular 19 mm (3/4 inch) outside diameter plate fin coil. The coil that is 3 rows deep has a face velocity of 1.5m/s (300 fpm) and has a depression ratio of 0.292 whereas a similar coil (adhering to the constraints listed above), 6 rows deep with a face velocity of 3.0m/s (600 fpm) has a depression ratio of 0.135. These values are encircled on Table 10.1, repeated below. To make the same point, curves of another manufacturer are reproduced below in Figure 10.5 (McQuay Inc. 1955) relating wet bulb depression ratio with face velocity and depth of coil. Again the relationship associated with face velocity is demonstrated:- the higher the wet bulb depression ratio the greater the capacity of the evaporator to dehumidify in relation to sensible cooling. This can be seen on examination of any of the four charts, (the upper right hand one for an 8 fin per inch evaporator is used in this example). A coil that is 3 rows deep and has a face velocity of 1.5m/s (300 fpm) would have a depression ratio of 0.27 whereas a similar 6-row deep coil with a face velocity of 3.0m/s (600 fpm) would have a depression ratio of 0.122. Thus both the table and the curves clearly indicate that under the constraints of this comparison the <u>LOWER VELOCITY COIL</u> has the larger depression ratio (bypass factor).

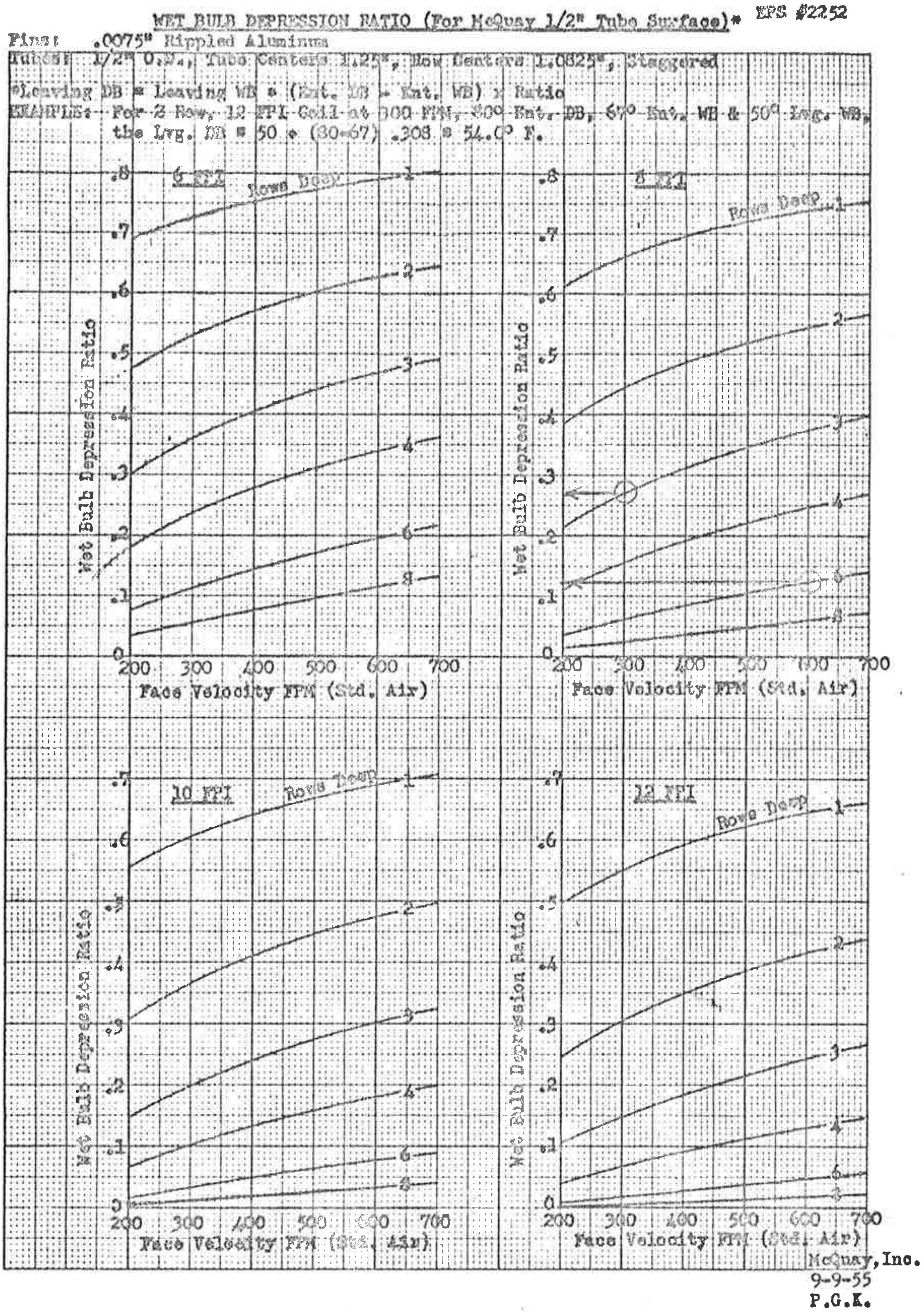


Fig. 10.5 Wet Bulb-Depression Ratio Related to Face Velocity and Depth of Coil

(Reproduced from McQuay (1955) with arrows added)

Ratio of Final to Initial Wet Bulb Depression
for Direct Expansion Coils

Rows of Wet Tubes	m/s	1.5	2.0	2.5	3.0	3.6	4.1	4.6	5.1	6.1
	fpm	300	400	500	600	700	800	900	1000	1200
2		0.440	0.472	0.497	0.512	0.527	0.543	0.554	0.566	0.583
3		0.292	0.323	0.346	0.368	0.387	0.399	0.411	0.423	0.445
4		0.194	0.221	0.244	0.264	0.281	0.295	0.307	0.320	0.340
5		0.129	0.153	0.172	0.188	0.204	0.217	0.228	0.239	0.259
6		0.085	0.104	0.121	0.135	0.148	0.160	0.170	0.179	0.198
7		0.057	0.071	0.085	0.097	0.108	0.118	0.126	0.135	0.151
8		0.038	0.049	0.060	0.069	0.078	0.086	0.094	0.101	0.114
10		0.016	0.023	0.030	0.036	0.042	0.047	0.052	0.057	0.067
12		0.007	0.011	0.015	0.018	0.022	0.025	0.029	0.032	0.039
14		0.003	0.005	0.007	0.009	0.012	0.014	0.016	0.018	0.023

TABLE 10.1 (Repeated) (Reproduced from Trane (1956)
With Metric Units Added)

Thus by means of Figure 10.4 it has been demonstrated

the STEEPER THE COIL CONDITION CURVE SLOPE the larger the bypass factor or wet bulb depression ratio

and by means of Table 10.1 and Figure 10.5 it has been demonstrated

the LOWER THE FACE VELOCITY the larger the bypass factor or wet bulb depression ratio.

It may therefore be concluded that under the constraints of this comparison

the STEEPER THE COIL CONDITION CURVE SLOPE

the LOWER THE FACE VELOCITY.

This conclusion points to a new system for dehumidifier selection for climate simulation and air conditioning that will be developed in Section 11.

Contrast the above conclusions with the following statement made in an authoritative text book (Carrier et al 1959a):

"The bypass factor decreases as the surface area and depth increase, and as *air velocity* decreases".

This may appear to contradict the above conclusion. However this is not the case. In Table 10.1, it can be seen that a 4-row deep coil having a face velocity of 3.0m/s (600 fpm) has a wet bulb depression ratio of 0.264 and the same coil at 1.5m/s (300 fpm) has a lower wet bulb depression ratio of 0.194. This bears out the text book statement quoted above. It is correct as an independent statement.

Nevertheless, the statement fails to constrain the change in velocity to conditions in which the mass flow rate of the air is kept constant. Dehumidifier selection should be based on a particular mass flow of air since it is a function of the temperature difference across the load ratio line. This is a fixed value in a particular design problem and establishes the mass flow value.

For a fixed mass flow of air, the 3.0m/s (600 fpm) coil should have half the face area of the 1.5m/s (300 fpm) coil.

Even though the higher velocity coil would have a greater capacity for heat transfer, with only half the face area it would still be necessary to use a deeper dehumidifier. If it is desired to compare two coils in the selection process under the conditions in which they have equal mass flow of air and equal total heat exchange surface, then the 3.0m/s (600 fpm) dehumidifier with half the face area of the 1.5m/s (300 fpm) would have twice the depth or 8 rows. The bypass factor reduces with coil depth as shown in equation (5). It will now be shown that the rate of increase of bypass factor with increase of air velocity is small in relation to the decrease resulting from the increase of coil depth. Allowance will be made for the increased total heat mass transfer, occurring due to an increased velocity, by increasing the depth of the coil only slightly and not in proportion to the decrease of the face area. Thus in the case of this example, when the face area is halved the depth will not be doubled to 8 rows, but only increased by 25 per cent to 5 rows deep. This increase would be far below the improvement in the total enthalpy change (approximately 20 per cent), that would occur due to the doubling of the face velocity.

Table 10.1 above can now be examined to see the relative effect.

The 1.5m/s (300 fpm) coil of 4 rows deep has a bypass factor (wet bulb depression ratio) = 0.194, compared with a 3.0m/s (600 fpm) coil of 8 rows deep the bypass factor is = 0.069; compared with a 3.0m/s (600 fpm) coil of 5 rows deep the bypass factor becomes = 0.188. (See Table 10.1).

Thus it can be seen that though the text book statement is correct, under the constraints that apply to climate simulation and air conditioning applications where a constant mass flow of air must be maintained, the lower face velocity coil will have the larger bypass factor or wet bulb depression ratio.

A further conclusion can be drawn by comparing the four wet bulb depression ratio curves of Figure 10.5. For any condition of rows of depth and face velocity, the less fins per unit length yields greater wet bulb depression ratios (bypass factors). Thus it is indicated that reducing the ratio of secondary fin area to primary tube area will result in a steeper coil condition curve.

For example for a 3-row deep coil at 1.5m/s (300 fpm) face velocity

6	fins per inch has a wet bulb depression ratio of	0.36
8	" " " " " " " " " "	0.27
10	" " " " " " " " " "	0.20
12	" " " " " " " " " "	0.15.

As in the case of higher air velocities, an increase in the number of fins per inch will increase the coil capacities resulting in more sensible cooling. Figure 10.6 (McQuay Inc. 1955), shows under the heading of "fin correction factor", that a coil having 6 fins per inch will have approximately three quarters of the capacity (0.765), of a coil with 12 fins per inch (1.0), when compared on the basis of refrigeration capacity per unit time per unit face area.

We can conclude that the reduction of the number of fins per inch and rows of depth will result in steeper coil condition curves. *However the total coil capacity would also be reduced.*

In determining which coil configuration will attain a steeper coil condition curve the relationship between the simultaneous sensible cooling and dehumidification processes must be considered. Sensible cooling will proceed with any temperature difference between the surfaces of the coil and the air stream. However, dehumidification will only proceed where the surface temperature of the coil is below the dew point temperature of the air stream passing over it. Consequently, the low primary tube temperature will be more effective for dehumidification than the higher fin temperatures. The fewer fins per unit length of primary surface, the smaller the centroidal area distance of the fins to the primary tube surface, the smaller the ratio of the fin area to the primary tube area the greater the dehumidification with respect to sensible cooling.

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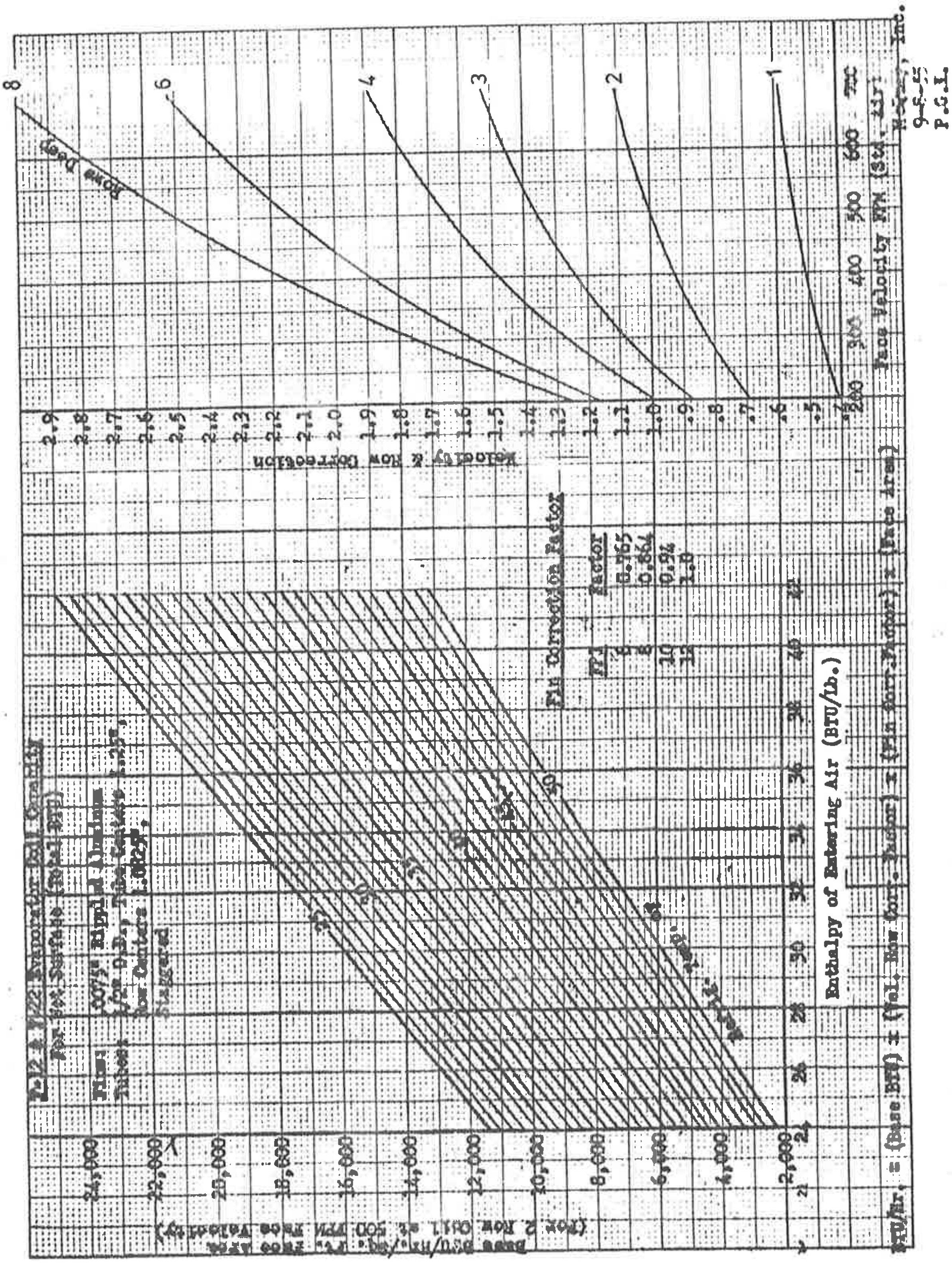


Fig. 10.6 Fin Correction Factor

(Reproduced from McQuay (1955))

10.7 An Example, Two Coil Performance Curves Calculated on Basis of Six Constraints

Performance of two coils was calculated using the manufacturer's selection data under the six conditions listed in Section 10.6. In this particular comparison, one coil has a face area 0.11 m^2 (1.2 square feet) and is 6 rows deep, the other 0.22 m^2 (2.4 square feet) and is 3 rows deep. The first coil handles an air stream with a face velocity of 3.0m/s (600 fpm), the other 1.5m/s (300 fpm). Both coils have 3.15 fins per cm (8 fins per inch) and both have a flow rate of 0.340 m^3/s (720 cfm). See Figure 10.6. The 3-row deep coil has the steeper slope.

Though the 6-row deep coil has the shallower slope it attained approximately the same exit humidity ratio but at a higher coil capacity involving more sensible cooling.

Thus in this example for the same change in humidity ratio, ΔW, the shallow coil will reach its coil leaving condition directly, whereas the deeper coil may reach its set point with excess refrigeration capacity followed by wasteful reheating as indicated in Figure 10.7 below.

The relativity of the heat and mass transfer processes has been stressed. To obtain the same amount of dehumidification the deeper coil will require 12.2 per cent more refrigeration capacity.

10.8 Conclusions

1. When simultaneous heat and mass transfer takes place between an air-water vapour stream, and a direct expansion heat exchanger surface under the six constraints of Section 10.6, the mass transfer rate will increase relative to the heat transfer rate with decrease of face velocity for a given direct expansion coil.

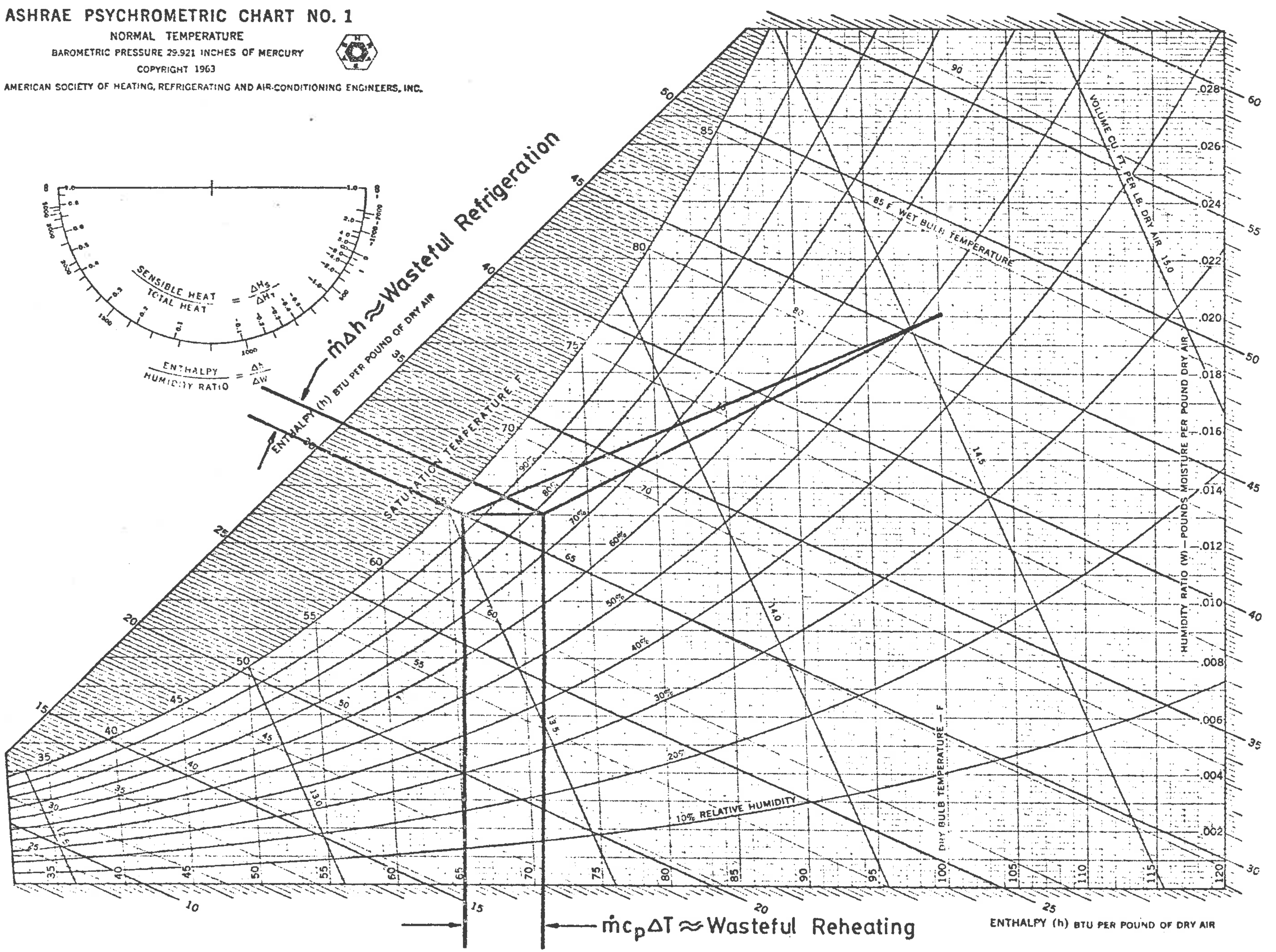


FIG 10-7 TWO COIL CONDITION CURVES CALCULATED ON BASIS OF SIX CONSTRAINTS

2. The mass transfer rate will increase relative to the heat transfer rate as the ratio of secondary fin area to primary surface area is reduced.
3. The mass transfer rate will increase relative to the heat transfer rate as the average distance from centroid of secondary fin area to the primary surface decreases. (Fin efficiency increases).

11. CRITERIA FOR SELECTION OF DEHUMIDIFIERS - A HEAT AND MASS TRANSFER ANALYSIS

The conclusions reached in the analysis based on industrial methods fail to reveal the actual path associated with simultaneous cooling and dehumidifying processes for a wetted direct expansion cooling surface. By adopting numerous simplifying approximations only straight line process paths with varying slopes are observable. There is no indication of the true nature of the curved process path - a most important requirement in climate simulation - where small temperature differences across the coil are preferred in order to offset loads acting over small temperature gradients.

It is the purpose of this section to improve on the industrial methods by way of a more basic analysis, using heat and mass transfer and enthalpy potential theory, and wherever possible reducing the approximations that are made in the name of simplification. Though selection is viewed with the special interest of climate simulation as in the unified approach, the findings of this section are also applicable to the air conditioning field.

The phytotron unit located at the Waite Agricultural Research Institute of the University of Adelaide, and a new teaching research heat-mass transfer laboratory in the Department of Mechanical Engineering were used to establish the data that forms the basis of this analysis. Both systems were designed by the writer.

Each system was run for the purpose of this project as a steady flow closed air cycle with no fresh air intake and the lighting load (simulating sunlight) in the off position.

At the Waite Institute Phytotron Unit a particular entering air condition was recorded and the data analysed. This data represents what will be referred to as Run 1, Datum Coil. The following information was obtained from the experimental data:

1. entering air enthalpy,
2. entering dry bulb temperature,
3. entering wet bulb temperature,
4. entering humidity ratio,
5. leaving air enthalpy,
6. leaving dry bulb temperature,
7. leaving wet bulb temperature,
8. leaving humidity ratio,
9. mass flow of air,
10. refrigerant temperatures and pressures.

In addition basic data on the direct expansion coil design was available. See Appendix II. The work of W.M. Kays and A.L. London "Compact Heat Exchangers" (1964) includes basic heat transfer and flow friction data for a surface configuration which very nearly resembles the direct expansion coil used in this project. In solving the Pierre equation (ASHRAE Handbook of Fundamentals 1977a p.2.22) it was necessary to make use of manufacturers' data to determine the combined coefficient of heat transfer through water layer metal and refrigerant film, h_i.

11.1 The Problem Outlined

It is desired to study the performance of direct expansion coils which will be employed in a wide range climatic system for simulation of temperature and humidity. In addition, one by-product of the study will be to assess the effect of the findings on air conditioning applications. The aim is to give particular attention to the selection of a coil configuration which will be most suited for simultaneously offsetting the sensible and latent heat loads from air and water vapour mixtures. In general, the aim is to select the best dehumidifier to satisfy the simultaneous cooling and dehumidifying requirements for offsetting sensible and latent heat loads.

Energy conservation and enlarging of the climatic range are the two important aspects under consideration. It is not the purpose of this study to emphasize compactness or to stress improved overall heat exchanger performance. If this were done, it would be found to go counter to the objectives of this investigation. The conclusions drawn from the study using commercial data in Section 10, (under the six constraints of the comparison made in Section 10.6) indicated that low velocity air streams constrained to be shallow in depth should be investigated. These of course would be expected to perform with less capacity than higher velocity air streams with deeper coils. This reduced performance is of secondary consideration since it is not the initial costs of the direct expansion coil that is important in this investigation but rather the running costs. In the field of climate simulation, economy of operation is a determining factor in the enlargement of the range of the system. Initial cost is of slight monetary advantage whereas energy savings can be considerable and are of continuous on-going advantage highlighted by an increasing energy crisis. Furthermore, it is not the initial cost of the dehumidifier that must be considered but that of the entire complex including the size of the refrigeration plant and the cooling tower. For the purposes of energy conservation it is not more dehumidification, nor less sensible cooling, that is required but rather the correct ratio of these two simultaneous processes. In most cases, the problem is associated with more sensible cooling occurring relative to the required dehumidification.

11.2 Conditions Posing Special Difficulties to the Solution

The problem includes a number of conditions which serve to make analysis difficult.

1. The direct expansion coil has a complex surface. For example, in this particular case a commercial, helically wound finned tube is used.
2. The analysis must include the presence of a water film covering both the primary and secondary surfaces of the heat exchanger.
3. The refrigerant flow within the tubes will pass through varying degrees of dryness fraction in the two phase region.
4. In the Waite Institute system the dehumidifier has six parallel paths; the distributer can not exactly apportion the refrigerant in equal flow streams to each path.
5. The refrigerant flow within the tubes will include varying degrees of superheat. (Thus a section of the direct expansion coil surface has less effect in the heat and mass transfer process. The effect of refrigerant in the superheat region flowing within the evaporator will be considerable because the numerous parallel paths will each contribute surfaces in this condition).
6. Along with the refrigerant, Freon 12, there will be a flow of oil in the miscible state which will affect the coefficient of heat transfer through the refrigerant film.

The industrial method assumes that there is a constant surface temperature common to each increment of area taken in the direction of the air flow. The analysis in this section proceeds with the more realistic assumption that each incremental area in the direction of air flow will drop in temperature. Goodman (1936) and many others since, (Wile 1953, Kusuda 1957a, Mizushina et al 1959), etc. have developed equations associated with enthalpy potential difference, which establish the relationships which do remain constant for each increment of area along the path of the air stream. These offer both a mathematical and a graphical solution to the heat mass transfer processes.

This analysis will take into account more closely, (unlike the straight line methods used by commercial manufacturers), the actual performance of the air stream using incremental areas in steps taken in a plane perpendicular to the bulk motion of the air stream, including the boundary layer in contact with the condensed water film over the primary and secondary surface of the evaporator. A means for determining an average value for the combined coefficient of heat transfer through the refrigerant film on the inside of the tube, the metal and the water film will be described.

The entire project was studied under the conditions of the twelve constraints enumerated below.

11.3 Basis Of Comparison - A Study Of Three Coils

The basis for comparison is listed in the form of *twelve constraints*.

1. *All have the same primary surface area, diameter, wall thickness and material.*
2. *All have the same secondary fin surface area, diameter, wall thickness and material.*
3. *All have equal heat exchange surfaces that are geometrically identical.*
4. *All will be compared under conditions of the same mass flow of air, specific enthalpy, dry bulb temperature, humidity ratio at inlet to the coil.*

5. *One coil has twice the face area of a datum coil.*
6. *One coil has half the face area of a datum coil.*
7. *All have the same refrigerant condenser pressure.*
8. *All have the same refrigerant evaporator pressure.*
9. *All have the same dryness fraction at inlet to the evaporator.*
10. *All have the same superheat condition leaving the evaporator.*
11. *All evaporator surfaces are completely wetted.*
12. *The refrigeration capacity and mass flow of the refrigerant will be varied to be compatible with constraints 7 to 10 inclusive.*

It is obvious that the three configurations will not have the same total heat plus mass transfer capacity. The selection of the twelfth constraint is discussed below in Sections 11.3.1 and 11.3.2.

Figure 11.1 describes the coil arrangements being compared.

11.3.1 The twelfth constraint poses a problem

One major problem in this comparative study was to ascertain the conditions for the twelfth constraint.

It was obvious that with the eleven constraints enumerated in Section 11.3, for the same mass flow of air, different heat and mass transfer capacities would be associated with the three coils being compared.

One basis of comparison could have been to adjust the mass flow of air of the deepest and shallowest coils so that the three configurations would have the same total heat capacity, the same mass flow of refrigerant and the same refrigerant temperature. This basis has the important advantage that the combined heat transfer coefficient through water layer, metal and refrigerant, h_i, for the deepest and the shallowest coils would be identical with that determined for the datum

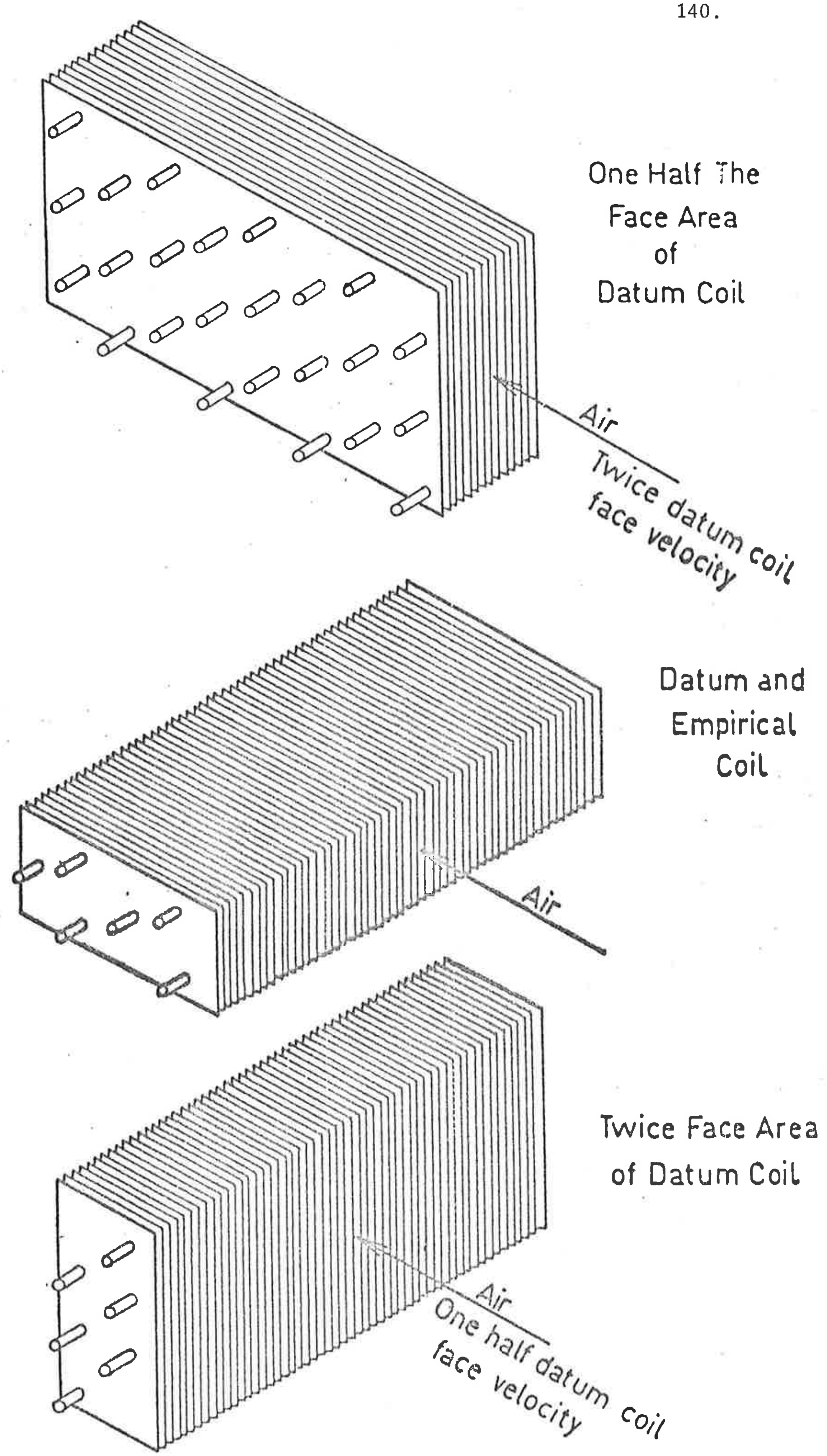


FIG 11.1 BASIS FOR COMPARISON OF THREE COILS

empirical coil. The reason for this is that the refrigerant capacity, refrigerant temperature, geometry and length of the evaporator surfaces beginning with the surface layer water temperature to the inside of the refrigerant tube would all be the same. Had this basis been selected, then criterion number 4 would be affected since the mass flow of air would be varied. This basis was rejected since in design practice the mass flow of air is determined by the loads and the temperature gradient to fit the problem. If this comparison is to yield any practical value for climate simulation or air conditioning design, it is necessary to maintain the same mass flow of air for each configuration.

Another basis for comparison would be to retain the same mass flow of air for all three configurations but to modify the largest and the smallest face area coils by adjusting the coil length. Thus the deepest coil would be made shallower since under the eleven constraints enumerated, it would have the highest total heat capacity. The shallowest coil with the least heat capacity would be made deeper until both coils had a heat capacity equal to the datum coil. Still another possibility was to increase the face area of the shallowest coil while at the same time reducing the face velocity to maintain the same mass flow of air. In such a scheme the deepest coil would have its face area decreased while at the same time increasing the face velocity to maintain the same mass flow of air. Both of these bases were rejected. They would have modified constraints 1, 2 and 3 and would have thus obscured the findings of this comparative analysis.

11.3.2 The twelfth constraint assumes the mass flow of the air stream is constant for the three coils

The basis of comparison that was finally selected as the twelfth constraint was to maintain the same mass flow of air, and accept the fact that each of the three coils will have different capacities. Obviously the deepest coil with the smallest face area would have the largest Reynolds number assignable and the highest refrigerant capacity, the shallowest coil the least. With this basis, the first eleven constraints would remain as listed. Thus the mass flow of refrigerant would alter to maintain constraints 7 to 11 inclusive to be identical for the three coils. In this way, each coil would have the same driving forces for heat and mass transfer, the same entering mass flow of air, the same entering enthalpy and the same constant evaporator temperature. Since the main purpose of this analysis is to assess the nature of the coil condition curve with particular attention to the ratio of the simultaneous processes of senisble and latent heat reduction, the fact that the analysis will involve different coil capacities does not detract from the information gained. On the contrary it adds to these findings, pointing out that one disadvantage in the use of shallow coils as per the twelve constraints is a spatial penalty and a higher capital cost heat exchanger. The reduced refrigeration capacity for the low velocity shallow coil need not be a disadvantage. Presumably if the coil has been selected to adhere to a desired coil condition curve, the reduced capacity is an advantage since it is indicative that overcooling is eliminated.

11.4 Thermodynamic Relationships Applicable to the Analysis

1. The enthalpy-potential equation for the heat transferred to the coil in a plane perpendicular to the air stream through an air film for an incremental area dA is

$$dq = \frac{h_{cow}}{c_{pm}} (H-H_s)\,dA \;. \qquad \ldots(1)$$

2. The equation representing the transfer of heat from a wetted surface of incremental area through the water film, metal and refrigerant film can be expressed as follows,

$$dq = \frac{h_i}{A_o/A_i} (t_s-t_r)\,dA \;. \qquad \ldots(2)$$

3. Equations 1 and 2 are equal and can therefore be expressed as

$$\frac{H_s-H}{t_s-t_r} = -\frac{c_{pm} h_i A_i}{h_{cow} A_o} \;. \qquad \ldots(3)$$

4. The Lewis Relationship is approximately equal to 1.

$$Le = \frac{h_{cow}}{h_{do} c_{pm}} \approx 1 \;. \qquad \ldots(4)$$

For air water vapour mixtures the value of 0.9 is closest to this application. Kusuda (1965) working with flat plates recommends for wetted surfaces this more precise value for the Lewis number given by the relationship

$$Le = \left(\frac{\alpha}{D}\right)^{2/3} \;.$$

For the saturated moist air at the wetted surface temperatures associated with this thesis Kusuda (1965) recommends a value of $(\alpha/D) = 0.854$. Though obtained from experiments using flat plates, the relationship involving vapour diffusivity applies as well to flow around cylinders and spheres and therefore Kusuda's recommendations will be used here. Thus the Lewis number in this study becomes $(0.854)^{2/3} = 0.900$. In this study lower velocities and lower Reynolds numbers are considered than conventionally used in dehumidifier applications. It may be questioned whether the value of 0.9 for the Lewis number will hold for these low velocities. Kusuda (1965) recommends for the case of natural convection that (α/D) be raised to an exponent of 0.48. The lowest velocity used here would of course be considerably more than for natural convection. Nevertheless if this value were to be used for conditions of minimum air flow the effect would be very slight. For example

$$\left(\frac{\alpha}{D}\right)^{0.48} = 0.927$$

rather than 0.900. Had this value been used in Section 11.7.6, h_{do}, the mass transfer coefficient would equal 0.048 rather than 0.049. It is therefore considered appropriate to use the value for the Lewis number of

$$\left(\frac{\alpha}{D}\right)^{2/3} = 0.9$$

throughout the analysis.

Thus

$$h_{cow} = 0.9\ (h_{do})(c_{pm})\ . \qquad \ldots(4a)$$

Substituting in (3) for h_{cow} we get

$$\frac{H_s - H}{t_s - t_r} = - \frac{h_i A_i}{0.9 h_{do} A_o} \quad \ldots(5)$$

This is the equation for the Tie Line Slope, a very important relationship in the comparative analysis that is made here.

11.5 Determination of the Tie Line Slope for the Datum Coil - The Left Hand Side of Equation 5 of Section 11.4

It will be shown in Section 11.7.1 that the value of h_i is very difficult to obtain. The laboratory phytotron unit at The Waite Institute and a laboratory teaching-research system built in the Mechanical Engineering Department offer a means by which the performance of the datum coil can be determined empirically and independently of the value of h_i. This is accomplished through the left hand side of the Tie Line Slope, equation 5 Section 11.4. Furthermore, once the Tie Line Slope is so obtained, a reliable average over-all value for h_i is also obtainable.

The equation is particularly adaptable to a semi-mathematical-graphical solution using the psychrometric chart for assessing energy transport conditions. Goodman (1936) in a very clear presentation, proved the fallacy of assuming that dehumidifying coils performed along a straight line path. He also showed that the surface temperature of a dehumidifier coil does not remain constant through the longitudinal depth of the coil. By separating his rows with sufficient space he was able to prove that there was a surface temperature drop in the direction of air flow. By both mathematical and graphical methods he developed a procedure for analysing a wetted surface and introducing the Tie Line Slope. Though his symbols are different his equation is similar and may be written

$$M = \frac{0.24 f_R}{B f_g} \qquad \ldots(6)$$

which is the right hand side of equation 3 Section 11.4

$$\frac{c_p h_i A_i}{h_{cow} A_o}$$

where, in Imperial units,

f_R = coefficient of heat transmission through refrigerant surface film (in lieu of h_i)

f_g = coefficient of heat transmission through air surface film (in lieu of h_{cow})

B = A_o/A_i

0.24 = c_p .

Though Goodman (1936) employed a conventional psychrometric chart, he devised a special monograph and table that associated wet bulb temperature and refrigerant temperature to surface temperature.

Above all, he expressed the characteristic which makes the Tie Line Slope such a very important factor in dehumidifier coil performance determination. This is - for every increment of surface, dA, over which an air stream flows there is a constant characteristic which permits both mathematical and graphical analysis of dehumidifier coil performance. That is the characteristic defined by the Tie Line Slope.

Other workers, (Merkel 1927; Mickley 1949; Mizushina et al 1959) have simplified the graphical problem by plotting the performance on a psychrometric chart with enthalpy as the ordinate.

In this analysis a conventional psychrometric chart is employed. However by the method of Kusuda (1957b) a Surface Temperature

Determination Line is superimposed on this conventional psychrometric chart.

The Tie Line Slope expresses a constant ratio. The numerator represents the steady heat and mass transfer occurring from the air to the wetted outside surfaces of the direct expansion coil at each infinitesimal area of complex surface in the direction of air flow. The denominator represents the equivalent amount of heat and mass transfer in heat transfer terms only. It represents the transfer of heat from the wetted outside surface above, through the water layer, metal and refrigerant film.

Knowing the numerator alone fails to reveal the relationship that exists between the heat and mass transferred since it is a total heat term. However, after the air dry bulb temperature or any other independent property of the air stream is defined on a psychrometric chart, a visual understanding of the relationship becomes apparent. Each increment of area in coil depth is associated with a surface temperature. If this surface temperature is lower than the dew point temperature of the air stream passing through this increment of area, then mass transfer is taking place. Otherwise, the coil is performing dry and no moisture is being condensed on the coil surface at the increment of area being considered.

The denominator is associated with surface and refrigerant (coolant) temperatures only. Clearly the refrigerant capacity when viewed from outside surface inwards is associated with only heat transfer.

Kusuda (1957b) in constructing a Surface Temperature Determination Line superimposed a plot of temperature at right angles to the enthalpy scale on a psychrometric chart. The entire transport process from the air temperature and its enthalpy to the refrigerant temperature and its corresponding fictitious 'air' enthalpy value are drawn to the same scale as the conventional chart upon which this plot is superimposed. The Surface Temperature Determination Line may be viewed as the pivot point at which the superimposed temperature ordinate can be transferred to the coordinates of the conventional psychrometric chart. This is done by way of lines of constant enthalpy which are common to both the conventional psychrometric chart and the superimposed enthalpy-temperature plot. Figure 11.2 below shows the relationship of the Surface Temperature Determination Line with the conventional psychrometric chart and the new imposed surface and refrigerant temperature axis. The method of plotting the Surface Temperature Determination Line is outlined by following the example for locating several of the temperature points, (5C, 10C and 30C), on this Surface Temperature Determination Line. The heavy construction lines marked with arrows shown on Figure 11.2 locate these three respective points.

Using the Surface Temperature Determination Line, S.T.D.L., the entire construction of a dehumidifying process path can be carried out on a single diagram.

This procedure is particularly suited to the comparative study being made here when there is a constant refrigerant temperature.

By this diagram the complex functional dependence between the entering enthalpy of the air stream, the wetted surface enthalpy, the entering dry bulb temperature and the wetted surface temperature can most effectively be seen.

The plot reveals whether the surface is completely wetted. If the straight line drawn in step 6 below slopes up to the right then the coil is completely wetted.

In this particular study there will be a trial and error solution. However, with the additional plot of temperature-enthalpy

PSYCHROMETRIC CHART

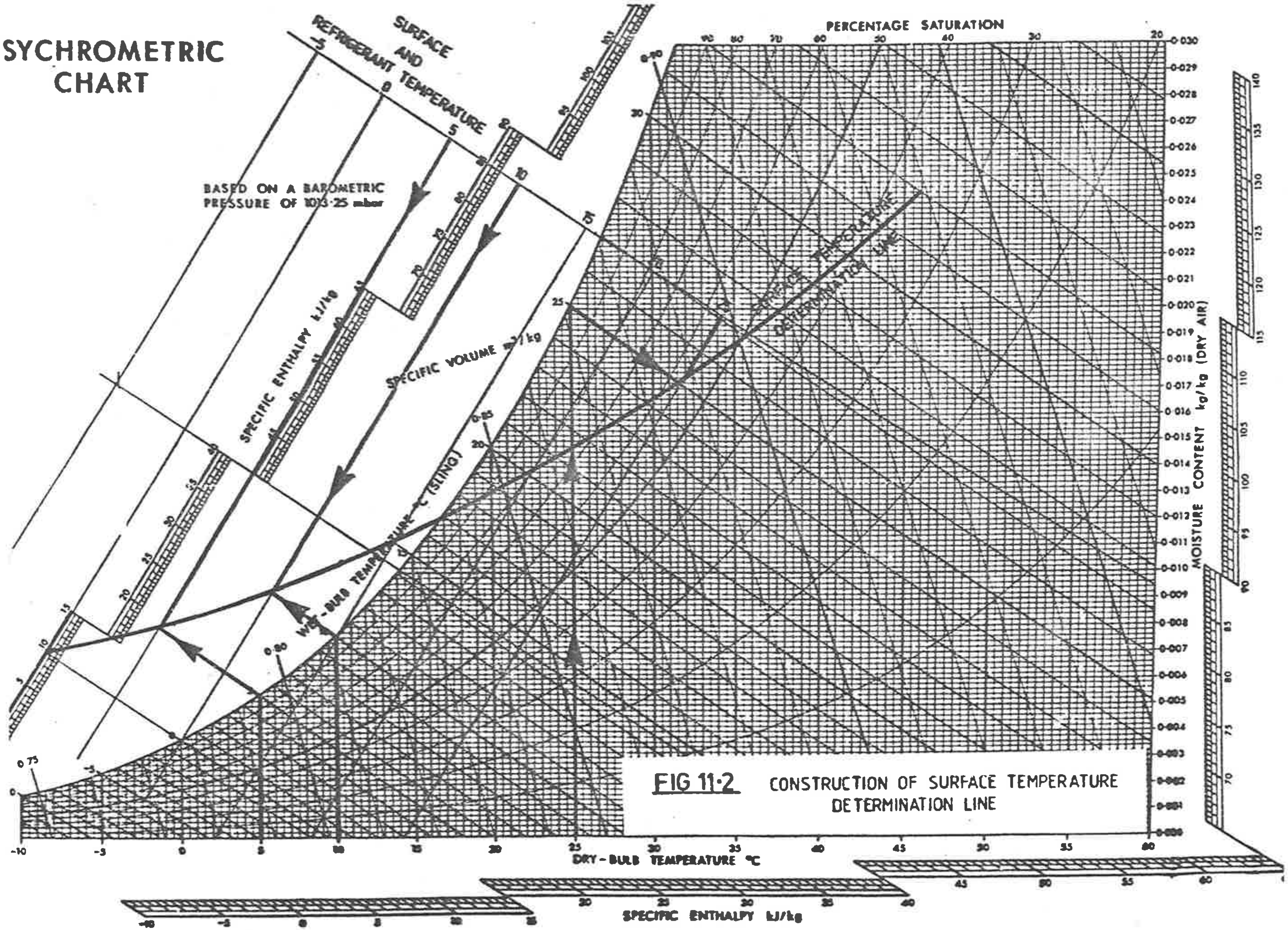


FIG 11·2 CONSTRUCTION OF SURFACE TEMPERATURE DETERMINATION LINE imposed on a conventional psychrometric chart the solution process is facilitated .

The trial and error solution of the Tie Line Slope equation on the conventional psychrometric chart that has both a temperature scale and a Surface Temperature Determination Line as per Kusuda (1957b) proceeds as outlined in the example below. Each numbered step is identified on Figure 11.3.

1. The entering condition, 1a, and leaving condition, 1b, determined empirically for a datum coil is located on the chart. The refrigerant temperature line, 1c, is drawn in.
2. Starting with the entering condition, a line is drawn upwards and to the left along a line of constant enthalpy to the refrigerant temperature line.
3. At the intersection of the entering enthalpy with the refrigerant temperature line of the newly imposed coordinates, a line is drawn at an assumed Tie Line Slope down until it intersects the Surface Temperature Determination Line at point 4.
4. With this point of intersection as a pivot, the conventional psychrometric chart section is entered along the line of constant enthalpy which passes through this point.
5. This line, labelled 5, is extended until it intersects with the saturation curve of the conventional psychrometric chart. This point represents the surface temperature corresponding to the entering condition differential area of coil.
6. A straight line, labelled 6, is then drawn connecting this surface temperature to the entering condition.

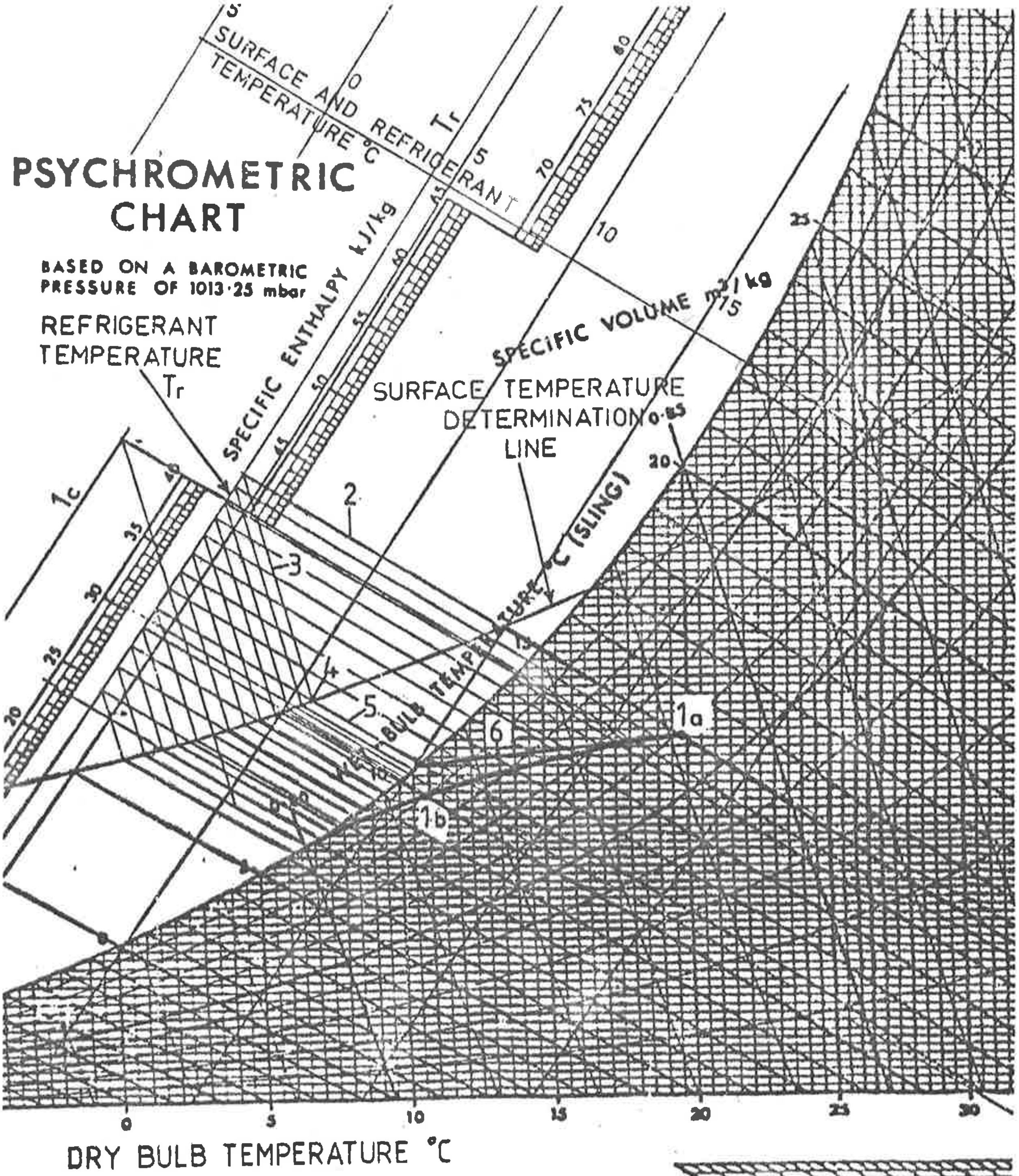


FIG 11·3 CONSTRUCTION OF COIL CONDITION CURVE

7. At some small arbitrary distance from that of the entering condition the process is repeated again a second time. This distance is so selected that a smooth coil condition curve would result when drawn through a series of points representing the incremental areas analysed along the path of air flow.
8. The analysis of the second differential increment of area would follow the same sequence of steps described for the entering condition. A new and lower surface temperature would be obtained. All Tie Line Slopes would be drawn parallel to the assumed Tie Line Slope for step 3.
9. The process would be continued until a curve drawn through each incremental point was found to pass through over or under the leaving condition which was determined empirically.
10. If it passes through this condition then the assumed Tie Line Slope was correct. If it passes over, it is too shallow, and if it passes under, it is too steep, and a new trial Tie Line Slope is to be assumed until the curve does pass directly through the leaving condition.

Usually three to four trials are all that is necessary and the left hand side of the Tie Line Slope equation is solved. The process described above is demonstrated for the entering and leaving conditions and refrigerant temperature of Run 1, for the datum coil. Only the trial giving the correct solution is shown. See Figure 11.3.

The Tie Line Slope obtained by this method for Datum Run 1

$= -2.50 . \frac{kJ}{kgK}$.

11.6 Confirmation of the Tie Line Slope Obtained for the Datum Coil With Log Mean Values

The accuracy of the Tie Line Slope obtained for the datum coil can now be tested by calculating directly the log mean wetted surface enthalpy and the log mean air enthalpy. Though the circuiting of the evaporator is a complex combination of cross flow and counter flow, the refrigerant temperature is very nearly constant and therefore log mean values apply (except for the superheat portion and effect of minor pressure drops). The value of the left hand side of the Tie Line Slope equation is:

$$\text{Tie Line Slope} = \frac{H_s - H}{t_s - t_r} \text{ .}$$

The log mean value for H and H_s is found knowing t_{s1} for H_1, t_{s2} for H_2 and knowing t_r. H_1, H_2 and t_r are empirical values. The values t_{s1} and t_{s2} are obtained from the curve which yields the surface temperature associated with each incremental area being analysed. The intersection of the first and last incremental coil condition curves with the saturation curve will give the values of t_{s1} and t_{s2}. The log mean air enthalpy will now be determined for Datum Run 1:

$$\Delta H_m = \frac{H_1 - H_2}{\ell n\left(\frac{H_1 - H_r}{H_2 - H_r}\right)} \rightarrow \Delta H_m = \frac{42.0-32.2}{\ell n\left(\frac{42.0-17.2}{32.2-17.2}\right)} = 19.6\ \frac{\text{kJ}}{\text{kg}}\ , \qquad \ldots(7)$$

which leads to

$$H_m = H_r + \Delta H_m \rightarrow H_m = 17.2 + 19.6 = 36.8\ \frac{\text{kJ}}{\text{kg}}\ .$$

The log mean wetted surface enthalpy is

$$\Delta H_{sm} = \frac{H_{s1}-H_{s2}}{\ell n\left(\frac{H_{s1}-H_r}{H_{s2}-H_r}\right)} = \Delta H_{sm} = \frac{28.2-23.7}{\ell n\left(\frac{28.2-17.2}{23.7-17.2}\right)} = 8.7\ \frac{kJ}{kg}\ , \qquad \ldots(8)$$

and hence

$$H_{sm} = H_r + \Delta H_{sm} \rightarrow H_{sm} = 17.2 + 8.7 = 25.9\ \frac{kJ}{kg}\ .$$

From psychrometric tables the values of H_{sm} determines t_{sm}.

$$t_{sm} = 8.5C\ .$$

Now returning to the left hand side of the Tie Line Slope equation

$$\frac{H_s-H}{t_s-t_r}$$

and substituting the log mean values just obtained we get

$$\frac{H_{sm}-H_m}{t_{sm}-t_r} \rightarrow \frac{25.9-36.8}{8.5-\ 4.2} = -2.5\ \ \frac{kJ}{kgK}$$

which is exactly equal to the Tie Line Slope found to yield the performance curve on the psychrometric chart. This indicates the accuracy obtainable by using the Tie Line Slope equation for determining the performance curve for air passing through a dehumidifier.

11.7 Determination of h_i, the Combined Coefficient of Heat Transfer Through Water Layer, Metal and Refrigerant Film, For Inside Surface of Datum Coil

In this section the value of the combined coefficient of heat transfer through the water layer, metal and refrigerant film, for inside surface h_i, will be determined for the datum coil. The datum coil Tie Line Slope is known, A_i and A_o are known from the manufacturer's coil data. Therefore, if the mass transfer coefficient for outside air h_{do}, for the datum coil is determined then h_i can be obtained from equation 5 of Section 11.6. Tie Line Slope (Datum) =

$$\frac{-h_i A_i}{0.9 h_{do} A_o} . \qquad \ldots(5)$$

It is first necessary to determine the value of h_{do}. This is developed in Sections 11.7.2 to 11.7.6 inclusive.

Based on the non-dimensional relationships plotted by Kays et al (1964) in the form,

$$St(Pr)^{2/3} = \phi(Re) , \qquad \ldots(9)$$

and knowing Reynolds number it is possible to ascertain the value of the product of the Stanton Number and the Prandtl Number raised to the two thirds power for a heat exchange surface similar to the one being used in the research system.

Key to the solution of the value of h_{do} is the determination of the convective heat transfer coefficient for the outside dry surface, h_{cod} developed in Section 11.7.4. The Stanton number includes h_{cod} as one of the properties which forms this dimensionless term. The value of h_{cod} through the Myers relationship gives a value for the convective heat transfer coefficient for outside wetted surfaces,

h_{cow}. In Section 11.7.6, h_{do} is obtained from the value determined for h_{cow} and through the Lewis relationship expressed by equation 4a, Section 11.4.

11.7.1 The problem of determining h_i, the combined coefficient of heat transfer for inside surface

It was indicated in Section 11.2 that there is considerable difficulty in obtaining the average value of h_i. Different values for h_i can be expected at different incremental areas along the path of flow of the refrigerant due to varying dryness fractions and refrigerant vapour in the superheated condition. The value of h_i under forced convection boiling varies considerably and there is very little information on boiling heat transfer in this area. ASHRAE Handbook of Fundamentals (1977b) highlights the problem. Figure 11.4 taken

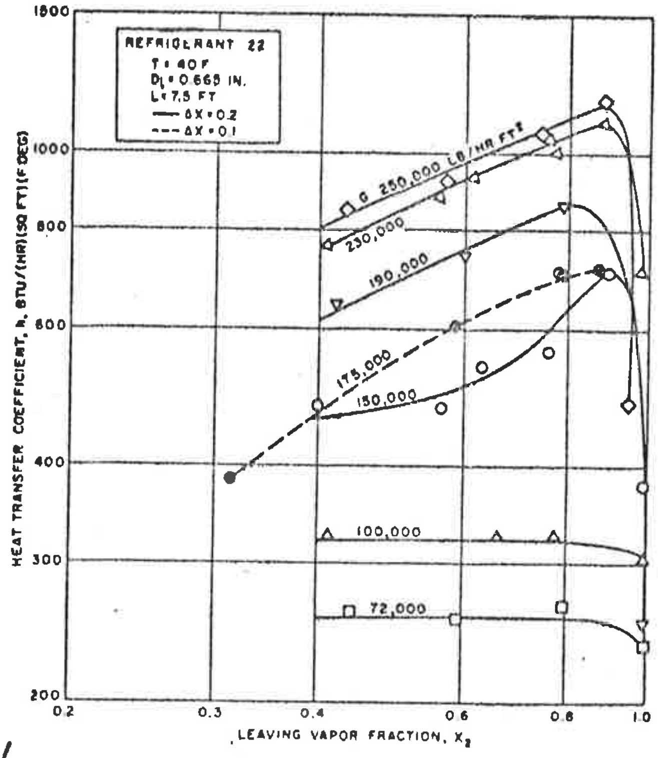


FIG 11·4

Heat Transfer Coefficient vs Vapor Fraction for Partial Evaporation from this reference indicates the enormous variation in the heat transfer coefficient with vapour fraction during partial evaporation, particularly during high mass velocities.

Pierre presents equations for determining the heat transfer coefficient of the refrigerant film at the inside surface of the tubes. This coefficient is the main component in determining h_i with a small contribution from the water layer and literally no contribution in significant figures from the metal. In this particular comparative study h_i for the datum coil will be determined independently. The Pierre equation, (ASHRAE Handbook 1977a) for Freon 12 with a superheat of 6C and a dryness fraction below 0.5C will be applied in determining h_i for the two non-empirical coils related to the experimental datum coil.

11.7.2 Calculation of the hydraulic diameter

Kays et al (1964) defines a value of Reynolds number based on an equivalent flow passage hydraulic diameter, $4r_h$, for the dehumidifier. In dimensionless terms this may be expressed as

$$\frac{4r_h}{L} = 4 \frac{A_c}{A} , \qquad \ldots(10)$$

where L is the flow length of the heat exchanger, A is the total heat transfer area, and A_c is the minimum free flow area. A number of research workers have suggested various ways of determining this characteristic length. The tube bundle is not a simple flat plate where the chord length can be used. Gunter and Shaw (1945); Kays, London and Lo (1954); and Jameson and Schnectady (1945) have explored this problem. The characteristic length (or more appropriately, the Equivalent Diameter) used in this work will be that used by Gunter and Shaw. According to them, Equivalent Volumetric Diameter is defined as

$$D_e = \frac{4 \text{ (Free Volume of tube bundle)}}{\text{(outside surface area of the tubes)}} \,. \qquad \ldots(11)$$

Imperial units will be used below since comparison is to be made with the data of Kays et al (1964) and that of the American manufacturer of the coil. These are given in imperial units. A sketch of the coil arrangement is shown in Figure 11.5. The Waite Institute coil data are as follows:

Tube diameter, $D = 5/8$ in. × 22G.

Fin height $= \frac{13}{32}$ inches.

Fin thickness, $t = 0.012$ inch.

No. of fins per inch, $N = 8$.

Secondary/primary surface = 11.

Total external surface (Sec. + Prim.)/internal surface

= 13.33.

Total external surface/Face area = 16.35 per row.

Area per foot of tube (internal = 0.149 ft^2.)

External surface ratio (External to Internal) = 13.2.

External surface to Face area per row = 16.2.

The direct expansion coil has 54 tubes made up of 3 rows deep of 18 tubes per row.

Referring to Figure 11.5.

Free Volume of tube bundle

= (Vol. of prism) - ½ (Vol. of 1 inch of finned tube length) ...(11a)

$$= \tfrac{1}{2}\, LH - \tfrac{1}{2}\left[\left(\frac{\pi}{4} \times D^2 \times 1\right) + Nt \times \frac{\pi}{4}(D_2^2 - D_1^2)\right] .$$

Outside surface area of tubes

$$= \tfrac{1}{2} \times \text{External Surface Ratio} \times \frac{\text{Area of fins/ft. of tube}}{12} \times 144 .$$

Substituting the values from coil data.

Free Volume of tube bundle

= 0.643 in^3.

Outside surface area of tubes

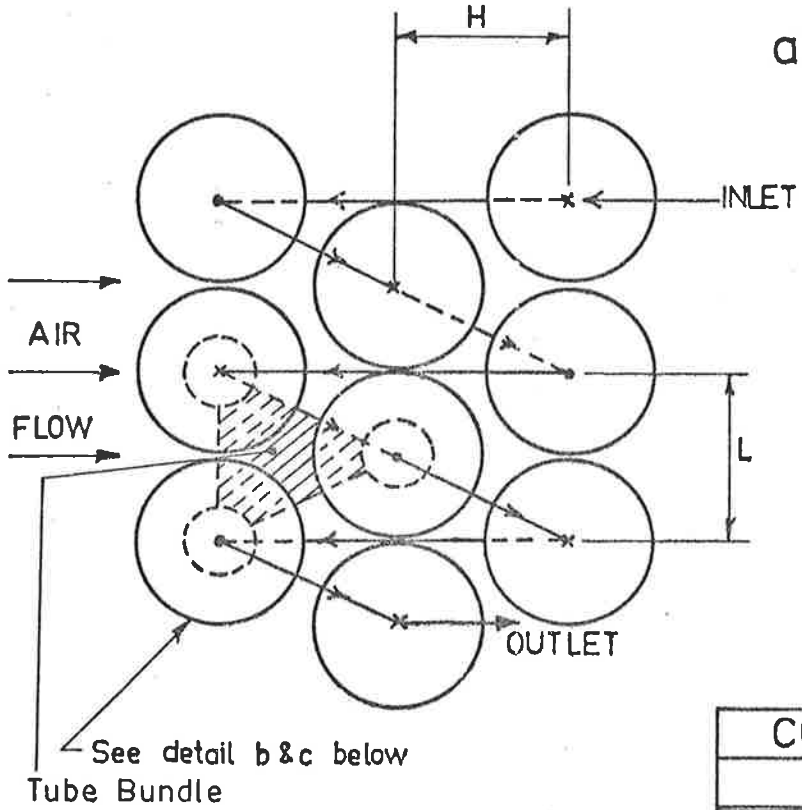
= 11.8 in^2.

Hence

D_e = 0.218 inch.

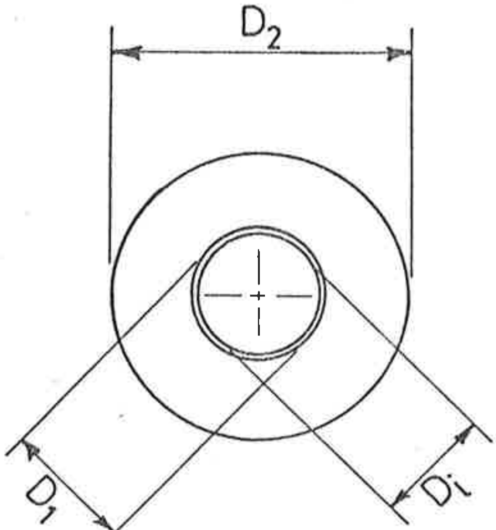
The Equivalent Diameter, D_e = 0.0182 ft. or 5.56×10^{-3} metres. This compares very closely with the Equivalent Diameter of Figure 10.80, Type A in Kays et al (1964) reproduced below as Figure 11.6. = 0.0179 feet or 5.48×10^{-3}m.

This method of obtaining the equivalent diameter is generally accepted for the staggered arrangement of the tube bundle where the centre to centre longitudinal distance, (dimension H in Figure 11.5a), in the direction of air flow is different from the centre to centre transverse distance perpendicular to the direction of flow, (dimension L in Figure 11.5a). However, it should be noted that though used here - the equivalent diameter for a deep coil of many rows would not be identical with that of a shallow coil. Lohrisch in Hsu (1962) indicated how the flow pattern of in line and staggered arrangements of tubes are affected by preceding rows. Under staggered arrangement the effect may not occur until the 3rd row in depth. Therefore, in this comparison of shallow to deep coils the equivalent diameter would be somewhat different.

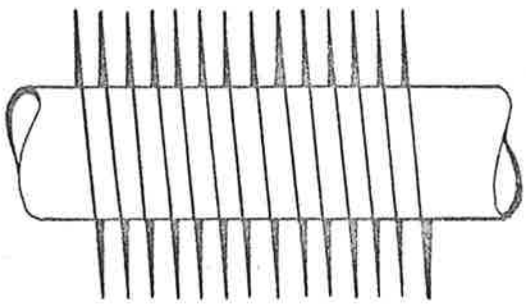


COIL DATA		
	m.m.	inches
L	34·9	1·375
H	31·75	1·250
D_i	14·47	0·57
D_1	15·87	0·625
D_2	36·4	1·437

Helical Fins 3·2 per c.m.
(8 per inch)



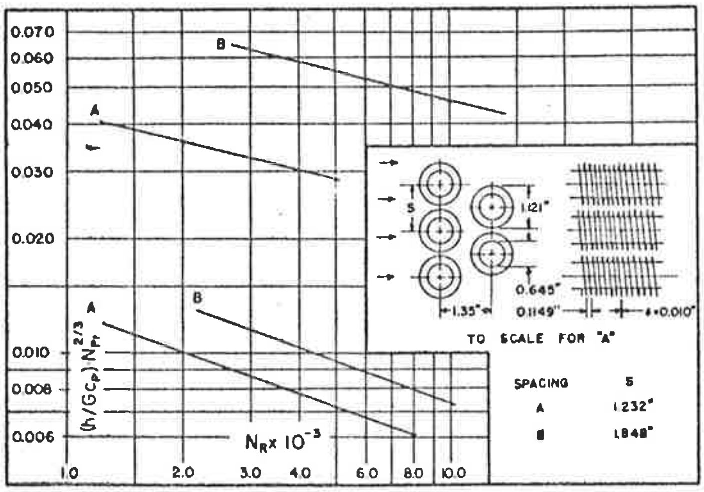
b) CROSS SECTION OF A FINNED TUBE



c) FACE VIEW OF A FINNED TUBE

Fig.11.5 COIL DATA AND ARRANGEMENT.

Fig. 10-80. Finned circular tubes, surfaces CF-8.7-5/8 J.
(*Data of Jameson.*)



Tube outside diameter = 0.645 in.
Fin pitch = 8.7 per in.
Fin thickness = 0.010 in.
Fin area/total area = 0.862

		A	B
Flow passage hydraulic diameter,	$4r_h$	= 0.01797	0.0383 ft.
Free-flow area/frontal area,	σ	= 0.443	0.628
Heat transfer area/total volume,	α	= 98.7	65.7 ft^2/ft^3

Note: Minimum free-flow area is in spaces transverse to flow.

FIG 11·6 HEAT TRANSFER TEST DATA

Reproduced from Figure 10·80 (Kays et al 1964)

11.7.3 Calculation of Reynolds number

Having determined the equivalent diameter it is now possible to assign a Reynolds number to the flow pattern through this extended surface. All values are known and yield

$$Re = \frac{D_e G_m}{\mu} \longrightarrow Re = \frac{(5.48\times10^{-3})(2.24)}{(0.018)(10^{-3})} = 682 \; .$$

11.7.4 Calculation of h_{cod}, the convective heat transfer coefficient for the outside dry surface

Having determined Reynolds number using an equivalent diameter, the value of the $(St)(Pr)^{2/3}$ number is known from Figure 10.80 in Kays et al (1964) and is equal to 0.0150.

In detail, for Run 1,

$$\left(\frac{h_{cod}}{G_m c_{pm}}\right)\left(\frac{c_{pm}\mu}{k}\right)^{2/3} = 0.0150 \; . \qquad ...(9)$$

The only unknown in the above relationship is h_{cod}, the convective heat transfer coefficient for the outside dry surface.

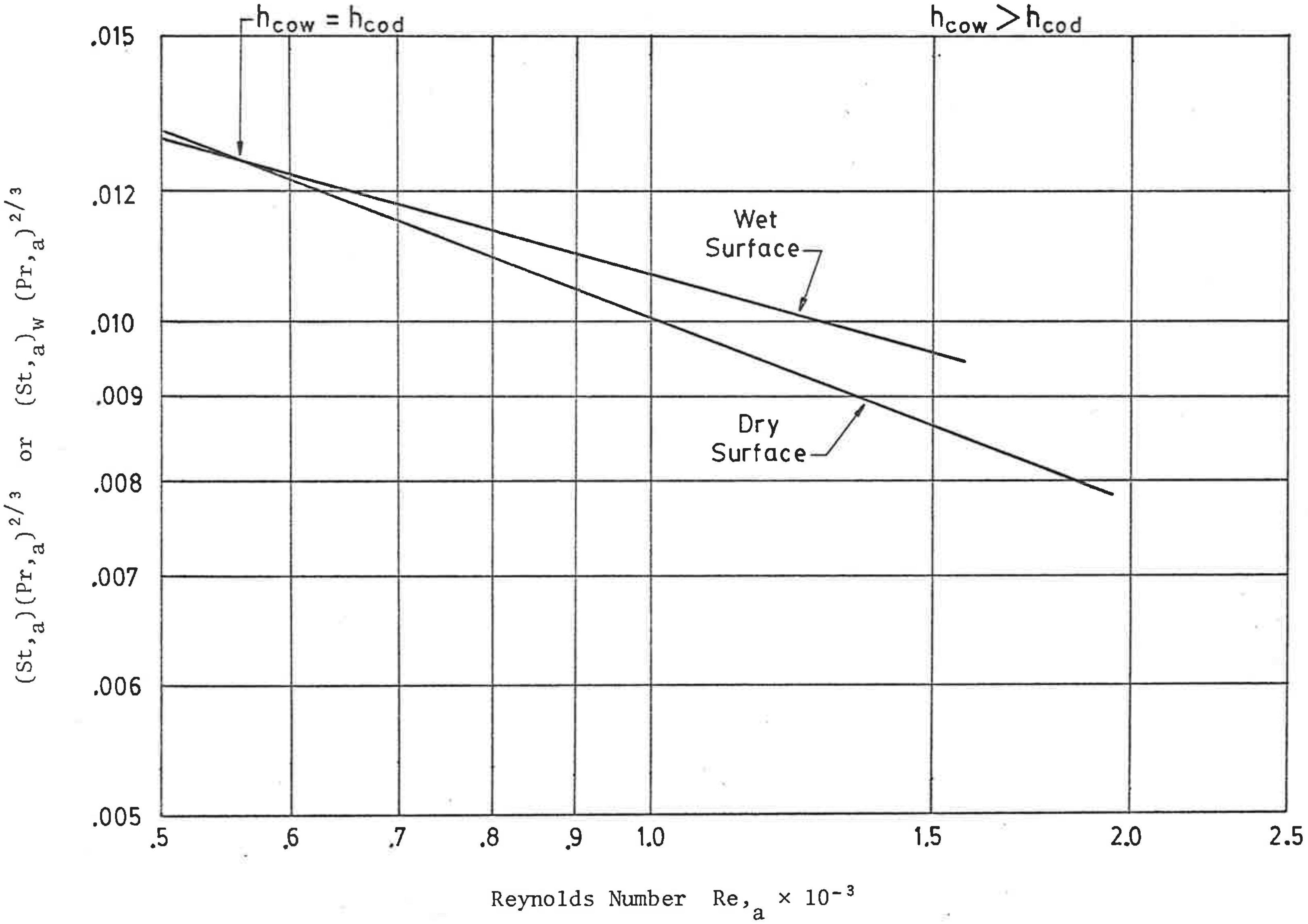
Rearranging the above equation and solving for h_{cod},

$$h_{cod} = \frac{f(Re)G_m c_{pm}}{(Pr)^{2/3}} \rightarrow h_{cod} = \frac{(0.0150)(2.24)(1.022\times10^3)}{0.8} = 43.09 \; \frac{W}{m^2K}$$

11.7.5 Calculation of h_{cow}, the heat transfer coefficient for outside wetted surface

The next step is to determine the value of h_{cow}, the heat transfer coefficient for outside wetted surface.

A Masters Thesis by R.J. Myers (1967a) establishes a relationship between the convective heat transfer coefficients for outside dry and wetted surfaces. Figure 11.7 is repeated from page 115 of his thesis representing the particular extended fin surface used by Myers.



N.B. Reproduced from Myers Figure 35 (1967) with extrapolation to the left and additional comment regarding h_{cow} and h_{cod}.

Fig. 11.7 Comparison of Dry Surface and Wet Surface Heat Transfer Coefficient Correlations

In a review, Myers (1967b) points to three references in the literature on the effect of dehumidification on the air side heat transfer coefficient. One opinion, a paper by Ware and Hacka (1960) concluded no significant differences existed. It would be of interest to study the details of the tests performed to see if they took place under conditions where the wet and dry surface curves of Figure 11.7 converge - and if extrapolated, cross. This would account for the insignificant differences found. Another paper, ARI Standards (1964) offers curves which indicate the wet surface coefficient is higher than the dry surface coefficient. A third paper by W.L. Bryan (1962) concluded that the wet surface coefficient was lower than the dry surface coefficient. However, as pointed out by Myers, the conclusions were based on a Lewis number greater than 1. This casts doubts on the validity of Bryan's conclusions. The work of Kusuda (1965) and of the general consensus to date point to a Lewis number of 1 or less than 1.

Professor J.L. Threlkeld (1970) has incorporated this information in his text book, finding Myers' work consistent with the procedures he himself describes within his text. Myers empirical arrangement and analysis appears to be well conceived and therefore the dry coil performance data of Kays et al (1964) has been converted in this thesis by the Myers factor for wetted surface conditions.

It is recognized that there is still considerable research necessary before these findings may be totally accepted for extended surfaces.

It is apparent that the conclusions presented here are valid both with and without Myers' (1967c) modification, though with the modification, further energy conservation is attainable and the Myers' work amplifies the effect of the recommendations suggested here.

The extended surface used in Myers' work was that of a plate fin type coil having the characteristics of Kays et al (1964) surface shown in Figure 10.83. This coil uses a smaller diameter primary tube than the one used for this analysis. In this paper helically wound fins are used similar to Kays et al Figure 10.80, type A spacing.

The calculations have included two recommendations made by Myers:

1. The extended surface minimum free flow area has taken into consideration the presence of a water film having a thickness assumed to be 1.016×10^{-4}m. (0.004 inches) over the fin area.
2. The Kays curve of Figure 10.80 has been modified to account for the presence of a water film. Though the same Reynolds number would hold for both dry and wet coils since the changed effect of decreased cut-off area A_i, is cancelled by the increased mass flow rate, G_m, the curve relating Reynolds number to the $(St)(Pr^{2/3})$ ordinate has been corrected to take into account the presence of a wetted surface.

It can be seen by examining Fig. 11.7 above that if the two curves of the Myers' coil are extended to the left, for this particular coil they will cross at about a Reynolds number of 550. Here the wetted surface coefficient is equal to the dry surface coefficient which would be very nearly the case for the shallowest coil of this comparison. The divergence of the curves at increasing higher air velocities affect this comparative study since for the same Reynolds number greater values for the $(St)(Pr^{2/3})$ ordinate would be obtained for a wetted surface as compared with the dry surface data of Kays et al (1964).

The higher air velocity deepest coil, due to the divergence, would result in a considerably larger value for the convective heat transfer coefficient for the outside wetted surface than for the outside dry surface. This in turn would result in a larger mass transfer coefficient for the air. This larger coefficient would be responsible for a higher Tie Line Slope for the deeper coil. This is the reason Myer's findings further emphasise the conclusions drawn in this thesis, pointing to the use of considerably lower face velocities for coils in applications involving low sensible heat ratio conditions and/or small temperature gradients, as is the case in climate simulator design. Possibly an increase in dropwise condensation may be responsible for the divergence of these two curves of Figure 11.7.

The empirical equations shown below were used by Myers to correlate the values of h_{cod} and h_{cow} respectively. Originally the equations were written in terms of $V^1_{f,std}$ standard airface velocities measured in feet per minute. They are expressed here in terms of $V_{f,std}$ measured in metres per second where $V_{f,std} = 0.00508\ V^1_{f,std}$. For this reason the constants differ from those originally quoted.

Heat transfer coefficient for a dry external surface:

$$h_{cod} = 6.353\ (V_{f,std})^{0.6307} \qquad \ldots(42 \text{ Myers 1967d})$$

Heat transfer coefficient for a wetted external surface:

$$h_{cow} = 6.785\ (V_{f,std})^{0.732} \qquad \ldots(102 \text{ Myers 1967e})$$

Combining the 2 equations above gives the relationship

$$\frac{h_{cow}}{h_{cod}} = 1.067\ (V_{f,std})^{0.101} \qquad \ldots(12.62 \text{ Threlkeld 1970})$$

Applying this equation to Run 1 Datum Coil:

$$\frac{h_{cow}}{43.09} = 1.067\ (0.919)^{0.101}$$

$$h_{cow} = 45.60\ \frac{W}{m^2K}\ .$$

11.7.6 Calculation of h_{do}, the mass transfer coefficient for outside surfaces

Knowing h_{cow} one can obtain h_{do}, the mass transfer coefficient for the outside surface through the value of the Lewis number derived from Kusuda (1965). Thus

$$h_{do} = \frac{h_{cow}}{0.9c_{pm}} \rightarrow h_{do} = \frac{45.60}{0.9(1022)} = 0.0494\ \frac{kg}{sm^2} \qquad \text{...(4a)}$$

11.7.7 Calculation of h_i, the combined coefficient of heat transfer through the water layer, metal and refrigerant film for the datum coil from data of Sections 11.7.2 to 11.7.6

The Tie Line Slope equation derived in Section 11.4 was written

$$\frac{H_s - H}{t_s - t_r} = -\frac{h_i A_i}{0.9h_{do}A_o}\ . \qquad \text{...(5)}$$

The value of the Tie Line Slope for the datum coil was determined from the left hand side of the equation by the methods of Section 11.5 above.

With h_{do} determined from Section 11.7.6 all values on the right hand side of the equation are known with the exception of h_i. Thus an 'average' value for h_i can be obtained for the total surface which is to serve as the datum coil in the research system. Thus

$$h_{i_{av}}\Big]_{DATUM} = -0.9 \text{ (Tie Line Slope) } (h_{do})(A_o/A_i)$$

$$h_{i_{av}}\Big]_{DATUM} = \frac{-0.9\ (-2.50)\quad (0.049)\quad (13.33)}{(-2.50)(0.049)(13.33)} = 1.47\times10^{3}\ \frac{W}{m^2K}\ .$$

11.8 Determination of h_i for the Two Non-Empirical Coils, the Shallowest and Deepest to Satisfy the Twelve Constraints

The twelfth constraint, established in Section 11.3, accepts a different refrigeration capacity and a different mass flow rate of refrigerant for the three coils of the comparison. The value of h_i for the datum coil will be different therefore for the shallowest and deepest coils of the comparison since, though the geometry is the same, the capacity differs. The shallowest and deepest coil used in this study are not evaluated empirically. For this to have been possible, condensing units of a size that was compatible with the new capacities would have been required and these would have had to maintain the datum coil refrigeration conditions, i.e.

- condenser pressure
- evaporator pressure
- dryness fraction at inlet to evaporator
- superheat condition leaving the evaporator as in constraints 7 to 10 inclusive.

(No alteration to the compact Waite Institute Phytotron Unit was possible).

The major component in the combined heat transfer coefficient is the coefficient through the refrigerant film, h_r. Perhaps the best means of determining h_i for the deepest and shallowest coil is to modify the h_i of the datum coil which was determined in Section 11.6.7 above. The Pierre equation which most closely fits the condition of operation of the condensing unit will be used for this purpose.

$$h_{r(avg)} = \frac{0.0082k_\ell}{d}\left[\left(\frac{G_r d}{\mu_\ell}\right)^2\left(\frac{J\Delta XH_{fg}}{L}\right)\right]^{0.4} \qquad \ldots(12)$$

Equation 12 above holds for the condition where the dryness fraction of the refrigerant entering the evaporator is less than 50 per cent and the superheat value is 6C. This is also the basis for the datum coil operating conditions. Though this equation is stated in terms of Imperial units it is not necessary to convert to metric units since in this problem ratios are being used. The only term that will vary from that of the datum coil is G_r due to the changed refrigeration capacity which is reflected in a change in the mass flow rate of refrigerant. All other terms in the equation remain the same due to the constraints listed above in this section.

From the manufacturer's data is has been determined that a shallower coil compatible with the twelve constraints would have a refrigeration capacity of 84 per cent of the datum coil; and for the deeper coil, the refrigeration capacity would be 119 per cent of the datum coil. Therefore the Pierre equation reduces for the shallow coil to

$$h_{i\ (shallow)} \approx h_r = \left[\left(\frac{G_{r\ (shallow)}}{G_{r\ (datum)}}\right)^2\right]^{0.4} h_{i\ (datum)} .$$

For Run 1:

$$h_{i\ (shallow)} \approx h_r = [(0.84)^2]^{0.4}(1.47\times10^3) = 1.28\times10^3\ \frac{W}{m^2K} .$$

For the deeper coil, it reduces to

$$h_{i\ (deep)} \approx h_r = \left[\left(\frac{G_{r\ (deep)}}{G_{r\ (datum)}}\right)^2\right]^{0.4} h_{i\ (datum)}$$

which for Run 1 becomes

$$h_{i\ (deep)} \approx h_r = [(1.19)^2]^{0.4}(1.47\times10^3) = 1.69\times10^3\ \frac{W}{m^2K}\ .$$

11.9 Determination of the Tie Line Slope for the Shallowest and Deepest Coil

The datum coil was available and was used as part of the research system. The Tie Line Slope was determined as described in Section 11.5. However, since the shallowest and deepest coils of this comparison are non-empirical, it is necessary to develop the Tie Line Slope by another method. The empirical information of the datum coil and the defining constraints of Section 11.3 will be used to determine the Tie Line Slope for the shallowest and deepest coils of this comparison.

The right hand side of the Tie Line Slope equation

$$\left[-\frac{h_i A_i}{0.9 h_{do} A_o}\right]$$

has been selected as being the more amenable to solution.

In Section 11.7 above, a method for obtaining the value of h_i was presented. The only remaining unknown is h_{do}. This is readily obtainable following step by step the solution used for the value of h_{do} of the datum coil, Sections 11.7.2 to 11.7.6 inclusive.

The equivalent diameter determined for the datum coil also applies to the shallowest and deepest coils. In the relationship

$$Re = \frac{D_e G_m}{\mu} ,$$

the only term that changes is G_m which is defined by constraints 5 and 6 of Section 11.3 as being respectively half and twice the value of G_m for the datum coil. This in turn changes the value of the Reynolds number to half and double respectively. Thus Reynolds number is known for the 2 fictitious coils.

Re for the shallowest coil = 341;

Re for the deepest coil = 1364.

Following the same procedure as in Section 11.6.4, h_{cod} is determined through Kays et al (1964) data relating Reynolds number to the product of the Stanton $(\text{Prandtl})^{2/3}$ and

$$h_{cod} = \frac{f(Re)G_m c_{pm}}{(P_r)^{2/3}} \rightarrow h_{cod}\Big]_{\substack{\text{shallowest}\\ \text{coil}}} = \frac{(0.0185)(1.12)(1.022\times10^3)}{0.8} = 26.47 \frac{W}{m^2K} ,$$

$$\rightarrow h_{cod}\Big]_{\substack{\text{deepest}\\ \text{coil}}} = \frac{(0.012(4.48)(1.022\times10^3)}{0.8} = 68.95 \frac{W}{m^2K} .$$

Through Myers relationship h_{cow} is determined as in Section 11.6.5:

$$h_{cow} = h_{cod}\,(1.067)(V)^{0.101} \rightarrow h_{cow}\Big]_{\substack{\text{shallowest}\\ \text{coil}}} = (26.47(1.067)(0.459)^{0.101} = \underline{26.1} \frac{W}{m^2K},$$

$$\rightarrow h_{cow}\Big]_{\substack{\text{deepest}\\ \text{coil}}} = (68.95)(1.067)(1.838)^{0.101} = 78.2 \frac{W}{m^2K}.$$

Finally h_{do} is determined as per Section 11.7.6 through the Lewis relationship with a Lewis number value of 0.9 as recommended by Kusuda (1965):

$$h_{do} = \frac{h_{cow}}{(0.9)(c_{pm})} \rightarrow h_{do}\Big]_{\text{shallowest coil}} = \frac{26.11}{(0.9)(1022)} = 0.0284\ \frac{\text{kg}}{\text{sm}^2}\ .$$

$$\rightarrow h_{do}\Big]_{\text{deepest coil}} = \frac{78.23}{(0.9)(1022)} = 0.0851\ \frac{\text{kg}}{\text{sm}^2}\ .$$

All the members of the right hand side of the Tie Line Slope equation for the two non-empirical coils have then been determined.

$$\text{Tie Line Slope} = -\frac{h_i A_i}{0.9 h_{do} A_o}$$

substituting values determined above,

$$\text{Tie Line Slope}\Big]_{\substack{\text{shallowest coil}\\ \text{Run 1}}} = -\frac{1.28\times10^3(1)}{0.9(0.0284)(13.33)} = -3.8\ \frac{\text{kJ}}{\text{kgK}}$$

$$\text{Tie Line Slope}\Big]_{\substack{\text{deepest coil}\\ \text{Run 1}}} = -\frac{1.69\times10^3/1000}{0.9(0.0848)(13.33)} = -1.7\ \frac{\text{kJ}}{\text{kgK}}\ .$$

These values should be compared with the Tie Line Slope of the datum coil determined in Section 11.5, namely

$$\text{Tie Line Slope}\Big]_{\substack{\text{datum coil}\\ \text{Run 1}}} = -2.5\ \frac{\text{kJ}}{\text{kgK}}\ .$$

11.10 Construction of the Coil Condition Curves for the Two Non-Empirical Coils

The coil condition curves for the shallowest and deepest coils of this comparison were constructed and compared with the curve obtained for the datum coil. The construction lines have been retained. Together these curves form a "Family of Curves". From this analysis a new method of dehumidifier selection will be developed in Section 11.13.

Presented below are four figures:

Figure 11.8 Coil Condition Curve, Shallowest Coil Run 1,

Figure 11.9 Coil Condition Curve, Datum Coil Run 1,

Figure 11.10 Coil Condition Curve, Deepest Coil Run 1,

Figure 11.11 A Family of Curves, Composite of 11.8, 11.9, 11.10.

From manufacturer's data such as Table 11.1 taken from Aerofin (1950), it is possible to locate the points along the coil condition curve which represents the leaving condition from each row in depth of a coil as is indicated on Figure 11.11.

CONSTANTS FOR MULTIPLYING TOTAL LOAD TO DETERMINE THE APPROXIMATE LOAD PER ROW WITH ALL COILS OPERATING AT SAME SUCTION TEMPERATURE

Percent of Load For	Rows Deep, in Direction of Air Flow				
	2	3	4	5	6
1st Row ...	.566	.417	.346	.305	.276
2nd Row ...	.433	.319	.265	.234	.216
3rd Row ...	..	.264	.220	.194	.178
4th Row ...	..	..	.168	.148	.133
5th Row ...	..	..	..	.119	.107
6th Row ...	..	..	..	..	.090

TABLE 11.1 (REPRODUCED FROM AEROFIN (1950))

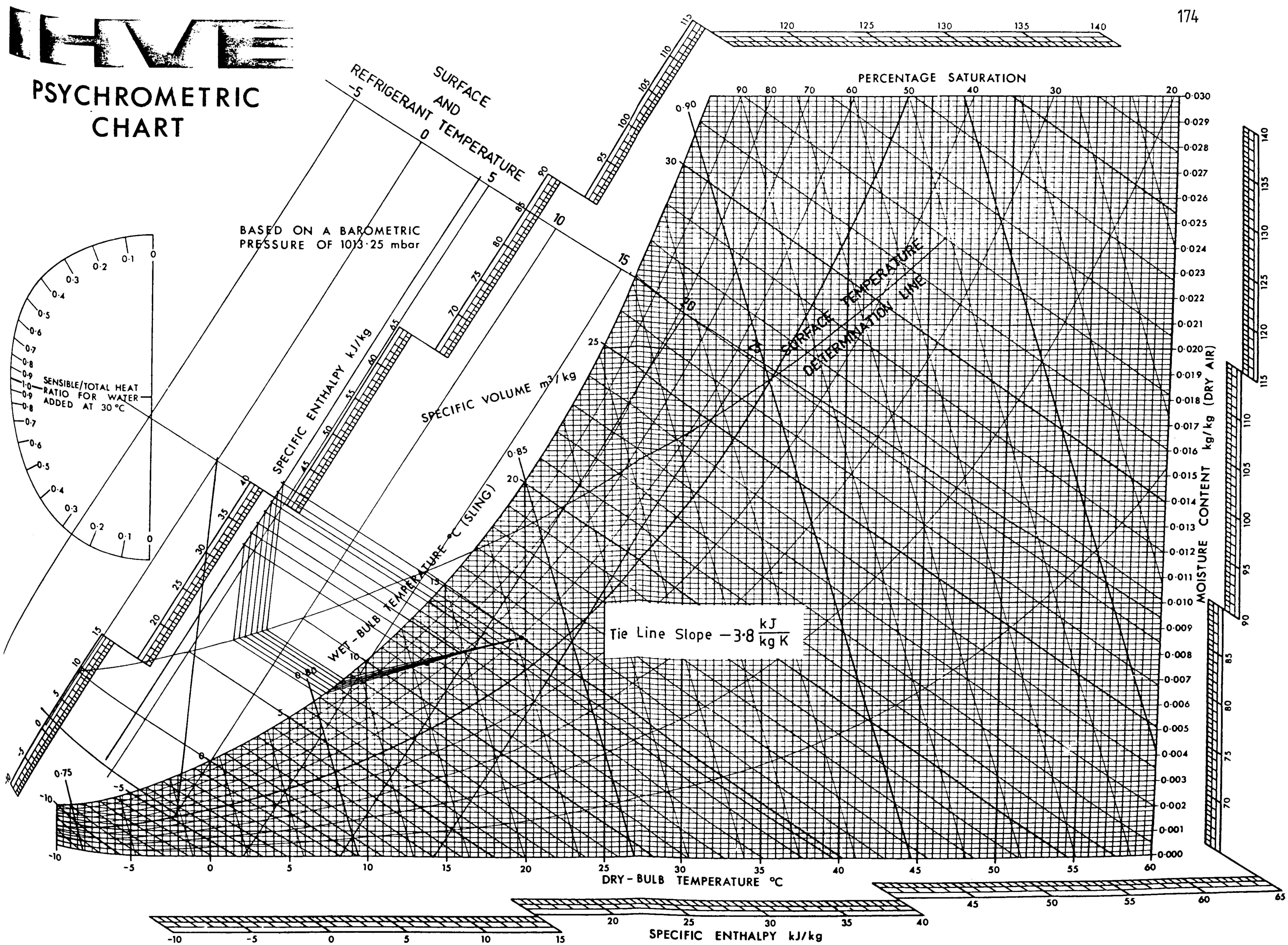
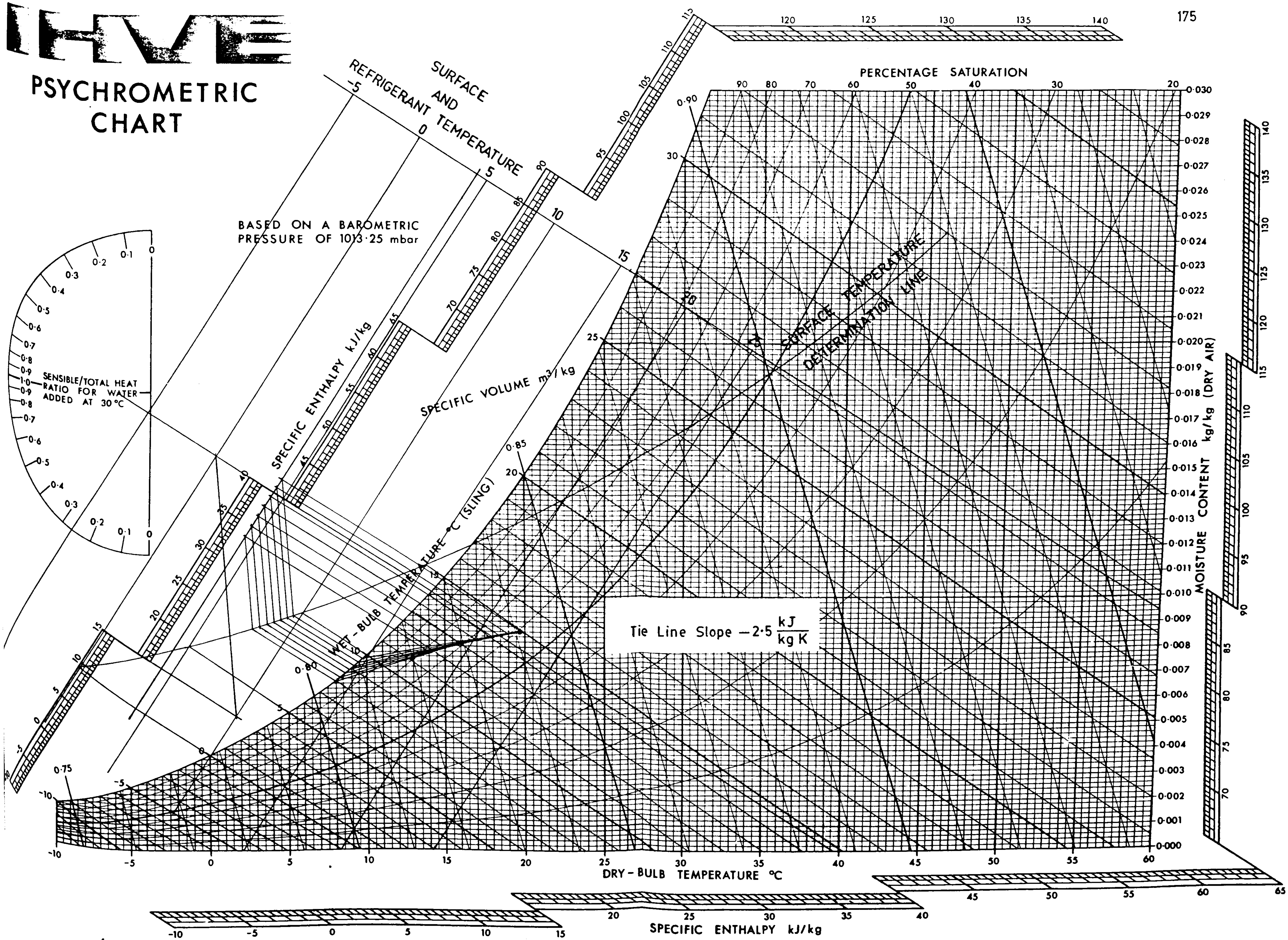


FIG 11·8 COIL CONDITION CURVE SHALLOWEST COIL, RUN 1



(REPEATED) FIG 11·9 COIL CONDITION CURVE DATUM COIL, RUN 1

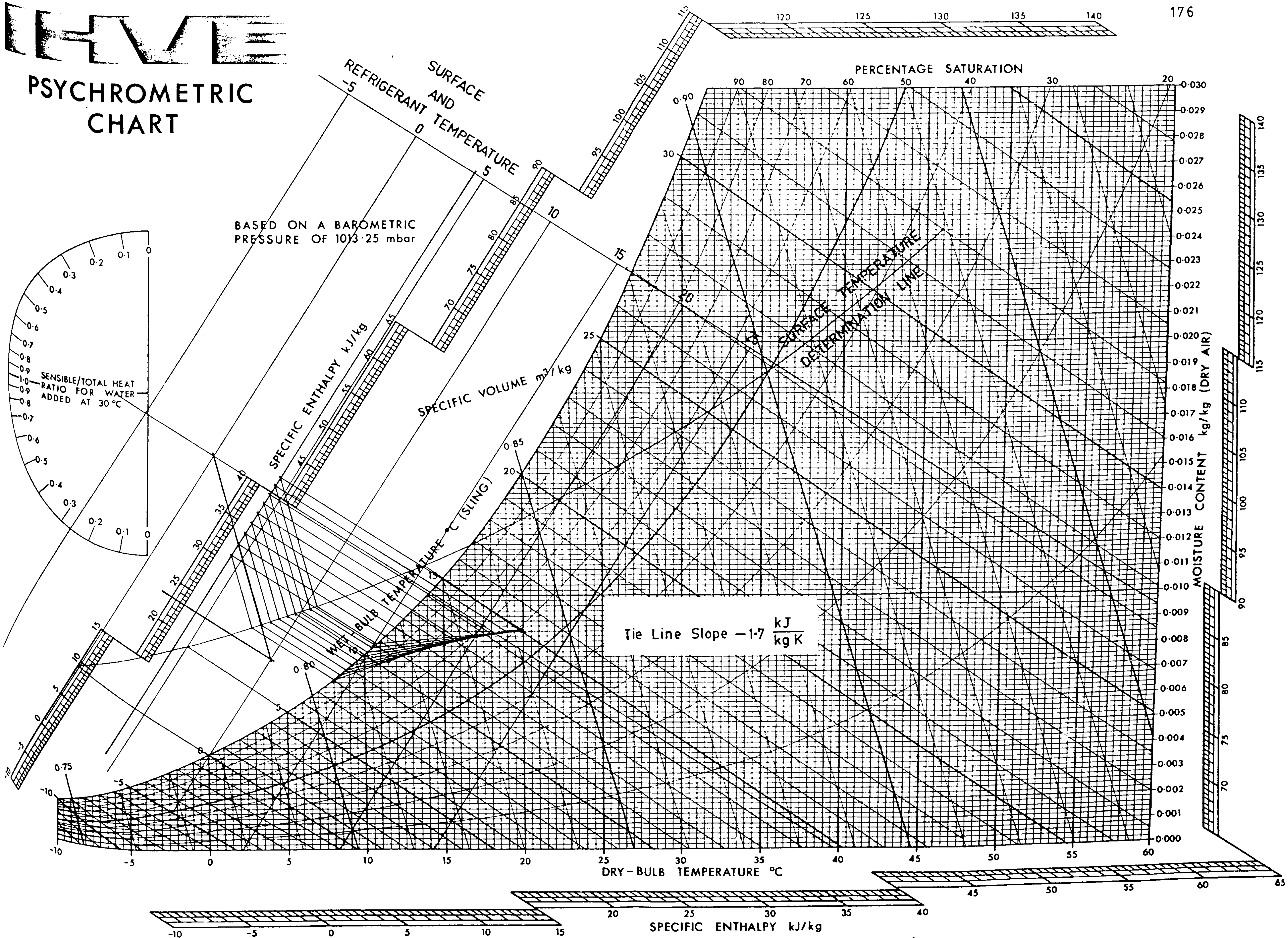


FIG 11·10 COIL CONDITION CURVE DEEPEST COIL, RUN 1

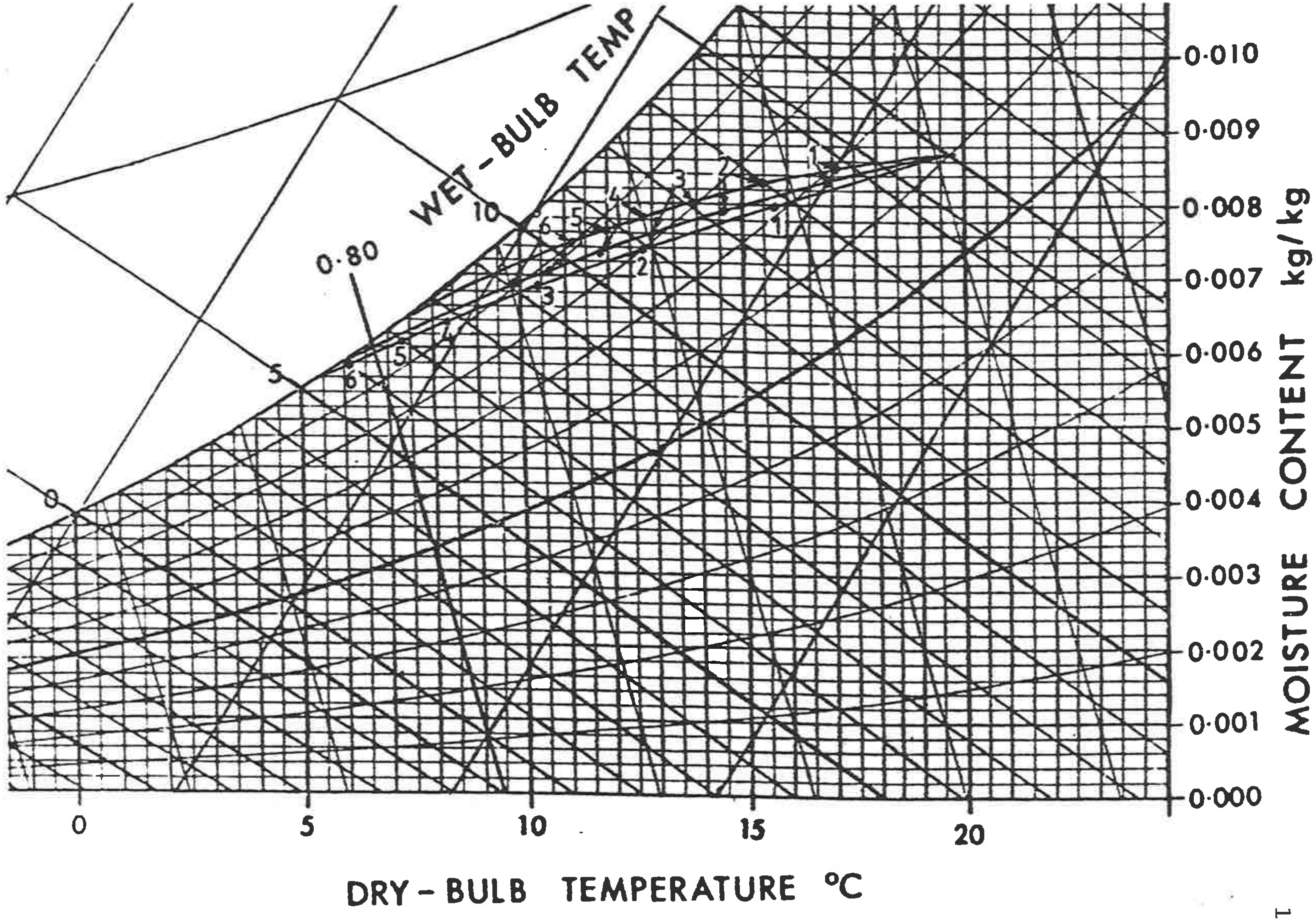


FIG 11·11 A FAMILY OF CURVES (COMPOSITE OF FIGS 11·8 11·9 11·10)

These curves reveal improved mass transfer to heat transfer at reduced air velocities. This occurrence is reflected in the decreasing Tie Line Slope. The value of the right hand side of the Tie Line Slope Equation is:

$$-(\text{Constant})(h_i/h_{do}) \quad \ldots(5)$$

Thus the improved mass transfer to heat transfer is related to the ratio of h_i to h_{do}.

Table 11.2 indicates for the condition of Run 1 the effect of decreasing the face velocity of the deepest coil in 2 steps, first by half to that of the datum coil and then the datum coil by half again to the shallowest coil. Associated with this decrease of face velocity, h_i decreased by only 13 per cent in step 1 and 13 per cent in step 2. On the other hand the value of h_{do} decreased by 42 per cent in step 1 and by 43 per cent in step 2. It is this relationship that is responsible for the significant reduction in Tie Line Slope and the related increase in ratio of simultaneous mass transfer to heat transfer with the reduction of face velocity.

	Face Velocity m/s	h_i W/m^2K	h_{do} kg/sm^2	$\frac{h_i}{h_{do}}$ $\frac{kWs}{kgK}$	Tie Line Slope $\frac{kJ}{kgK}$
Deepest Coil	1.84	1.69×10^3	0.0848	19.9	-1.7
Datum Coil	0.92	1.47×10^3	0.0494	29.8	-2.5
Shallowest Coil	0.46	1.28×10^3	0.0283	45.2	-3.7

TABLE 11.2 GREATER DECREASE IN h_{do} OVER h_i RELATED TO IMPROVED RATIO $\frac{h_i}{h_{do}}$ WITH DECREASE OF FACE VELOCITY

11.11 Confirmation of the Tie Line Slope Obtained for the Datum Coil With Log Mean Values

Following the procedure used for the datum coil in Section 11.6, the log mean air enthalpy for the shallowest coil is determined:

$$\Delta H_m = \frac{H_1 - H_2}{\ell n\left(\frac{H_1 - H_r}{H_2 - H_r}\right)}$$

$$\Delta H_m\Big]_{\substack{\text{shallowest}\\ \text{coil}}} = \frac{42.0-33.5}{\ell n\left(\frac{42.0-17.2}{33.5-17.2}\right)} = 20.2\ \frac{kJ}{kg}$$

$$H_{1m} = H_{sr} + \Delta H_m \rightarrow H_{1m} = 17.2 + 20.2 = 37.4\ \frac{kg}{kg}\ .$$

The log mean wet surface temperature is

$$\Delta H_{sm} = \frac{H_{s1} - H_{s2}}{\ell n\left(\frac{H_{s1} - H_{sr}}{H_{s2} - H_{sr}}\right)} \rightarrow \Delta H_{sm} = \frac{(26.0-23.0)}{\ell n\left(\frac{26.0-17.2}{23.0-17.2}\right)} = 7.1\ \frac{kJ}{kg}$$

and

$$H_{sm} = H_{sr} + \Delta H_{sm} \rightarrow H_{sm} = 17.2 + 7.1 = 24.3\ \frac{kJ}{kg}$$

which gives $t_{sm} = 7.7C$.

$$\text{Tie Line Slope}\Big]_{\substack{\text{shallowest}\\ \text{coil}}} = \frac{H_{sm} - H_{1m}}{t_{sm} - t_r} \rightarrow \frac{24.3-37.4}{7.7-\ 4.2} = -3.7\ \frac{kJ}{kg}$$

This agrees closely with the Tie Line Slope obtained from the analysis of Section 11.9.

$$= -3.8\ \frac{kJ}{kg}\ .$$

Following the same procedure for the deepest coil,

$$\Delta H_m\Big]_{\substack{\text{deepest}\\\text{coil}}} = \frac{42.0-30.2}{\ell n\left(\frac{42.0-17.2}{30.2-17.2}\right)} = 18.2\ \frac{kJ}{kg}$$

which gives

$$H_{1m} = H_{sr} + \Delta H_m \rightarrow H_{1m} = 17.2 + 18.2 = 35.4\ \frac{kJ}{kg}$$

also

$$\Delta H_{s,m} = \frac{H_{s1}-H_{s2}}{\ell n\left(\frac{H_{s1}-H_{sr}}{H_{s2}-H_{sr}}\right)} \rightarrow \Delta H_{sm} = \frac{31.1-24.2}{\ell n\left(\frac{31.1-17.2}{24.2-17.2}\right)} = 10\ \frac{kJ}{kg}$$

which gives

$$H_{sm} = H_{sr} + \Delta H_{sm} \rightarrow H_{sm} = 17.2 + 10.0 = 27.2\ \frac{kJ}{kg}$$

and t_{sm} = 9.0C. Thus

$$\text{Tie Line Slope}\Big]_{\substack{\text{deepest}\\\text{coil}}} = \frac{H_{sm}-H_{1m}}{t_{sm}-t_r} \rightarrow \text{Tie Line Slope} = \frac{27.3-35.43}{9\ -\ 4.2} = -1.7\ \frac{kJ}{kgK}$$

This agrees with the Tie Line Slope obtained from the analysis of Section 11.9.

$$= -1.7\ \frac{kJ}{kgK}\ .$$

11.12 Part Load Performance

The performance characteristics displayed in the comparative analysis of Section 11.3 were associated with design load characteristics for wide range climate simulation. Some of the conclusions were also stated to be applicable to air conditioning. Considering both areas, the question arises how best to design dehumidifier coils for part load conditions. This question was discussed in Sections 5 and 6 of this paper with respect to systems for climate simulation. In air conditioning practice one of the most common arrangements, when chilled water coils are employed, is to bypass chilled water around the coil as a means of maintaining the desired conditions in the air conditioned space during part load operation. See Figure 11.12.

This existing practice frequently goes counter to the change in air conditioning load characteristics during part load performance. During marginal weather the transmission sensible heat loads will reduce or actually become negative and cancel part of the internal sensible heat loads. However, the latent heat loads from people and infiltration will remain the same. The result is a reduced sensible heat ratio during part load conditions. (Carrier et al p.449, 1959b). A steeper coil condition curve is usually required, that is, one which has a LOWER TIE LINE SLOPE. As will be shown, for the Figure 11.12 arrangement, it becomes shallower. As a consequence the space conditions are either not maintained or a system may be employed using wasteful overcooling accompanied by wasteful reheating.

11.12.1 Conventional system performance during part load conditions illustrated

In this section by way of an illustrative example a comparative study will be made using the performance of the datum coil during Run 1 as representing a dehumidifier operating at full load conditions.

The objective will be to assess the change in performance of the dehumidifier from full load to part load operating conditions of 65 per cent.

Though an evaporator is investigated, Figure 11.12 represents the typical arrangement for chilled water coil application. (The principles that are developed here are applicable to both direct expansion and chilled water coils). This arrangement is in such common usage that it is shown here to highlight the serious problem that arises in conventional air conditioning system application.

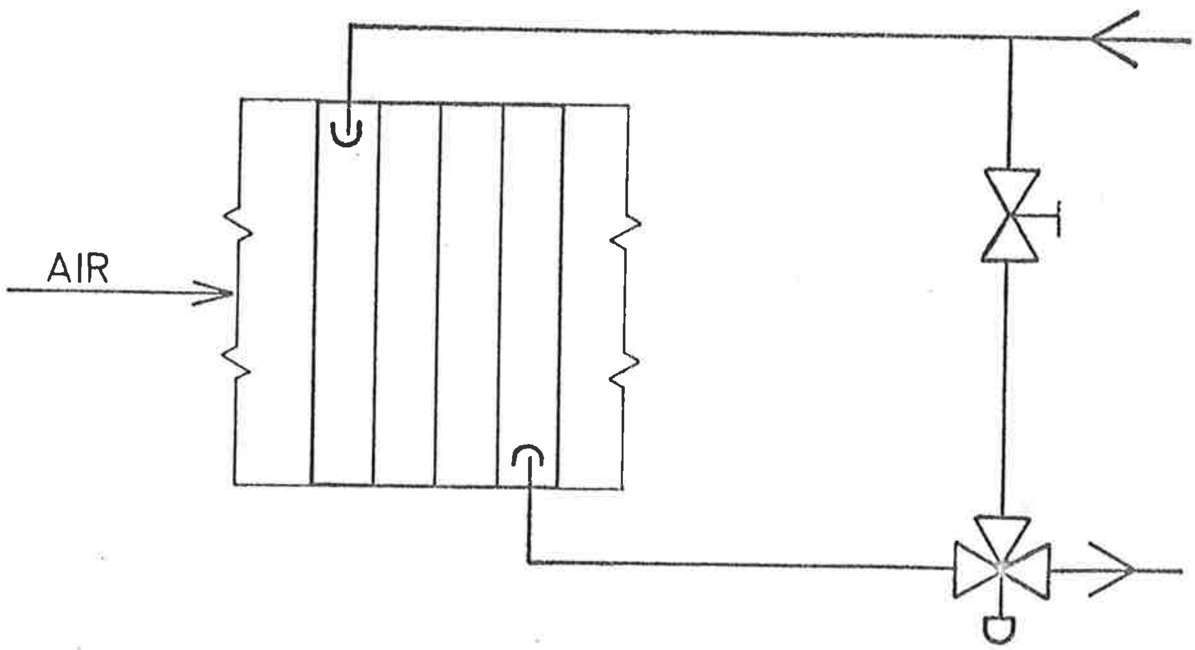


FIG 11·12 A CHILLED WATER COIL CONVENTIONALLY CIRCUITED

The known "full load" Tie Line Slope value for the datum coil Run 1 will be compared with the 65 per cent part load value.

The conditions of comparison with a part load operating period is to maintain in accord with conventional practice, the same mass flow rate of air as for full load. Under these conditions, the mass transfer coefficient for outside air, h_{do}, will remain the same. Part load operation will be obtained through bypassing refrigerant until its flow rate is compatible with constraints 7 to 10 inclusive of Section 11.3. (In order to obtain the same evaporator temperature as for full load, assume a smaller rated condensing unit during part load operation.) Thus the only variation that will take place to the Tie Line Slope equation, (4a) below, during part load operation will be a reduced value for h_i, the combined coefficient of heat transfer through water layer metal and refrigerant film. Thus

$$\text{Tie Line Slope} = -\frac{h_i A_i}{.9 h_{do} A_o} \quad \text{...(4a)}$$

The extent of this reduction can be approximately determined by the Pierre Equation, (ASHRAE Handbook 1977a),

$$\Delta h_{i(avg)} \approx \Delta h_{r(avg)} = \frac{0.0082 k_\ell}{d} \left[\left(\frac{\Delta G_r d}{\mu_\ell} \right)^2 \left(\frac{J \Delta x H_{fg}}{L} \right) \right]^{0.4} \quad \text{...(12)}$$

Similar to the use of the Pierre equation in Section 11.8, though it is stated in terms of Imperial units, it is not necessary to convert to metric units since in this problem ratios are being used. Here we are considering the ratio of $h_{i(avg)}$ part load to $h_{i(avg)}$ full load.

Since G_r is reduced to 0.65 of the full load value and is the only value in the Pierre equation which changes, $h_{i(\text{Part Load})}$ can be calculated as follows

$$h_{i(\text{Part Load})} \approx h_r\Big]_{\text{part load}} = \left[\left(\frac{G_{r(\text{part load})}}{G_{r(\text{full load})}}\right)^2\right]^{0.4} h_{i(\text{Full Load})}$$

$$h_{i(\text{Part Load})} \approx h_r\Big]_{\text{part load}} = [(0.65)^2]^{0.4}\ 1.47\times10^3 = 1.04\times10^3\ \frac{W}{m^2K}$$

$$\text{Tie Line Slope}\Big]_{\substack{\text{part load}\\(\dot{m}_a\,=\,\text{full load})}} = -\frac{h_{i(\text{Part Load})}A_i}{0.9\ h_{do}\ A_o}$$

$$\text{Tie Line Slope}\Big]_{\substack{\text{part load}\\(\dot{m}_a\,=\,\text{full load})}} = -\frac{(1.04)}{0.9(0.0494)(13.33)} = -1.8\ \frac{kJ}{kgK}.$$

This is to be compared with Tie Line Slope$_{(\text{Full Load})} = -2.5\ \frac{kJ}{kgK}$.

Thus during part load operation of about 65 per cent of full load the conventional method of controlling the coolant results in a Tie Line Slope that is increased by 28 per cent of the full load Tie Line Slope rather than - as is often required - being decreased.

11.12.2 An economical solution for offsetting low sensible heat ratio load patterns during part load operation

One of the main conclusions that can be drawn from this research project is that air stream velocity is an important factor in determining the slope and shape of the coil condition curve.

It follows from the results of this study that a steeper coil condition curve can be obtained for part load operating conditions if the air flow rate is controlled so that it is reduced in relation to prevailing part load operating condition.

The following example shows the effect of reducing the air flow rate with load reduction for the 65 per cent part load condition.

Again the Datum Tie Line Slope for Run 1, $-2.5\ \frac{kJ}{kgK}$, will be used.

From manufacturer's data for the coil in question it was determined that for the part load condition of 65 per cent, the air flow rate should be reduced to 50 per cent of full load value in order to retain the same evaporator temperature.

Determination of $h_{do(Part\ Load)}$

The change in air flow rate indicates a change in the value of the Stanton number due to a change in the value of G_m by 50 per cent.

$$G_m\Big]_{(part\ load)} = \frac{G_m\Big]_{(full\ load)}}{2} = 1.12\ \frac{kg}{(s)(m^2)}.$$

Also Reynolds number is reduced by 50 per cent. Hence

$$Re\Big]_{(part\ load)} = \frac{Re\Big]_{(full\ load)}}{2} = 342$$

$$f(Re)\Big]_{(part\ load)} = St\ Pr^{2/3} = 0.0185$$

and therefore ($\dot{m}_a$ reduced)

$$h_{cod}\Big]_{\substack{(part\ load)\\(\dot{m}_a\ reduced)}} = \frac{f(Re)G_m c_{pm}}{(Pr)^{2/3}} \rightarrow h_{cod}\Big]_{\substack{(part\ load)\\(\dot{m}_a\ reduced)}}$$

$$= \frac{(0.0185)(1.12)(1.022)(10^3)}{0.8} = 26.47\ \frac{W}{m^2K}$$

$$h_{cow}\Big]_{\substack{\text{(part load)}\\ (\dot{m}_a \text{ reduced})}} = h_{cod}\,(1.067)(V)^{0.101} = (26.47)(1.067)(0.459)^{0.101}$$

$$= 26.12\ \frac{W}{m^2K}$$

$$h_{do}\Big]_{\substack{\text{(part load)}\\ (\dot{m}_a \text{ reduced})}} = .0284\ \frac{kg}{sm^2}$$

(N.B. The solution for h_{cod}, h_{cow} and h_{do} follow the same procedure as for Section 11.9).

The value of $h_{i(\text{Part Load})}$ was determined in Section 11.12.1 for a 65 per cent reduction of the refrigerant flow. This also applies here and therefore,

$$h_i\Big]_{\text{(part load)}} = 1.04\times10^3\ \frac{W}{m^2K}\ .$$

Consequently the Tie Line Slope for 65 per cent load conditions where the refrigerant flow is reduced by 65 per cent and the air flow is reduced by 50 per cent becomes,

$$\text{Tie Line Slope}\Big]_{\substack{\text{(part load)}\\ (\dot{m}_a \text{ reduced})}} = -\frac{h_iA_i}{0.9\ h_{do}A_o} \qquad \ldots(5)$$

$$\text{Tie Line Slope}\Big]_{\substack{\text{(part load)}\\ (\dot{m}_a \text{ reduced})}} = -\frac{(1.04)}{(0.0284)(13.33)} = -2.8\ \frac{kJ}{kgK}$$

With conventional practice of maintaining the same air flow rate during part load as during full load the Tie Line Slope during part load operating conditions increased for this example from

$$-2.5 \quad \text{to} \quad -1.8 \ \frac{\text{kJ}}{\text{kgK}} \ ,$$

whereas with the reduction of air flow rate, the Tie Line Slope for the same dehumidifier decreased from

$$-2.5 \quad \text{to} \quad -2.8 \ \frac{\text{kJ}}{\text{kgK}} \ .$$

The decrease in Tie Line Slope means deeper dehumidification.

It can thus be seen that by applying some of the principles developed here it is possible to achieve a higher standard of desired operating setting for air conditioning systems during part load operation and to avoid wasteful overcooling and reheating.

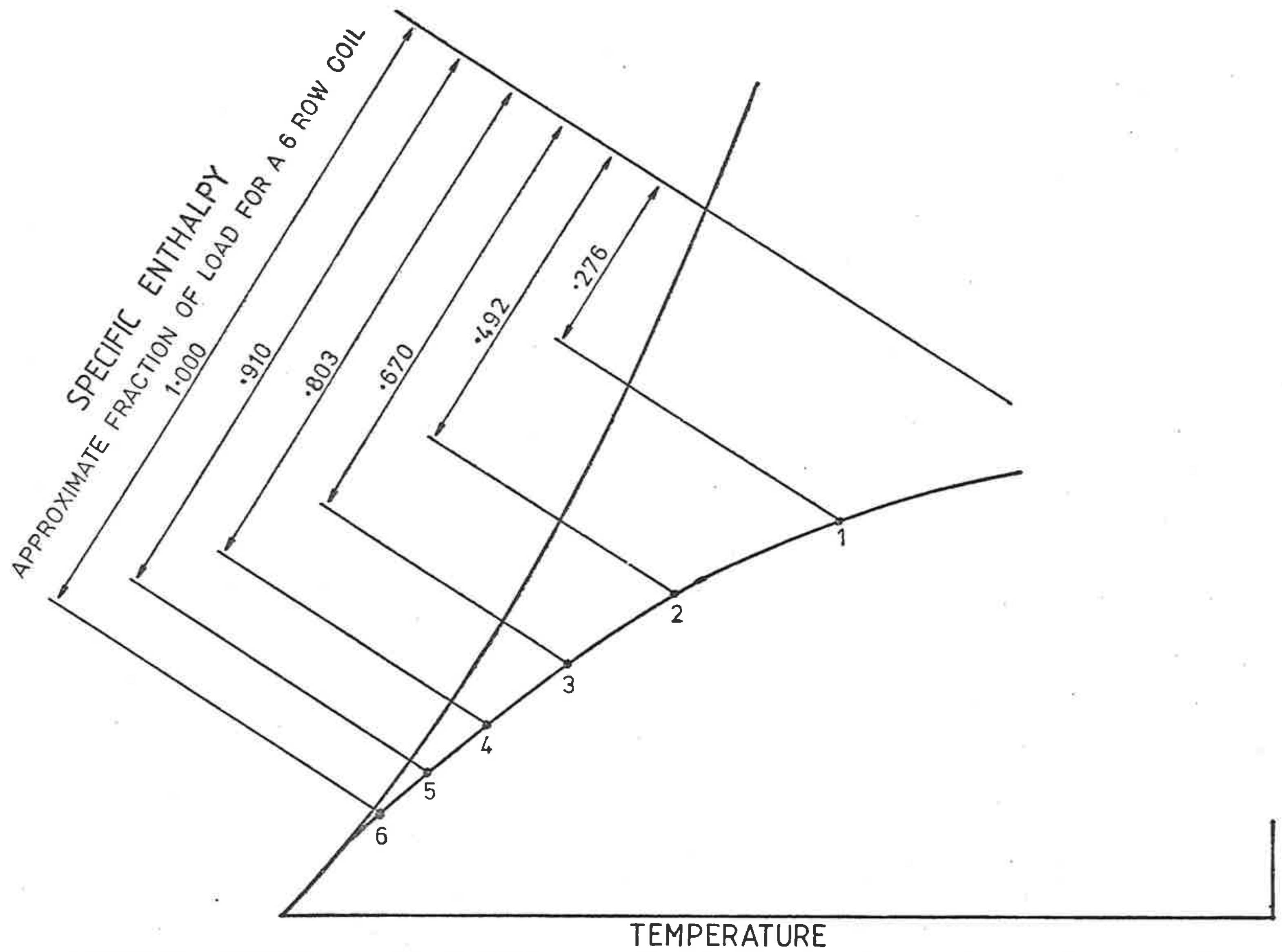
11.13 A New Method of Dehumidifier Selection

A heat and mass transfer analysis has revealed that one of the main factors affecting the nature of the coil condition curve is that of the face velocity of the air stream.

By means of a comparative analysis three dehumidifier coils and their coil condition curves have been determined. Each of the three coil condition curves have been developed fully from some arbitrary entering condition to the final surface temperature of the most downstream increment of wetted surface area in the direction of air flow. Together they form a family of curves.

It is possible as in Figure 10.1, repeated below, to indicate the condition of the air stream in its passage through the coil by locating along each curve the number of rows in depth that apply. (See also Figure 11.11 above).

The three curves have been defined through the twelve constraints that form the basis of comparison. See Section 11.3. The



(REPEATED) FIG 10·1 AIR SIDE COIL PERFORMANCE resulting Tie Line Slopes, though consistent with:

the same primary surface area of constraint number 1

the same secondary fin surface area of constraint number 2

the equal heat exchange surface area of constraint number 3

the doubling and halving of face areas of constraints number 5 and 6,

represent conditions all along the coil condition curves and therefore are independent of these limitations.

It is not the shallowness or the depth of a coil per se which determines the coil condition curve since a curve will represent a particular Tie Line Slope from a single row in depth to an infinity of rows. However in a comparison based on the constraints of Section 11.3, when comparing coils on the basis of the same mass flow rate, the shallower coil with the lower velocity will have the lower Tie Line Slope. The restrictiveness of the comparison was due to the non-empirical means by which the Tie Line Slope of the shallowest and the deepest coil had to be determined. However, once determined, the above enumerated restrictions no longer had to apply in developing the full coil condition curves for these non-empirical dehumidifier coils.

11.13.1 Common properties of the family of coil condition curves

The common properties that identify the family of curves presented in Figure 11.8 are as follows:

1. *All curves use the identical type of extended surface heat exchangers having the same general pattern, tube spacing, tube arrangement, fins per unit length of tube, secondary fin to primary tube surface area, tube diameter, wall thickness and material.*

2. *All curves have the same intensive property of air, specific enthalpy, dry bulb-temperature and humidity ratio at inlet to the dehumidifier.*
3. *All have the same refrigerant condenser pressure.*
4. *All have the same refrigerant evaporator pressure.*
5. *All have the same dryness fraction at inlet to the evaporator.*
6. *All have the same superheat condition leaving the evaporator.*
7. *All evaporator surfaces are completely wetted.*
8. *All have a face area which is adjusted to retain the same mass flow of air.*

11.13.2 Distinguishing properties within each family of coil condition curves

1. *The curves differ from each other due to the face velocity and the related Reynolds number assignable to the dehumidifier complex.*
2. *The highest face velocity coil has the maximum curvature and the shallowest slope. The maximum face velocity is limited by that velocity above which dry performance commences.*
3. *The lowest face velocity coil approaches a straight line and has the maximum slope.*
4. *Though for the sake of simplicity only three curves are shown, any number of curves can be developed. The curvature and slope of all other curves considered would be located between the maximum and minimum face velocity curves, progressively having decreasing curvature and increasing slope as the face velocity is decreased.*

11.13.3 Significance of the family of curves

The variables in dehumidifier coil selection are very numerous. Apart from the major criterion associated with the slope of the coil condition curve, i.e. the face velocity of the air stream, there are:

the temperature of the coolant,

the depth of coil,

the geometry of the coil including the ratio of outside to inside surface,

the operating set point,

the dew point temperature of the operating set point and

the temperature difference across the dehumidifier coil.

With the exception of the geometry of the coil, which will be consistent for every "Family of Curves Chart", the parameters listed directly above can be depicted on a psychrometric chart. As such, given the charts covering various coil designs for a particular inlet condition, a designer can select that coil which best fits the problem. On identifying the required coil condition curve the designer also will have available

the coil size,

the face area and

the rows of depth.

From the same chart, if marked with the Tie Line Slope construction lines, the log mean specific enthalpy and the log mean surface temperature can be calculated.

It is tempting to speculate that improved standards could be introduced in those fields which use dehumidifiers such as climate simulation and air conditioning, if manufacturers accept that the "Family of Curve Charts" for their products should be determined by a certified or independent testing organization. An appropriate body in Australia to certify testing organizations and laboratories would be the National Association of Testing Authorities (NATA). The designer could have a broader choice in finding the best dehumidifier and would not be restricted to the geometric pattern of one manufacturer specializing in one type of design. It may ultimately be desirable to store and regularly update a large number of Family of Coil Condition Curve Charts on an interactive computer system, preferably written within a computer network, for ready access by designers.

Alternatively, the number of charts required could be reduced to manageable proportions by assigning correction factors to allow for variations in fin spacing, condensing temperatures etc..

11.14 Energy Saving Demonstrated

The emphasis of Sections 10 and 11 of this thesis has been placed on the operating costs. It is therefore fitting to examine the significance of the findings in this context. This will be done by presenting three problems which will be associated with Run 1 at the Waite Institute Phytotron Unit.

The highest and the lowest face velocity coils of Run 1 will be examined from the point of view of evaluating the effect of the variation of face velocity, through the use of a Family of Coil Condition Curve Chart, in selecting a dehumidifier.

For this purpose three design problems will be solved.

In each problem all eight common properties of Section 11.13.1 are satisfied by the dehumidifier - thus one single Family of Curves Chart will suffice. The common entering condition and the three different desired leaving conditions are tabulated below.

Entering condition:	specific enthalpy	42.0 kJ/kg
	dry bulb temperature	19.9C
	humidity ratio	0.0086

Desired leaving conditions:	Problem 1	Problem 2	Problem 3
Specific enthalpy	39.1 kJ/kg	33.8 kJ/kg	25.5 kJ/kg
Dry bulb temperature	17.9C	14.3C	8.8C
Humidity ratio	0.0083	0.0076	0.0065

AT LOWEST FACE VELOCITY   0.46 m/s (90.5 feet/minute)

AT HIGHEST FACE VELOCITY   1.84 m/s (363.0 feet/minute)

It is to be noted that the *highest* face velocity is *below* the minimum conventionally used in air conditioning practice and that the *lowest face velocity* is at *one fourth of this value.*

Both coils have the same mass flow of air.

In each problem the coil condition curve of the low face velocity coil passes directly through the desired leaving condition.

In each case the air stream of the higher face velocity coil must be overcooled in order to achieve sufficient dehumidification.

In each case the air stream of the higher face velocity coil will be reheated to reach the design condition.

The problem is to determine the per cent of energy wasted by using the higher face velocity coil rather than the low face velocity coil.

Figure 11.13 represents 2 members of the family of curves chart of figure 11.11 indicating all the values used in the solution of the 3 problems worked below.

Assume that the coefficient of performance of the refrigeration cycle is 3.0.

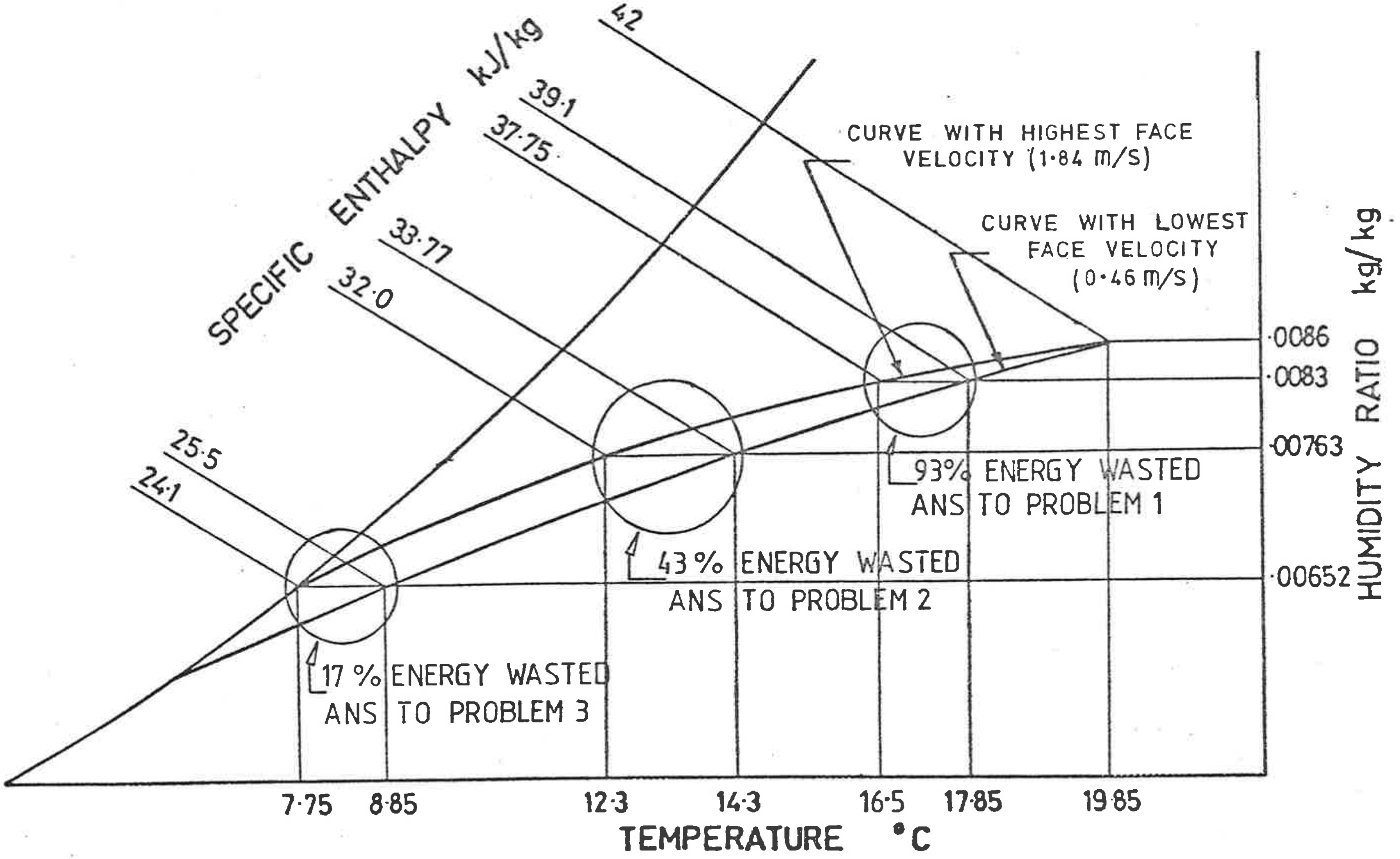


Fig. 11·13 3 COMPARITIVE PROBLEMS DEMONSTRATING EFFECT OF FACE VELOCITY ON OPERATING COSTS.

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11.14.1 Solution to three problems

With reference to Figure 11.13, each problem considers the ratio of the specific enthalpy change, $\Delta H_{(\text{excess cooling})}$, (enthalpy change associated with the excess sensible cooling that was required in order to sufficiently offset the latent heat loads) to $\Delta H_{(\text{necessary cooling})}$, (enthalpy change required if the dehumidifier is ideally selected to offset exactly both the sensible and latent heat loads). In each problem a multiplier, 2, appears in the calculations in order to represent the double penalty due to both overcooling and reheating. *

$$\text{Per Cent of Energy Wasted} = \frac{\Delta H_{(\text{excess cooling})}}{\Delta H_{(\text{necessary cooling})}} \times 2$$

Problem 1 $$= \frac{(39.1-37.8)2}{(42.0-39.1)} = 93 \text{ per cent}$$

Problem 2 $$= \frac{(33.8-32.0)2}{(42.0-33.8)} = 43 \text{ per cent}$$

Problem 3 $$= \frac{(25.5-24.1)2}{(42.0-25.5)} = 17 \text{ per cent}$$

---

*Though the coefficient of performance of the refrigeration cycle is 3, a double penalty only has been considered on the assumption that the refrigeration compressor is operated from an electrical source whereas the reheater is operated directly from a thermal energy source. If the reheater is also operated from an electrical source then the multiplier of 2 in the 3 problems should be changed to 4.

11.14.2 Comments on problems

Problem 1 is a type encountered in climate simulation where only 2C is the permissible temperature gradient across the conditioned space as for example in the case of a phytotron unit. Problem 2 involves a medium temperature gradient of about 5.5C. Problem 3 represents a typical conventional air conditioning application with 11C temperature difference across the coil and the leaving condition near saturation.

These problems highlight the effect of the qualitatively different face velocities that are recommended by the findings of this thesis as compared to existing practice.

Run 1 represents a common entering condition encountered in the field of air conditioning, yet as can be seen from the results, very substantial energy savings in operating costs would be realized for low sensible heat ratio conditions if the dehumidifier were selected to use an unconventional low face velocity, such as the 0.5 metres per second used in these problems above.

It is interesting to note that the highest and lowest velocity coil condition curves of Run 1 do not appear to be very drastically different from each other. See Figure 11.11. However the large energy savings are based on this small difference. This difference is based on the assumption that the highest face velocity coil of a "Family of Curves" is fully wetted. Actually test data described in Section 12 indicate that the shallowest coil condition curve of a "Family of Curves" is limited by a maximum face velocity which is frequently below conventional velocities. Coil condition curves with higher face velocities than this are partially dry and therefore do not belong to the same "Family of Curves". For these curves there is a marked separation from the curves having fully wetted surfaces. Therefore the energy savings are frequently greater than indicated by the solution to the 3 problems above, particularly in climate simulation with its low temperature gradient requirement.

11.15 Experimental Confirmation of the Theoretical Performance Characteristics of Dehumidifiers Developed in this Section

The non-empirical curves developed for the shallowest and deepest dehumidifiers using the comparative analysis of Section 11.3 have been empirically confirmed in a research project using the new Heat and Mass Transfer Laboratory in the Department of Mechanical Engineering in the University of Adelaide which was commenced during the final months of the preparation of this thesis. This confirmation is described in Section 12.

11.16 Conclusions

1. Air stream velocity and the related Reynolds number (assignable to the dehumidifier complex) have been found to be the major operative factors for a system of dehumidifier selection in air conditioning and climate simulation applications.
2. As the velocity of an air stream over a dehumidifier is reduced the slope of the coil condition curve becomes steeper.
3. As the velocity of an air stream over a dehumidifier is reduced the curvature of the coil condition curve reduces towards that of a straight line.
4. As the velocity of an air stream over a dehumidifier is reduced, the number of rows of depth required to reach near saturation conditions is reduced (when compared on the basis of Section 11.3).

5. The straight line characteristic of coil condition curves assumed in industrial design methods as described in Section 10, does not hold for the range of air velocities employed in air conditioning applications, [namely 2 metres per second (400 feet per minute) to 3.5 metres per second (700 feet per minute) approximately].

6. As the velocity of an air stream is increased over fully wetted dehumidifier surfaces, the coil condition curves become shallower and a point is reached which marks the maximum face velocity for a family of curves. This maximum velocity has been found to frequently occur below the minimum face velocity used in conventional air conditioning practice. (See Section 12). Above this velocity the coil is no longer fully wetted. When this occurs the first increment of depth in the direction of air flow has zero slope. This characteristic increases the energy penalty that has been described for conditions of fully wetted surfaces.

7. The conventional design approach used in air conditioning and climate simulation can result in large energy penalties and failure to attain desired conditions for full load and/ or part load operation particularly when low sensible heat ratios prevail.

8. The conventional design approach towards spatial arrangements particularly where low sensible heat ratios prevail, must be re-examined in the light of potential energy savings due to reduced cooling and reheating, reduced fan power, reduced size of refrigeration equipment and cooling tower and their reduced weights and costs.

9. From an examination of part load conditions frequently present in conventional air conditioning applications there is a strong case indicating energy savings and improved performance if the air flow rates are controlled to reduce with decrease of the loads.

10. A major criterion in dehumidifier coil selection is to find the extended surface which has a coil condition curve that best fits the load ratio line of the problem. Maximum energy conservation is then obtained.

11.17 Recommendations

Resulting from the findings of this thesis, a new system of dehumidifier selection has been described that is not consistent with existing air conditioning practice. Though the system would be more rigorous in adhering to the actual heat and mass transfer performance and in eliminating many of the assumptions and approximations used at present, it would offer a simpler method for determining the best dehumidifier, in terms of overall system performance. However the use of this system entails considerable work. Initially it would be necessary to establish agreed standards of testing involving manufacturers, standards associations and engineering societies. Approximations used by the commercial community (Section 10) should be reassessed. This implies a vast program. In addition this program would have to consider the effect of the dehumidifier on the total system of climate simulation and/or air conditioning. Space is also a valuable factor that must be assessed in the final selection process. The consulting engineer is therefore involved. Usually the consulting engineer does not include the selection of the dehumidifier as an important part in the design of a system, leaving the choice to the manufacturer to whom he specifies only the functional requirements.

Considerable work must be done to realize the large energy savings and improved air conditioning performance that would be derived from the implementation of a system of selection using the "Family of Curves Charts" or equivalent.

The entire philosophy of coil selection with emphasis on increased surface density for gas-liquid heat exchangers should be reassessed when the coil in question is a dehumidifier. The increase of fin to primary tube surface may result in optimizing heat transfer but will reduce mass transfer in relation to heat transfer.

The basic relationships leading from Reynolds number to the $(St)(Pr)^{2/3}$ value on through to the Lewis number relationship have revealed that there is a decreased mass transfer to heat transfer ratio with increase of face velocity. As examined in Section 11.10 this is indicated by the value of h_i changing at a lesser rate than h_{do} with change of face velocity. As a consequence an increased Tie Line Slope value (or negatively decreased Tie Line Slope value), is obtained. An increased Tie Line Slope reveals a decrease in the ratio of mass transfer to heat transfer. In dehumidifier applications it is important to recognize that though increased air flow velocity will improve heat transfer it will reduce the ratio of mass transfer to heat transfer.

12. EXPERIMENTAL CONFIRMATION OF THE CONCLUSIONS - DEVELOPED IN SECTIONS 10 AND 11

12.1 Introduction

In Sections 11.6 and 11.11 the Tie Line Slope compatibility with log mean values was demonstrated and very close agreement was obtained. Nevertheless it was considered desirable to establish empirical verification that the type of non-empirical curves developed for the shallowest and the deepest coils of the comparative analysis in Section 11.3 was correct. The first projects carried out on the heat and mass transfer research system in the Department of Mechanical Engineering were to confirm the findings of Sections 10 and 11.

12.2 The Research Systems

The datum empirical coil data of Section 11, referred to as Run 1, was gathered in 1977 at the Waite Institute Phytotron Unit which has since been dismantled due to severe corrosion caused by use of untreated Adelaide water. Furthermore, though the system was excellent in its performance characteristics (see Figure 7.1 and Appendix I), it did not have some of the sophisticated design, equipment and instrumentation necessary for a research system. It was therefore decided to build a new teaching and research system within the Department of Mechanical Engineering and in February 1979 this new system was completed.

12.2.1 Air flow characteristics

The air flow pattern, its control and measurement, are important to the programs planned for this teaching and research system. As indicated in Section 7.3.1 considerable attention was given to this aspect in the design of the system.

12.2.2 The requirement to maintain six properties constant

In order to verify the findings of Sections 10 and 11 it was necessary to maintain the following properties constant while testing the dehumidifier performance for each member of a family of curves:

1. entering dry bulb temperature
2. entering wet bulb temperature
3. condensing pressure
4. evaporator pressure
5. dryness fraction of the refrigerant entering evaporator
6. superheat leaving evaporator.

(Although the Waite Institute Phytotron Unit was capable of maintaining to very close tolerances a desired temperature and humidity upstream of the direct expansion coil, it was not able simultaneously to repeat and hold constant all of the above common properties for each different face velocity).

12.2.3 The mechanism to maintain constancy of the six properties

The new research system has the capacity to hold the six properties enumerated above constant within close limits. Control is obtained in the following manner:

1. Entering dry bulb temperature is maintained constant for any desired set-point dry bulb temperature over a climatic range by means of a proportional controller as in the basic climate simulator design described in Section 4.2.3.
2. Entering wet bulb temperature is maintained constant indirectly through control of a cavity dew point temperature, for any desired wet bulb temperature over a climatic range,

by means of a proportional controller also as in the basic climate simulator design described in Section 4.2.3.

3. Condensing pressure is maintained constant for any selected operating conditions over a climatic range by means of a proportional controller which automatically bypasses the condensing water around the condenser to obtain the desired operating setting, Figure 7.8.

4. Evaporator pressure is maintained constant for any selected pair of dry bulb and wet bulb temperatures over a climatic range, for the required air flow rate, by means of a number of manual adjustments available including

   (a) manual variable speed adjustment of compressor rpm, Figure 7.9;

   (b) condensing water inlet temperature regulation by means of manual valves which control the flow of condensing water supplying and bypassing the condenser and also the air flow at the cooling tower fan discharge, Figure 7.8.

5. Dryness fraction of the refrigerant entering evaporator is set manually by adjusting a vapour bypass around the liquid suction heat exchanger, Figure 7.8, until the desired subcooled temperature of the refrigerant entering the thermostatic expansion valve is obtained. The subcooled temperature is monitored by print out point 1 on the left hand twenty-four point recorder chart, Figure 7.9.

6. Superheat leaving evaporator is set by adjusting the spring pressure within the thermostatic expansion valve. The superheat temperature is indirectly monitored by print out point 3 on the left hand twenty-four point recorder chart, Figure 7.9.

It is essential that *all* these properties remain constant in the generation of the 'family of curves'. Different superheat values or different dryness fractions for different members of a 'family of curves' would fail to give a true evaluation of the relative performances at the differing air velocities. Such a comparison would be equivalent to comparing evaporators of *differing* sizes since the heat and mass transfer performance of the two phase portion of each evaporator would not be the same in all cases. A higher dryness fraction combined with a higher superheat value would be equivalent in effect to a much smaller evaporator. Different evaporator pressures would indicate different enthalpy potentials since the refrigerant temperatures would be different. Similarly differences in the air entry conditions would change the enthalpy potential. Different head pressures would result in different volumes of new charge after the gases in the clearance volume of the compressor had expanded. Thus *all* of these properties must be kept constant for each member of a 'family of curves'.

The Department of Mechanical Engineering research system is an application of the unified approach in a very sophisticated form. The requirements to verify the findings of Sections 10 and 11 go far beyond the requirements of a system of climate simulation which is only required to deliver a desired combination of dry bulb temperature and humidity over a specified range.

12.2.4 Wide range of refrigeration capacities and other system requirements

In addition to maintaining the six properties constant for the widely differing velocities of a 'family of curves', in order to verify the findings of Sections 10 and 11 the research system must accommodate the wide refrigeration capacity range that will occur when a single evaporator is subjected to the variations in air velocity,

entering specific enthalpy values and refrigerant temperatures.
The following are required:

(a) A manually controlled variable speed motor for the air system fan.

(b) A manually controlled variable speed motor for the refrigeration compressor.

(c) A cooling tower with built-in flexibility. A centrifugal backward curved fan with a manual damper at the fan discharge was installed. At some of the very low face velocity settings it is necessary to completely shut down the cooling tower fan during low wet bulb temperature weather.

(d) A drain pan with provision for measuring condensed water flow rate from the dehumidifier.

(e) Insulated air duct and refrigeration piping.

(f) A sealed closed air cycle capable of reaching steady state conditions.

12.3 Problems to be Overcome to Confirm Conclusions of Sections 10 and 11

It was not practical to build a research system that would adhere to the constraints of Section 11.3.

12.3.1 The research system employs one dehumidifier

The Department of Mechanical Engineering System employs a fixed dehumidifier. It is therefore not possible to satisfy the requirement in the experiment to halve or double the depth of the dehumidifier as the face velocity of the coil is halved or doubled. However this problem is not serious. The Tie Line Slope characteristic is independent of the depth of the coil. (See Section 11.13). It is therefore possible to obtain data from the fixed depth research coil and then, by means of the manufacturer's data, such as Table 11.1 of Section 11.10, obtain the outlet condition for any desired depth on the coil condition curve, Figure 10.1. One difficulty arising from the inability to vary the dehumidifier configuration is that when the face velocity is halved, and the depth of the coil is not halved (Section 11.3), an outlet condition very close to the saturation line of the psychrometric chart results. This is a point where the high and low face velocity coil condition curves converge, as indicated on Figure 11.11, and therefore a reduced accuracy in the construction of the coil condition curves by way of the Tie Line Slope method results.

12.3.2 Limited range

The range of air velocities used to construct non-empirically the shallowest and deepest coils, as in the comparative analysis of Section 11.3, was 4 to 1. The Department of Mechanical Engineering System is limited to about 2 to 1 by limitations in the speed of the air supply fan and the speed of the open type compressor. There is also a maximum limit of 1000 kPa gauge imposed by the refrigerant head pressure controller.

12.4 Extension of the Research Project to Include Part Dry and Dry Coil Performance

It would be wrong to confine the findings of Section 11 to completely wetted coils only. Design requirements include coils with performances over a complete range of sensible heat ratios. Although Section 11, in establishing clear relationships due to change in face velocity only, confined itself to entirely wetted surfaces, it was found when the research system operated over the narrower range of face velocities of 2 to 1, the coil would frequently pass from completely wetted to partly wetted performance. At these conditions the differences between the slopes of the high and low velocity coil condition curves increased over the findings of Section 11 for completely wetted surfaces only. Indeed, as will be demonstrated by the results of Runs 3 and 7, wide divergence of the coil condition curves from completely dry to a partially wetted condition are obtained. This highlights the point made in the first paragraph of Section 3.2.1 with regard to the range of dehumidification for climate simulation. Obviously the performance of Runs 3 and 7 indicate that under higher face velocity, the conditions of these runs, (which is low by conventional standards), the research coil has no capacity to dehumidify. The range over which it can control humidity is limited to the higher humidity ratios and lower dry bulb temperatures.

Thus consideration of dry and part dry performance in this section further supports the importance of the findings of Section 11, conclusion 6 of Section 11.16 regarding part dry performance.

12.5 The Confirming Tests

The confirming tests cover two independent research projects. However the entry conditions and constraints were so selected as to closely inter-relate the two projects so as to link the verification to the basic theory. Both projects study the performance of six pairs of different entry conditions to the dehumidifier. In Project I one member of each pair has twice the velocity of the other.

12.5.1 Project I: to study the effect of different air velocities across a dehumidifier coil

The following properties will be kept constant for each member of the six pairs of runs considered:

entry dry bulb temperature

entry wet bulb temperature

evaporator pressure

condenser pressure

entry dryness fraction of refrigerant to the evaporator

superheat condition of refrigerant leaving the evaporator.

. The face velocity of one member of the pair will be twice that of the other member of the pair.

The performance data of the six pairs are listed in Table 12.1. The entry and leaving conditions are connected by a straight line on the psychrometric chart of Figure 12.1. The black lines represent the higher velocity runs and the red lines represent the lower velocity runs.*

The actual path of the coil condition curve including Tie Line Slope construction lines can be observed in psychrometric charts with figure numbers as listed in Table 12.2.

TABLE 12.2 PROJECT I, COIL CONDITION CURVES FOR RUNS 2 TO 7

PAIR NUMBER	RUNS NUMBERS	FIGURE NUMBERS **	
		HIGH VELOCITY MEMBER	LOW VELOCITY MEMBER
1	2	12.2 HV	12.2 LV
2	3	(not required, dry performance (See Fig. 12.1)	12.3 LV
3	4	12.4 HV	12.4 LV
4	5	12.5 HV	12.5 LV
5	6	12.6 HV	12.6 LV
6	7	(not required, dry performance (See Fig. 12.1)	12.7 LV

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* A section of the recorder charts of the high and low velocity members of Run 7 are shown in Figures 7.12A to D.

** All figures listed in Table 12.2 appear at the end of Section 12.

TABLE 12.1 PROJECT I, PERFORMANCE DATA OF SIX PAIRS OF TEST RUNS

A study of the effect of variation of air velocity across a dehumidifier coil

RUN NUMBER	PROPERTY VARIED FOR MEMBERS OF EACH PAIR	PROPERTIES MAINTAINED CONSTANT FOR MEMBERS OF EACH PAIR OF TEST RUNS							LEAVING CONDITIONS		
	Face Velocity m/s	eh_a kJ/kg	edbt C	ewbt C	Evaporator Pressure kPa gauge	Condenser Pressure kPa gauge	Dryness Fraction at Entry to Evaporator	Superheat Leaving Evaporator C	ℓdbt C	ℓwbt C	ℓw g/kg
2HV	1.86	32.8	14.0	11.4	200	875	0.22	5.3	8.5	8.0	6.5
2LV	0.93								6.5	6.0	5.6
3HV	1.86	32.8	17.0	11.6	200	875	0.22	5.3	9.2	8.0	6.2
3LV	0.93								6.7	6.0	5.5
4HV	1.36	41.5	19.7	14.8	230	975	0.20	4.7	11.3	10.1	7.0
4LV	0.68								9.0	8.4	6.7
5HV	1.71	64.0	29.2	22.0	295	960	0.20	5.8	18.8	17.4	11.9
5LV	0.86								15.8	15.3	10.7
6HV	1.71	64.0	25.7	21.9	295	960	0.20	5.8	18.0	17.5	12.3
6LV	0.86								15.6	15.3	10.7
7HV	1.36	41.5	23.0	15.0	230	975	0.20	4.7	11.8	10.2	7.0
7LV	0.68								8.8	8.2	6.5

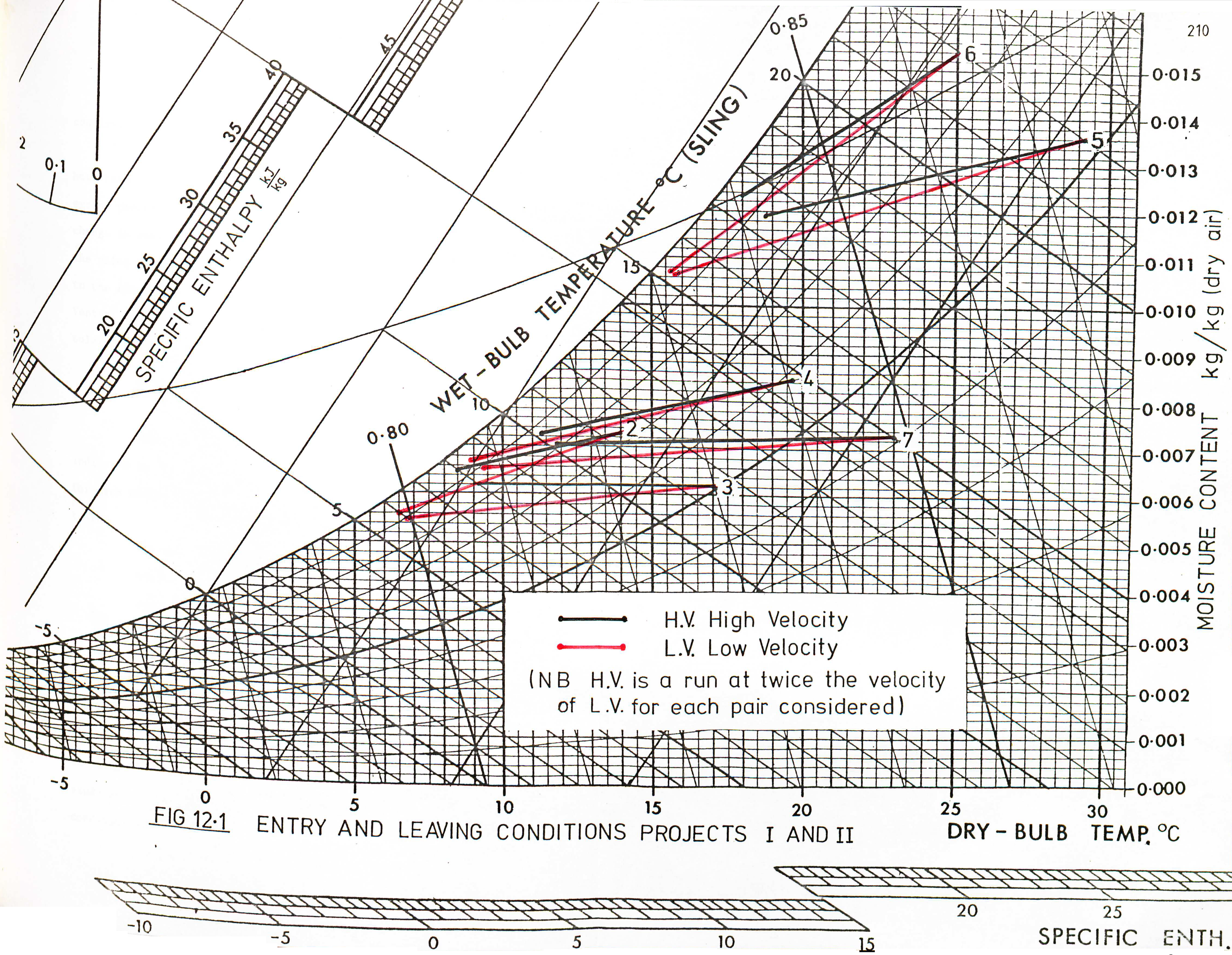


FIG 12·1 ENTRY AND LEAVING CONDITIONS PROJECTS I AND II

In every case the conclusions of Sections 10 and 11 are confirmed.

The humidity ratio changes indicated by the Assmann psychrometer have been checked by the independent means afforded by the teaching system procedure described in Section 8.1, paragraphs 4d and 4e. The change in humidity ratio across the dehumidifier may be compared with the value of the ratio of rate of water condensed at the dehumidifier to the flow rate of *dry air* obtained from the pressure reading at the Venturi tube. It was found that these two ratios agreed within a tolerance equivalent to $\pm$ 0.1C on the Assmann readings.

12.5.2 Project II: to study dehumidifier coil performance in relation to enthalpy potential theory

This experiment is described in general in Section 9.1. As indicated in Section 9.1 the following properties will be kept constant for each member of the six pairs of runs considered in these confirming tests:

air flow rate

condensing temperature

evaporator temperature

superheat condition leaving evaporator

dryness fraction entering evaporator

entering air enthalpy.

The dry bulb temperature of each member of a pair will be different. The entry and leaving conditions, connected by straight lines, share the same Figure 12.1 of Project I. The six pairs that make up this study are listed in Table 12.3 with the figure number identifying coil condition curve to which each member belongs.

TABLE 12.3 PROJECT II, IDENTIFICATION OF TEST PAIRS

PAIR	RUN	FIGURE
1	High Velocity Run 2 with High Velocity Run 3	12.2 HV 12.1 (Dry)
2	Low Velocity Run 2 with Low Velocity Run 3	12.2 LV 12.3 LV
3	High Velocity Run 4 with High Velocity Run 7	12.4 HV 12.1 (Dry)
4	Low Velocity Run 4 with Low Velocity Run 7	12.4 LV 12.7 LV
5	High Velocity Run 5 with High Velocity Run 6	12.5 HV 12.6 HV
6	Low Velocity Run 5 with Low Velocity Run 6	12.5 LV 12.6 LV

An examination of the performance data for the six pairs of test runs associated with Research Project II, Table 12.4, and the lines connecting the entry and leaving conditions for each pair, Figure 12.1, reveal that for both high and low velocity runs, for dry, part wetted and fully wetted performance of the coil, each pair of coils started *with the same* entry enthalpy condition and ended with the *same* leaving enthalpy condition. This confirms the enthalpy potential theory for all runs examined including the ones that exhibited heat transfer only. In the case of Run 4HV, compared with *dry* Run 7HV there was a deviation of 0.2 C. There was also a deviation of 0.2 C when Run 4LV was compared with Run 7LV. All other pairs tested for adherence to enthalpy potential theory agreed with $\pm$ 0.05 C which is well within the resolution capability of the instruments and imperfections in the total system steady flow operation. This included Run 2HV compared with dry Run 3HV.

This is an interesting development. Enthalpy potential is concerned with the difference between the enthalpy of unsaturated air and the enthalpy of air at the temperature of a wetted surface, yet for High Velocity Run 3 the leaving enthalpy for this dry run is, within reading accuracy, that of the High Velocity Run 2 which has a completely wetted surface. It is to be noted that High Velocity Dry Run 7 is at a lower face velocity than High Velocity Dry Run 3. This information will be left to be investigated by a future research programme. However it may be of interest to indicate that it is related to the work of Myers (1967) discussed in Section 11 and in general the effect of a wetted film can be studied in this heat and mass transfer research system.

TABLE 12.4 PROJECT II, PERFORMANCE DATA OF SIX PAIRS OF TEST RUNS

A study of dehumidifier performance in relation to enthalpy potential theory

<table>
<tr><th rowspan="2">RUN NUMBER</th><th colspan="2">PROPERTIES VARIED FOR MEMBERS OF EACH PAIR</th><th colspan="6">PROPERTIES MAINTAINED CONSTANT FOR MEMBERS OF EACH PAIR OF TEST RUNS</th><th colspan="3">LEAVING CONDITIONS</th></tr>
<tr><th>edbt C</th><th>ewbt C</th><th>Face Velocity m/s</th><th>Evaporator Pressure kPa gauge</th><th>Condenser Pressure kPa gauge</th><th>Dryness Fraction at Entry to Evaporator</th><th>Superheat Leaving Evaporator C</th><th>eh_a kJ/kg</th><th>ℓh_a kJ/kg</th><th>ℓdbt C</th><th>ℓwbt C</th></tr>
<tr><td>2HV</td><td>14.0</td><td>11.4</td><td rowspan="2">1.86</td><td rowspan="2">200.</td><td rowspan="2">875.</td><td rowspan="2">0.22</td><td rowspan="2">5.3</td><td rowspan="2">32.8</td><td rowspan="2">24.9</td><td>8.5</td><td>8.0</td></tr>
<tr><td>3HV</td><td>17.0</td><td>11.6</td><td>9.2</td><td>8.0</td></tr>
<tr><td>4HV</td><td>19.7</td><td>14.8</td><td rowspan="2">1.36</td><td rowspan="2">230.</td><td rowspan="2">975</td><td rowspan="2">0.20</td><td rowspan="2">4.7</td><td rowspan="2">41.5</td><td rowspan="2">30.2</td><td>11.3</td><td>10.1</td></tr>
<tr><td>7HV</td><td>23.0</td><td>15.0</td><td>11.8</td><td>10.2</td></tr>
<tr><td>6HV</td><td>25.7</td><td>21.9</td><td rowspan="2">1.71</td><td rowspan="2">295</td><td rowspan="2">960</td><td rowspan="2">0.20</td><td rowspan="2">5.8</td><td rowspan="2">64.0</td><td rowspan="2">49.5 (±0.10)</td><td>18.0</td><td>17.5</td></tr>
<tr><td>5HV</td><td>29.2</td><td>22.0</td><td>18.8</td><td>17.4</td></tr>
<tr><td>2LV</td><td>14.0</td><td>11.4</td><td rowspan="2">0.93</td><td rowspan="2">200</td><td rowspan="2">875</td><td rowspan="2">0.22</td><td rowspan="2">5.3</td><td rowspan="2">32.8</td><td rowspan="2">20.7</td><td>6.5</td><td>6.0</td></tr>
<tr><td>3LV</td><td>17.0</td><td>11.6</td><td>6.7</td><td>6.0</td></tr>
<tr><td>4LV</td><td>19.7</td><td>14.8</td><td rowspan="2">0.68</td><td rowspan="2">230</td><td rowspan="2">975</td><td rowspan="2">0.20</td><td rowspan="2">4.7</td><td rowspan="2">41.5</td><td rowspan="2">25.7 (±0.15)</td><td>9.0</td><td>8.4</td></tr>
<tr><td>7LV</td><td>23.0</td><td>15.0</td><td>8.8</td><td>8.2</td></tr>
<tr><td>6LV</td><td>25.7</td><td>21.9</td><td rowspan="2">0.86</td><td rowspan="2">295</td><td rowspan="2">960</td><td rowspan="2">0.20</td><td rowspan="2">5.8</td><td rowspan="2">64.0</td><td rowspan="2">43.0 (±0.05)</td><td>15.6</td><td>15.3</td></tr>
<tr><td>5LV</td><td>29.2</td><td>22.0</td><td>15.8</td><td>15.3</td></tr>
</table>

Project I is related to Project II in that every member of each pair of runs was a member of *both* projects. Consequently if the performance of Project II is accepted to adhere to enthalpy potential theory so too must the performance of Project I.

All the research data presented above are repeatable. Many of the tests were performed twice.

12.6 Conclusion

The research system has been demonstrated to have the capacity to maintain six properties constant under conditions where a seventh property is varied, a necessary requirement to conduct Projects I and II.

The Assmann psychrometer readings have been checked by independent means to be compatible with direct measurement of moisture content change.

Project II has been shown to yield results consistent with enthalpy potential theory.

Since the individual test runs of Project I are identical with those of Project II, it follows that the test results of Project I, are also consistent with enthalpy potential theory and hence may be judged to be reliable.

The performance data of Project I confirms the conclusions of Sections 10 and 11.

The results of Project I are therefore offered as reliable evidence in confirmation of the findings presented in Sections 10 and 11.

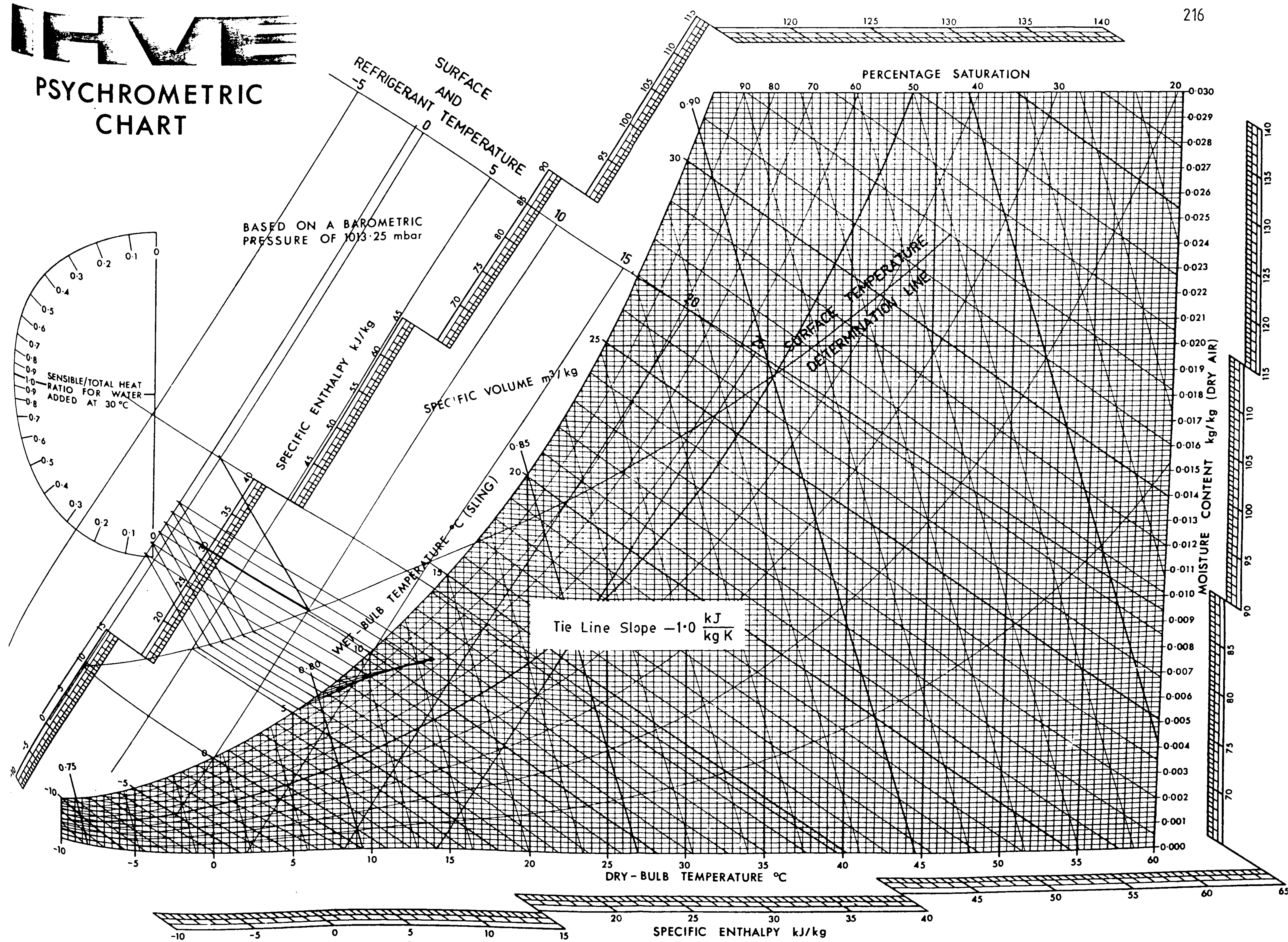


FIG 12·2 H.V. HIGH VELOCITY RUN 2

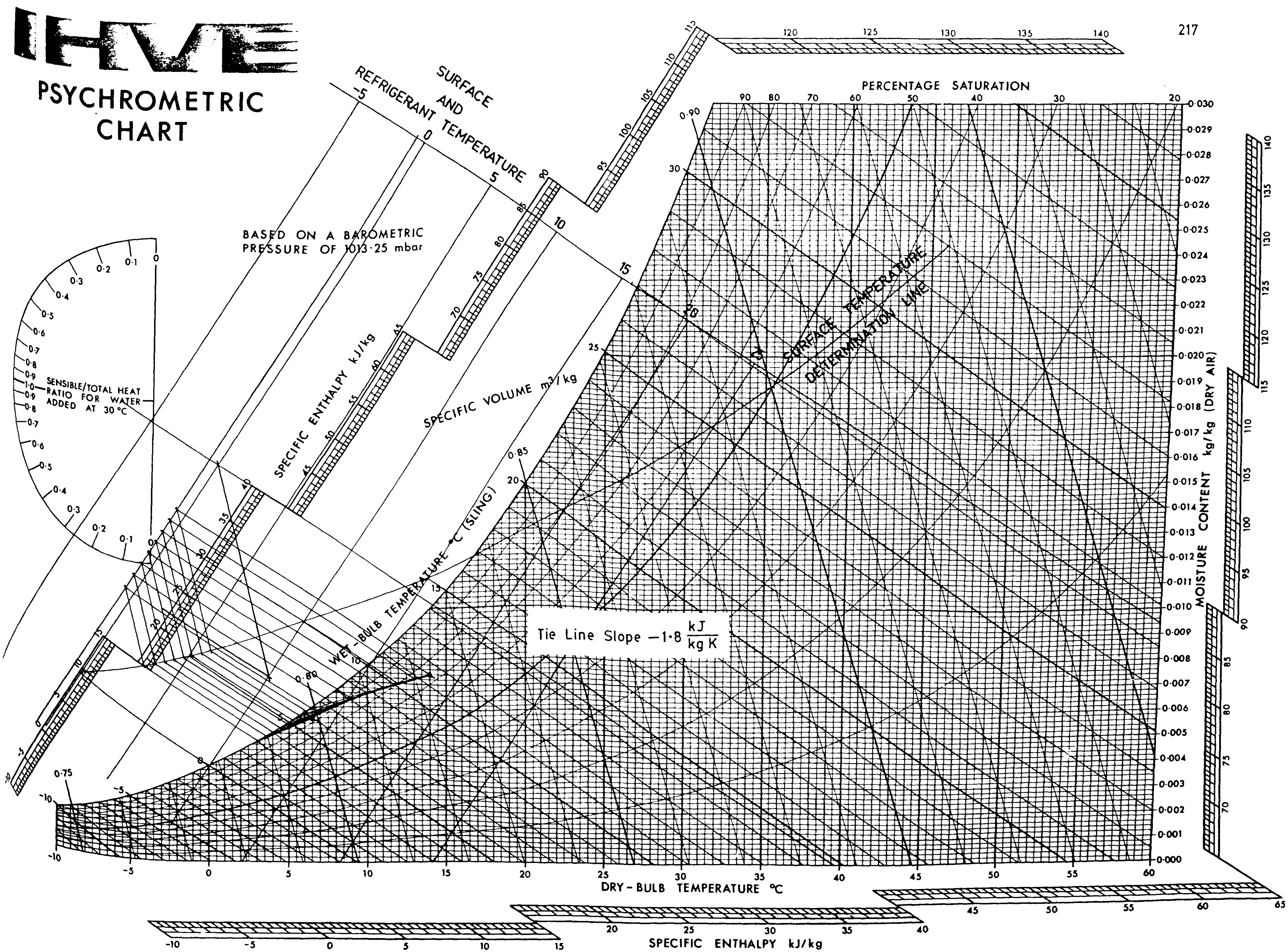


FIG 12·2 L.V. LOW VELOCITY RUN 2

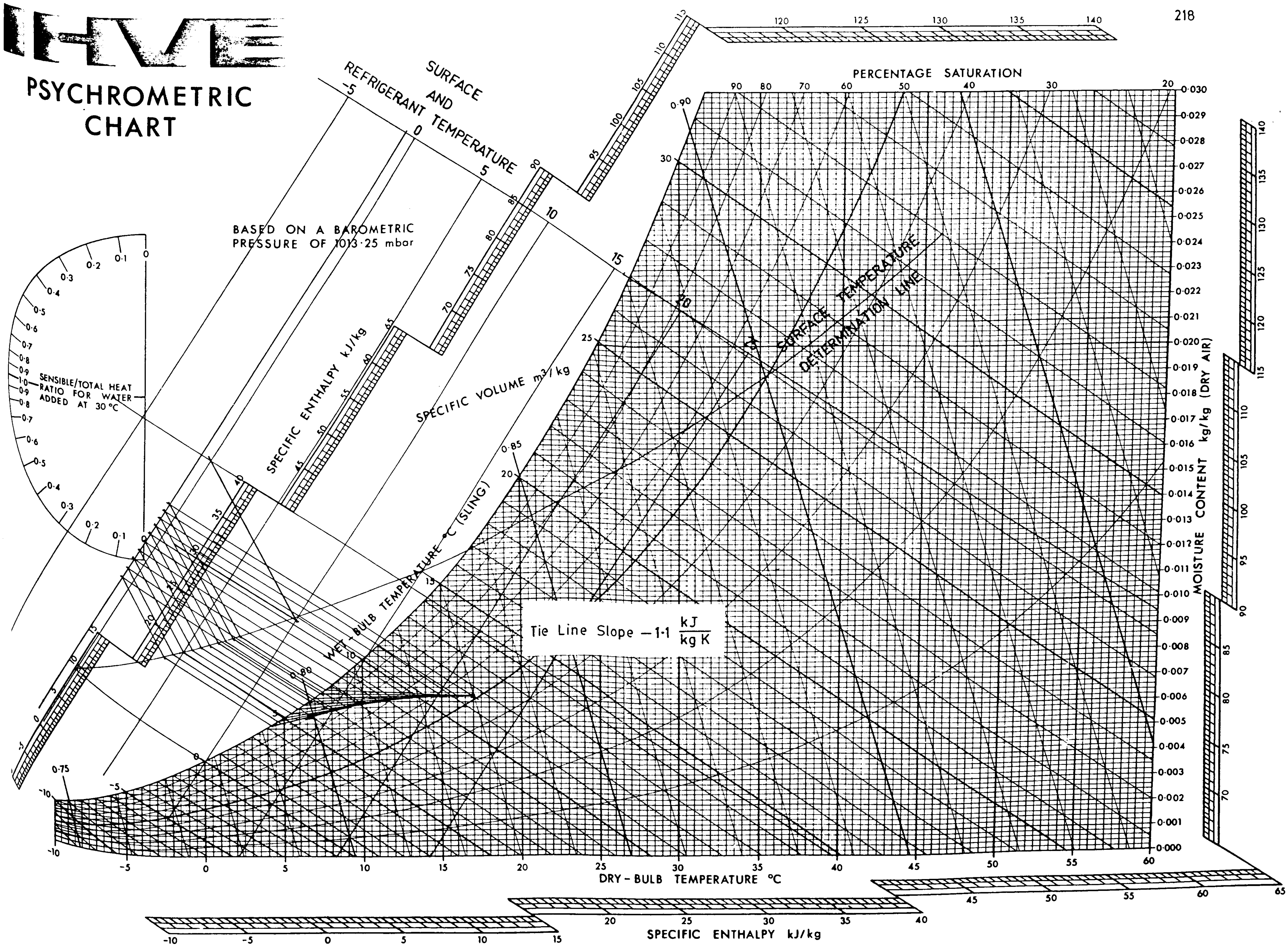


FIG 12·3 L.V. LOW VELOCITY RUN 3

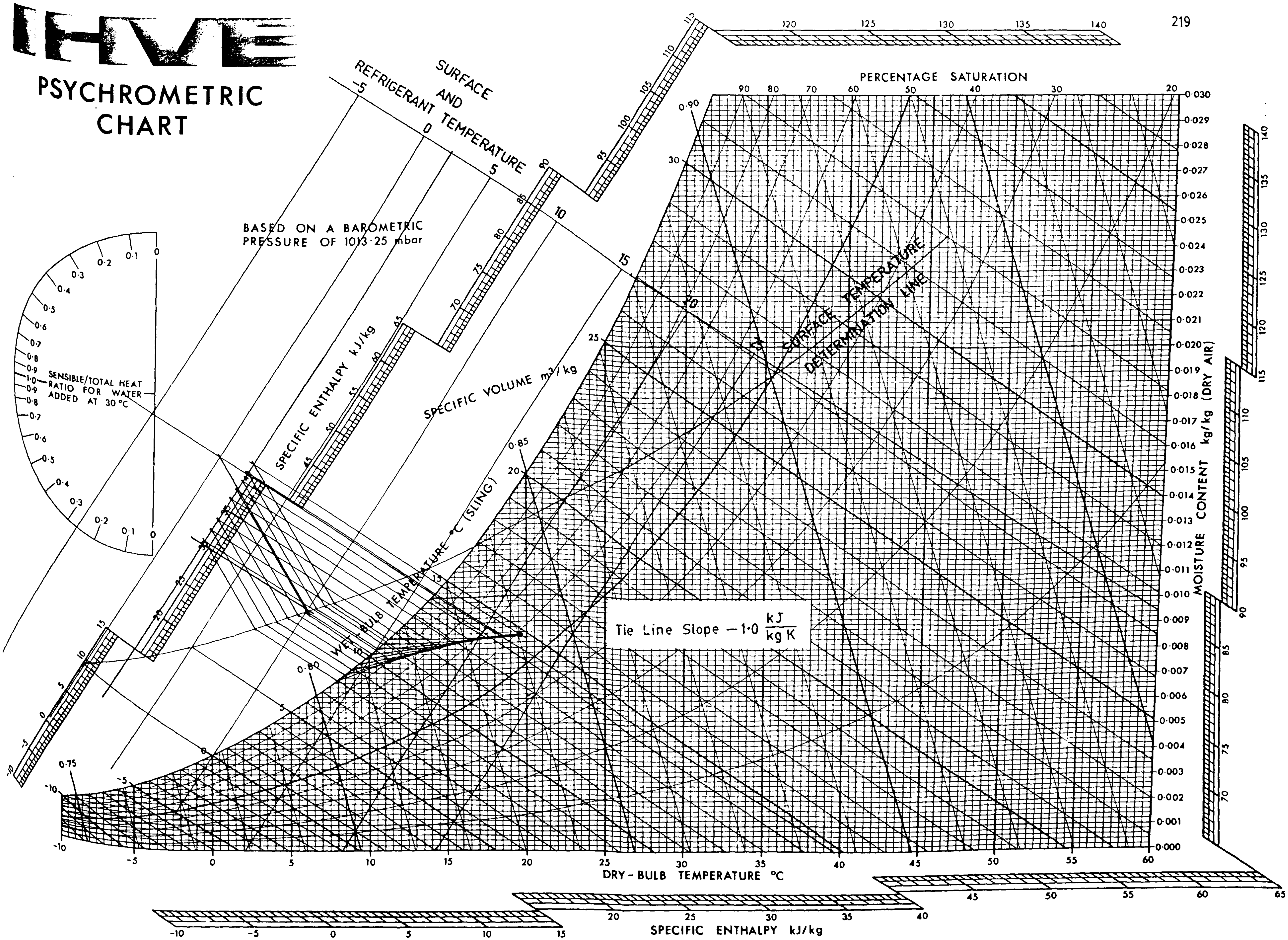


FIG 12·4 H.V. HIGH VELOCITY RUN 4

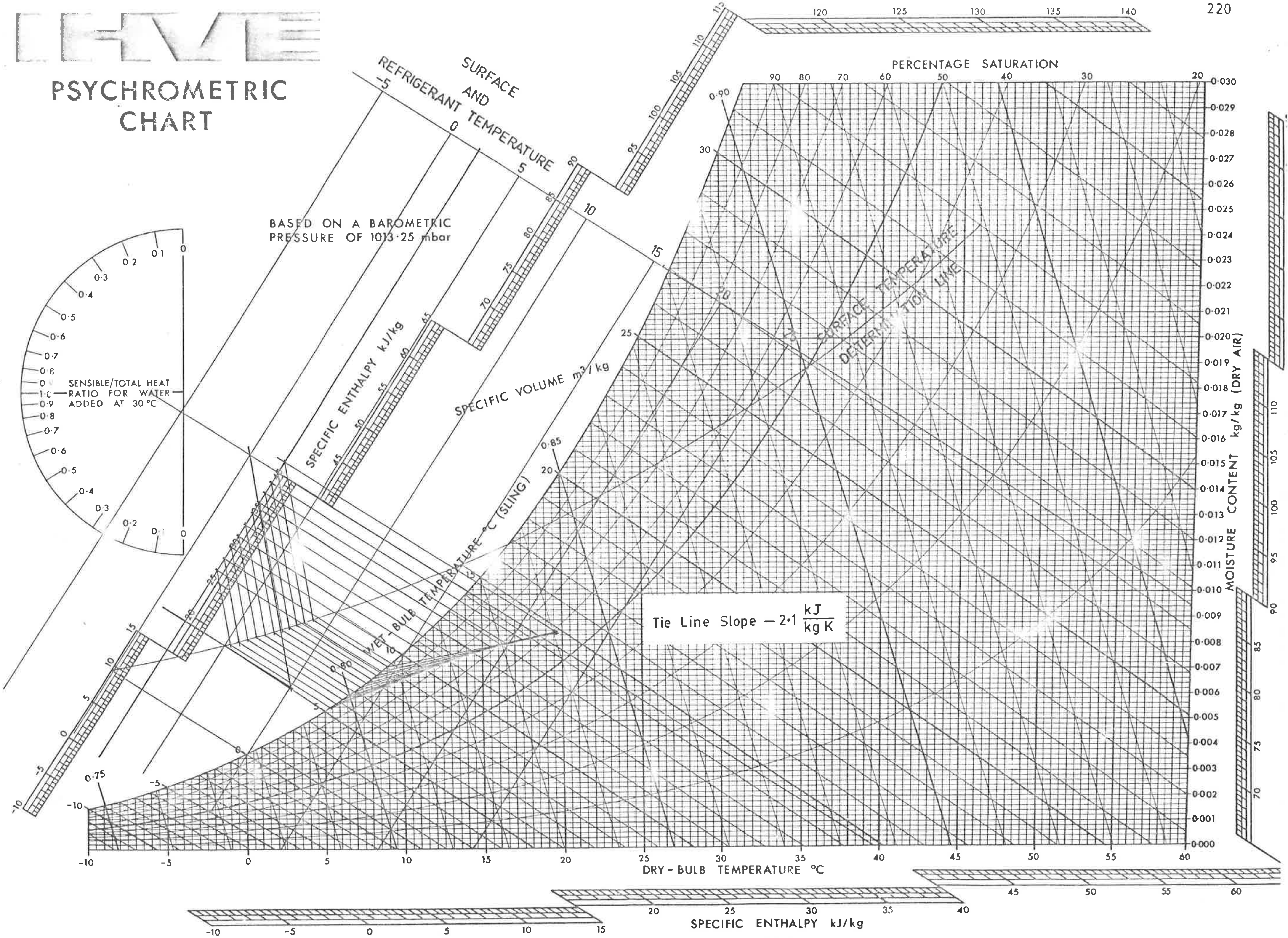


FIG 12·4 L.V. LOW VELOCITY RUN 4

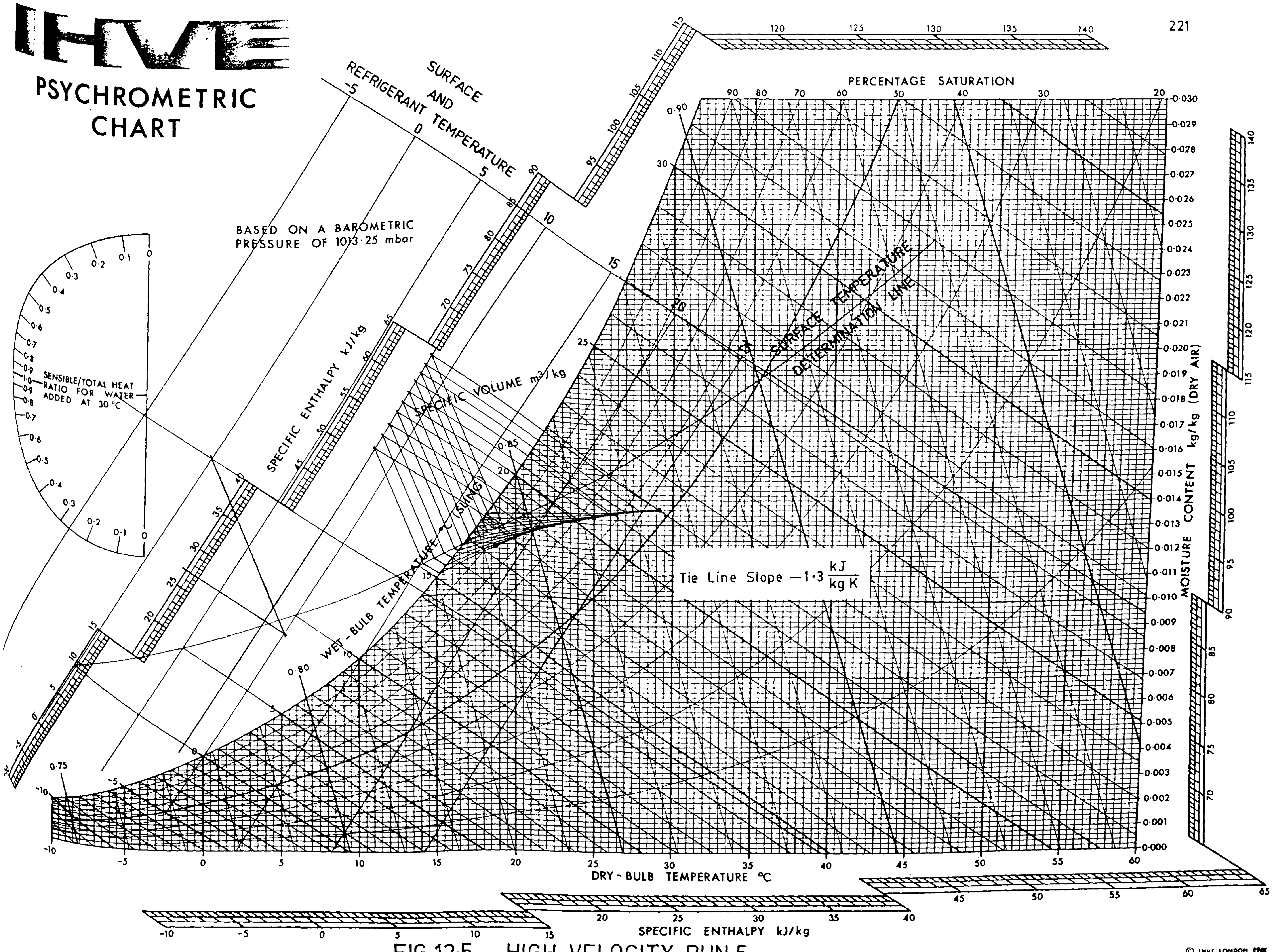


FIG 12·5 H.V. HIGH VELOCITY RUN 5

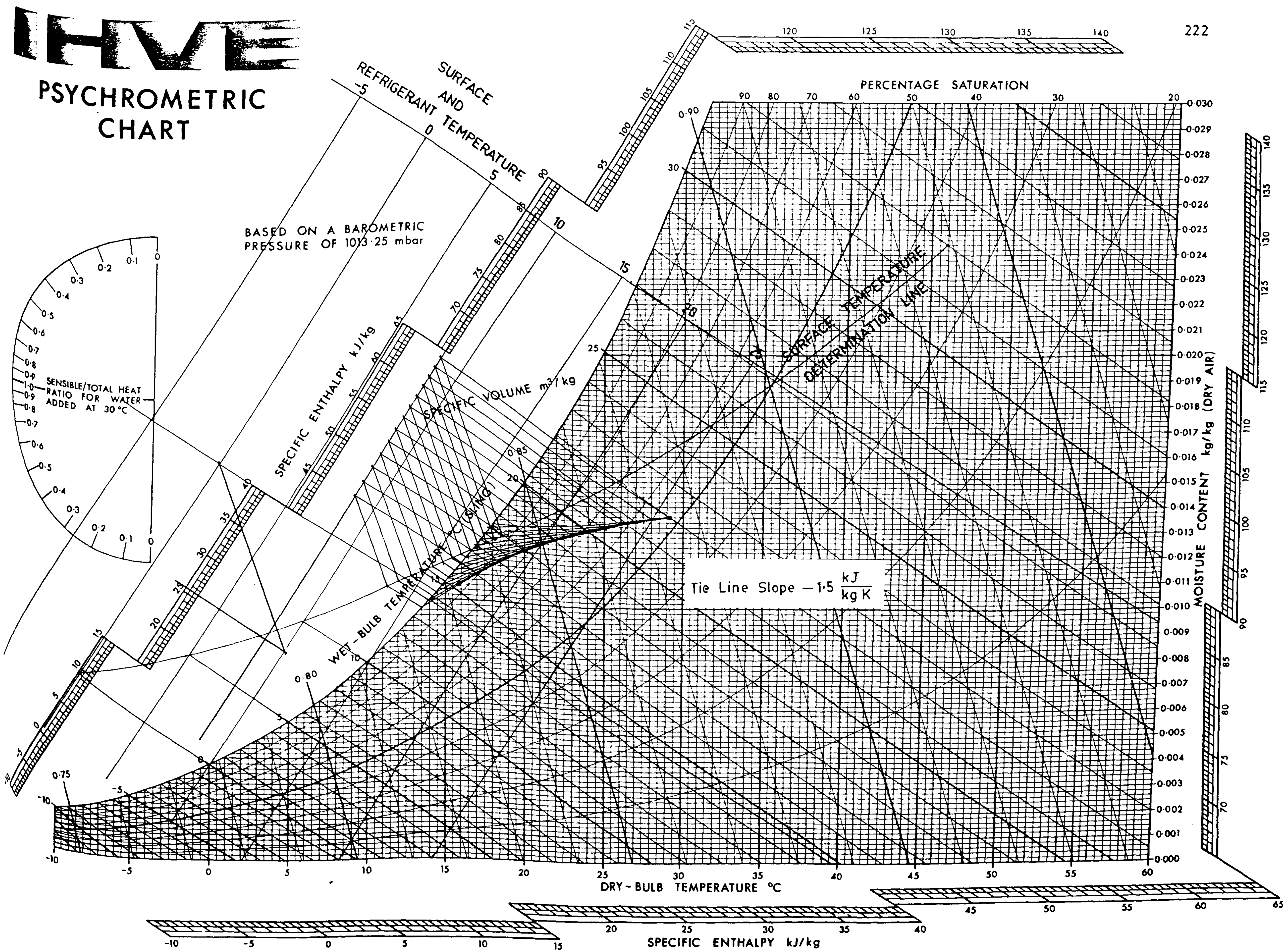


FIG 12·5 L.V. LOW VELOCITY RUN 5

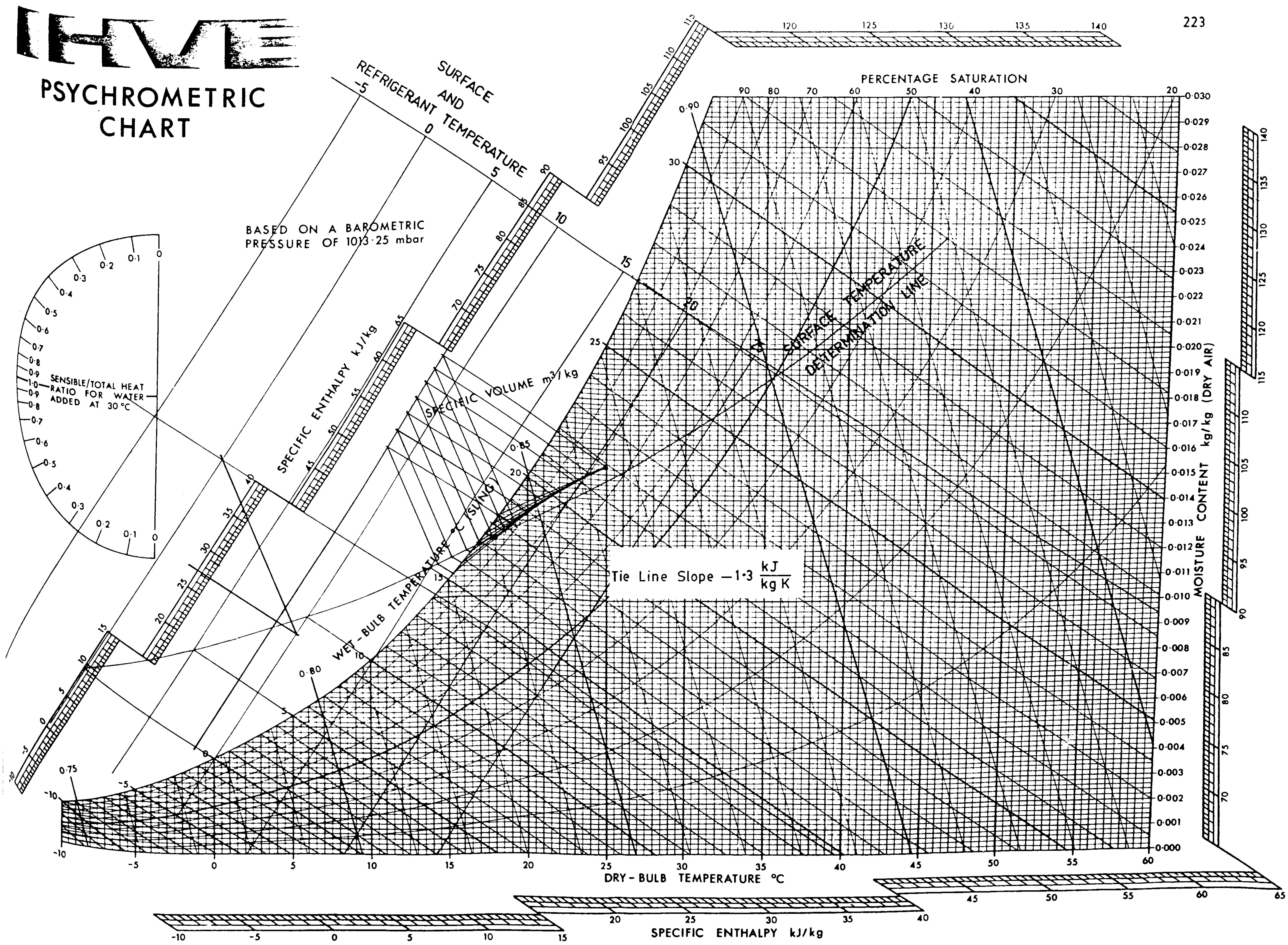


FIG 12·6 H.V. HIGH VELOCITY RUN 6

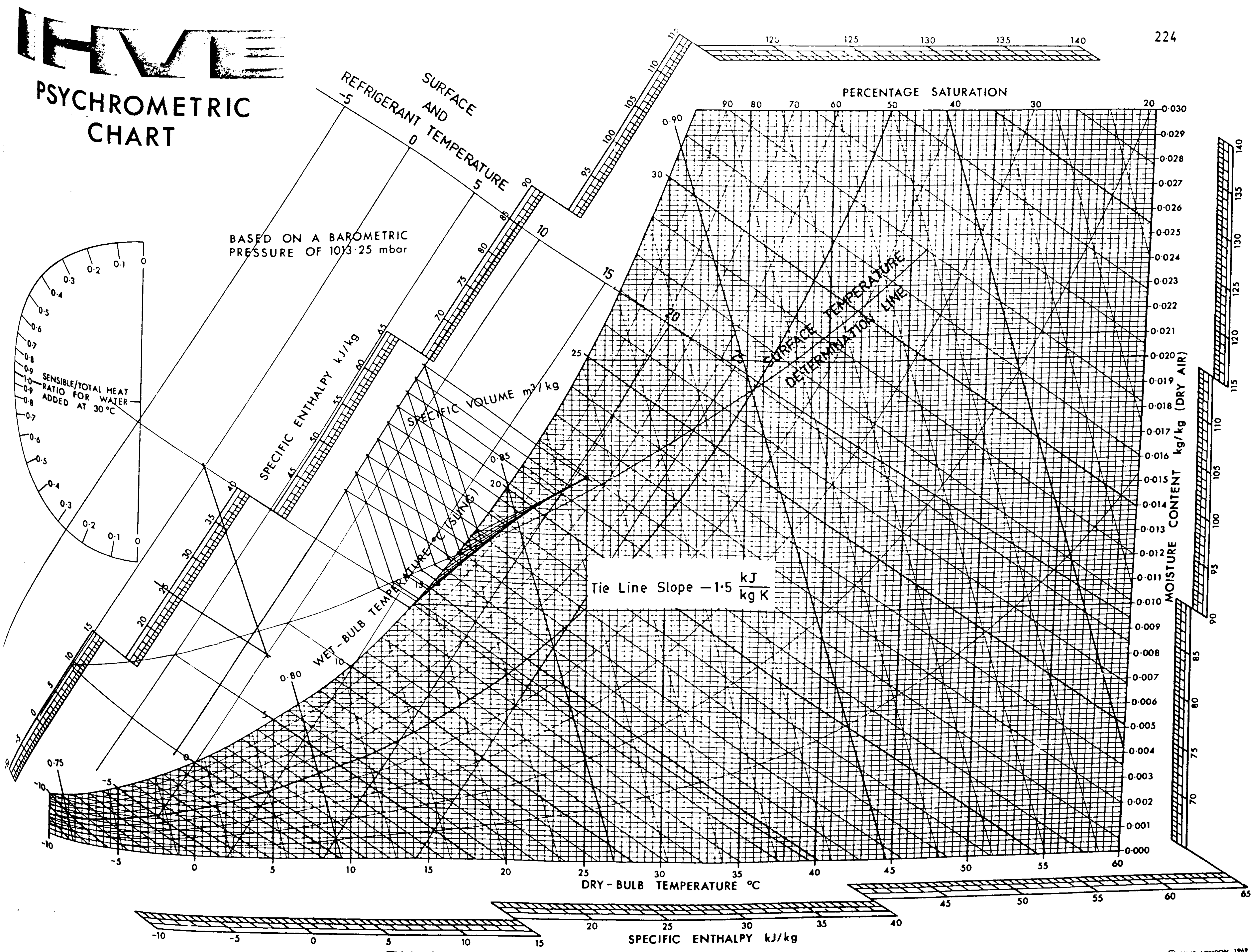


FIG 12·6 L.V. LOW VELOCITY RUN 6

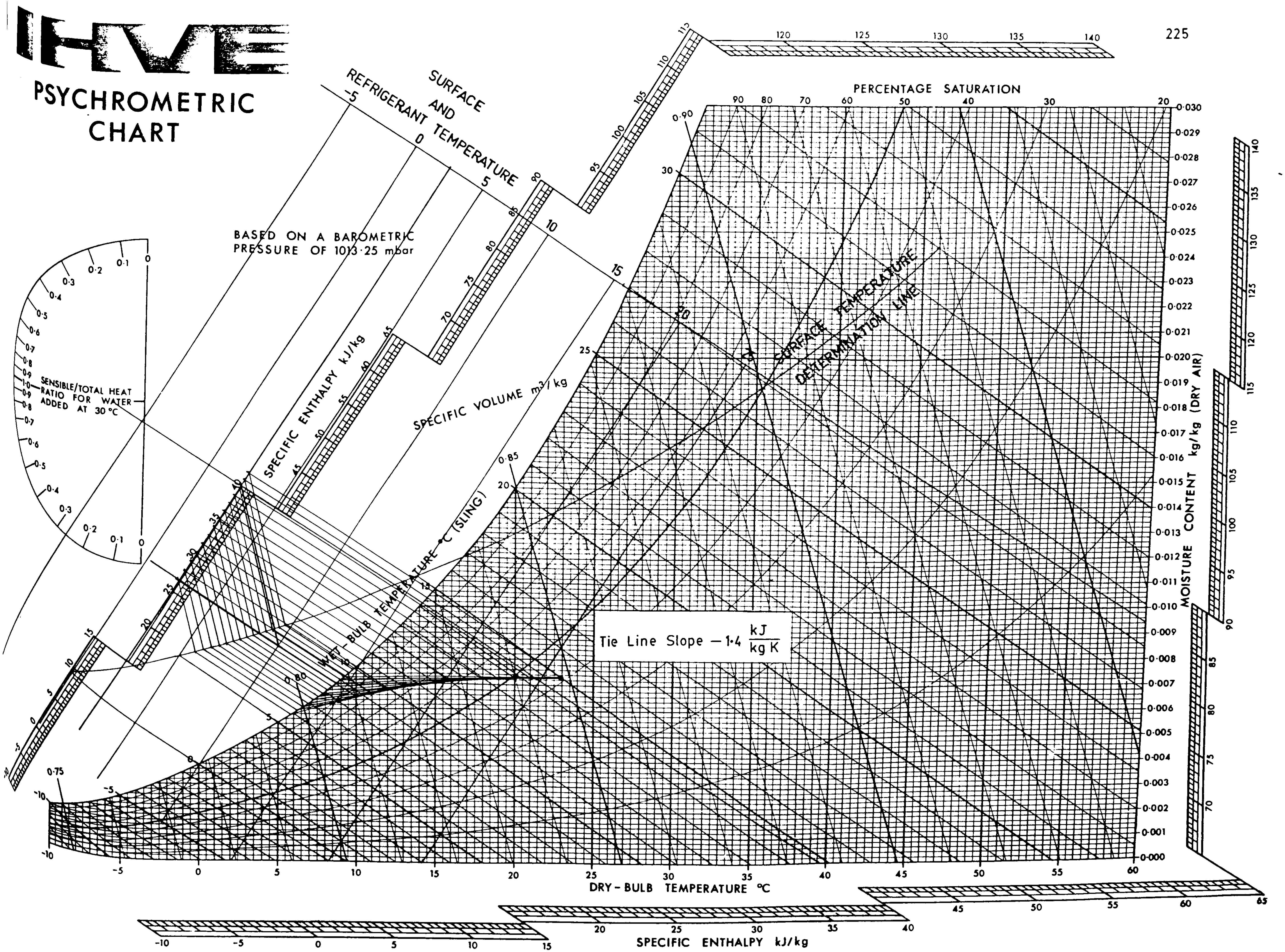


FIG 12·7 L.V. LOW VELOCITY RUN 7

13. CONCLUSIONS

In this thesis the need for climate simulation in teaching and research in the physical and life sciences has been established and the problems associated with the engineering of climate simulators have been analyzed. Two main avenues have been explored. The first relates to those aspects which are peculiar to climate simulators, such as the requirement to simulate *any* realistic temperature and humidity combination within a climatic range and to change automatically from *one* combination of temperature and humidity to *any other* combination of temperature and humidity. In Sections 3 to 6 inclusive a unified system of design has been presented which was shown to be applicable over a wide range of specifications for climate simulation. Design requirements for this area were shown to be in marked contrast to the design requirements of conventional air conditioning, which is essentially free of range requirements.

The essentials of a basic engineering design (Shaw 1964) for climate simulation have been described and this basic design has been shown to be adaptable to accommodate the differences in the detailed specifications of users from various fields. One embodiment is as a low energy system, easily operable by non-engineers, yet having the capacity to change over automatically between different temperature and humidity settings in accordance with some prescribed program. It is applicable to a wide range temperature-humidity system as well as to a narrow range temperature-only system.

The aims of the unified approach have been expressed as a set of specifications. They have been demonstrated by application to several systems whose performance has been investigated. All these systems, on start-up or changeover, automatically establish steady flow conditions for all operating settings within the particular design range, and achieve these settings with constant evaporator surface temperatures. As a consequence all of the systems built to the unified approach exhibit stable performance and maintain both temperature and humidity operating settings to close tolerances.

The second avenue explored in this thesis was the selection of the dehumidifier. The method used enabled the system to achieve stable operation within a wider range, including lower humidity ratios and markedly reduced the energy requirements. The dehumidifier has been shown to be a very important component of this system because of its dual role of offsetting both the sensible and latent heat loads. In environmental chambers having ranges which extend to high dry bulb temperatures combined with low humidity ratio settings the conventional selection methods used by industry are very wasteful of energy, particularly when the temperature gradient in the conditioned space is required to be small. Manufacturers serving the air conditioning industry have adopted numerous simplifying approximations to dehumidifier selection. Only straight line process paths with varying slopes are admitted. There is no indication of the true nature of the curved path of the coil condition curve. The knowledge of the actual process path in climate simulation is essential because small temperature differences across the coil are preferred in order to offset thermal loads acting over small temperature gradients within the controlled space.

In Section 10 the heat and mass transfer relationships involved in dehumidifier selection have been discussed using industrial methods and terminology. In Section 11 a more basic analysis, using heat and mass transfer and enthalpy potential theory was carried out. The path of the coil condition curve was constructed using the Tie Line Slope method. Though selection was first viewed with the special interest of climate simulation, the findings of this section have been shown to be applicable to the air conditioning field.

From a study of dehumidifiers under constraints, (Section 11.3), which established the important parameters influencing selection, it can be concluded that:

1. Air stream velocity and the related Reynolds number have been found to be the major factors in the systematic selection of dehumidifiers both for air conditioning and climate simulation.
2. As the velocity of an air stream over a dehumidifier is reduced the slope of the coil condition curve becomes steeper, the curvature of the coil condition curve reduces towards that of a straight line, and the number of rows of depth required to reach near saturation conditions is reduced.
3. The straight line characteristic of coil condition curves assumed in industrial design methods fails to reveal the major factors which are pertinent to the best selection of a dehumidifier. As the velocity of an air stream is increased over fully wetted dehumidifier surfaces, the coil condition curves become shallower and a point is reached which marks the maximum face velocity for a 'family of curves'. This maximum velocity has been found frequently to occur below the minimum face velocity used in conventional air conditioning practice. Above this velocity the coil is no longer fully wetted. When this occurs, since the first increment of depth in the direction of air flow has zero slope the energy penalty that has been described for conditions of fully wetted surfaces is increased where low sensible heat ratio conditions prevail. The conventional

industrial design approach and spatial arrangements, particularly where low sensible heat ratios prevail, need to be re-examined in the light of potential energy savings due to reduced cooling and reheating, reduced fan power, reduced size of refrigeration equipment and cooling tower and their reduced weights and costs. Despite criticism of conventional methods it is of importance to note that some of the approximate methods used in the commercial field are acceptable in conventional air conditioning applications. An examination of the test data depicted on the psychrometric chart of Figure 12.1 indicates that when the straight lines connecting inlet and outlet conditions of the test runs having the same face velocity and entry enthalpy condition are extended, they very nearly meet at the saturation curve. The temperature at this point is referred to by some in the commercial field as the apparatus dew point. It represents the coil surface temperature.

4. From an examination of part load conditions frequently present in conventional air conditioning applications the case has been made for variable volume systems, indicating energy savings and improved performance if the air flow rates are reduced with decrease of load.

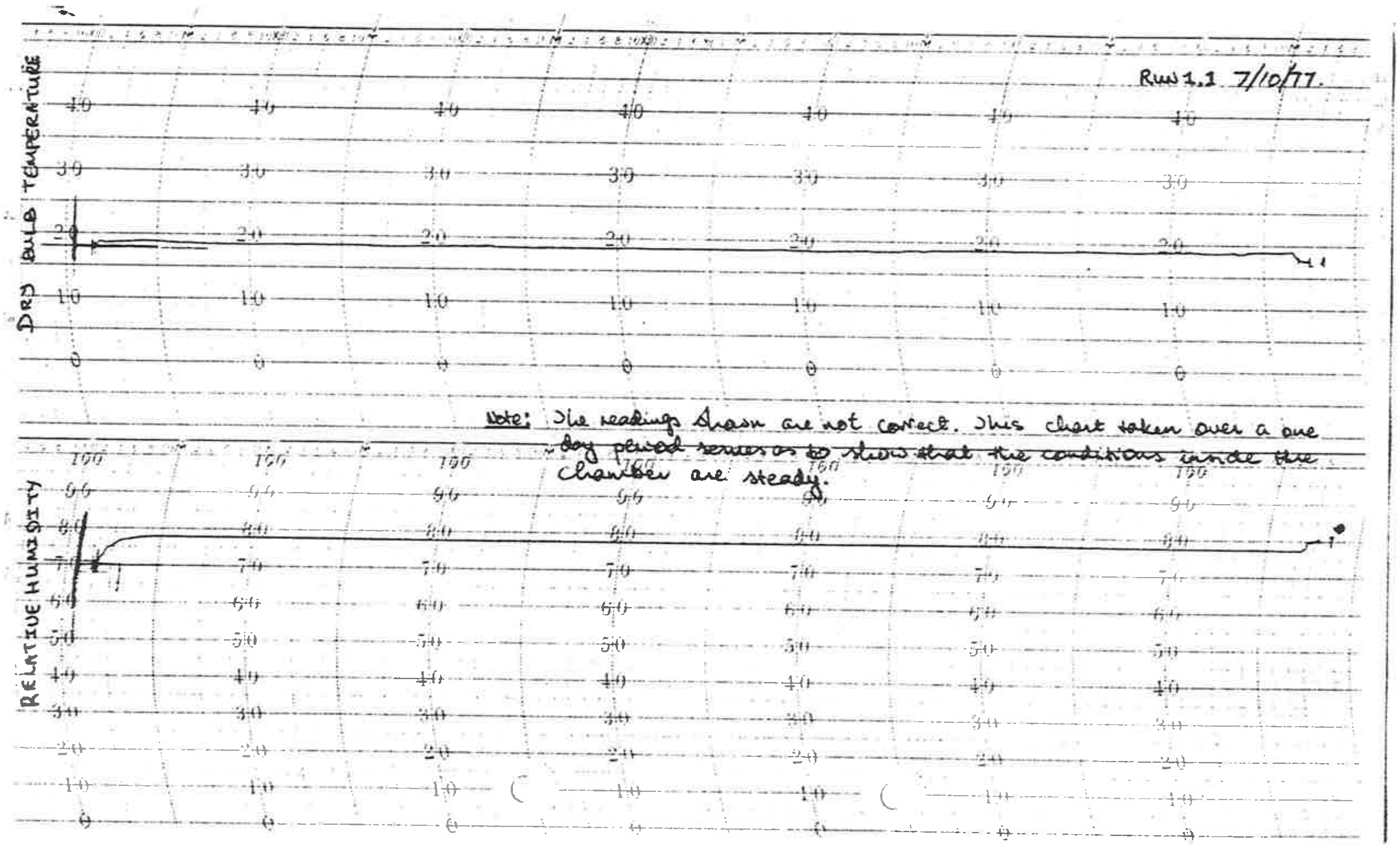
5. Maximum energy conservation is ensured in dehumidifier coil selection when the extended surface has a coil condition curve that best satisfies the load ratio line of the problem. Therefore the entire philosophy of coil selection with emphasis on increased surface density for gas-liquid heat exchangers should be reassessed when the coil in question is a dehumidifier. The increase of fin to primary tube surface may give optimum heat transfer but will reduce mass transfer

in relation to heat transfer. The basic relationships leading from Reynolds number to the $(St)(Pr)^{2/3}$ value on through to the Lewis number relationship have revealed that there is a decreased mass transfer to heat transfer ratio with increase of face velocity. As examined in Section 11.10 this is indicated by the value of h_i changing at a lesser rate than h_{do} with change of face velocity. As a consequence an increased Tie Line Slope value is obtained. An increased Tie Line Slope reveals a decrease in the ratio of mass transfer to heat transfer. In dehumidifier applications it is important to recognize that though increased air flow velocity will improve heat transfer it will reduce the ratio of mass transfer to heat transfer.

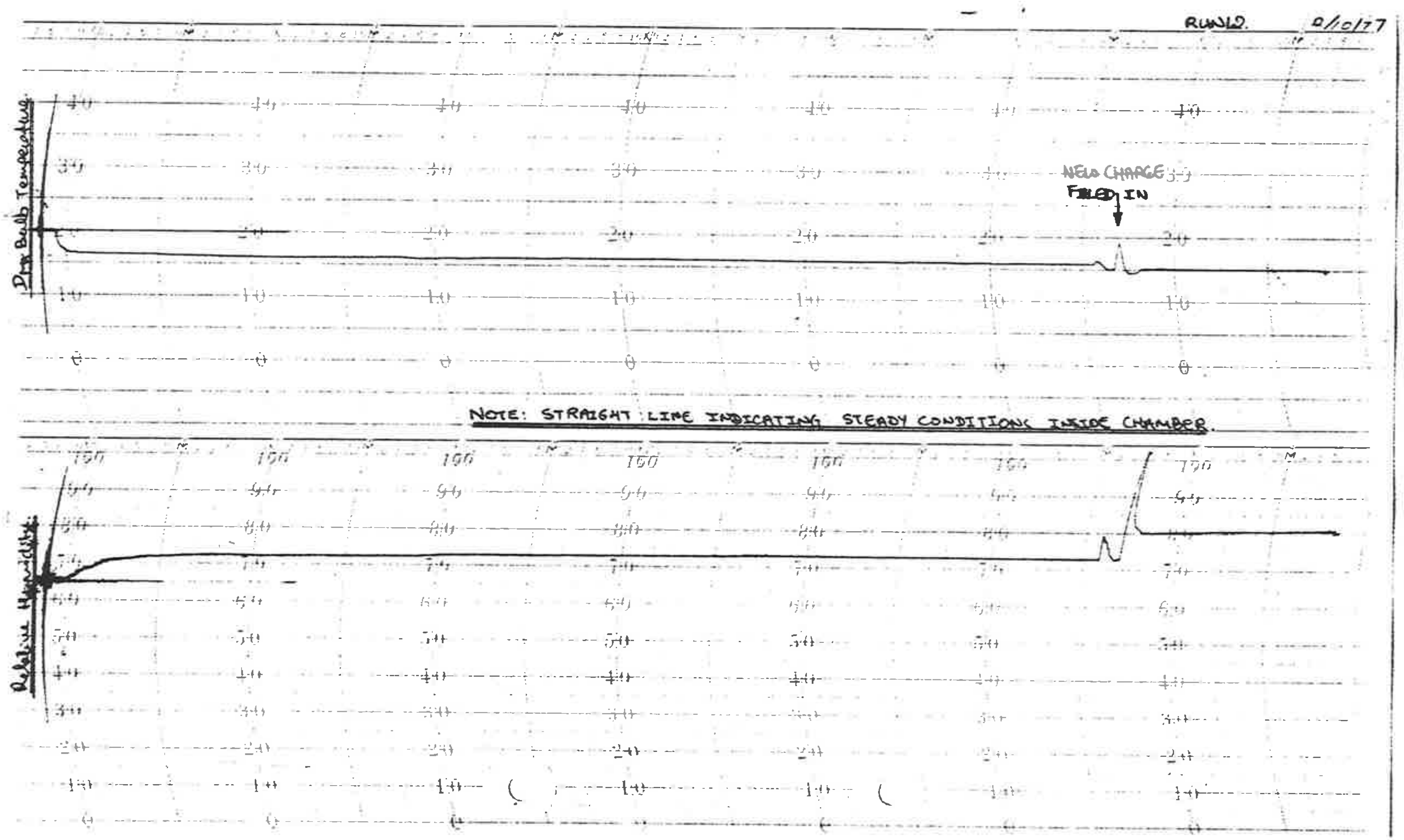
Resulting from the findings of this thesis, a method of dehumidifier selection has been described which differs from existing air conditioning practice. The proposed selection method acknowledges the actual heat and mass transfer performance and eliminates many of the assumptions and approximations used at present. It is direct and simple to use and leads ideally to selection of the optimum dehumidifier in terms of overall system performance. However if its widespread use is to be promoted, cooperation will be necessary between standards associations, engineering societies and manufacturers.

The effectiveness and validity of the unified design method for climate simulators espoused in this thesis has been demonstrated by analysis and interpretation of actual experimental data from physical plant which was designed using the unified system. Additionally the flexibility of this type of plant for teaching and for research in the physical and life sciences has been indicated.

Further areas of research which lie beyond the immediate aims of this thesis have become apparent. These include studies of the effect of dry, part-dry and wetted transfer surfaces, investigation of enthalpy potential and further optimisation of the proposed method of dehumidifier selection. The apparatus built to demonstrate the findings of this thesis is also eminently suitable for fundamental heat and mass transfer research and for performance testing of commercial heat and/or mass transfer components.



APPENDIX IA THERMOHYDROGRAPH OF RUN 1.1, WAITE INSTITUTE PHYTOTRON UNIT



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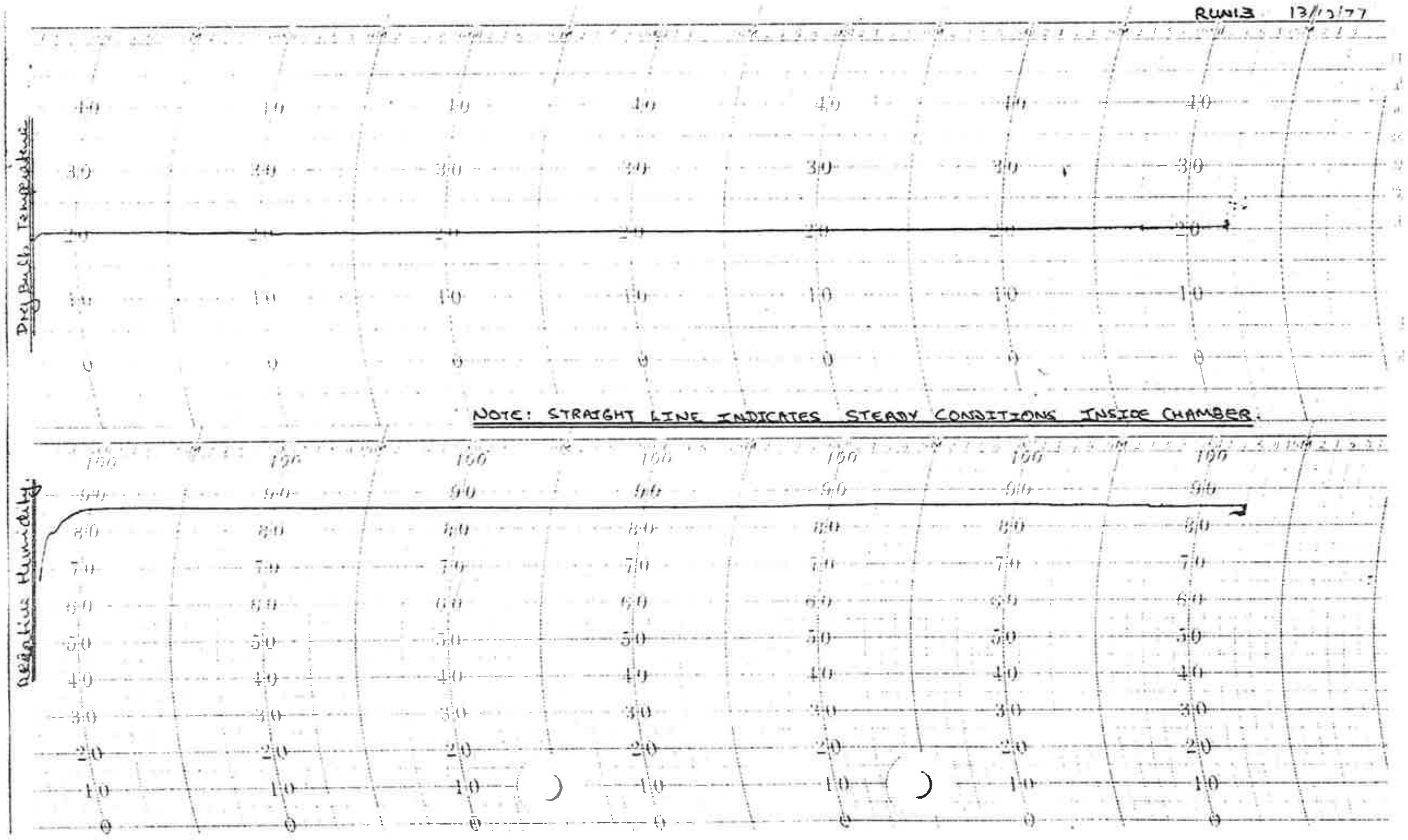

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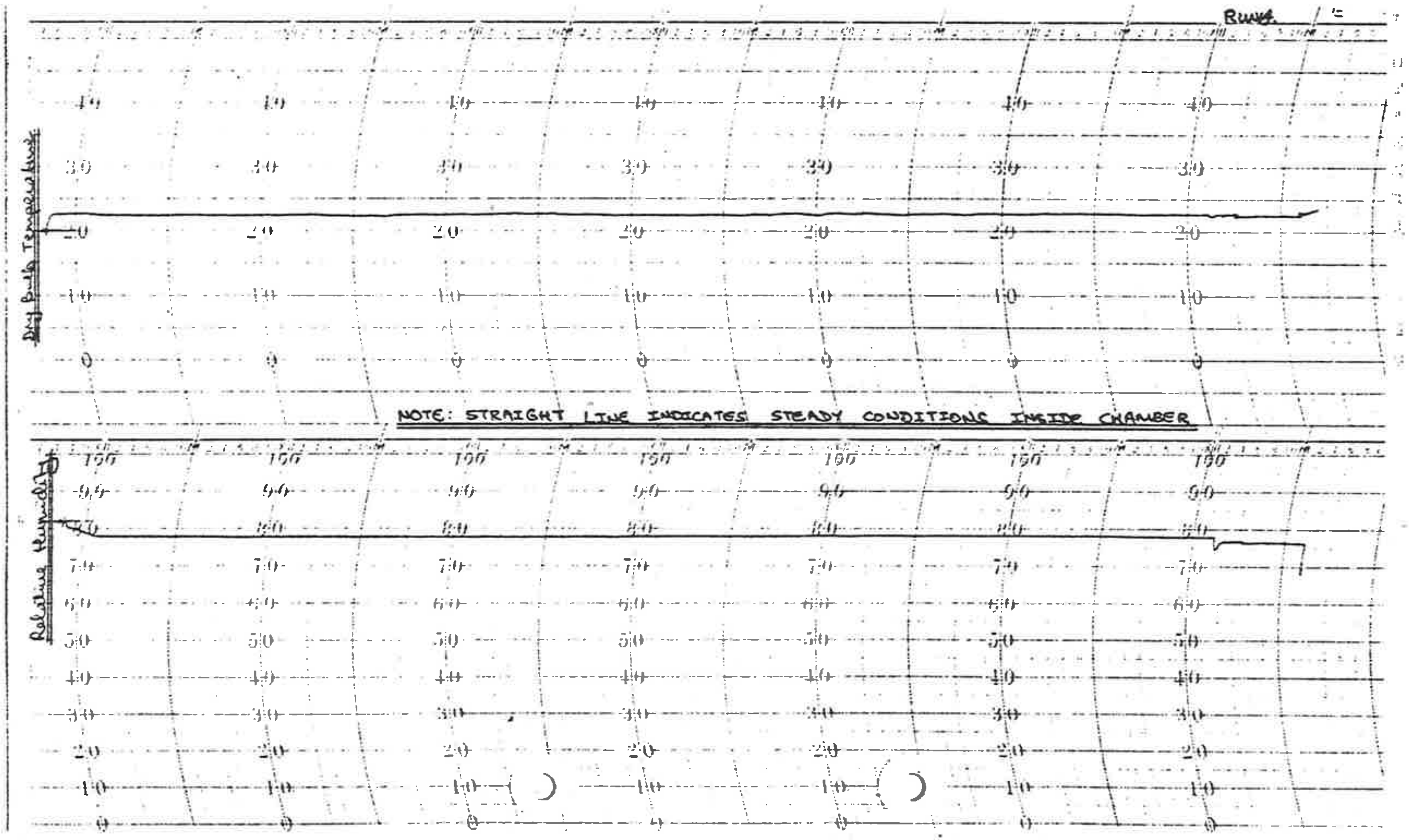

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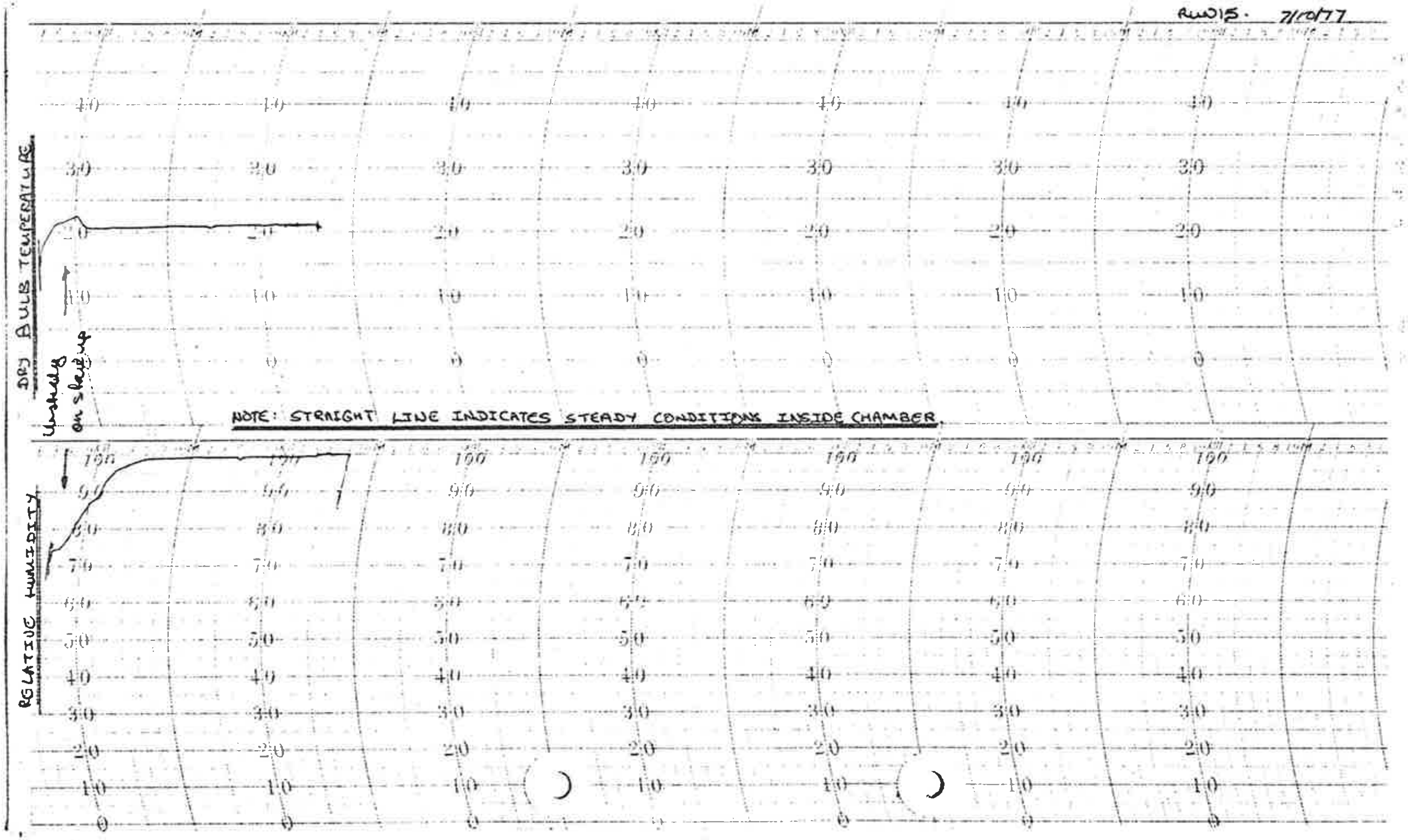

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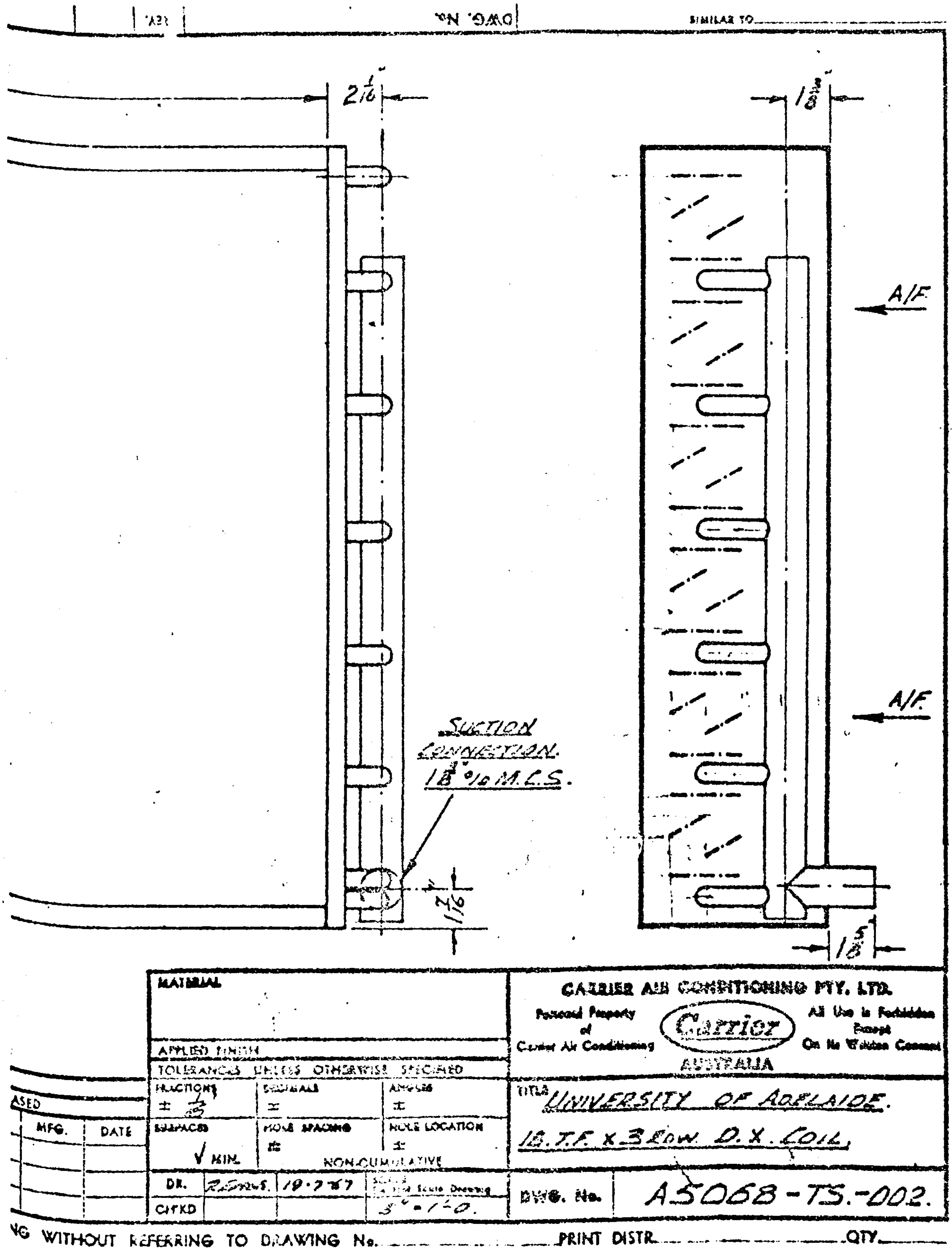
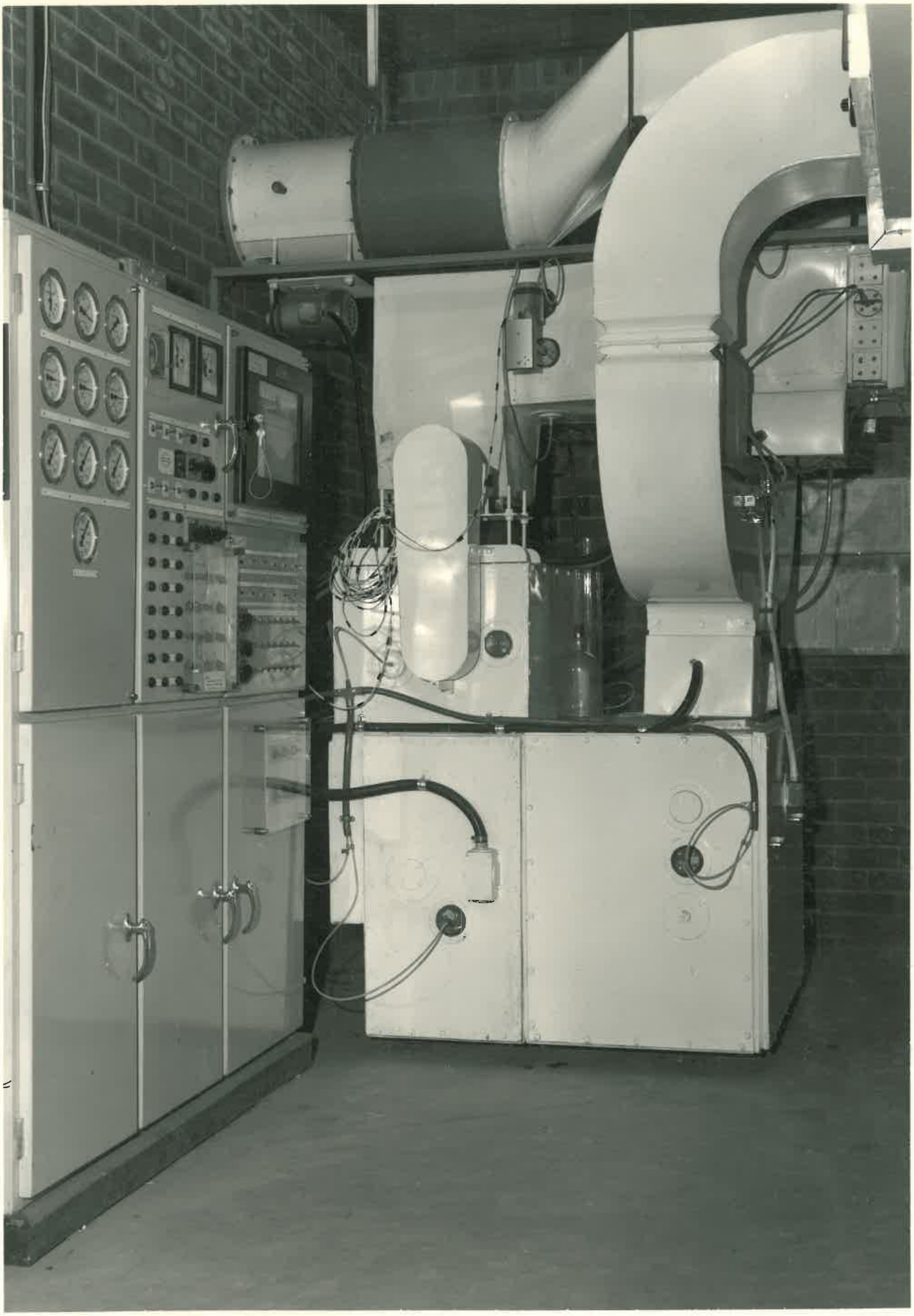


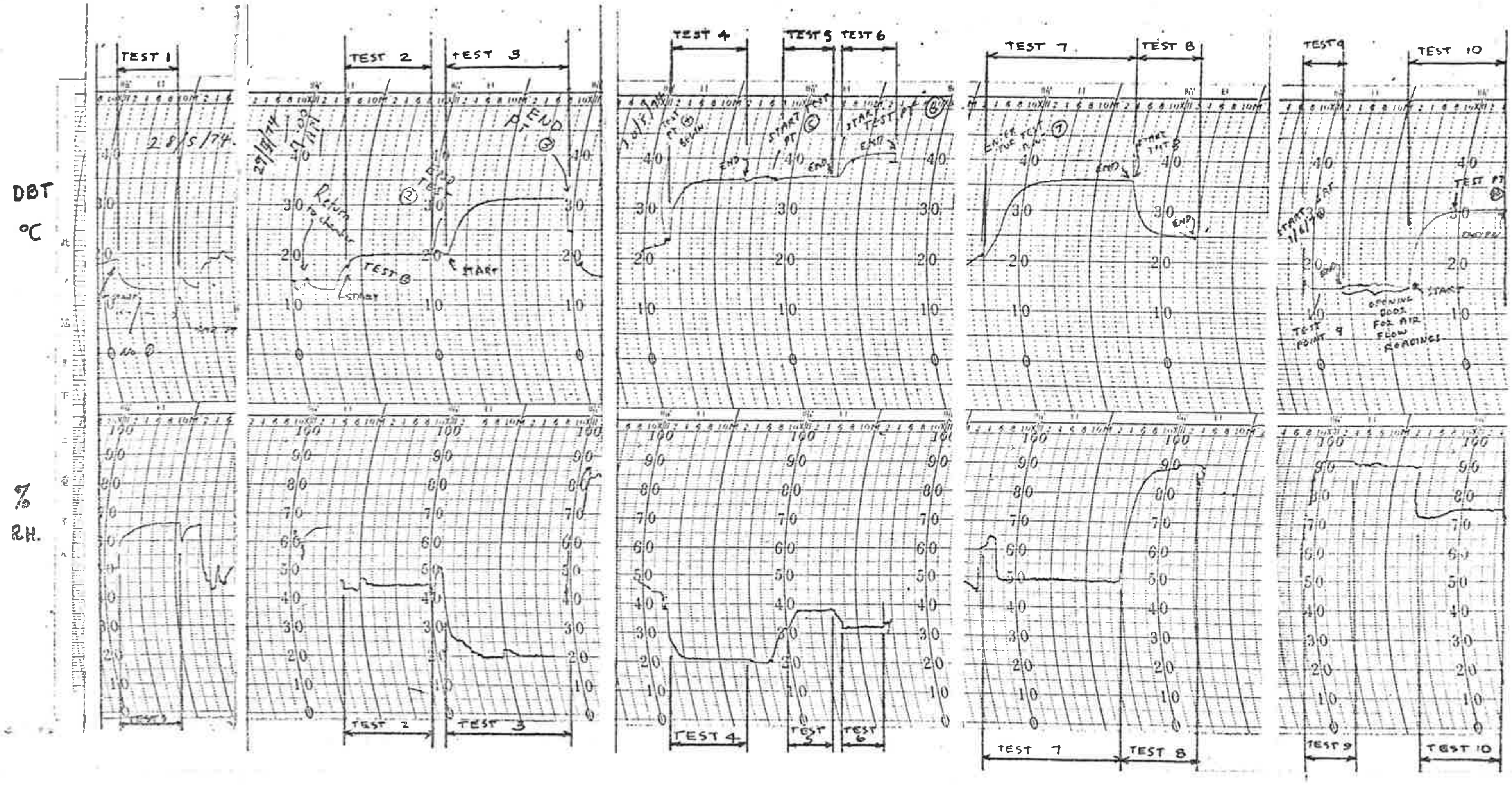
APPENDIX II DEHUMIDIFIER WAITE INSTITUTE PHYTOTRON UNIT



APPENDIX III CONTROL PANEL AND AIR TREATMENT SYSTEM
WAITE INSTITUTE PHYTOTRON UNIT

APPENDIX IV PLANT GROWTH CHAMBER
WAITE INSTITUTE PHYTOTRON UNIT

APPENDIX V   A TWO DIRECT EXPANSION COIL AND TWO HERMETIC CONDENSING UNIT SYSTEM



APPENDIX VI PERFORMANCE OF COMMERCIAL FACE AND BYPASS SYSTEM

(11) 402,202

COMMONWEALTH OF AUSTRALIA

PATENT SPECIFICATION (21) 57,747/65.

Class (52) 31.4; 90.9; 29.7.

Int. Cl. (51) A01g; F24f.

Application Number (21) 57,747/65.
Lodged (22) 20th April, 1965.
(Accompanied by a Provisional Specification)

Complete Specification entitled (54) PHYTOTRON.

Lodged (23) 13th April, 1966.
Accepted (44) 1st April, 1970.
Published (41) 19th October, 1967.

Convention Priority (30)

Applicant (71) ALLAN SHAW.

Actual Inventor (72) ALLAN SHAW.

Related Art (56)	120,751(12,249/43)	90.9
	22,721/35.	90.9
	105,910(1283/37)	90.9.

The following statement is a full description of this invention, including the best method of performing it known to me

12688/70 109-3D-23/4/70-15P.C. W. G. Murray, Government Printer, Canberra

AUSTRALIAN PATENT

The claims defining the invention are as follows :-

1. A controlled atmosphere chamber defined by walls of a cabinet, including a fan to draw a gas vapour or gas/vapor mixture through the chamber, a refrigeration system having an evaporator unit with a cooling heat exchange surface in the path of flow of the gas, vapor or gas/vapor mixture, heating means in said path, and control means responsive to changes in at least one of the sensible heat portion and latent heat portion of the true-load (as defined herein) to maintain a series of set points, characterized in that said control means are responsive to true-load changes and that the total loads on the refrigeration system are maintained substantially constant for each control set point.

(20th April, 1965)

2. A controlled atmosphere chamber according to claim 1 further characterized in that the heating means include a preheater disposed upstream of the cooling heat exchange surface. (20th April, 1965)

3. A controlled atmosphere chamber according to either claim 1 or claim 2 wherein said control means are responsive to changes in both the sensible heat portion and the latent heat portion, further characterized in that the evaporator is of such capacity that the heat exchange surface dehumidifies the gas/vapor mixture when passing it to a moisture content equal to or less than the moisture control set point of the chamber. (20th April, 1965)

4. A controlled atmosphere chamber according to claim 3 further comprising humidifying means and further characterized

in that the cooling and dehumidifying, and the heating and humidifying means are of minimum capacity for the range of set points within the psychrometric limits of operation.

(20th April, 1965)

5. A controlled atmosphere chamber according to any one preceding claim further characterized by a reheater disposed downstream of the cooling heat exchange surface controllably connected to a dry bulb sensor in the said chamber responsive to true-load changes, the reheater constituting temperature control means by correcting for over cooling of air passing through the evaporator, by which dry bulb temperature operating conditions are maintained over a range of dry bulb temperatures.

(20th April, 1965)

6. A controlled atmosphere chamber according to any one preceding claim further characterized by a humidifier disposed downstream of the cooled heat exchange surface, and controllably connected to a dew point sensor in the said chamber, responsive to true-load changes, the humidifier constituting humidity control means by which the dew point temperature (humidity ratio) operating conditions are maintained over a range of dew point temperatures.

(20th April, 1965)

7. A controlled atmosphere chamber according to any one preceding claim further characterized by a preheater disposed upstream of the cooled heat exchange surface, the preheater being of sufficient capacity to supply sufficient temperature to the gas/vapor mixture to avoid frosting.

(20th April, 1965)

8. A controlled atmosphere chamber according to any

one preceding claim wherein the chamber is defined by the walls of a cabinet, a plenum in the base of the cabinet and a register in the side walls near the top of the cabinet, the upper portion of the plenum being defined by at least one perforated plate, characterized in that the gas, vapor or gas/vapor mixture enters the cabinet through the plenum in the base thereof and leaves the cabinet from the register. (20th April, 1965).

9. An environment control system comprising:

a recirculating fluid circuit including cabinet walls defining a chamber which defines the environmental space to be controlled and a fan for driving a substantially constant volume flow of gas and/or vapor through the circuit; a closed steady flow, vapor compression refrigeration system having an evaporator, with a cooling heat exchange surface in the circuit and compressor with means for operating it continuously and at a constant speed when the system is operating to maintain a desired chamber condition; and control means responsive to changes in at least one of the sensible heat portion and the latent heat portion of the true-load (as hereinbefore defined) and operable independently of the refrigeration system to control the heat portion or portions to which it is responsive in the chamber to any of a plurality of presettable desired values for the or each such heat portion by the controlled addition of heat to the gas and/or vapor, the arrangement being operable such that, on starting-up or upon a change of desired value, the control means will act on the gas and/or vapor in the fluid circuit to drive it to a condition in which the then desired chamber condition is obtained in the chamber, during which action the total load on the refrigeration system and the temperature of the refrigerant will naturally and automatically reach steady state values which are functions of the desired chamber condition by virtue of the effect upon the refrigeration system of the change in inlet conditions to the evaporator produced by the action of the control means upon the gas and/or vapor, the control means then functioning to maintain the preset desired chamber condition without adjustment of the constant volume flow of the gas and/or vapor or of the steady state refrigeration system in which said steady state values of total heat and temperature remain substantially constant regardless of true load changes.

(20th April, 1965)

10. A method of controlling the environment in a chamber to any one of a plurality of conditions of at least one of the sensible heat and latent heat, the chamber being defined by cabinet walls, and being in a fluid circuit containing the evaporator of a closed vapor-compression refrigeration system, and wherein, when operating at any one of a plurality of desired environment conditions in said chamber, gas and/or vapor is passed through the circuit with a constant volume flow, the refrigeration system is operated continuously in a steady state with a total load upon it and a refrigerant temperature which are constant and independent of changes in true load (as hereinbefore defined), and at least one of the sensible heat and latent heat of the true-load is sensed and controlled by control means independent of the refrigeration system to maintain the desired environment condition by

the controlled addition of heat to the gas and/or vapor, whilst, on commencing the control or upon a change of the desired condition, the control means is caused to vary the amount of heat it adds to the gas and/or vapor to drive it to the then desired condition inthe chamber, during which action the continuously running refrigeration system naturally and automatically moves towards and reaches a steady state condition corresponding to the then desired chamber condition by virtue of the effect of the changes in the inlet conditions of the evaporator produced by the action of the control means on the gas and/or vapor.

(20th April, 1965)

11. A system according to claim 9 or a method according to claim 10, wherein the control means comprises a heating means downstream of the heat exchange surface and upstream of the chamber to maintain a preset dry bulb temperature in the chamber. (20th April, 1965)

12. A system or method according to claim 11 wherein the heating means comprises a bank of electrical heater wires.

(20th April, 1965)

13. A system or method according to claim 11 wherein the heating means comprises a hot water, or steam, heat exchange coil. (20th April, 1965)

14. A system according to any one of claims 9 or 11 to 13 or a method according to any one of claims 10 to 13 wherein the control means comprises humidifying means downstream of the heat exchange surface and upstream of the chamber to maintain a preset dew point temperature in the chamber.

(20th April, 1965)

15. A system or method according to claim 14 wherein the humidifying means comprises a bank of electrical heater wires. (20th April, 1965)

16. A system or method according to claim 14 wherein the humidifying means comprises a steam jet humidifier. (20th April, 1965)

17. A system according to any one of claims 9 or 11 to 16 or a method according to any one of claims 10 to 16 and having preheater means upstream of the heat exchange surface and downstream of the chamber. (20th April, 1965)

18. A system or method according to claim 17, wherein the preheater means has dry bulb temperatures sensing means downstream of the preheater means and upstream of the heat exchange surface and the control means comprises regulating means for regulating the preheater means in accordance with the sensing means. (20th April, 1965)

19. A system or method according to claim 17 or 18, wherein the preheater means comprises a bank of electrical heater wires. (20th April, 1965)

20. A system or method according to claim 17 or 18, wherein the preheater means comprises a hot water or steam, heat exchange coil. (20th April, 1965)

21. A system or method according to any one of claims 12, 15 or 19, wherein at least one wire in the bank is connected to motor driven vatiable transformer. (20th April, 1965)

22. A system or method according to claim 21, wherein the or each bank includes wires controlled by switches to be

placed progressively into or out of circuit when the transformer hasbeen driven to its upper or lower voltage limit respectively. (20th April, 1965)

23. A system according to any one of claims 9 or 11 to 22 or a method according to any one of claims 10 to 22, wherein the control means can be preset for any chamber condition in a predetermined area of the psychrometric chart, the system or method being operable with substantially constant volume flow of gas and/or vapor and substantially constant mass flow of refrigerant and with a substantially constant total heat laod upon the refrigeration system to maintain a steady state chamber condition corresponding to any one of the values in said predetermined area. (20th April, 1965)

24. A system or method according to claim 23, wherein the area of the chart extends at least from 55°F. to 85°F. dry bulb temperature and at least from 40°F. to 80°F. dew point temperature. (20th April, 1965)

25. A system or method according to claim 23 or 24, wherein the control means and the refrigeration system have substantially the minimum capacity necessary for achieving the chamber conditions of all of said preset values.

(20th April, 1965)

26. A system according to any one of claims 9 or 11 to 25 or a method according to any one of claims 10 to 25, wherein an air inlet vent is disposed downstream of the chamber and a spill damper is disposed upstream of the chamber.

(20th April, 1965)

27. A system according to any one of claims 9 or 11 to 26, or a method according to any one of claims 10 to 26, wherein the control means has separate setting means, whereby,

more than one chamber condition can be preset, there being change-over means by which the control means can be changed from one setting to the other. (20th April, 1965)

28. A system or method according to claim 27, wherein there is a timer for operating the change-over means according to a presettable time programme. (20th April, 1965.)

29. A system according to any one of claims 9 or 11 to 28, or a method according to any one of claims 10 to 28, wherein the chamber has a plenum by which the gas and/or vapor will be introduced into the chamber by way of at least one perforated plate in the plenum wall. (20th April, 1965)

30. A system according to any one of claims 9 or 11 to 29, or a method according to any one of claims 10 to 29, wherein the chamber has a plenum in its base, by which the gas and/or vapor will be introduced upwardly into the chamber, the upper portion of the plenum being defined by at least one perforated plate. (20th April, 1965)

31. A system or method according to claim 30, wherein the chamber has a register in the side wall at the upper region of the chamber for the exit of gas and/or vapor.

(20th April, 1965)

32. A system according to any one of claims 9 or 11 to 31, or a method according to any one of claims 10 to 31, wherein a chamber dew point temperature below 32°F. is attainable. (20th April, 1965)

33. A system according to any one of claims 9 or 11 to 32, or a method according to any one of claims 10 to 32, when operating under a steady state chamber condition, wherein the gas and/or vapor has a constant volume flow rate; the gas and/or vapor inlet state to the cooling heat exchange surface, the refrigeration system condensing temperature and the refrigerant temperature are substantially constant; the compressor of the condensing unit runs continuously, all the gas and/or vapor circulates past the cooling heat exchange surface; and a thermostatic expansion valve exists in the refrigeration system and constitutes the only automatic controller on the operation of the refrigeration cycle. (20/4/65)

34. A controlled atmosphere chamber constructed substantially according to the embodiment described in the specification with reference to and as illustrated in the accompanying drawings. (20/4/65)

Dated this 27th day of January, 1970.

ALLAN SHAW,

By his Patent Attorneys,
R.K. MADDERN & ASSOCIATES.

PATENT SPECIFICATION

DRAWINGS ATTACHED

1,147,025

1,147,025

Date of Application and filing Complete Specification: 19 April, 1966. No. 17139/66

Application made in Australia (No. 57747) on 20 April, 1965.

Complete Specification Published, 2 April, 1969.

Index at Acceptance:—**F4** V (B3D, B4A, B4B, B4C, B4D,); **A1** E (3, 8); **G3** R (2A1, 36A1, 36D1, 36FX, 36G3, 36H1, 38).

Int Cl.:—**F 24** f 5/00.

COMPLETE SPECIFICATION

Phytotron

I, Allan Shaw, a citizen of the United States of America, of 72, Cross Road, Myrtle Bank, State of South Australia, Commonwealth of Australia, do hereby declare the invention, for which I pray that a patent may be granted to us, and the method by which it is to be performed, to be particularly described in and by the following statement:—

This invention relates to a controlled chamber, that is, a space defined by boundaries within which the properties of dry bulb temperature and/or dew point temperature can be controlled over a range of settings.

Controlled chambers have been used heretofore to a large extent for plant growth studies, and when used as such as herein termed "phytotrons". However controlled chambers extend to include chambers for both the control and the testing and rating of performance, and are for example used by space scientists, physicists, building scientists, biologists, zoologists, geneticists and engineers. For example they may be used for a controlled animal house, or a space suit using oxygen and water vapor mixtures at low pressure, or for the testing of expansion coils or electronic equipment and with this invention if desired in these uses the controlled settings can be subjected to periodic variations.

This invention is applicable to "chambers" in the broadest sense of the term; the chamber may be an insulated section of duct work, a room, a series of rooms, an animal house, a space suit, a space capsule, a calorimeter or the like.

The application of this invention extends to the general phenomena of heat and mass transfer of any gas and any vapor mixture (including the limiting cases wherein the gas content or vapor content approaches zero) and is not necessarily confined to air/water vapor mixtures.

This invention is applicable to any mixture pressure and is not necessarily confined to barometric pressure at sea level.

Phytotrons and similar chambers have been required to meet numerous specifications of the users. These specifications vary widely from narrow to wide ranges of operating settings; from temperature only to simultaneous temperature and humidity settings; from a single fixed operating condition to an operating condition which automatically changes over to simulate both day and night conditions or other periodicity programming arrangements; from a one hundred per cent recirculating system with carbon dioxide injection to systems specifying a fixed percentage of fresh air and spill; from systems operating to broad tolerances to systems requiring close tolerances with minimal deviation between points within the plant occupied space; from systems that are set up laboriously with long periods required for manual adjustment to quick start automatic systems.

Heretofore phytotrons and similar chambers have been operated by systems which have not been able to economically meet the requirements of the users. They have failed to function properly when wide ranges of operating settings were specified, when automatic change-over between day and night operating settings was specified, when control of both dry bulb and dew point temperature was specified, when fresh air introduced from the outside of the chamber was specified, when close tolerance control was specified, or when automatic start-up procedure was specified.

Existing systems have failed to meet the needs of the agronomist and scientist. This failure has increased as these demands have

[Price 4s. 6d.]

UNITED KINGDOM PATENT

high or low pressure compressor cut out, change to another compressor speed and again repeat the same pattern of programmed inlet conditions as for the original compressor speed setting. Instrumentation measuring air flow and leaving coil dry bulb and dew point temperatures will establish the coil performance for standard stipulated conditions and also for any desired operating inlet condition within a broad range. When used for rating and testing, the direct expansion coil would be installed in a manner which would facilitate quick removal and replacement in accordance with good engineering practice.

On the other hand, the usual control chambers used for testing materials or performance of a complete self-contained system such as a fan-coil unit could simply replace the phytotron chamber. The dry bulb and dew point sensors would always be located at the point where the gas/vapor mixture is controlled. For the testing of heat and mass transfer surfaces the sensors would be located directly upstream of the test surface, with the required instrumentation giving the performance downstream of the test surface.

This invention can combine this use for testing direct expansion coils for varying coil entry conditions by controlling the entry condition to the direct expansion coil with its use for controlling a separate cabinet adapted for testing self-contained units, building materials, electronic equipment all together in one installation. This combination would simply require a procedure of activating only those sensors of the control system in the area being controlled. This combination would serve as a very important tool to scientists and university laboratories. It could not only be a material testing and heat transfer surface testing system but could be effectively used for demonstrating the principles of refrigeration and for laboratory experiments in applied thermo-dynamics.

Modifications can also be made by the use of devices and components other than those described herein. For example, an analagous pneumatic control system is wholly applicable in lieu of the electric control system described in particular. Heating components for the preheat, reheat and humidifying functions may be hot water coils, steam coils, steam jet humidifiers etcetera. It need not be electric as described for the above embodiment.

The specification of the user of this invention with reference to automatic changeover requirements can be very varied in programming, including rising or falling temperature or humidity or compressor capacity. It need not be simply a change-over between day and night conditions as described for the particular embodiment.

The heat exchange surface need not be a finned evaporator coil.

The specifications of the user may allow for a selection of components so that the minimum refrigerant temperature is high enough to avoid frosting and therefore the preheater of the embodiment may be omitted. (For example a very narrow range of settings in the high temperature and humidity region may result in refrigerant temperatures being high enough to avoid occurrence of frosting).

The reduction of the non-true load to a practical minimum within the terms of the engineering system makes it economically feasible to provide for narrow range systems and systems requiring control of temperature only. For example should the user be interested in a temperature-humidity controlled chamber operative in the higher range of humidities, say above 50% relative humidity rather than the wide range within the bordered area A of Fig. 2, all other requirements of the user being identical to this embodiment presented herein detail, a smaller capacity system with lower running costs would result. Inlet condition 1 of Fig. 2 determines the basis for selecting the refrigeration system of this embodiment, the limiting factor in this case is to meet the humidifying true load of the system. For this example, where the range of humidities required by the user is above 50% relative humidity inlet condition number 6 of Fig. 2 would determine the basis for selecting the refrigeration system and would have a refrigeration capacity of approximately one fourth of the system described in this embodiment.

The variable element and fixed elements of each bank of heaters are a specific example for this embodiment and the specific user's requirements establishing this embodiment. It is possible to have a user's requirement for a phytotron or for a controlled chamber wherein the tolerances are very broad, wherein less sensitive controllers may warrant, due to the slow rate of true load change, the use of on-off control. For example if the controllers have a sensitivity which will result in a response only after, say, a $\pm\frac{3}{4}$°F. change has occurred in the cabinet and if the change in true load is such that a rise or fall of $\frac{3}{4}$°F. would not occur for at least one hour, the use of a variable element would not be necessary and a more economical solution using on-off controllers would operate every hour or more to respond to the slow change of true load.

WHAT I CLAIM IS:—

1. An environment control system comprising: a recirculating fluid circuit includ-

ing a chamber which defines the environmental space to be controlled and a fan for driving a substantially constant volume flow of gas and/or vapour through the circuit; a close, steady flow, vapour compression refrigeration system having an evaporator with a cooling heat exchange surface in the circuit and a compressor with means for operating it continuously and at a constant speed when the system is operating to maintain a desired chamber condition; and control means responsive to changes in at least one of the sensible heat portion and the latent heat portion of the true load (as hereinbefore defined) and operable independently of the refrigeration system to control the heat portion or portions to which it is responsive in the chamber to any of a plurality of presettable desired values for the or each such heat portion by the controlled addition of heat to the gas and/or vapour, the arrangement being operable such that, on starting-up or upon a change of desired value, the control means will act on the gas an/or vapour in the fluid circuit to drive it to a condition in which the then desired chamber condition is obtained in the chamber, during which action the total load on the refrigeration system and the temperature of the refrigerant will naturally and automatically reach steady state values which are functions of the desired chamber condition by virtue of the effect upon the refrigeration system of the change in inlet conditions to the evaporator produced by the action of the control means upon the gas and/or vapour, the control means then functioning to maintain the preset desired chamber condition without adjustment of the constant volume flow of the gas and/or vapour or of the steady state refrigeration system in which said steady state values of total heat and temperature remain substantially constant regardless of true load changes.

2. A method of controlling the environment in a chamber to any one of a plurality of conditions of at least one of the sensible heat and latent heat, the chamber being in a fluid circuit containing the evaporator of a closed vapour-compression refrigeration system, and wherein, when operating at any one of a plurality of desired environment conditions in said chamber, gas and/or vapour is passed through the circuit with a constant volume flow, the refrigeration system is operated continuously in a steady state with a total load upon it and a refrigerant temperature which are constant and independent of changes in true load (as hereinbefore defined), and at least one of the sensible heat and latent heat of the true load is sensed and controlled by control means independent of the refrigeration system to maintain the desired environment condition by the controlled addition of heat to the gas and/or vapour, whilst, on commencing the control or upon a change of the desired condition, the control means is caused to vary the amount of heat it adds to the gas and/or vapour to drive it to the then desired condition in the chamber, during which action the continuously running refrigeration system naturally and automatically moves towards and reaches a steady state condition corresponding to the then desired chamber condition by virtue of the effect of the changes in the inlet conditions of the evaporator produced by the action of the control means on the gas and/or vapour.

3. A system according to claim 1 or a method according to claim 2, wherein the control means comprises a heating means downstream of the heat exchange surface and upstream of the chamber to maintain a preset dry bulb temperature in the chamber.

4. A system or method according to claim 3, wherein the heating means comprises a bank of electrical heater wires.

5. A system or method according to claim 3, wherein the heating means comprises a hot water, or steam, heat exchange coil.

6. A system according to any one of claims 1 and 3 to 5, or a method according to any one of claims 2 to 5, wherein the control means comprises humidifying means downstream of the heat exchange surface and upstream of the chamber to maintain a preset dew point temperature in the chamber.

7. A system or method according to claim 6, wherein the humidifying means comprises a bank of electrical heater wires.

8. A system or method according to claim 6, wherein the humidifying means comprises a steam jet humidifier.

9. A system according to any one of claims 1 and 3 to 8, or a method according to any one of claims 2 to 8 and having preheater means upstream of the heat exchange surface and downstream of the chamber.

10. A system or method according to claim 9, wherein the preheater means has dry bulb temperature sensing means downstream of the preheater means and upstream of the heat exchange surface and the control means comprises regulating means for regulating the preheater means in accordance with the sensing means.

11. A system or method according to claim 9 or 10, wherein the preheater means comprises a bank of electrical heater wires.

12. A system or method according to claim 9 or 10, wherein the preheater means comprises a hot water, or steam, heat exchange coil.

13. A system or method according to any one of claims 4, 7 and 11, wherein at least

one wire in the bank is connected to a motor driven variable transformer.

14. A system or method according to claim 13, wherein the or each bank includes wires controlled by switches to be placed progressively into or out of circuit when the transformer has been driven to its upper or lower voltage limit respectively.

15. A system according to any one of claims 1 and 3 to 14 or a method according to any one of claims 2 to 14, wherein the control means can be preset for any chamber condition in a predetermined area of the psychrometric chart, the system or method being operable with substantially constant volume flow of gas and/or vapour and substantially constant mass flow of refrigerant and with a substantially constant total heat load upon the refrigeration system to maintain a steady state chamber condition corresponding to any one of the values in said predetermined area.

16. A system or method according to claim 15, wherein the area of the chart extends at least from 55°F to 85°F dry bulb temperature and at least from 40°F to 80°F dew point temperature.

17. A system or method according to claim 15 or 16, wherein the control means and the refrigeration system have substantially the minimum capacity necessary for achieving the chamber conditions of all of said preset values.

18. A system according to any one of claims 1 and 3 to 17, or a method according to any one of claims 2 to 17, wherein an air inlet vent is disposed downstream of the chamber and a spill damper is disposed upstream of the chamber.

19. A system according to any one of claims 1 and 3 to 18, or a method according to any one of claims 2 to 18, wherein the control means has separate setting means, whereby more than one chamber condition can be preset, there being change-over means by which the control means can be changed from one setting to the other.

20. A system or method according to claim 19, wherein there is a timer for operating the change-over means according to a presettable time programme.

21. A system according to any one of claims 1 and 3 to 20, or a method according to any one of claims 2 to 20, wherein the chamber has a plenum by which the gas and/or vapour will be introduced into the chamber by way of at least one perforated plate in the plenum wall.

22. A system according to any one of claims 1 and 3 to 21, or a method according to any one of claims 2 to 21, wherein the chamber has a plenum in its base, by which the gas and/or vapour will be introduced upwardly into the chamber, the upper portion of the plenum being defined by at least one perforated plate.

23. A system or method according to claim 22, wherein the chamber has a register in the side wall at the upper region of the chamber for the exit of gas and/or vapour

24. A system according to any one of claims 1 and 3 to 23, or a method according to any one of claims 2 to 23, wherein a chamber dew point temperature below 32°F is attainable.

25. A system according to any one of claims 1 and 3 to 24, or a method according to any one of claims 2 to 24, when operating under a steady state chamber condition, wherein the gas and/or vapour has a constant volume flow rate; the gas and/or vapour inlet state to the cooling heat exchange surface, the refrigeration system condensing temperature and the refrigerant temperature are substantially constant; the compressor of the condensing unit runs continuously, all the gas and/or vapour circulates past the cooling heat exchange surface; and a thermostatic expansion valve exists in the refrigeration system and constitutes the only automatic controller on the operation of the refrigeration cycle.

26. An environmental chamber system substantially as hereinbefore described with reference to the accompanying drawings.

27. A method of environmental chamber control substantially as hereinbefore described with reference to the accompanying drawings.

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Printed for Her Majesty's Stationery Office by The Tweeddale Press Ltd., Berwick-upon-Tweed, 1969. Published at the Patent Office, 25 Southampton Buildings, London, W.C.2, from which copies may be obtained.

United States Patent Office

3,478,817
Patented Nov. 18, 1969

3,478,817
ENVIRONMENTAL SPACE CONDITIONING CHAMBER

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Filed Apr. 18, 1966, Ser. No. 543,352
Claims priority, application Australia, Apr. 13, 1966, 57,747/66
Int. Cl. F24f 3/14; G05d 22/00
U.S. Cl. 165—21 19 Claims

ABSTRACT OF THE DISCLOSURE

A controlled chamber is described in which humidity and temperature are independently controlled, the control means being such as to react to changes in the conditions within the chamber and in such a manner as to maintain substantially constant the total load upon a cooling system associated with the chamber.

This invention relates to a controlled chamber, that is, a space defined by boundaries within which the properties of dry bulb temperature and dew point temperature can be controlled over a range of settings.

Controlled chambers have been used heretofore to a large extent for plant growth studies, and when used as such are herein termed "phytotrons." However controlled chambers extend to include chambers for both the control and the testing and rating of performance, and are for example used by space scientists, physicists, building scientists, biologists, zoologists, geneticits and engineers. For example they may be used for a controlled animal house, or a space suit using oxygen and water vapor mixtures at low pressures, or for the testing of expansion coils or electronic equipment and with this invention if desired in these uses the controlled settings can be subjected to periodic variations.

This invention is applicable to "chambers" in the broadest sense of the term; the chamber may be an insulated section of duct work, a room, a series of rooms, an animal house, a space suit, a space capsule, a calorimeter or the like.

The application of this invention extends to the general phenomena of heat and mass transfer of any gas and any vapor mixture (including the limiting cases wherein the gas content or vapor content approaches zero) and is not necessarily confined to air/water vapor mixtures.

This invention is applicable to any mixture pressure and is not necessarily confined to barometric pressure at sea level.

Phytotrons and similar chambers have been required to meet numerous specifications of the users. These specifications vary widely from narrow to wide range operating settings; from temperature only to simultaneous temperature and humidity settings; from a single fixed operating condition to an operating condition which automatically changes over to simulate both day and night conditions or other periodicity programming arrangements; from a one hundred percent recirculating system with carbon dioxide injection to systems specifying a fixed percentage of fresh air and spill; from systems operating to broad tolerances to systems requiring close tolerances with minimal deviation between points within the plant occupied space; from systems that are set up laboriously with long periods required for manual adjustment to quick start automatic systems.

Heretofore phytotrons and similar chambers have been operated by systems which have not been able to economically meet the requirements of the users. They have failed to function properly when wide ranges of operating

settings were specified, when automatic change-over between day and night operating settings was specified, when control of both dry bulb and dew point temperature was specified, when fresh air introduced from the outside of the chamber was specified, when close tolerance control was specified, or when automatic start-up procedure was specified.

Existing systems have failed to meet the needs of the agronomist and scientist. This failure has increased as these demands have become more refined. The result has been the use of expensive complex systems, unstable in performance and frequently requiring the attention of skilled operators and loss of considerable time to set up and to maintain conditions within desired limits.

The main object of this invention is to provide a controlled chamber which is more economical and stable than heretofore in meeting the user's specifications.

The invention may be said to consist of a controlled chamber including a fan to draw a gas, vapor or gas/vapor mixture through the chamber, a condenser and evaporator unit containing refrigerant and operable on a refrigeration cycle having a cooled heat exchange surface in the path of flow of the gas, vapor or gas/vapor mixture, heating means in said path, and control means, characterized in that said control means are responsive to true load changes and that the total loads on the refrigeration system are maintained substantially constant for each control set point.

This invention is capable of providing a simpler economically feasible system of engineering for a controlled chamber which will be capable of maintaining the chamber to the desired limits of temperature and humidity conditions (each or together) and within the desired range (narrow or broad) even though the mixture controlled is subject to random rates of heat and mass transfer.

The invention can also provide control means, if desired, to automatically change the temperature and humidity according to some desired program, thus permitting automatic periodicity control within a range of settings.

This further feature is desirable when the chamber is to be used as a phytotron for the study of plant growth. The desired change for a phytotron is to pass between simulated "day" and "night" conditions.

This further feature if desired, when it is used as a means to control the inlet conditions of gas vapor mixtures to heat and mass transfer surfaces for the purpose of testing and rating of performance, may be combined with additional programming facilities. For example, when the chamber is to be used for testing and the rating of the performance of direct expansion coils, the automatic periodicity control if desired may not only take the form of varying the temperature and humidity of the gas/vapor mixture at the inlet to the direct expansion coil placed in the chamber but may include the varying of the capacity of the compressor or varying automatically the constant rate of gas/vapor flow. Thus test data could be indicated or recorded downstream of the direct expansion coil, giving the coil performance not only at a standard condition but over a wide range of operating conditions.

I have found that this invention can be best described by introducing a division of the "total loads" on the refrigeration system into "true loads" and "non-true loads."

It is possible to devise an engineering system wherein the controls are responsive to true load changes only, and to arrange total loads on the refrigeration system to be substantially constant. The characterizing feature of this invention is therefore, as said above, that the control means are responsive to true load changes and total loads are maintained substantially constant. If the non-true

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the system may need revision. Thus for example if a direct expansion coil is to be tested for varying conditions, this may take the place of the coil 129 and the cabinet 100 may be replaced and relocated to be a simple insulated duct extending fore and aft of the coil 129 which is to be tested. The preheater and its controls, the fresh air and spill vents and accessories would no longer be required. A system as described in the above embodiment however could be automatically programmed to change between different preset inlet conditions along the constant compressor speed (see FIG. 3) (or other compressor capacity control means) and with varying set points upstream of the coils 129. As the capacity of the system reaches a minimum or maximum condensing unit capacity a programmer may on say high or low pressure compressor cut out, change to another compressor speed and again repeat the same pattern of programmed inlet conditions as for the original compressor speed setting. Instrumentation measuring air flow and leaving coil dry bulb and dew point temperatures will establish the coil performance for standard stipulated conditions and also for any desired operating inlet condition within a broad range. When used for rating and testing the direct expansion coil would be installed in a manner which would facilitate quick removal and replacement in accordance with good engineering practice.

On the other hand, usually control chambers used for testing materials or performance of a complete self-contained system such as a fan-coil unit would simply replace the phytotron chamber with one suited for the needs of the test. The dry bulb and dew point sensors would always be located at the point where the gas vapor mixture is controlled. For the testing of heat and mass transfer surfaces the sensors would be located directly upstream of the test surface, with the required instrumentation giving the performance downstream of the test surface.

This invention can combine the use for testing of direct expansion coils for varying entry conditions by controlling the inlet condition to the direct expansion coils as described above with its use for controlling a separate cabinet adapted for testing self-contained units, building materials, electronic equipment all together in one installation. This combination would simply require a procedure of activating only those sensors of the control system in the area being controlled. This combination would serve as a very important tool to scientists and university laboratories. It could not only be material testing and heat transfer surface testing system but could be effectively used for demonstrating the principles of refrigeration and for laboratory experiments in applied thermodynamics.

The invention can be satisfied by the use of devices and components other than described herein. For example, an analogous pneumatic control system is wholly applicable in lieu of the electric control system described in particular. Heating components for the preheat, reheat and humidifying functions may be hot water coils, steam coils, steam jet humidifiers et cetera. It need not be electric as described for the above embodiment.

The specification of the user of this invention with reference to automatic change-over requirements can be very varied in programming, including rising or falling temperature or humidity or compressor capacity. It need not be simply a change-over between day and night conditions as described for a particular embodiment.

The heat exchange surface need not be limited to a finned evaporator coil, but circulation of chilled water or brine to a heat exchange coil may be employed as an adjunct to the evaporation system.

The specifications of the user may allow for a selection of components so that the minimum refrigerant temperature is high enough to avoid frosting and therefore the preheater of the embodiment may be omitted. (For example a very narrow range user's requirement in high temperature and humidity region may result in refrigerant temperatures being high enough to avoid occurrence of frosting.)

The reduction of the non-true load to a practical minimum within the terms of the engineering system of this invention, makes this invention economically feasible for narrow range systems and systems requiring control of temperature only. For example should the user be interested in a temperature-humidity controlled chamber operative in the higher range of humidities, say above 50% relative humidity rather than the wide range within the bordered area of FIG. 2, all other requirements of the user being identical to this embodiment presented herein in detail, a smaller capacity system with lower running costs would result. Inlet condition 1 of FIG. 2 determines the basic for selecting the refrigeration system of this embodiment, the limiting factor in this case to meet the humidifying true load of the system. For this example, where the range of humidities required by the user is above 50% relative humidity inlet condition number 6 of FIG. 2 would determine the basis for selecting the refrigeration system and would have a refrigeration capacity of approximately one fourth of the system described in this embodiment.

The variable element and fixed elements of each bank of heaters are a specific example for this embodiment and the specific user's requirements establishing this embodiment. It is possible to have a user's requirement for a phytotron or for a controlled chamber wherein the tolerances are very broad, wherein less sensitivity controllers may warrant, due to the slow rate of true load change, to use on-off control. For example if the controllers have a sensitivity which will result in a response only after, say, a $\pm 3/4^\circ$ F. change has occurred in the cabinet and if the change in true load is such that a rise or fall of $3/4^\circ$ F. would not occur for at least one hour, the use of a variable element would not be necessary and a more economical solution using an on-off controllers would operate every hour or more to respond to the slow change of true load.

What I claim is:

1. An environment system controlling heat level of an environmental space which contains both sensible and latent heat,

a closed, steady flow refrigeration system of the vapor compression type having an evaporator unit and a compressor, said compressor including means for being operated continuously and at a constant speed when the system is being used for maintaining an environment to a preset level of at least one of the sensible heat portions or the latent heat portion of its total heat level,

a chamber which defines the environmental space to be controlled,

a gas/vapor circulating system for flowing gas/vapor in the system over a heat exchange surface of the evaporator unit and through said chamber, and including a fan means to move gas/vapor through said circulating gas/vapor system and over said heat exchange surface at a steady flow rate,

heating means positioned external to the space defined by said chamber and in a path of flow of said circulating gas/vapor system,

adjustable control means operatively associated with said chamber responsive solely to true load changes which consist of variations in at least one of the sensible heat levels or latent heat level in said chamber,

said control means being operable independently of said refrigeration system to control at least one of said heat portions to any one of a plurality of presettable desired values, by the controlled addition of heat from said heating means to the circulating gas/vapor

so that environmental conditions for the chamber are detected and maintained without any adjustment of said steady flow refrigeration system, the total load on the refrigeration system and the temperature of the refrigerant in the evaporator for any given set point being independent of the variation of the true loads on the system, and the total load on the refrigeration system and the temperature of the refrigerant in the evaporator being a constant value that is a function of the location of the set point within the range.

2. The improvement according to claim 1 wherein the heating means include a preheater disposed upstream of the cooling heat exchange surface.

3. A controlled chamber according to claim 2 wherein the evaporator is of such capacity that the heat exchange surface dehumidifies the gas/vapor mixture when passing it to a moisture content equal to or less than the moisture control set point of the chamber.

4. A controlled chamber according to claim 3 wherein the cooling and dehumidifying, and the heating and humidifying surfaces are of minimum capacity for the range of set points within the psychometric limits of operation.

5. A controlled chamber according to claim 3 further including a reheater disposed downstream of the cooling heat exchange surface, the reheater constituting temperature control means by which dry bulb temperature operating conditions are maintained over a range of dry bulb temperatures.

6. A controlled chamber according to claim 3 further including a humidifier disposed downstream of the cooling heat exchange surface, the humidifier constituting humidity control means by which the dew point temperature (humidity ratio) operating conditions are maintained over a range of dew point temperatures.

7. A controlled chamber according to claim 3 further including a preheater disposed upstream of the cooled heat exchange surface, the preheater being of sufficient capacity to supply sufficient temperature to the gas/vapor mixture to avoid frosting.

8. A controlled chamber according to claim 2 wherein the chamber is defined by the walls of a cabinet, a plenum in the base of the cabinet and a register in the side walls near the top of the cabinet, the upper portion of the plenum being defined by at least one perforated plate, the gas, vapor or gas/vapor mixture entering the cabinet through the plenum and leaving the cabinet from the register.

9. The improvement according to claim 5 wherein the gas/vapor mixture has a constant volume flow rate; the gas/vapor mixture inlet state to the cooling heat exchange surface, the refrigeration system condensing temperature, the refrigerant temperature, and the reheated gas/vapor mixture downstream of the evaporator does not vary during any one operating setting; the compressor of the condensing unit is arranged to run continuously; all the gas/vapor mixture circulates past the cooling heat exchange surface, a thermostatic expansion valve exists in the refrigeration system and constitutes the only automatic controller on the operation of the refrigeration cycle; and a uniform air flow pattern passes through the chamber.

10. The improvement according to claim 1 wherein sensible and latent heat levels for said chamber are separately controlled and maintained, and including separate sensors and separate control means for detecting and adjusting temperature and humidity.

11. A controlled chamber comprising a cabinet means and an air conditioning unit provided with a fan arranged to continuously circulate a gas/vapor mixture through the cabinet means, a humidifier in the path of gas/vapor flow, a reheater also in said path, the air conditioning unit having a vapor compression type of refrigeration system including an evaporator coil which is disposed to intersect the gas/vapor mixture being circulated through the cabinet means, the evaporator coil being upstream of the reheater, a preheater upstream of the evaporator, a dry bulb temperature sensing element disposed in the path of gas/vapor flow between the preheater and the evaporator to thereby sense the temperature of the gas/vapor mixture downstream of the preheater, preheater control means coupled to the dry bulb temperature sensing element to thereby regulate the temperature of the gas/vapor mixture downstream of the preheater, reheat control means adjustable over a wide range of control set points to set a temperature condition to be maintained in said cabinet, a dry bulb temperature sensor operatively associated with the controlled cabinet means and coupled to the reheat control means, a reheater controlled by the reheat control means to thereby control dry bulb temperature within the cabinet means, a dew point sensor operatively associated with the cabinet means, and dew point control means coupled to the dew point sensor to thereby control the dew point temperature within the cabinet means, said dew point control means being adjustable over a range of control set points.

12. A controlled chamber according to claim 11 further comprising a motor driven variable transformer, the preheater, the reheater and the humidifier each comprising a bank of electrically heated conductors, each bank of electrically heated conductors being provided with at least one element controlled by the motor driven variable transformer.

13. A controlled chamber according to claim 12 further comprising a plurality of switches, each bank of electrically heated conductors including sections which are controlled by the switches thereby being placed progressively into or out of circuit after the transformer of the respective bank is driven to its upper or lower voltage limit respectively.

14. A controlled chamber according to claim 11 wherein the refrigeration system includes a compressor, the compressor being arranged to run continuously.

15. A controlled chamber according to claim 13 further comprising clock programming means, the preheat control means, reheat control means and dew point control means each having day and night settings, the clock programming means selecting their day and night set points consecutively.

16. A controlled chamber according to claim 15 further comprising an air plenum across the lower portion of said cabinet and a light section containing one or a plurality of lamps across its upper portion.

17. A controlled chamber according to claim 11 further comprising an air inlet valve disposed upstream of the heater means and a manual spill damper disposed downstream of the fan.

18. A process for separately controlling temperature and humidity conditions, in an enclosed environment which allows a simulation of conditions such as climatic, plant growing conditions over wide ranges, comprising the steps of:

continuously circulating a gas/vapor mixture through an enclosed environment, and

simultaneously intersecting the gas/vapor flow at an evaporator unit of a continuously operating vapor compression type of refrigeration system in a manner such that during start-up of the total system and during change-over the gas/vapor and the refrigeration flow systems naturally and automatically will be moving towards and reaching a steady flow state wherein the total load on the refrigeration system and the temperature of the refrigerant in the evaporator for any given set point will be independent of the variations of the true loads on the system, whereby following the establishing of steady flow conditions for the gas/vapor and refrigerant paths across the evaporator, the system arrangement and preselection will result in a cooling of the gas/vapor passing through the evaporator to a level which is sufficiently low to anticipate all sensible heat gains of the system and which, at the same time, will result in a dehumidifying of the gas/vapor passing through the evaporator unit to a level which is sufficient to anticipate all latent heat gains of the system,

thereupon maintaining a set dry bulb temperature in the enclosed environment by controlling only a reheating means which intersects the gas/vapor flow and which is responsive only to true load variations in the sensible heat load, and

simultaneously maintaining a set humidity in the enclosed environment by controlling only a humidifier which intersects the gas/vapor flow and which is responsive only to true load variations in the latent heat load.

19. A process for controlling heat load in an environment to any one of a plurality of preset conditions by controlling at least one of the sensible heat portion and the latent heat portion of the total heat level of a chamber to allow a simulation of conditions such as climatic, plant growing conditions over wide ranges, comprising the steps of:

continuously circulating a gas/vapor mixture through an enclosed environment, and

simultaneously intersecting the gas/vapor flow at an evaporator unit of a continuously operating vapor compression type of refrigeration system in a manner such that during start-up of the total system and during change-over the gas/vapor and the refrigeration flow systems naturally and automatically will be moving towards and reaching a steady flow state wherein the total load on the refrigeration system and the temperature of the refrigerant in the evaporator for any given set point will be independent of the variations of the true loads on the system, whereby following the establishing of steady flow conditions for the gas/vapor and refrigerant paths across

the evaporator, the system arrangement and preselection will result in a reduction of the energy level of the gas/vapor passing through the evaporation to a level which is sufficiently low to anticipate all heat gains for at least that portion of the total heat level which it is desired to control, said heat level being at least one of the sensible heat portion and the latent heat portion, thereupon maintaining at least one of the following:

(1) a set sensible heat level in the enclosed environment by controlling heating means which intersect the gas/vapor flow and which is responsive only to the load variations in the sensible heat load, or

(2) a set latent heat level in the enclosed environment by controlling a humidifying means which intersects the gas/vapor flow and which is responsive only to true load variations in the latent heat load.

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U.S. Cl. X.R.

236—44

(51) Int. Cl.: F 24 f, 5/00

BUNDESREPUBLIK DEUTSCHLAND

DEUTSCHES PATENTAMT

(52) Deutsche Kl.: 36 d, 1/56

(10) (11) # Patentschrift 1 604 253

(21) Aktenzeichen: P 16 04 253.8-16 (N 28415)

(22) Anmeldetag: 20. April 1966

(43) Offenlegungstag: 20. April 1972

(44) Auslegetag: 12. Oktober 1972

(45) Ausgabetag: 10. Mai 1973

Patentschrift stimmt mit der Auslegeschrift überein

Ausstellungspriorität: —

(30) Unionspriorität

(32) Datum: 20. April 1965

(33) Land: Australien

(31) Aktenzeichen: 57747-65

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(54) Bezeichnung: Einrichtung zur Erzeugung bestimmter Feuchtigkeits- und/oder Temperaturbedingungen in einer Untersuchungskammer

(61) Zusatz zu: —

(62) Ausscheidung aus: —

(73) Patentiert für: Shaw, Allan, Adelaide (Australien)

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FEDERAL REPUBLIC OF GERMANY PATENT

APPENDIX VIID p.1

Patentansprüche:

1. Einrichtung zur Erzeugung bestimmter Feuchtigkeits- und/oder Temperaturbedingungen in einer Untersuchungskammer, durch welche hindurch ein im wesentlichen konstanter Strom eines Gases, Dampfes oder Gasgemisches geleitet wird, mit einem mittels einer Wärmeaustauschfläche auf diesen Gas- bzw. Dampfstrom einwirkenden Kühlsystem, Vorrichtungen zum Aufheizen und zum Befeuchten des Gas- bzw. Dampfstromes und mit einer Regeleinrichtung zur Regelung der genannten Feuchtigkeits- und/oder Temperaturbedingungen, dadurch gekennzeichnet, daß diese Regeleinrichtung (**122**, **123**, **139**, **140**) sowohl auf die latente als auch auf die freie (fühlbare) Wärme in dem Gas- bzw. Dampfstrom innerhalb der Untersuchungskammer (**100**) ansprechende Fühlermittel (**122**, **123**) aufweist und von dem Kühlsystem (**126**, **129**) unabhängig ist und daß mittels der Regeleinrichtung der zu regelnde Zustand, welcher ein Bereich von Feuchtigkeits- und/oder Temperaturwerten ist, durch zusätzliche Wärmezufuhr bzw. -abfuhr (**131**, **136**) erzeugbar ist, während die Gesamtlast des Kühlsystems einen konstanten Wert beibehält bzw. bei Sollwertänderungen bezüglich des zu regelnden Zustandes in einen entsprechenden anderen konstanten Wert übergeht.

2. Einrichtung nach Anspruch 1, dadurch gekennzeichnet, daß die Wärmeaustauschfläche ein Verdampfer (**129**) ist und daß die Heizvorrichtung einen stromaufwärts von dieser Wärmeaustauschfläche gelegenen Vorwärmer (**128**) aufweist.

3. Einrichtung nach Anspruch 1 oder 2, dadurch gekennzeichnet, daß die Kapazität des Verdampfers (**129**) so gewählt ist, daß an seiner Wärmeaustauschfläche eine Entfeuchtung des vorbeiströmenden Dampf-Gas-Gemisches stattfindet, durch welche dieses auf einen dem eingestellten Sollwert der Untersuchungskammer (**100**) gleichen oder unter diesem Wert gelegenen Feuchtigkeitsgehalt gebracht wird.

4. Einrichtung nach einem der Ansprüche 1 bis 3, dadurch gekennzeichnet, daß stromabwärts von der Wärmeaustauschfläche des Verdampfers (**129**) ein Nachheizer (**131**) angeordnet ist, dessen Heizflächen für den Sollwertbereich innerhalb der psychrometrischen Betriebsgrenzen (Fig. 2) nur mit minimaler Energie zu versorgen sind, und daß dieser Nachheizer eine Temperatursteuereinrichtung zur Erzeugung bestimmter Trockentemperatur-Betriebsbedingungen (**123**) innerhalb eines Trockentemperatur-Einstellbereiches dient.

5. Einrichtung nach einem der Ansprüche 1 bis 4, dadurch gekennzeichnet, daß zur Erzeugung bestimmter Werte des Wassergehaltes innerhalb eines Taupunkttemperatur-Einstellbereiches (**162**, **163**) eine Befeuchtungsvorrichtung (**135**, **136**) dient, welche stromabwärts von der Wärmeaustauschfläche des Verdampfers (**129**) angeordnet ist.

6. Einrichtung nach Anspruch 2, dadurch gekennzeichnet, daß die Kapazität des Vorwärmers (**128**) so gewählt ist, daß dieser dem vorbeiströmenden Dampf-Gas-Gemisch zur Verhinderung der Reifbildung durch Erzeugung konstanter Betriebsbedingungen innerhalb des Kühlsystems ausreichend Wärme zuführt.

7. Einrichtung nach einem der Ansprüche 2 bis 6, dadurch gekennzeichnet, daß ein auf die stromabwärts von dem Vorwärmer (**128**) auftretende Trockentemperatur des Dampf-Gas-Gemisches ansprechendes Trockentemperatur-Meßorgan (**130**) vorgesehen ist, welches mittels eines Vorwärmreglers (**138**) auf diesen Vorwärmer im Sinne einer Regelung der Temperatur des Dampf-Gas-Gemisches stromabwärts von dem Vorwärmer einwirkt, und daß die den Fühlermitteln (**122**, **123**) zugeordneten Teile der Regeleinrichtung ein Taupunkt-Regler (**140**) und ein Nachheizregler (**139**) sind.

8. Einrichtung nach einem der Ansprüche 2 bis 7, dadurch gekennzeichnet, daß der Vorwärmer (**128**) der Nachheizer (**131**) und die Befeuchtungsvorrichtung (**135**, **136**) jeweils eine Anordnung elektrischer Heizleiter (z. B. **149**, **151**) aufweisen und daß jede dieser Heizleiteranordnungen jeweils mindestens ein Heizelement (z. B. **149**) enthält, welches jeweils von einem motorbetätigten (**147**, **154**, **155**) Regeltransformator (**148**) gesteuert wird (Fig. 4).

9. Einrichtung nach Anspruch 8, dadurch gekennzeichnet, daß jede der Heizleiteranordnungen (**149**, **151**) Teilabschnitte (**151**) aufweist, die mittels Schalter (**150**) nacheinander zu- bzw. abschaltbar sind, nachdem der jeweilige Transformator (**148**) seine obere bzw. untere Grenzspannung erreicht hat (Fig. 4).

10. Einrichtung nach einem der Ansprüche 7 bis 9, dadurch gekennzeichnet, daß der Vorwärm-Regler (**138**), der Nachheizregler (**139**) und der Taupunktregler (**140**) Eingabeeinrichtungen (**158**, **159**, **160**, **161**, **162**, **163**) zur Einstellung der Tag- und Nachtsollwerte sowie eine Zeitsteuereinrichtung (**141**) enthalten, mittels welcher diese Sollwerte jeweils aufeinanderfolgend einstellbar sind (Fig. 1 und 4).

11. Einrichtung nach einem der Ansprüche 2 bis 10, dadurch gekennzeichnet, daß stromaufwärts von der Heizvorrichtung (**128**) ein Zuluft-Einlaß (**142**) angeordnet ist und daß stromabwärts von einem den Strom des Dampf-Gas-Gemisches erzeugenden Gebläse (**134**) ein von Hand einstellbares Auslaßventil (**143**) angeordnet ist (Fig. 1).

12. Regeleinrichtung nach einem der Ansprüche 1 bis 11, dadurch gekennzeichnet, daß der auf die Wärmeaustauschfläche auftreffende Dampf-Gas-Gemischstrom, die Kühlsystemkondensationstemperatur, die Kältemitteltemperatur und der aufgeheizte Dampf-Gas-Gemischstrom stromabwärts vom Verdampfer (**129**) für einen bestimmten Betriebssollwert jeweils konstant sind, daß ferner der Kompressor des Kühlsystems fortwährend im Einschaltzustand ist, daß weiter der gesamte Dampf-Gemisch-Strom an der Wärmeaustauschfläche vorbeigeführt ist und daß schließlich als einziges automatisches Regelorgan in dem Kühlsystem ein thermostatisches Entspannungsventil vorgesehen ist.

THE UNIVERSITY OF ADELAIDE and ALLAN SHAW

application for Commonwealth Provisional Protection No. PD8226 dated 28th March, 1979 covering the above invention in the name of ALLAN SHAW and THE UNIVERSITY OF ADELAIDE including payment of Patent Office fees and all usual costs and charges — — —

COMMONWEALTH OF AUSTRALIA

PATENTS ACT 1952-1969

PROVISIONAL SPECIFICATION FOR THE INVENTION ENTITLED:

"METHOD OF AIR CONDITIONING"

This invention is described in the following statement:

PROVISIONAL AUSTRALIAN APPLICATION ARISING FROM FINDINGS OF SECTION 11

This invention relates to a method of air conditioning and has as its main object the conservation of energy as well as the improvement of air conditioning system performance.

Removal of humidity (latent heat) from air by chemical means is already known and this forms no part of the invention. For many reasons dehumidification by passing air over a low temperature extended heat exchange surface is preferred, but throughout the full range of air inlet temperature from $4^{\circ}C$ to $45^{\circ}C$ and the outlet temperature from $-2^{\circ}C$ to $44^{\circ}C$, and the range of inlet moisture content from 0.004 to 0.022 and the outlet humidity ratio from 0.004 to 0.021 (kg. of moisture per kg of dry air) air conditioning has not been achieved without in many instances, overcooling the air in order to offset the humidity load. This is because present practice has not been able to obtain from the air conditioning system a performance which achieves one of the major aims of an air conditioning system, that is a coil condition curve which is compatible with the load ratio line. The methods which have been adopted heretofore have frequently made it necessary to overcool to sufficiently dehumidify the air to offset the latent heat loads, resulting with reference to a psychrometric chart, in the outlet end of the coil condition curve being at a dry bulb temperature which is less than the required temperature at the inlet condition of the load ratio line. Frequently reheating is employed to rectify these conditions.

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frequently it may be recommended to use a coil condition curve during full load operation which has a higher negative tie line slope than is necessary, though resulting in an acceptable effective temperature. Thus part load conditions would be further improved.

A consideration of the above embodiment will reveal the following:

1. Air stream velocity and Reynolds number of the coil complex is one of the major operative factors in determining the coil condition curve of a dehumidifier.

2. As the velocity of an air stream and the Reynolds number over a dehumidifier surface varies from high to low so does the slope of the coil condition curve vary from shallow to steep.

3. As the velocity of an air stream and the Reynolds number over a dehumidifier surface varies from high to low so does the curvature of the coil condition curve vary from a considerate curvature towards that of a straight line.

4. The assumed straight line characteristic of coil condition curves by industrial methods as described above does not hold for the range of air velocities employed in air conditioning applications, (3.5 metres per second (700 feet per minute) down to 2 metres per second (400 feet per minute).

5. Conventional design approach used in air conditioning and climate simulation can result in large energy penalties and failure to attain desired conditions for full load and/or part load operation when dehumidification is required.

6. Conventional design approach towards spacial arrangements where dehumidification is required must be re-examined in the light of energy savings due to reduced cooling and reheating, reduced fan power, reduced size of refrigeration equipment and cooling tower their piping, conduit and accessories and their reduced weights and costs.

7. From an examination of part load conditions frequently present in conventional air conditioning applications there is a strong case pointing to the use of variable air flow rates varying proportionally with the size of the loads.

8. From an analysis of dehumidifier performance a new method of air conditioning has been derived.

9. This system can be implemented by way of a simplified method of selecting the best dehumidifier applicable to a design problem as developed through a family of curves or some equivalent method such as tabulation.

10. A major objective in air conditioning design is the dehumidifier coil selection. Apropos the total air conditioning problem, the aim is to find the extended surface which has a coil condition curve that best fits the load ratio line. Maximum energy conservation is then obtained.

11. In an air conditioning complex running costs may far outweigh initial costs as a criterion.

12. In dehumidifier selection it is often detrimental to achieving the major criteria enumerated above to select air velocity coils, in the order of 2m/s or greater even though such coils have improved heat transfer performance.

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13. In determining design for air conditioning application the entire philosophy of coil selection for air conditioning with emphasis on increased surface density for gas-liquid heat exchangers has been reassessed for a dehumidifier. The increase of fin to primary tube surface may result in optimizing heat transfer but will reduce mass transfer in relation to heat transfer. The increase of air flow velocity will increase heat transfer. However, again, it will reduce mass transfer in relation to heat transfer.

14. In determining dehumidifier design for air conditioning application, a new system is recommended with the face velocities different and with the coil surfaces characterised by a lower range of Reynolds number than presently used in existing air conditioning practice.

Dated this 28th day of April 1979.

THE UNIVERSITY OF ADELAIDE and
ALLAN SHAW,

By their Patent Attorneys,
R.K. MADDERN & ASSOCIATES

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