THE PREPARATION OF A COMMON TECHNIQUE FOR SELECTING COUNTERFLOW AND CROSSFLOW COOLING TOWERS.

BY

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A thesis submitted to the University of Aston in Birmingham as a requirement for the Degree of Master of Science.

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July 1971.

S U M M A R Y

The industrial cooling tower is a device for cooling water by energy transfer into a passing air stream. Such a device is mechanically simple but analysis and selection are relatively difficult because of the many possible combinations of operating conditions. For these latter reasons a comprehensive selection chart is fundamental to the correct sizing of cooling towers. Ideally it is the purpose of the selection chart to represent all probable cooling duties whilst remaining easy to use. In practice more than one chart is necessary to satisfy these requirements. Unfortunately, few detailed selection techniques have been published and no standard method of presentation formulated, although most attempts have been based upon the Merkel Method. Some Authors have used the development of the Merkel Method in the form of the Merkel Diagram and its derivatives to plot selection curves. Others, mainly within the cooling tower industry use the Merkel Method in combination with their own techniques of presentation. For evaluation of the Merkel Method, knowledge of the bulk air enthalpy and the equilibrium enthalpy temperature relationship is required.

It is the object of this work to consider the published cooling tower theory and selection techniques, to explore the theoretical analysis of cooling towers and ultimately to propose easily used common techniques for presenting selection data for counterflow and crossflow cooling towers. Each method is considered for its accuracy, scope and usefulness with particular reference to its day-to-day use in industry. A short section of the work has been devoted to surveying the enthalpy temperature relationships which appear in the literature as well as developing

several new ones. The relationships were considered for accuracy over the desired temperature range and the time required for computation and as a result one of the available relationships and also one of the newly developed ones were selected for use in the computer program.

To obtain the large amount of information necessary to plot the selection charts, computing methods were used to evaluate the enthalpy temperature relationships, the energy balance and driving force equations and hence the Number of Diffusion Units (NDU).

ACKNOWLEDGEMENTS.

The author would like to express his thanks to the following:

Mr. A.R. Cooper for his invaluable help and supervision throughout the period of this research.

Dr. J. Kirkaldy for his advice and encouragement.

Carter Thermal Engineering Limited for their support. Their interest has been appreciated.

To the Computer Center for their service and advice.

Mr. N. Roberts and the technical staff of the Chemical Engineering Department for their assistance and co-operation.

My wife, Miss S. A. Edwards and Mrs. J. Griffiths for their part in preparing this thesis.

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INTRODUCTION

In the past 30 years the rapid growth of industry, particularly the oil and chemical industry and more recently the widespread adoption of air conditioning systems has resulted in a great increase in demand for water cooling units. This upward trend still continues particularly as conservation of water is now a matter of economics and national concern, in many cases justifying investment in water cooling equipment.

Increasing cooling tower demand is also connected with a change in the type of equipment specified. Spray ponds and natural draught cooling towers have long been superseded by mechanical draught towers in all, except the largest applications where natural draught towers still remain economic. Mechanical draught towers use fans to increase their air throughput, and the corresponding higher heat and mass transfer rate brings about significant reductions in the tower size. The use of fans also allows high pressure drop packings to be used which in turn give increased air water contact resulting in physically smaller towers. The smaller towers require lower pumping heads which with the development of more efficient fans and tower packings has led to lower overall power requirements. It is the mechanical draught cooling tower with particular reference to equipment having a counterflow or crossflow pack arrangement which forms the subject of discussion in this work. Until 1955 the counterflow arrangement predominated but recently, the crossflow tower has become more widely accepted.

With the widespread use of cooling towers has come the need to provide a ready means of selecting and comparing the performance of the many types of tower packings available, so that the true economics based upon both capital cost and running cost can be quickly assessed.

To satisfy the needs of the user and supplier, selection charts for predicting cooling tower performance for a multitude of duties should be both comprehensive and easy to use. In the absence of standardised methods of presenting this data most tower manufacturers keep their selection technique secret. Some do publish performance charts but these in the main are of limited application because of their sales orientation.

Experimental tests have been carried out on cooling tower packs and a large quantity of data obtained enabling empirical formulae to be developed for relating the mass transfer coefficient to the packing characteristic represented by the air and water flow rates. It is not intended to discuss these formulae in the present work which is solely concerned with the NDU integral for providing results which can be used in the preparation of suitable selection charts, for any type of packing including those mentioned above. Furthermore it is not the purpose of this work to introduce a new cooling tower theory. In fact, most workers have long adopted the basic theory first developed by Merkel; it is their individual techniques for analysing and presenting performance data which is of interest.

The various methods available are reviewed to establish those which can be used with ease and reliability by persons whose knowledge of cooling towers and their operating principles, is limited.

In trying to arrive at information that has wide application, original methods of presenting the selection data have been developed.

The differential equations, when first developed by Merkel for representing the counterflow cooling tower process and as used by many other authors since, were originally solved using graphical techniques. Recently the use of the computer has enabled more accurate and detailed information to be obtained.

It is important to note that the mechanism of overall energy transfer is the same for both the counterflow and the crossflow process. The difference between the processes arises from the different temperature and enthalpy distributions within the cooling tower pack. For instance, the analysis of a counterflow tower is a one-dimensional problem because the water and air conditions are considered functions of only their vertical position in the tower, for given flow rates and energy transfer coefficients. In contrast the crossflow tower must be treated as a two-dimensional problem because the horizontal air flow results in temperature variations across the tower as well as over its height. Thus the principles applied to the counterflow system have in recent years been applied to crossflow towers where the crossflow process can be represented by partial differential equations.

The process equations for both the counterflow and crossflow cooling towers express the energy balance and the enthalpy difference driving force between the air and water phases. The advent of computers has enabled these equations to be evaluated easily and quickly.

In order that the driving force can be found and the differential equations analysed, the relationships for the enthalpy of the saturated water film and the enthalpy of the bulk air have to be established. Equilibrium at the air-water interface is assumed and hence the enthalpy of the saturated water film is represented by an equilibrium line which can be expressed empirically or in the form of a fundamental equation made up of two principal components, namely the sensible heat of air and the enthalpy of water vapour. The many derivations of both types of equation are considered, to establish the one most suited to cooling tower analysis by computer. Desirable features of this equation include sufficient accuracy for the temperature range $50^{\circ}F$ to $135^{\circ}F$ and a term which enables variations in atmospheric pressure to be accommodated. The selected enthalpy temperature equation is then used for evaluation of the equations representing the cooling tower process.

The content of the thesis is divided into six principal subjects for discussion, namely:

i) Introduction	
ii) Literature Survey	Chapters One and Two
iii) The Enthalpy Temperature Relationship	Chapters Three and Four
iv) Counterflow Selection Technique	Chapters Five to Eight
v) Crossflow Selection Technique	Chapters Nine to Twelve
vi) Conclusions	Chapter Thirteen

CHAPTER ONE

LITERATURE SURVEY : ENTHALPY TEMPERATURE RELATIONSHIP

The first requirement for the study of any process of direct contact between air and water is the phase equilibrium line. This will be referred to as the enthalpy temperature relationship. For the concise statement of enthalpy, four conditions need to be defined for the working fluid. These are the pressure, temperature and physical state of the fluid in question and the choice of datum.

The curve relating the equilibrium enthalpy condition of a system of air water contact is a non-linear function of the water temperature. The enthalpy temperature relationship is generally expressed as a fundamental equation or an empirical equation, the latter being represented as an exponential function or a polynomial expression.

A simple exponential of the form :

$$h_L = \exp f(t_L) \tag{1.1}$$

is used by Mikyska and Reinisch (13) who found it suitable for showing how the enthalpy can be calculated from temperature. This equation is accurate only for short temperature ranges, say 70°F to 90°F. Suitable accuracy over a longer range can be obtained from the relationship used by Fuller (1) :

$$h_L = \exp(\text{constant} + f(t_L)) \tag{1.2}$$

For accuracy over even greater range, Gardner (14) and (15) expressed the enthalpy equation in terms of a linear relationship combined with an exponential relationship :

$$h_L = E + Bt + F\exp f(Dt_L) \tag{1.3}$$

It is interesting to note that this equation is a further development of the one used by Mikyska and Reinisch.

Polynomial expressions of the second order are suggested by Zivi and Brand (2) :

$$h_L = \varepsilon t_L^2 + \beta t_L + \gamma \tag{1.4}$$

but these are found to apply only for short temperature ranges.

During the course of the present work higher order polynomials using the forward backward difference formulae have been developed from tabulated values.

The fundamental equation linking the specific enthalpy of saturated air and its temperature can be expressed for 1 lb sample of dry air as :

h = sensible heat of air and enthalpy of water vapour present. (1.5)

The many authors who considered the fundamental equation, applied their own assumptions and approximations to the final details of the equation. The equations can be considered as three separate parts, namely :

a) Sensible heat of 1 lb dry air

b) Mass of water vapour present in the 1 lb of dry air

c) Enthalpy of vaporisation plus the rise in sensible heat of the vapour in the 1 lb of dry air.

All the fundamental equations, either directly or indirectly, are based upon a convenient enthalpy datum, namely zero Fahrenheit ($0^{\circ}F$) or zero Centigrade ($0^{\circ}C = 32^{\circ}F$). The former is the datum commonly used in the United States whilst the $32^{\circ}F$ datum is that generally employed in the United Kingdom. The different choices of datum cause significant differences in the actual enthalpy values. The more important enthalpy difference, commonly referred to as driving force is not influenced significantly. Nevertheless, care must be taken when using enthalpy tables, to ensure that they are based upon the appropriate temperature datum. Both Lichtenstein (3) and Zahn (12) use a zero Fahrenheit datum for the enthalpy temperature relationship. The work in this paper has been based on the $32^{\circ}F$ temperature datum.

As shown above the fundamental equation consists of two specific heats, a latent heat and superheat term. Zahn shows that the specific heat of dry air can be represented by a second order polynomial function of temperature but this specific heat is often taken as a constant value for cooling tower application.

The specific heat of water vapour is also considered a constant by many workers: it is given the value 0.45 by Gurney and Cotter (7), McKelvey and Brooke (10) and the British Standard (5). The appropriate summation of these specific heats gives the specific heat of moist air commonly called the humid heat.

The Smithsonian Tables (6) give an equation for latent heat as a function of the temperature at which vaporisation takes place. Gurney and Cotter use the value 1035 Btu/lb (assuming vaporisation at about 100° F) and McKelvey and Brooke the value 1075 Btu/lb (assuming vaporisation at 32° F) to quote two examples of the many values in use.

The difference in temperature between the bulk water and the saturated interfacial film is considered small enough to assume that the film has the bulk water temperature and so the effect of superheat can be neglected.

The British Standard (5) shows an equation relating partial pressure to the mass of vapour in the moist air. Both Mikyska and Reinisch (13), and Zahn (12) quote an exponential equation for the vapour pressure, whilst MacDonald (4) uses a third order polynomial.

Because of the importance of the enthalpy temperature relationship in the analysis of cooling towers, a full discussion of the various enthalpy temperature equations and the corresponding variables together with their cumulative effect on accuracy is considered in Chapter Three.

CHAPTER TWO

LITERATURE SURVEY : COOLING TOWER PERFORMANCE CURVES

2.1 General

There can be no question of the desirable and useful nature of a general correlation for an air water evaporative cooling system applicable to a wide range of possible operating conditions. Such a correlation must contain all those factors which influence the performance of a cooling tower.

Jackson (9) prepared the following list of factors which affect the performance of mechanical draught towers.

Item No.	Quantity	Symbol
1	Mass rate of water flowing	L
2	Cooling Range	Δt
3	Outlet Water temperature	t_{L2}
4	Wet bulb temperature of the inlet air	t_{WB}
5	Mass rate of air flowing per hour	G
6	Water to air ratio	L/G
7	Tower characteristic	KaV/L
8	Height of packing in tower	y
9	Atmospheric pressure	P

The specification of a cooling tower is such that items 1 to 3 in the above table are usually specified by the client as is the wet bulb temperature, which will be dictated by the tower location in combination with economic considerations. An optimum height of packing 'y' and mass air flow G will also be selected on economic grounds and/or site restrictions. If the tower characteristic KaV/L is known, the one remaining variable is the water to air ratio L/G or G, as L is nominated. Like the wet bulb temperature, the atmospheric pressure P will be dependent upon the tower location.

So that all probable cooling tower selections are accommodated, a series of graphs will need to be prepared arranged so that some of the above parameters are held constant. Obviously the more parameters which the graph can accommodate the more universal it will become.

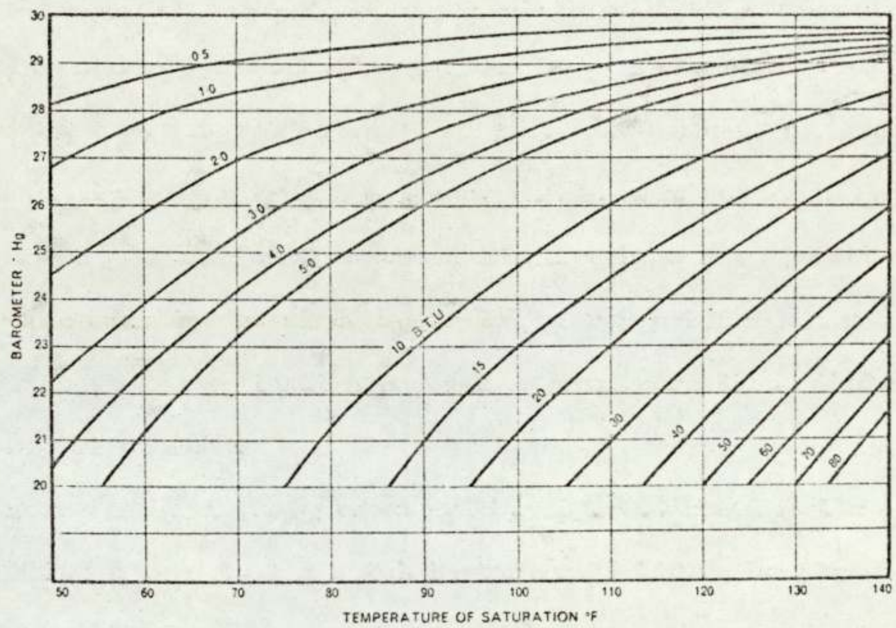
In recent times the widespread application of cooling towers, at various heights above sea level, has led to the need for performance charts, based on various atmospheric pressures, to be prepared since a change in altitude can have a significant effect upon the tower size. Increasing altitude makes a given tower more effective. In the past, authors considered the effect of varying atmospheric pressure to be relatively unimportant when compared with the other assumptions employed. The use of computers and more sophisticated numerical methods, enable more accurate selections to be prepared such as those required by a more competitive market which demands towers sized accurately for specific duties particularly as their application becomes more universal and their locations more widespread.

Stanford and Hill (23) give a graph, Fig. 2.1 showing the effect of altitude on enthalpy which in turn will influence the driving force available.

2.2 Historical background

One of the first performance charts to be devised was the Merkel Cooling Diagram which resulted from Merkel's work on the study of evaporative cooling towers. In 1925 Frederick Merkel developed the fundamental differential equation for the cooling process, which today forms the basis of analysis for cooling tower performance and the method for many of the more recently developed charts. The equation is fully developed in Chapter Five but can be written as :

$$\int_{t_{L2}}^{t_{L1}} c_L/(h_L - h_G)\,dt = KaV/L \qquad (2.1)$$



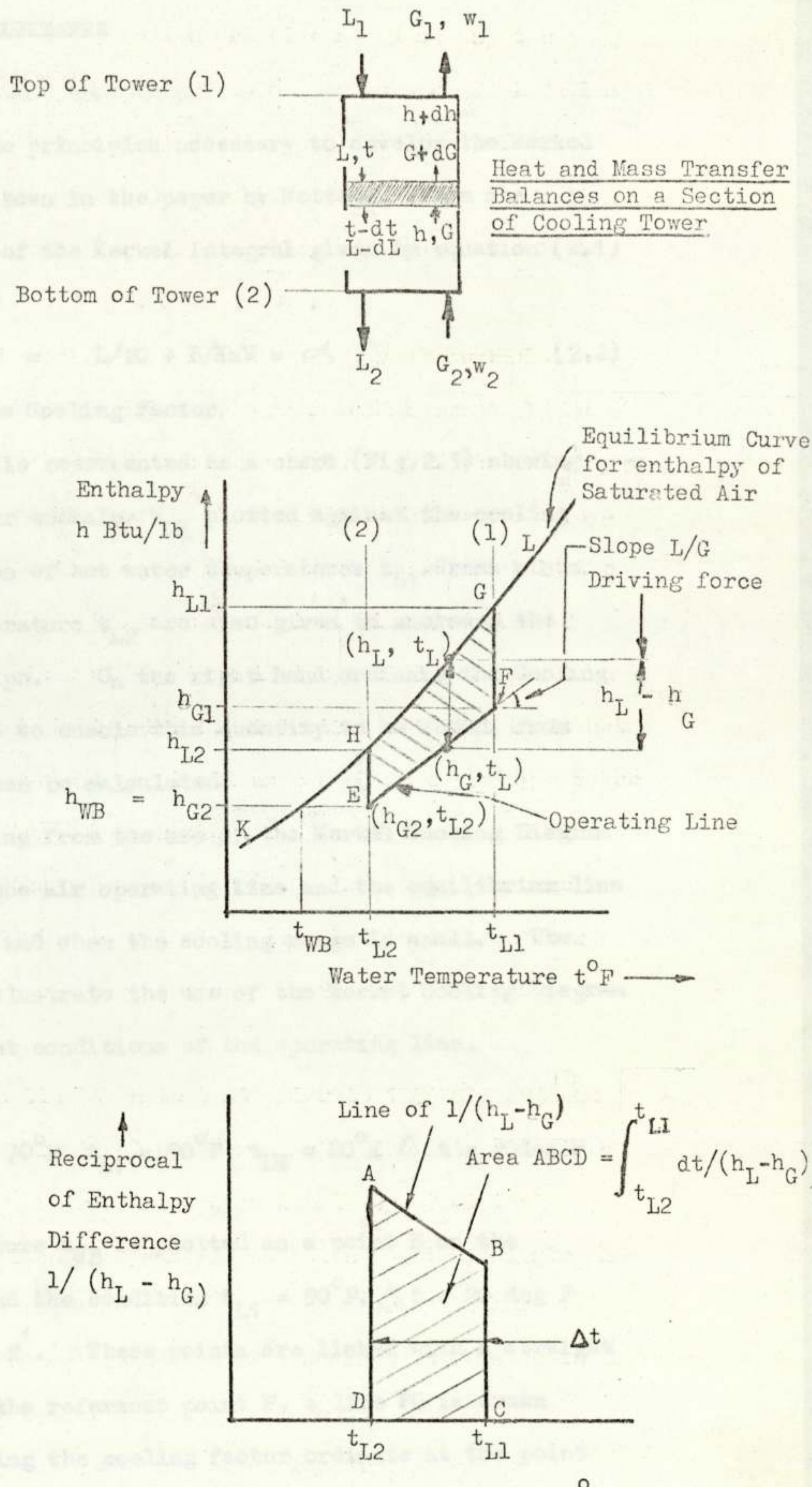
CURVES SHOWING THE INCREASE IN ENTHALPY
FOR REDUCED BAROMETRIC PRESSURE

Fig 2.1

Both sides of the equation are dimensionless and the value of this integral is referred to as the Number of Diffusion Units (NDU). The left hand side of the equation contains the thermodynamic conditions for the cooling process. The equation is shown graphically on Fig. 2.2 where integration of the left hand side is represented by the area ABCD.

To solve the area ABCD for a known tower characteristic KaV/L, would be tedious as trial and error techniques would be necessary. Some early authors use approximate methods normally adopting the analogy with the heat exchanger to introduce a log-mean potential. The log-mean potential would be correct if the enthalpy potential ($h_L - h_G$) were a straight line function of temperature t_L. This would be true if the enthalpy temperature relationship were straight (Fig. 2.2 illustrates that this is not the case) but reasonable approximations can be made giving a value for the log-mean potential over short ranges of the equilibrium line, where straightness can be assumed. As the temperature increases the rapid rise in enthalpy will dictate that an ever reducing temperature range can be used in order that the approximations are valid.

It is interesting to note that prior to 1955 all authors used manual and graphical methods for solving the integral given by equation (2.1). Thereafter the availability of digital computers made detailed analysis and solution a quick, simple and accurate feature of subsequent work. The paper by Fuller (1) published in 1956 appears to be one of the first to illustrate the use of computer techniques and the more sophisticated numerical methods which could now be employed.



COUNTERFLOW PROCESS DIAGRAM

Fig 2.2

2.3 Counterflow Performance

2.3.1 Discussion

A review of the principles necessary to develop the Merkel Cooling Diagram are shown in the paper by Nottage, where the approximate solution of the Merkel Integral given by equation (2.1) is shown to be :

$$(h_{LM} - h_2)/G\Delta t \quad = \quad L/2G + L/KaV = \alpha \qquad (2.2)$$

where α = Merkel's Cooling Factor.

This equation is represented as a chart (Fig.2.3) showing a family of curves for enthalpy h_{LM} plotted against the cooling range Δt and a series of hot water temperatures t_{L1}. Cross plots of outlet water temperature t_{L2} are also given to increase the usefulness of the graph. On the right hand ordinate the Cooling Factor α is plotted to enable this quantity to be found, from which the L/G value can be calculated.

Errors resulting from the use of the Merkel Cooling Diagram will be slight when the air operating line and the equilibrium line are roughly parallel and when the cooling range is small. The following examples illustrate the use of the Merkel Cooling Diagram for two very different conditions of the operating line.

Example 1

$t_{WB} = 65^\circ F \quad t_{L2} = 70^\circ F \quad t_{L1} = 90^\circ F \quad t_{LM} = 80^\circ F \quad \Delta t = 20 \deg F$

$L/G = 0.5$

The wet bulb temperature t_{WB} is plotted as a point E on the enthalpy ordinate, and the condition $t_{L1} = 90^\circ F$, $\Delta t = 20$ deg F on the grid as point F'. These points are linked with a straight line EF'. Through the reference point P, a line PQ is drawn parallel to EF' cutting the cooling factor ordinate at the point Q, where $\alpha = 0.68$.

From equation (2.2) :

$$\alpha = L/2G + L/KaV$$

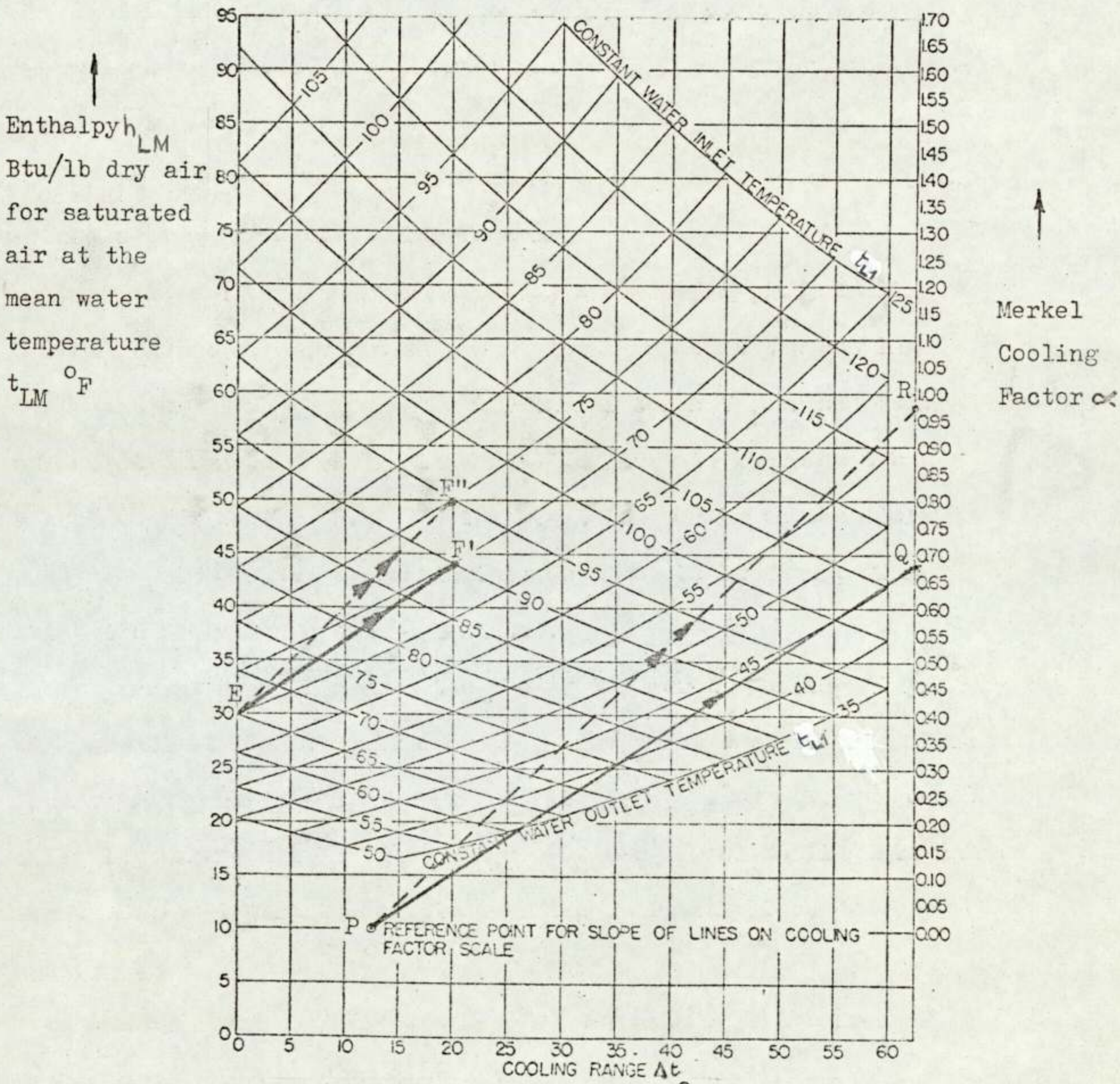


Diagram corresponds to a 0°F Enthalpy Datum

(h_{65} = 30 Btu/lb)

MERKEL COOLING DIAGRAM

Fig 2.3

For the values $L/G = 0.5$ and $\alpha = 0.68$:

$KaV/L = 2.32$

The correct NDU (KaV/L) value as found by computer is 1.28

Using the Mean Driving Force Method (see equation 5.24) the NDU value is 1.32

Note the discrepancy between the NDU values and also the difference in slope between the air operating line and the equilibrium line as shown on Fig. 2.4

Example 2

$t_{WB} = 65^{o}F$ $t_{L2} = 75^{o}F$ $t_{L1} = 95^{o}F$ $t_{LM} = 85^{o}F$ $\Delta t = 20$ deg F

$L/G = 1.0$

The wet bulb temperature t_{WB} is plotted as the point E on the enthalpy ordinate, and the condition $t_{L1} = 95^{o}F$, $\Delta t = 20$ deg F on the grid as point F". These points are linked with a straight line EF". Through the reference point P, a line PR is drawn parallel to the line EF" cutting the cooling factor ordinate at the point R where $\alpha = 0.98$. From equation (2.2) and the values $L/G = 1.0$ and $\alpha = 0.98$ give :

$KaV/L = 2.1$

The correct NDU value as found by computer is 2.0

Similarity in these two values can be related to the fact that the operating line EF" and the equilibrium line KL are approximately parallel as shown on Fig. 2.4. Using the Mean Driving Force Method, the NDU value is 2.03.

It can be seen from the examples considered, that the Merkel Cooling Diagram can be applied with confidence only when the operating line is approximately parallel to the equilibrium line. For all other conditions of the air operating line, errors in the NDU value can be significant. It can also be shown that the same type of error will be obtained when the cooling range exceeds $20^{o}F$.

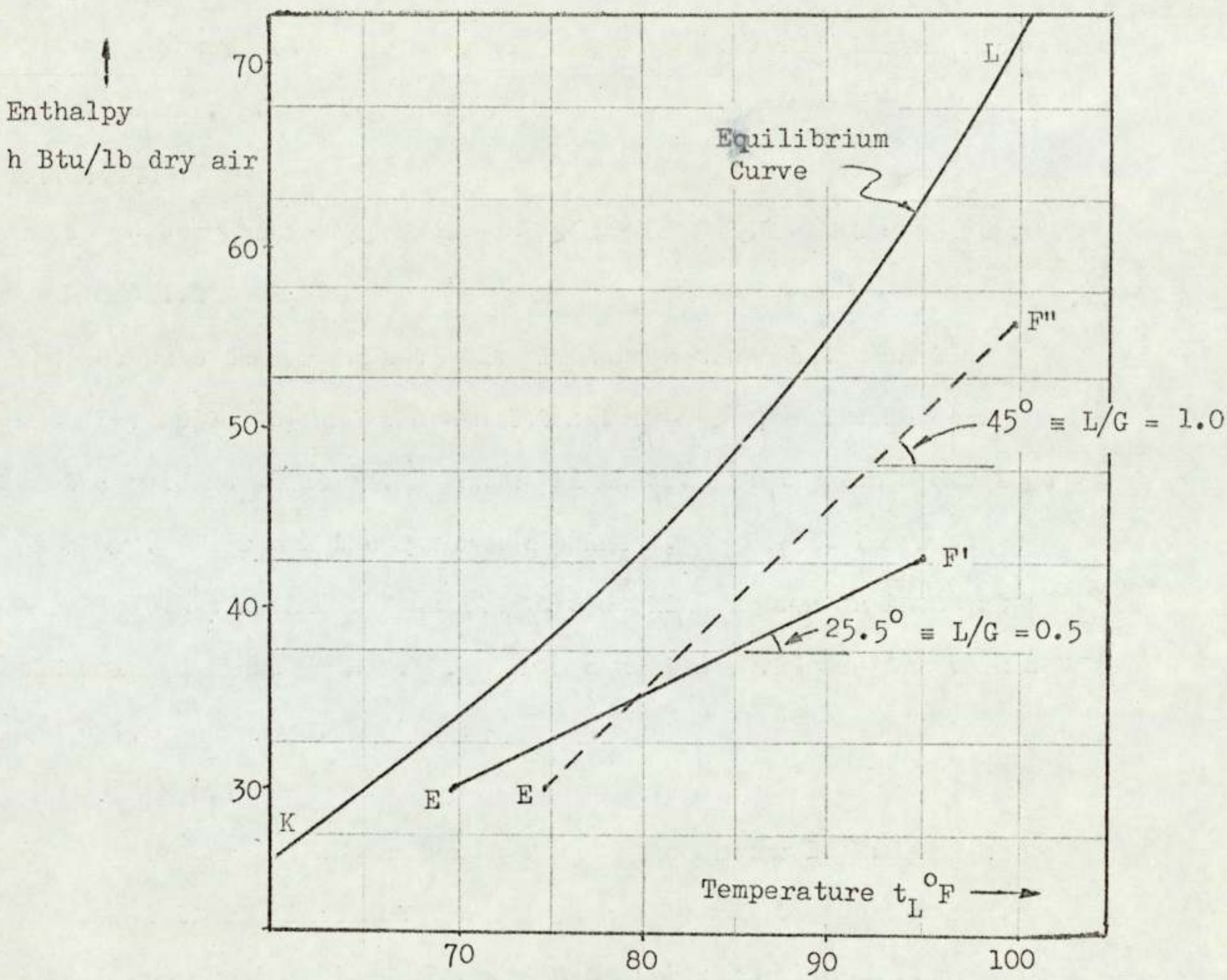


Chart corresponds to a 0°F
Temperature Datum for Enthalpy

CHART SHOWING EXAMPLES OF THE OPERATING LINES

Fig 2.4

To enable the ratio L/G to be identified directly on the Merkel Diagram, Wood and Betts (24) in 1950 developed a Modified Cooling Diagram so that the parameters L/G and KaV/L could be shown separately. The parameters L/G and KaV/L are shown as ordinates, one on each side of the graph, see Fig 2.5. The scale of wet bulb temperature t_{WB} is actually a suitable graduated enthalpy scale h_{WB}. The graph is plotted in oblique ordinates to give it a more convenient shape and so make it easier to read.

Again, errors will be slight when the air operating line and the equilibrium line are roughly parallel, and when the cooling range is small. It is to be noted that Wood and Betts limit their graph to a 20 deg F temperature range whilst Nottage limited his graph to a temperature range of 60 deg F, the latter giving rise to significant errors.

The following examples illustrate the use of the Wood and Betts version of the Modified Cooling Diagram.

Example 1

$t_{WB} = 65^\circ$ F $\quad t_{L2} = 70^\circ$ F $\quad t_{L1} = 90^\circ$F $\quad t_{LM} = 80^\circ$ F $\quad \Delta t = 20$ deg F.

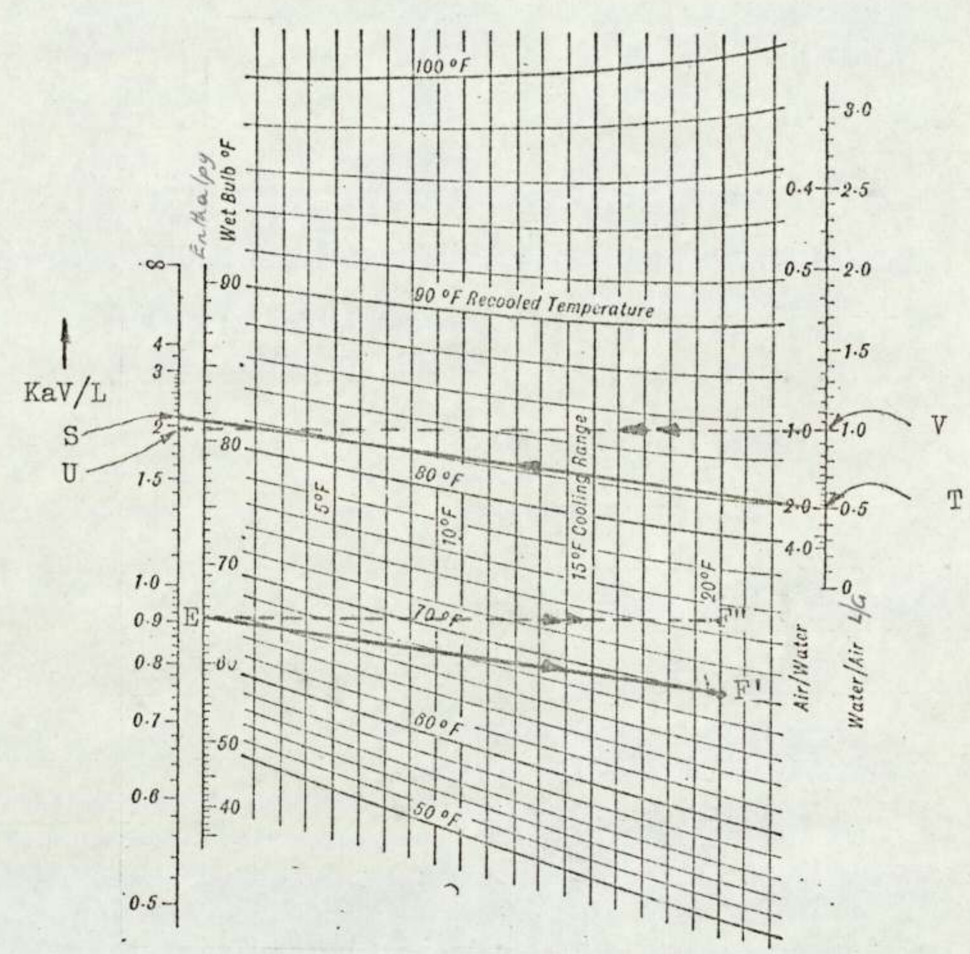
L/G = 0.5

The wet bulb temperature t_{WB} is plotted as a point E on the enthalpy ordinate of Fig 2.5 and the condition $t_{L2} = 70^\circ$ F, $\Delta t = 20$ deg F on the grid as point F'. These points are linked with a straight line EF'.

From the known value of L/G = 0.5, a straight line TS is drawn parallel to the line EF'. The corresponding value of the NDU (KaV/L) is read from the left hand ordinate as :

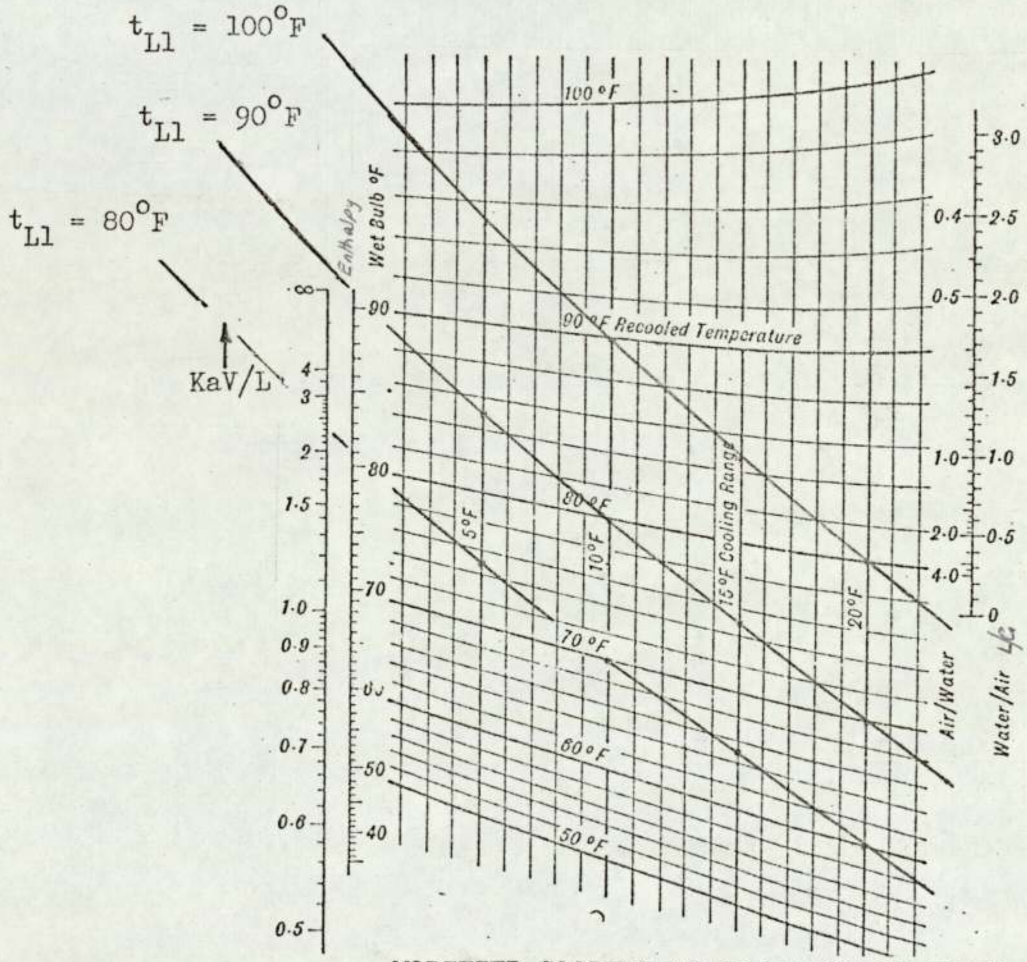
KaV/L = 2.1

This value compares with the Nottage value for the same example, both of which are incorrect. It is of interest to note the difference in slope between the operating line EF' and the equilibrium line KL on Fig 2.4.



MODIFIED COOLING DIAGRAM

Fig 2.5



MODIFIED COOLING DIAGRAM SHOWING LINES OF CONSTANT INLET WATER TEMPERATURE

Fig 2.6

Example 2:

$t_{WB} = 65^{o}F$ $t_{L2} = 75^{o}F$ $t_{L1} = 95^{o}F$ $t_{LM} = 85^{o}F$ $\triangle t = 20$ deg F

$L/G = 1.0$

The wet bulb temperature t_{WB} is plotted as the point E on the enthalpy ordinate and the condition $t_{L2} = 75^{o}$ F and $\triangle t = 20$ deg F on the grid as point F''. These points are linked with a straight line EF''.

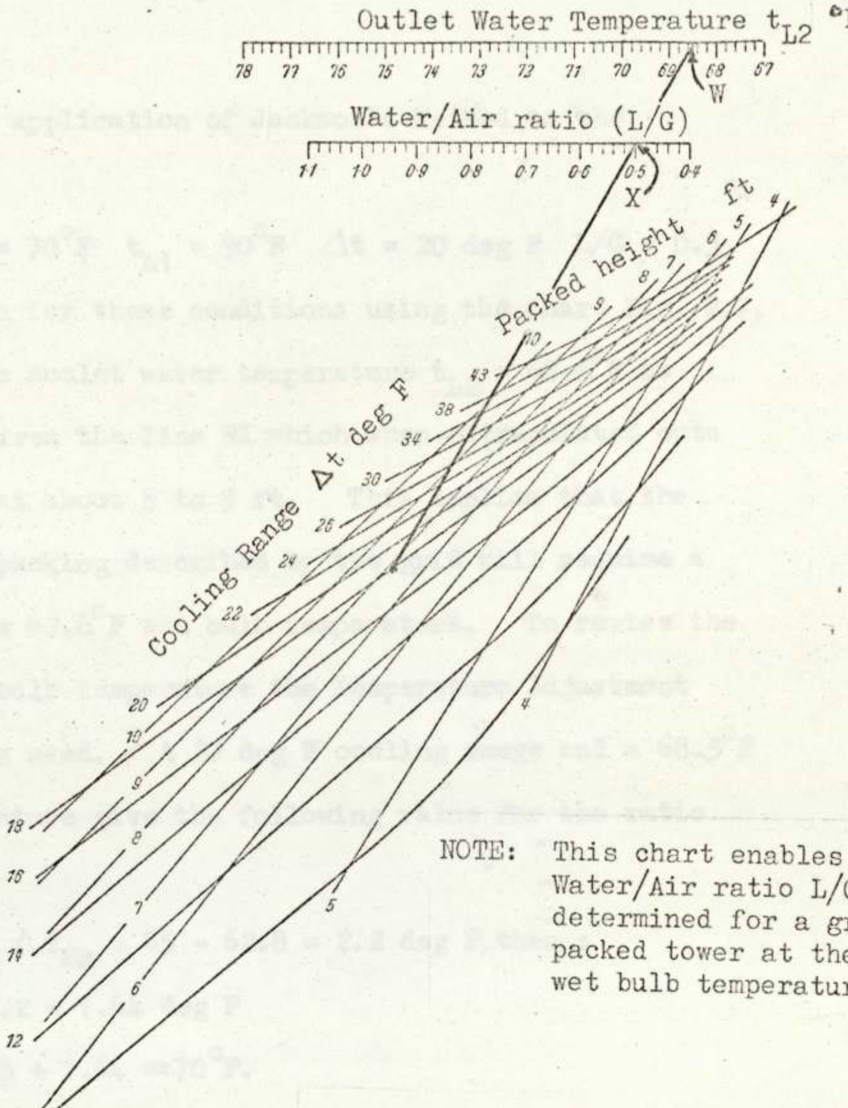
A line VU parallel to EF'' is drawn through the known value of $L/G = 1.0$. The corresponding NDU value (KaV/L) is read from the left hand ordinate as :

$KaV/L = 1.95$

This value compares favourably with that of the Nottage example and that found using a computer program and the Mean Driving Force Method.

The Wood and Betts graph can be made easier to use by adding lines of inlet water temperature. The redrawn graph is shown as Fig 2.6.

The Cooling Chart Fig 2.7 developed by Jackson (9) in 1951 using the Mean Driving Force Method to solve the integral shows similarities to the Modified Cooling Diagram described by Wood and Betts (24). Again oblique co-ordinates are used to plot a series of curves showing successive values of the cooling range $\triangle t$ and cross plots of constant pack height. The abscissa is calibrated to give the corresponding recooled temperature t_{L2} and water to air ratio L/G. This graph is plotted for a standard wet bulb temperature of 62.8^{o} F. For other wet bulb temperatures, Jackson devised a method utilising a Temperature Adjustment Diagram Fig 2.8 to adjust the outlet water temperature condition to other wet bulb temperatures and so extend the usefulness of his Cooling Chart Fig 2.7.

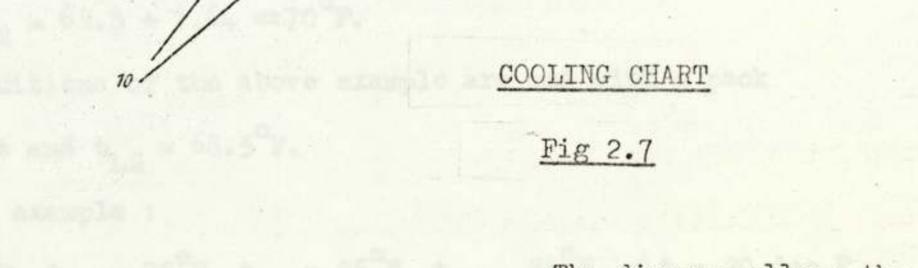


NOTE: This chart enables the Water/Air ratio L/G to be determined for a grid-packed tower at the standard wet bulb temperature 62.8°F

COOLING CHART

Fig 2.7

The diagram allows the use of the chart shown on Fig 2.7 to be extended to the designof towers for which the air inlet wet bulbtemperature is not 62.8°F



t_{L2}* = Outlet Water Temperature when the wet bulb temperature, t_{WB} = 62.8 °F (°F)

Δt_{WB} = Difference between the actual wet bulb temperature and 62.8 °F (deg F)

Δt_{L2} = Corresponding difference in outlet water temperatures (deg F)

TEMPERATURE ADJUSTMENT DIAGRAM

Fig 2.8

Consider the application of Jackson's Method to the example :

$t_{WB} = 65^{o}F$ $t_{L2} = 70^{o}F$ $t_{L1} = 90^{o}F$ $\Delta t = 20$ deg F $L/G = 0.5$

To obtain a solution for these conditions using the chart Fig. 2.7, a first guess of the outlet water temperature $t_{L2} = 68.5^{o}F$ is suggested. This gives the line WX which when extrapolated cuts the 20 deg F range at about 8 to 9 ft. This implies that the performance of the packing described by the grid will require a depth of 9 ft at the $62.8^{o}F$ wet bulb temperature. To revise the duty to a $65^{o}F$ wet bulb temperature the Temperature Adjustment Diagram, Fig. 2.8 is used. A 20 deg F cooling range and a $68.5^{o}F$ outlet water temperature give the following value for the ratio

$\Delta t_{L2}/\Delta t_{WB} = 0.73$

At $t_{L2} = 68.5^{o}F$ and $\Delta t_{WB} = 65 - 62.8 = 2.2$ deg F, then :

$\Delta t_{L2} = 0.73 \times 2.2 = 1.64$ deg F

therefore $t_{L2} = 68.5 + 1.64 \simeq 70^{o}F$.

Thus the conditions of the above example are met with a pack depth of 9 ft and $t_{L2} = 68.5^{o}F$.

Consider the example :

$t_{WB} = 65^{o}F$ $t_{L2} = 75^{o}F$ $t_{L1} = 95^{o}F$ $t_{LM} = 85^{o}F$ $\Delta t = 20$ deg F

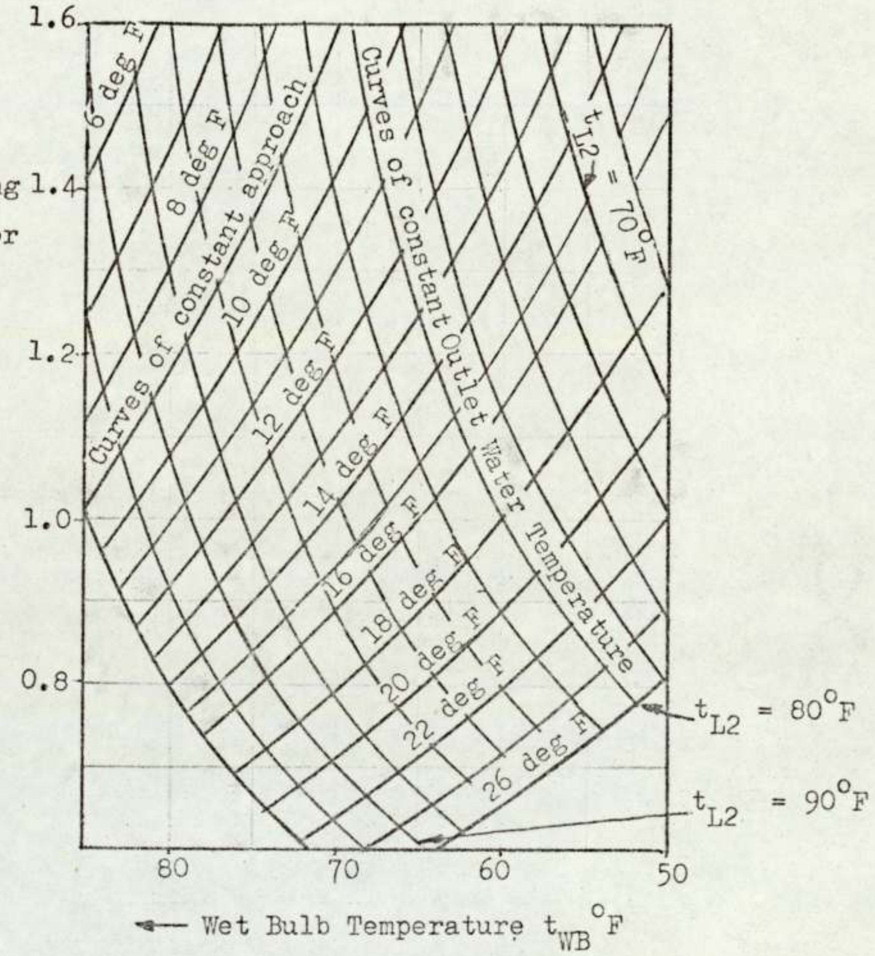
$L/G = 1.0$

Because of the relatively large L/G value for the graph, it can be seen that this condition is not plottable on Fig. 2.7 and hence highlights one of the shortcomings of the Jackson Cooling Chart. Besides the limited scope of the L/G value which can be accommodated, it is also worth noting that the maximum recooled temperature t_{L2} which can be applied is $78^{o}F$.

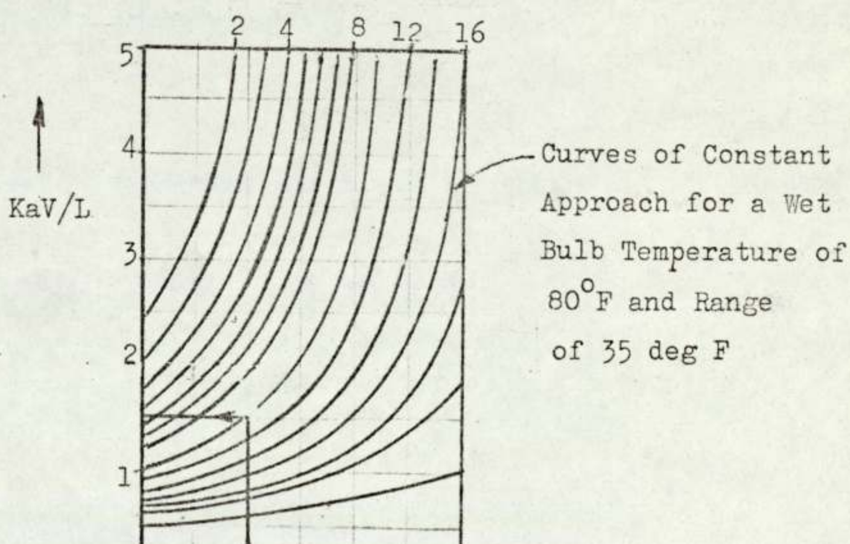
Jackson also shows a chart similar to Fig 2.7 with the height lines calibrated in transfer units. This enables the water/air ratio for any type of cooling tower to be determined provided the number of transfer units is known.

The A.S.H.R.A.E. Guide and Data Book (21) presents a series of charts obtained by using a small temperature increment for which the approximate NDU value is calculated from $\Delta t/(h_{LM} - h_{GM})$. This relationship uses the driving force at the mean water temperature for the interval. NDU values for all successive temperature intervals are calculated and the resulting summation gives the total NDU and thus the value of the Integral for the conditions considered. This technique is then adopted for many other conditions and the results plotted to give the Rating Chart, Fig 2.9. The term Rating Factor is used as the ordinate, wet bulb temperature t_{WB} as the abscissa with lines of constant outlet water temperature t_{L2} and cross plots of constant approach shown upon them. They are useful for checking the alternative cooling duties possible with a particular cooling tower. The ratio, tower area (ft^2) to water flow rate (gpm) at the condition $t_{L1} = 90^oF$, $t_{L2} = 80^oF$, $t_{WB} = 70^oF$ is made unity and all other duties are related to this condition by a term called the Rating Factor.

Graphical methods are employed by Lichtenstein (3) who considered the arithmetic methods used by others to be both tedious and inaccurate. Lichtenstein in 1943 co-ordinated his graphical results into curves, forming a curve book which could be used to predict the performance of a cooling tower under widely varying conditions of operation. From the many variables, Lichtenstein plots the tower characteristic KaV/L as the ordinate and L/G as the abscissa with curves of various approaches on a graph for a particular range. The performance curves are plotted for ranges from 8 to 50 deg F in intervals of 2 deg to 20 deg F and intervals of 5 deg from 20 deg F to 50 deg F giving 13 graphs per wet bulb temperature. To restrict the number of Performance



RATING CHART FOR 15deg F Range Fig 2.9



LICHTENSTEIN PERFORMANCE CHART

Fig 2.10

Charts a constant value of the air flow rate G is chosen for optimum power requirements. Lichtenstein has also shown that for a given cooling tower a plot of KaV/L on log-log paper with G as a parameter results in a closely spaced family of nearly straight parallel lines, as shown on Fig 2.12.

The Performance Chart shown in Lichtenstein's paper is that for an 80^{o}F air wet bulb temperature and a temperature range of 35 deg F. See Fig 2.10. Considering an example with a 10 deg F approach and an L/G = 1.0 the corresponding NDU (KaV/L) value from the chart is :

KaV/L = 1.5

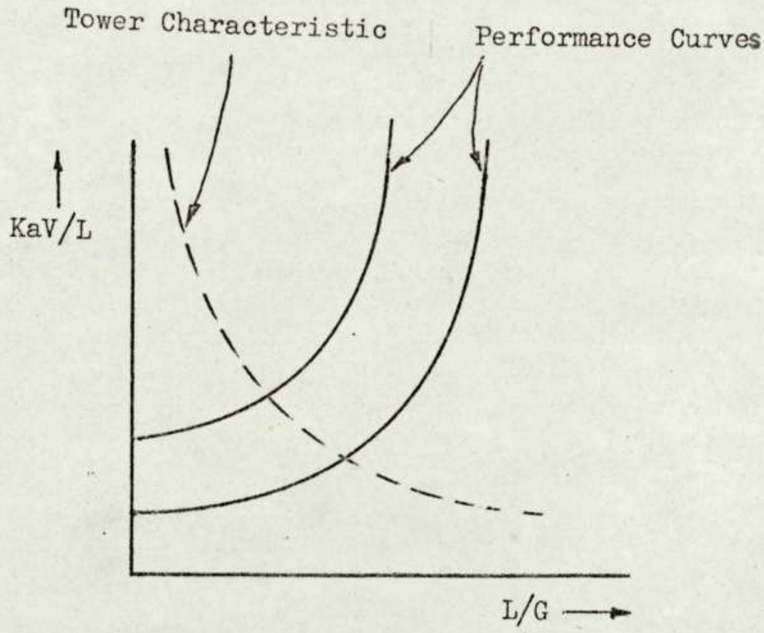
Using the Mean Driving Force Method the NDU value obtained is :

NDU = 1.51

Lichtenstein's method is an easy to apply direct means of finding the NDU value, suffering from the disadvantage that a great number of charts are necessary to cover all selection possibilities.

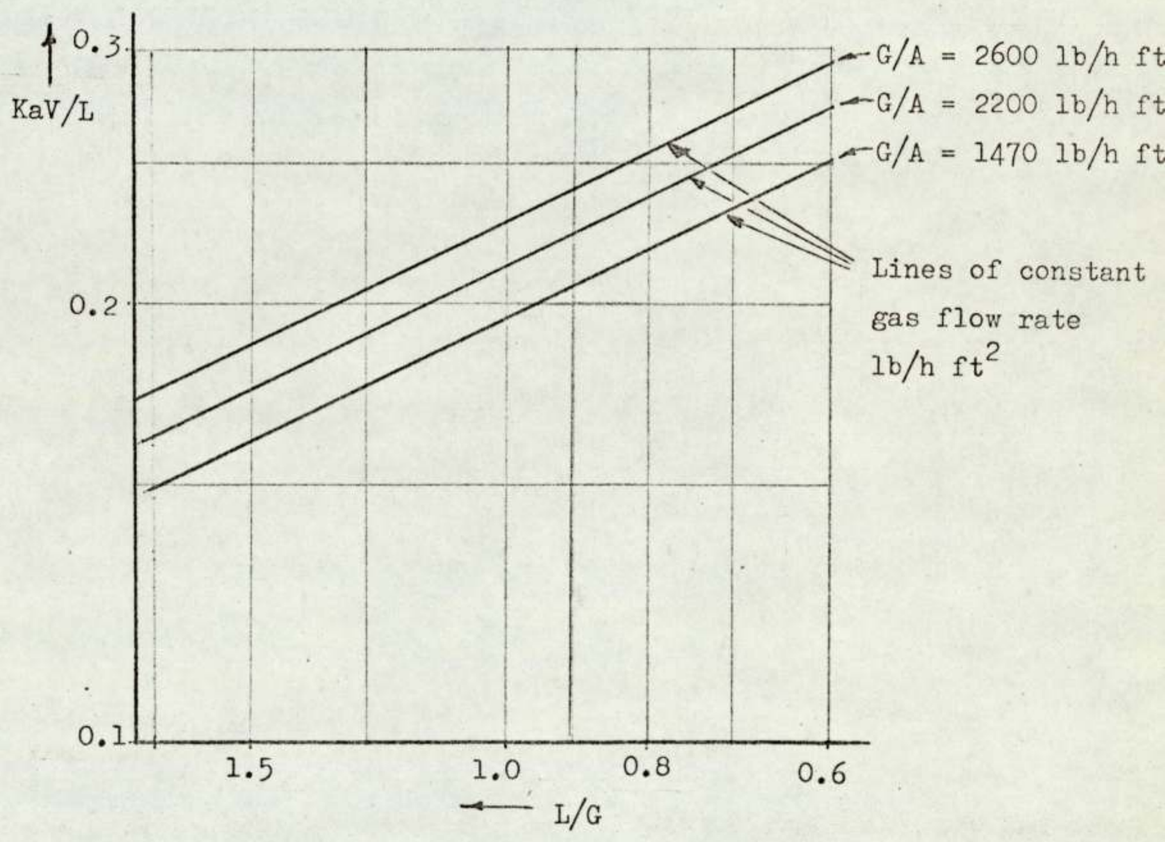
Having represented the solution of a particular integral as a point on the Performance Chart, Lichtenstein (3) then shows how the charts can be fully utilised. From the appropriate Performance Charts, values for a particular duty and corresponding L/G are read and plotted as a curve on a Tower Selection Graph; this is referred to as the performance curve and an example is shown on Fig. 2.11. The chosen tower characteristic KaV/L is also plotted on the graph. The intersection of these two lines indicates the L/G ratio at which the chosen tower will operate for the conditions specified by the performance curve.

From the literature surveyed this method of presentation would seem to be one of the most suitable for use in the cooling tower industry. It satisfies most of the requirements for a performance chart mentioned earlier and as further evidence of



LICHTENSTEIN TOWER SELECTION GRAPH

Fig 2.11



TYPICAL COUNTERFLOW TOWER CHARACTERISTICS

Fig 2.12

its usefulness, it has been incorporated into the recent British Standard (25) for Cooling Towers. This Standard uses the Lichtenstein Tower Selection Graph for showing on-site test results and comparison with the manufacturers characteristic curve so that a tower capability can be readily determined.

In 1956 Fuller (1) used a digital computer to solve Simpson's Rule applied to the evaluation of the Merkel Integral. The program is developed to process data from a test unit to obtain a heat balance around a test apparatus, and to establish its characteristic KaV/L for a variety of air and water mass flow rates; he does not give an example of his performance chart.

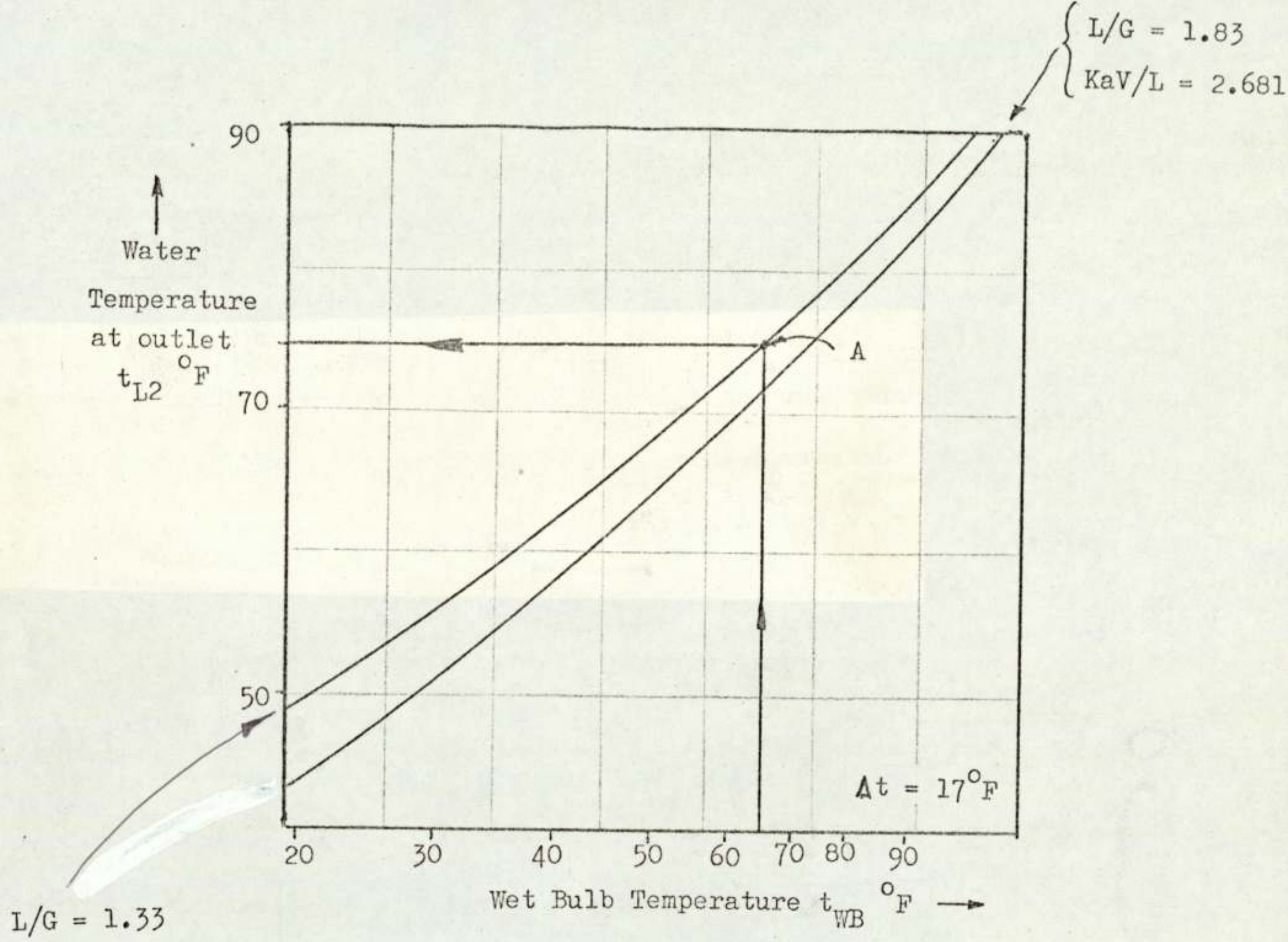
A paper presented in 1967 by Mikyska and Reinisch (13) also uses a digital computer to compute cooling curves. With a nominated tower characteristic, various design points are calculated using Simpson's Rule on 30 trapezoidal elements to evaluate the Merkel Integral. A plot of outlet water temperature t_{L2} versus an enthalpy scale marked off in wet bulb temperature t_{WB} is shown on Fig. 2.13 for curves of constant L/G and NDU, for a nominated range. A series of graphs is prepared for different values of Δt from 17 to 25 deg F.

The Mikyska and Reinisch chart, see Fig. 2.13 is of restricted use because a separate chart is required for each set of temperature conditions. To illustrate how it can be utilised the following example for a 65°F wet bulb temperature is considered. Using the upper performance curve the conditions read from the chart are as follows :

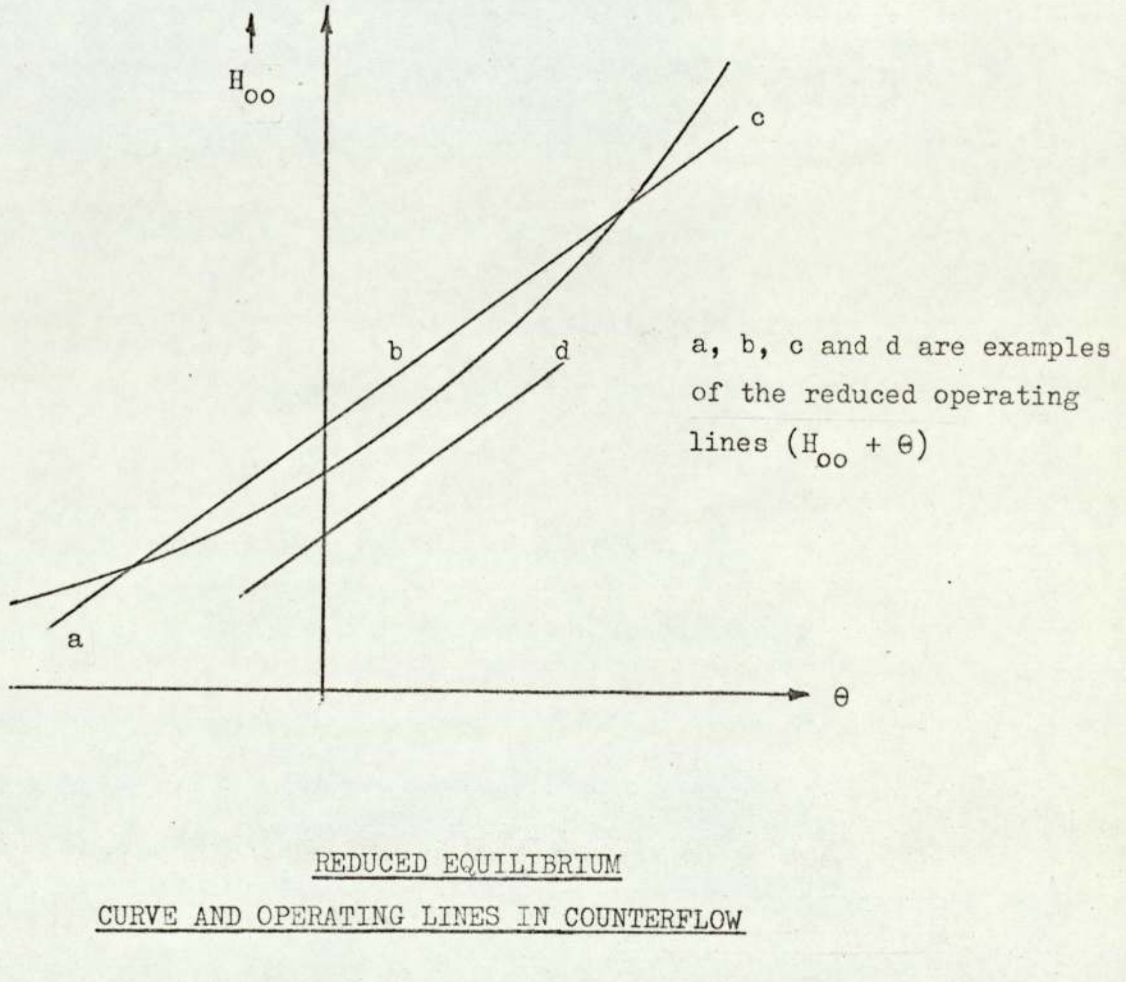
$t_{WB} = 65^{o}F \quad \Delta t = 17$ deg F for $L/G = 1.33$

For an NDU value = 2.183 the corresponding outlet water temperature is:

$t_{L2} = 75^{o}F$



COOLING CURVES
Fig 2.13



REDUCED EQUILIBRIUM
CURVE AND OPERATING LINES IN COUNTERFLOW

Fig 2.14

This value of NDU is somewhat less than that calculated using the Mean Driving Force Method for the above example. This gives :

$$NDU = 2.5$$

In 1966 Gardner (14) and (15) presented charts for the determination of the Number of Transfer Units in countercurrent flow where the equilibrium line can be represented by

$$h_L = E + Bt + F\exp(DT) \qquad (2.3)$$

for B, E, F and D as constants chosen to make a suitably accurate fit with the equilibrium line. He expresses the number of transfer units as :

$$NTU = c_L L/GR' \int_0^{\theta} d\theta/(\exp(\theta) - H_{oo} - \theta) = (c_L L/GR')I \qquad (2.4)$$

$$\text{for } \theta = D\,(t - t_o) = Dt - \log_e\,(R'/FD) \qquad (2.5)$$

$$R' = c_L L/G + B \qquad (2.6)$$

$$H_{oo} = (D/R')\,(h_G - E - Bt) - \theta \qquad (2.7)$$

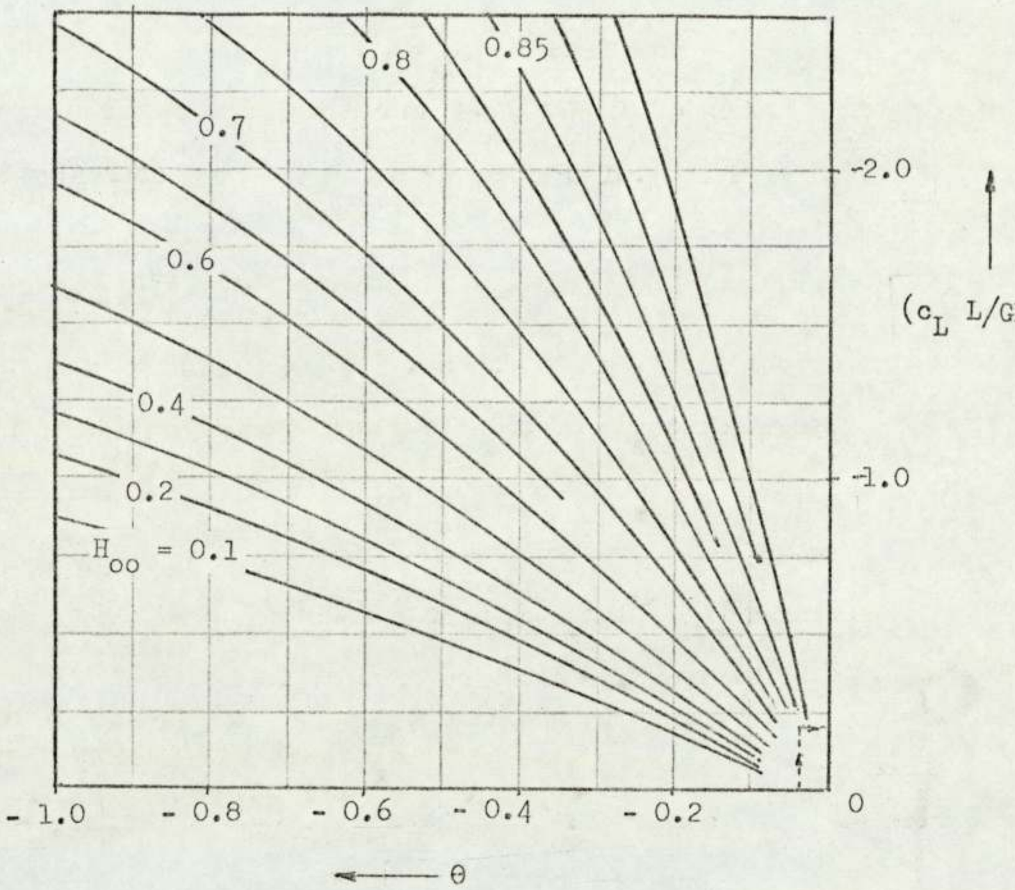
For $h_G = h_{WB} = h_{G2}$, use θ_2 value calculated together with temperature t_{L2} for H_{oo} and

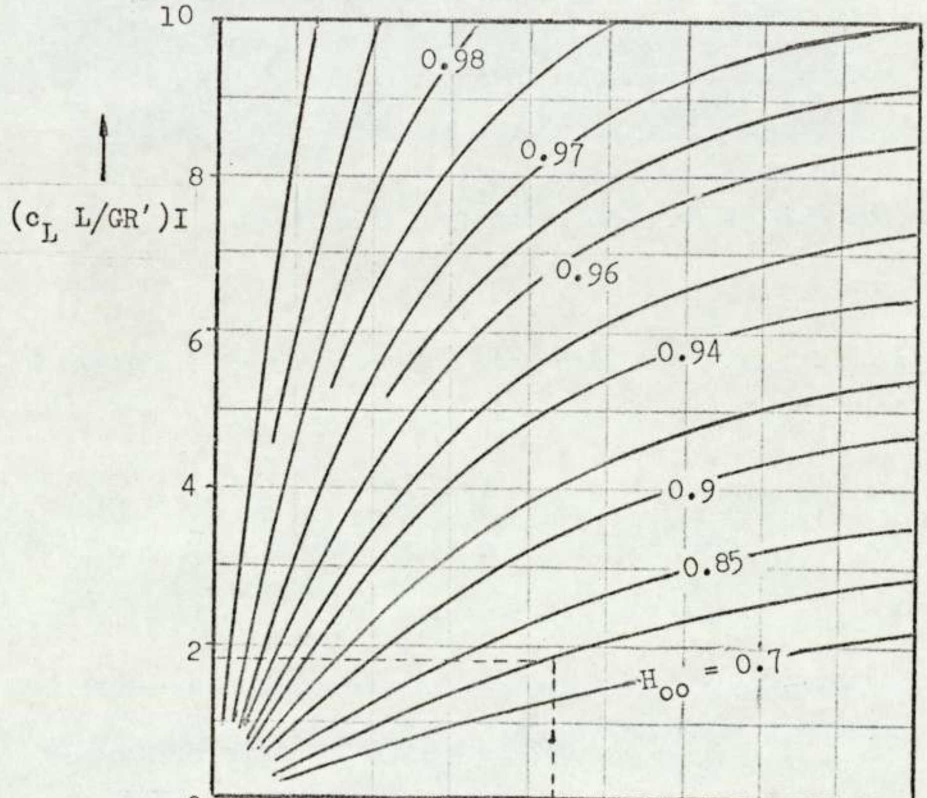
for $h_G = h_{G1}$, use θ_1 value calculated together with temperature t_{L1} for H_{oo})

where a reduced curve of the form exp θ is used to represent the equilibrium line.

The integral is evaluated for a wide range of conditions and plotted as a series of curves, see Fig. 2.15 with θ as the abscissa and $(c_L L/GR')I$, as the ordinate for lines of constant H_{oo}. To obtain the desired accuracy, the many curves are arranged in a series of charts.

The equation (2.7) is also developed into a more general solution to represent a curved equilibrium line with four possible operating lines as shown on Fig. 2.14.





REDUCED PERFORMANCE CHARTS

Fig 2.15

Consider the Gardner method applied to the example :

$t_{WB} = 65^{o}F$, $t_{L2} = 75^{o}F$, $t_{L1} = 95^{o}F$, $t_{LM} = 85^{o}F$, $\Delta t = 20$ deg F,

$L/G = 1.0$ and $h_{WB} = 22.6$ Btu/lb.

Using the equations (2.5), (2.6) and (2.7) together with the constants E = -10, B = 0, D = 0.02352 and F = exp1.954, substitution gives :

$R' = L/G = R$ for $B = 0$

$\theta_1 = (0.02352 \times 95) - \log_e (1.0/0.02352\exp 1.954)$

$\theta_1 = 2.23300 - 1.79840 = 0.4346$

$\theta_2 = 0.02352 \times 75 - \log_e (1.0/0.02352\exp 1.954)$

$\theta_2 = 1.76300 - 1.79840 = -0.0354$

$H_{oo} = (0.02352/1.0)(22.6 - (-10)) - (-0.0354)$

$H_{oo} = 0.7670 + 0.0354 = +0.8024$

From Fig. 2.15 the integral value at :

$\theta_2 = -0.0354$ and $H_{oo} = 0.8024$ is $I_2 = -0.15$ where $R' = L/G$ and

$c_L = 1.0$. The integral value at :

$\theta_1 = 0.4346$ and $H_{oo} = 0.8024$ is $I_1 = 1.85$

Now since Gardner's paper is based upon the Number of Transfer Units the corresponding value of the integral is :

$NTU = 1.85 - (-0.15) = 2.0$

To obtain the equivalent NDU value the NTU is divided by the L/G ratio. Thus :

$NDU = NTU/(L/G) = 2.0/1.0 = 2.0$

The computer value is :

$NDU = 2.0$

The Mean Driving Force value is :

$NDU = 2.03$

The many mathematical operations necessary to solve this integral using the Gardner Method are tedious and prone to human error. Unless adequate care is taken mistakes can be made in reading the graphs. This is particularly true for the previous example. For the numerous rapid selections which a performance chart should provide, this method is considered too slow and involved to apply frequently besides being subject to human error.

To complete the analysis of the Gardner method a computer program was written, see Fig. 2.16 to give the Gardner parameters θ_1, θ_2 and H_{oo} for known operating conditions.

Manufacturers of cooling towers prepare selection charts usually based upon their own unique methods of presentation developed from the analysis of the Merkel Integral. The Mean Driving Force Method is used by Stanford and Hill (23) of Carter Thermal Engineering Limited in their work on Performance Charts. Fig. 2.17 shows an example of their chart which applies to a particular tower characteristic for which a series of curves at constant recooled temperature t_{L2} is plotted for co-ordinates of water loading per unit area (Q/A) and the water inlet temperature t_{L1}. Each chart applies to a particular wet bulb temperature t_{WB} and unit mass flow rate of air G/A.

An example of the use of this chart is as follows. Consider the cooling duty :

$$t_{L1} = 95^oF \quad t_{L2} = 75^oF \text{ at } t_{WB} = 65^oF \text{ and } L/G = 1.0.$$

The operating condition is given by point A on Fig 2.17 which corresponds to a water loading per unit area of about 168 gall/h ft^2 (namely L/A = 1680 lb/h ft^2) for the particular tower considered. Knowing the water quantity to be cooled Q, and the liquid loading Q/A = 168, the area of tower A required can be calculated.

A similar chart, see Fig 2.18 is plotted by Film Cooling Towers. They show curves of constant approach on a chart with

PROGRAM

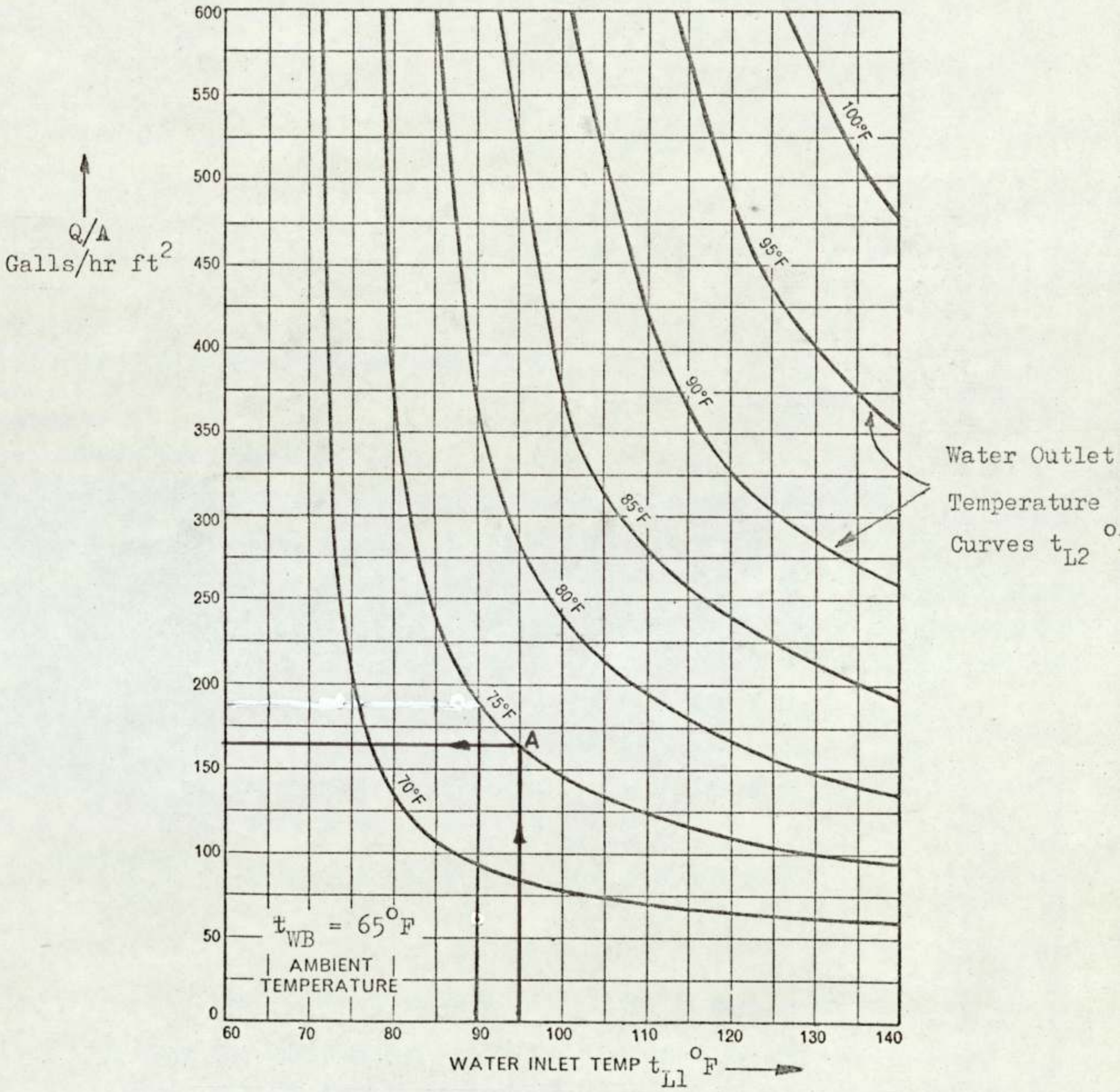
```
::TITLE TAPE 123 M A PENDERY JULY 1968
::GARDNERS NTU METHOD
SETV CDH(3)LS(2)T(4)
SETF EXP LOG
SETR 2
2)LINE
SPACES 6
TITLE TAPE 123(04).... M A PENDERY AUG 1968
LINE
LINE
SPACES 1
TITLE T ON
SPACES 5
TITLE T OFF
SPACES 5
TITLE L/G
SPACES 5
TITLE H WB
SPACES 5
TITLE TH1
SPACES 5
TITLE TH2
SPACES 5
TITLE HOO
LINE
1)READ T1
READ T2
READ L::RATIO OF LIQUID FLOW L/G
READ H
C=EXP 1.954
CHECK C
D=L/0.02352
D=D/C
CHECK D
D=LOG D
CHECK D
T3=0.02352*T1
S1=T3-D
CHECK S1
T4=0.02352*T2
S2=T4-D
CHECK S2
H1=0.02352/L
H2=H+10
H3=H1*H2
H3=H3-S1
CHECK H3
LINE
PRINT T2,3:1
SPACES 2
PRINT T1,3:1
SPACES 2
PRINT L,1:2
SPACES 2
PRINT H,2:1
SPACES 2
PRINT S1,1:4
SPACES 2
PRINT S2,1:4
SPACES 2
PRINT H3,1:4
JUMP @1
STOP
START 2
```

PRINTOUT OF RESULTS

T ON	T OFF	L/G	H WB	TH1	TH2	HOO
92.0	75.0	2.00	22.6	-0.7251	-0.3252	1.1084
99.5	80.0	0.89	24.9	0.2022	0.6609	0.7201
74.5	70.0	2.32	18.9	-0.9911	-0.8852	1.2841
78.0	70.0	1.35	18.9	-0.4496	-0.2614	0.9531
80.5	70.0	1.06	18.9	-0.2078	0.0392	0.8490
85.5	70.0	0.77	18.9	0.1119	0.4764	0.7709
93.5	70.0	0.54	18.9	0.4667	1.0194	0.7921
106.0	70.0	0.39	18.9	0.7921	1.6388	0.9508
92.0	75.0	0.62	26.6	0.4461	0.8460	0.9423
92.0	75.0	0.62	24.9	0.4461	0.8460	0.8778
92.0	75.0	0.62	22.6	0.4461	0.8460	0.7906
92.0	75.0	0.50	22.6	0.6612	1.0611	0.8723
92.0	75.0	1.00	22.6	-0.0319	0.3679	0.7987
92.0	75.0	1.50	22.6	-0.4374	-0.0375	0.9485
72.0	70.0	2.67	22.6	-1.1316	-1.0845	1.4188

COMPUTER PROGRAM FOR USE WITH GARDNER'S METHOD

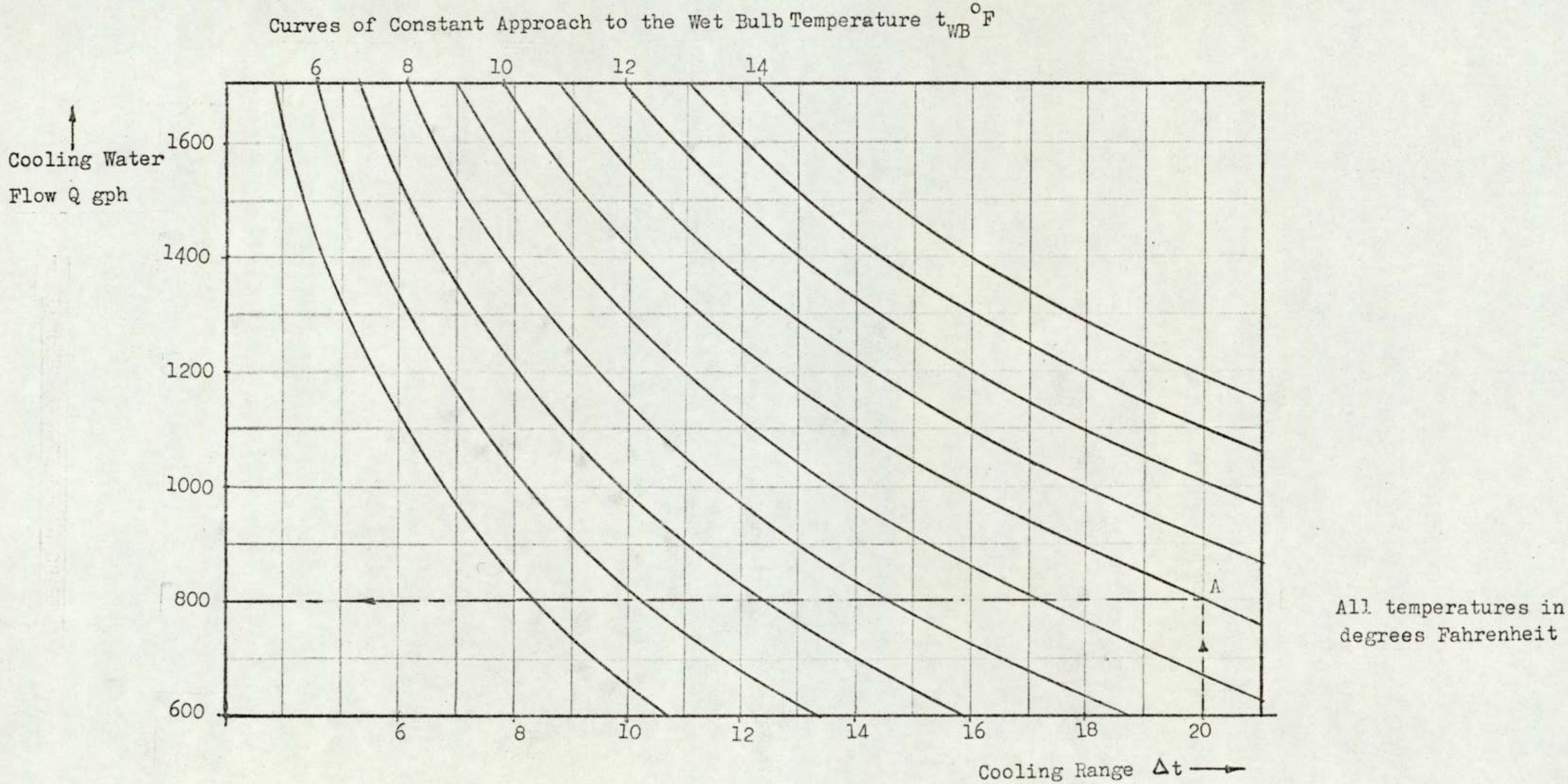
Fig 2.16



TYPICAL COMMERCIAL SELECTION CHART FOR A

PARTICULAR COOLING TOWER

Fig 2.17



COMMERCIAL SELECTION CHART

Fig 2.18

Fig 2.18

co-ordinates of water loading and cooling range Δt for a particular size of cooling tower. An example illustrating the use of this chart is as follows. Consider the cooling duty :

$t_{L1} = 95^\circ$ F, $t_{L2} = 75^\circ$ F, $\Delta t = 20$ deg F

$t_{WB} = 65^\circ$ F, Approach = 10 deg F

The operating condition is given by the Point A on Fig 2.18 which corresponds to a water flow of 800 gallons per hour, for a particular configuration and size of cooling tower.

Yet another design chart is used by Gurney and Cotter (7), see Fig 2.19 who plotted lines of constant range Δt and wet bulb temperature t_{WB} for a graph with the water outlet temperature t_{L2} and Duty Conversion Factor as ordinates. The use of this Approximate Design Chart is shown in the following example for the cooling duty given by :

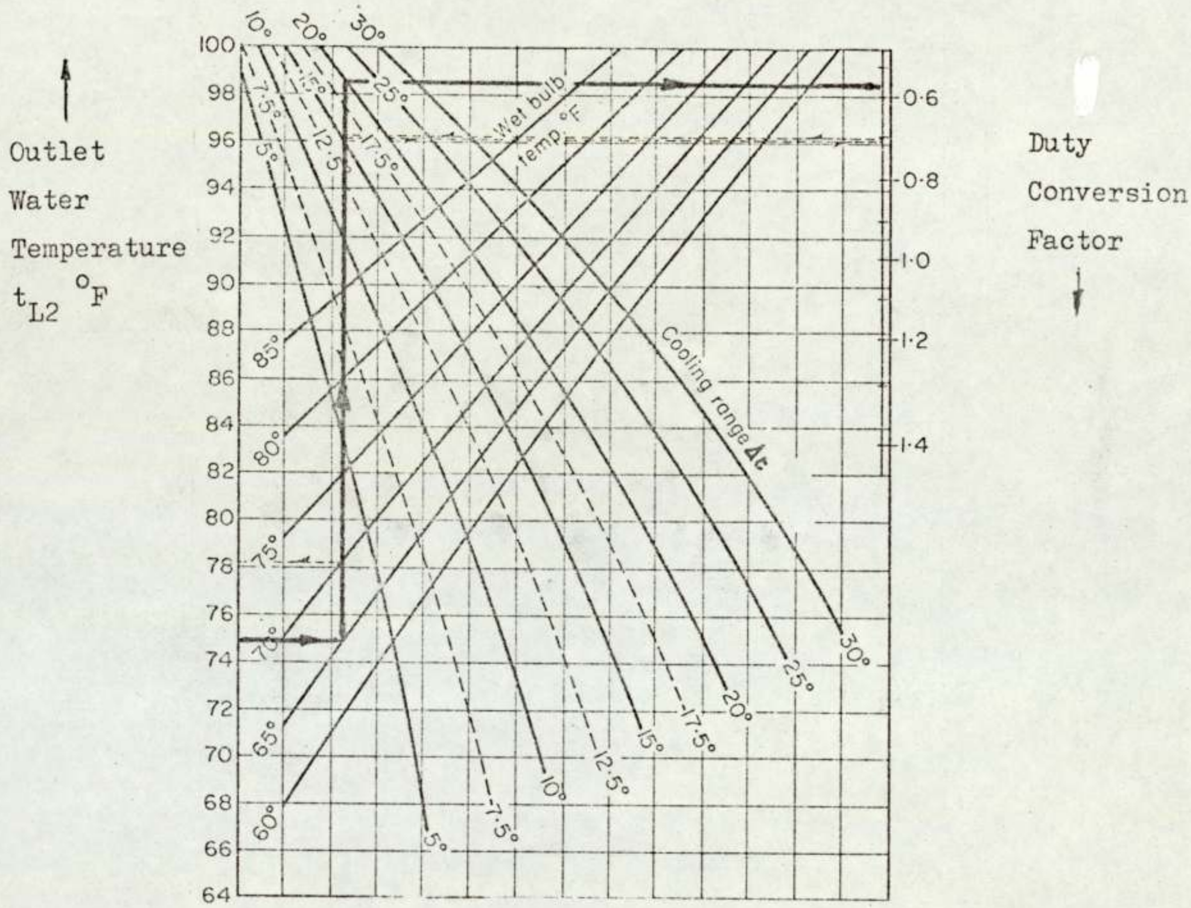
$t_{L1} = 95^\circ$ F $\quad t_{L2} = 75^\circ$ F $\quad \Delta t = 20$ deg F

$t_{WB} = 65^\circ$ F $\quad$ Approach = 10 deg F

Referring to Fig 2.19 and starting on the left hand scale with an outlet water temperature from the tower of $t_{L2} = 75^\circ$ F, a horizontal line is drawn to cut the 65° F wet bulb temperature line. A vertical line is drawn from this intersection to meet the 20 deg F cooling range line, and then horizontally to give a duty conversion factory (D.C.F.) equal to about 0.57. This D.C.F. value represents a given tower design.

McKelvey and Brooke (10) mention performance curves used by the Marley Company of U.S.A. The application of these curves is based upon the use of a rating factor. However further consideration is not given because the method applies to one particular design of cooling tower.

The performance charts prepared by the Industrial Companies are based upon providing a graph describing a particular type of



APPROXIMATE DESIGN CHART

Fig 2.19

tower, which give rapid duty selections. For a variety of cooling tower configurations and packings a great many of these charts are necessary. To reduce the number of charts it is thus desirable to have a more general presentation method which can be applied to any cooling tower design. Those charts giving an NDU (KaV/L) value obviously satisfy this need.

2.3.2 Summary of Counterflow Selection Methods

Author	Number of charts	Scope	Comment
Nottage	1	Various wet bulb temperatures and operating conditions.	Read a Merkel Cooling Factor. Satisfactory results obtained only when the operating line is approximately parallel to the equilibrium line. Range limited to 20 deg F maximum.
Wood & Betts	1	Accommodates the operating conditions for a variety of wet bulb temperatures.	Read a NDU value. Range limited to 20 deg F maximum.
Jackson	1	Applicable to one wet bulb temperature, namely 62.8° F.W.B. Restricted by small range of L/G ($0.4 \leqslant L/G \leqslant 1.1$) and outlet water temperature ($67 \leqslant t_{L2} \leqslant 78$).	Oblique co-ordinates make it difficult to read. Incorporates various pack heights but required additional chart to enable conversions to other wet bulb temperatures to be made.

Author	Number of charts	Scope	Comments
A.S.H.R.A.E.	15 (guess)	Require a separate chart for each temperature range but accommodates a variety of wet bulb temperatures.	Relates a standard set of conditions to the Rating Factor.
Lichenstein	13 per WBT	Thirteen charts per wet bulb temperature (WBT), each one for a constant approach temperature.	These charts are used to plot the performance curve against the tower characteristic. Intersection of the curve establishes the operating conditions.
Mikyska and Reinisch	Many	A separate chart is required for each set of temperature conditions.	Uses a computer program to give results which are plotted on a unique performance chart for each set of operating conditions.
Gardner	10	The reduced process equations are shown as the parameters of the graphs	Interpolation can produce significant errors. Method indirect and not easy to use.
Carter Thermal.	20 per type of tower	Unique performance chart required for each pack section per wet bulb temperature and constant gas flow rate.	Easy to use, applies to a range of base areas of one type of cooling tower.
Gurney & Cotter	1	Limited range of application.	Allows performance to be expressed as a Duty Conversion Factor which can be related to the pack performance of the cooling tower concerned.

Author	Number of charts	Scope	Comment
Film Cooling Towers	Many	Unique performance charts required for each pack section and tower size. Applicable to any wet bulb temperature. Approach limited to 14 deg F.	Assumes a constant gas flow rate.

2.3.3 Accuracy

Although numerical analysis and computer methods enable accurate solution of the integral this has not always been achieved. Nottage (8) in 1941 maintains that the inaccuracy involved in the substitution :

$$(h_{LM} - h_2)c_L \triangle t = L/2G + L/KaV \qquad (2.2)$$

is of the same order as that accompanying the log-mean enthalpy which presupposes that both lines on the driving force diagram are approximately straight. Wood and Betts (24) in 1950 made further qualifications by indicating that the error will be small when the operating line and equilibrium curve are roughly parallel and the cooling range $\triangle t$ is small.

The assumption that the equilibrium curve is straight for a limited temperature range is not considered by Jackson (9) to be entirely suitable. He found the method presented by Carey and Williamson (27) to be more satisfactory. This depends upon the use of a chart for reading the enthalpy correction factor 'f' to correct the driving force value. The use of this method gives more accurate results provided the range for which it applies is restricted to about 30 deg F and a maximum inlet water temperature to 125°F.

Lichtenstein (3) in 1943 considered the approximations used by Nottage as unsatisfactory. This led him to use the more accurate though tedious graphical integration method to prepare a series of charts.

The A.S.H.R.A.E. Guide and Data Book (21) of 1967 uses the trapezoidal rule applied to small temperature increments of 1 deg F for which the enthalpy difference at the inlet and outlet of each increment is determined. This is an approximate method but sufficiently accurate for the requirements of the rating charts.

Fuller (1) used Simpson's Rule to solve the integral with the temperature range divided into 10 equal temperature increments to obtain a suitable degree of accuracy.

Mikyska and Reinisch (13) used Simpson's Rule and iterative means to obtain graphs for a set of known conditions. To achieve the required accuracy necessitated three to nine iterations for every computed point.

With Gardner's method the accuracy is dependent upon the fit of the polynomial equation to the enthalpy temperature equilibrium curve, which with the chosen constants is accurate to 0.1% of the enthalpy for the specified temperature range 60 to 90°F. The method also depends upon accurate interpretation. This range of temperature is somewhat limited for normal commercial requirements where a range of 50°F to 135°F would be considered necessary.

2.3.4 Summary

From the charts and graphs described certain facts emerge. These are :

a) Merkel's integral is the accepted means of representing the cooling process, and this integral is evaluated by one of the following three methods :

b) Graphical Integration.

c) Summation of incremental values.

d) Modification to the integral by use of a second degree polynomial to approximate the equilibrium curve.

The accuracy required for solving the integral will be influenced by the method employed to plot the results and the technique for using the graph. With the use of numerical methods of integration and the advent of computers the degree of accuracy can easily be controlled and regulated to suit the graphical presentation required. These methods readily enable additional parameters to be accommodated within the calculation, including the effect of a varying atmospheric pressure which to date has been ignored by other workers but it is included in the present work.

Another desirable feature of a performance chart is the ease and speed with which it can be used. For this reason the variables and parameters plotted should be those which directly influence the tower performance and not be obscure parameters as employed with Gardner's (14) and (15) method. Ideally, all variables should be represented on the minimum number of performance charts compatible with ease of use and the required accuracy. It is also desirable that the charts be of universal application.

Much of the computer work included the use of the enthalpy temperature relationship represented by a forward backward difference equation. This proved a most satisfactory method for calculating enthalpies from a known temperature but for other pressure conditions it would be necessary to analyse fresh data and evaluate alternative forward backward difference equations.

Since industry, particularly in South Africa and Central America is located high above sea level some means of accommodating the atmospheric pressure is deemed necessary. The fundamental equation includes the atmospheric pressure making the equation ideal for a general computer program capable of giving performance data at any of the desired atmospheric pressures. The use of this equation can enable this effect to be investigated to see what significance it has on cooling tower selection.

From the methods reviewed, the charts presented by Lichtenstein (3) are considered the most suitable for further development towards universal performance charts incorporating those variables listed by Jackson (9) mentioned earlier, to obtain the minimum number of useful charts.

The Lichtenstein graph includes the parameters NDU, L/G, approach but requires a large number of graphs to cover all the likely temperature ranges (Δt) and wet bulb temperatures (t_{WB}). It is therefore the intention to develop a series of charts capable of containing more information and yet retaining the accuracy and ease of use, associated with the Lichtenstein graph, for both the counterflow and crossflow process.

2.4 Crossflow Performance

2.4.1 Discussion

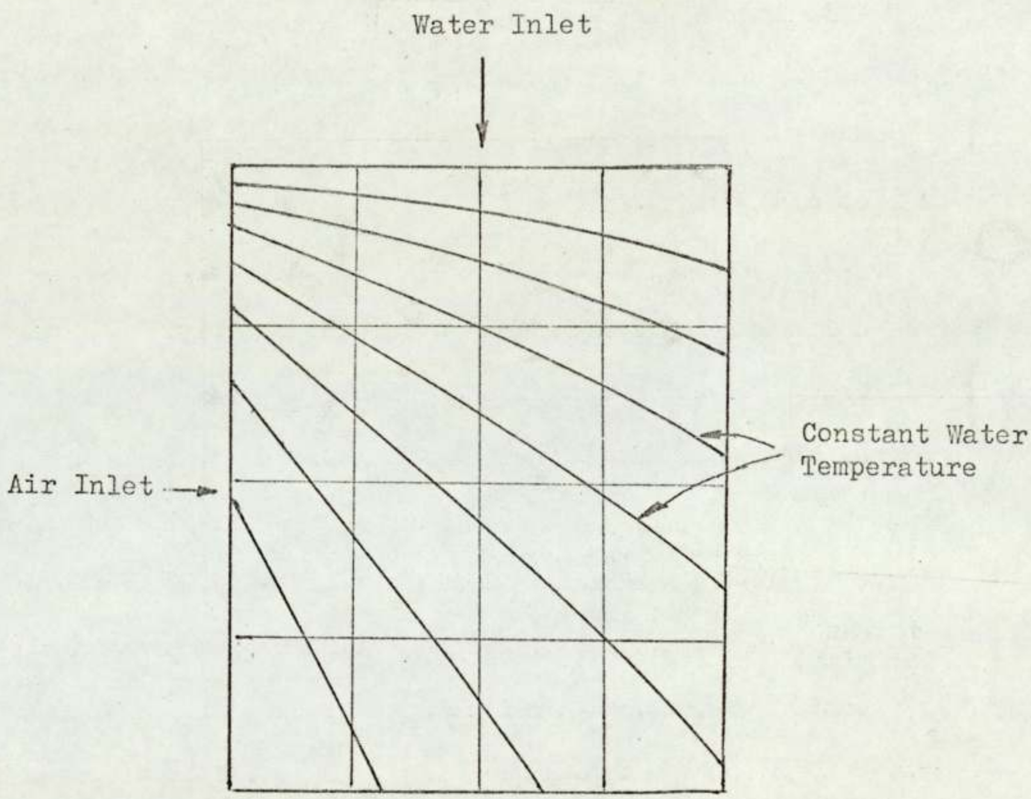
The crossflow cooling tower is a development of the counterflow tower. In contrast to the rising air and falling water of the counterflow arrangement, the crossflow tower has air flowing horizontally and water falling vertically. The counterflow tower has much greater use than the crossflow one, although the latter has the advantage of low overall height and low power consumption but generally it suffers from a reduced cooling performance per unit volume of packing, and usually a larger base area than an equivalent counterflow tower.

Since 1956 more attention has been focused upon the crossflow cooling tower. Whilst there is nothing revolutionary in the crossflow arrangement, interest has probably been generated by a more selective customer, a wider range of cooling tower application, and the use of computers for obtaining performance data. This last comment is of particular significance since the calculation of crossflow performance data was a very tedious task using hand calculation methods.

In 1956 Fuller (1) made one of the first references to the computer evaluation of the crossflow cooling tower. His computer program for the crossflow correlation is an extension of the counterflow theory but using two simultaneous differential equations, one in the direction of water flow and the other in the direction of air flow.

To appreciate the process within a section of crossflow tower it must be understood that conditions across a horizontal section of packing are not constant as in a counterflow tower. Observation shows that the air flowing horizontally moves towards progressively warmer water as the water falls towards colder air, as shown graphically on Fig. 2.20. It is this two directional change in conditions which makes the crossflow analysis more difficult than the counterflow one. Gurney and Cotter (7) concluded that the temperature and enthalpy gradients in a crossflow tower are so complicated that the number of diffusion units cannot be calculated using the counterflow method and to obtain a solution they use the mean driving force method applied to each elemental part of a pack.

Gurney and Cotter (7) and the A.S.H.R.A.E. Guide and Data Book (21) before them, Zammer (30) and, Zivi and Brand (2-) give a description of this technique. In detail they consider



GRAPH SHOWING LINES OF CONSTANT WATER TEMPERATURE

Fig 2.20

a cross section of packing broken down into elemental volumes of unit width, with the dimensions dx and dy arranged in size so that the overall L/G ratio (L/A)/(G/A') applies to each element. The section is then divided up to have the same number of elements in both directions. The calculations start at the top left hand corner of Fig. 2.21. The temperature change in each element is determined by trial and error from the energy transfer equation :

$$t_{Li} - t_{Lo} = NDU\left((h_{Li} - h_{Gi}) + (h_{Lo} - h_{Go})\right)/2 \qquad (2.9)$$

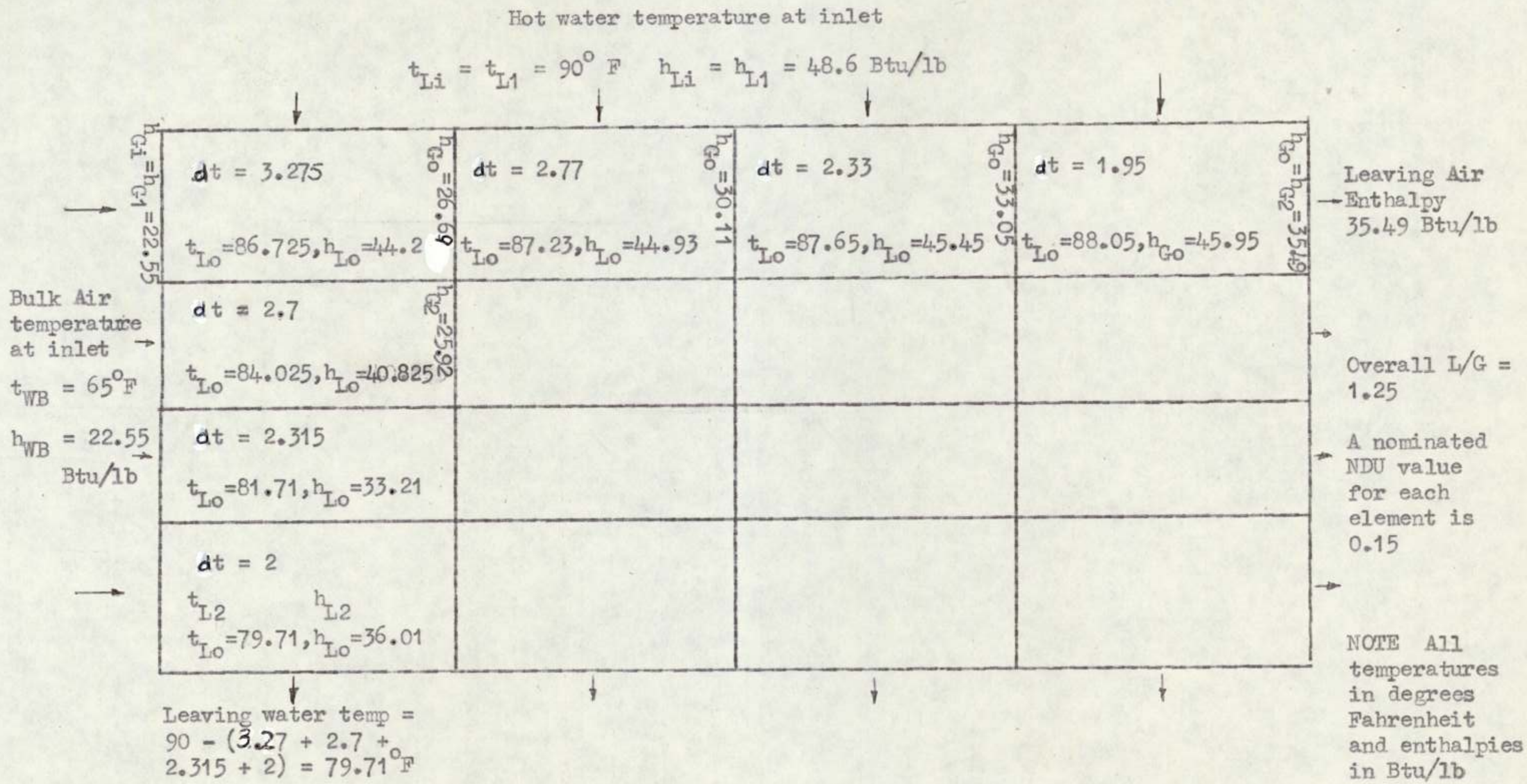
and the heat balance equation :

$$L\,(t_{Li} - t_{Lo}) = G\,(h_{Gi} - h_{Go}) \qquad (2.10)$$

the suffix 'i' indicating inlet conditions to the element and suffix 'o' indicating the outlet conditions from the element for a known NDU value. The calculated outlet conditions are then used for calculations on adjacent elements in order to find their air and water leaving conditions. This procedure continues down and across the pack for all of the elements. The outlet temperatures from the end elements give leaving air and water profiles as shown on Fig. 2.22 which are similar to those of the heat exchanger, described by Rogers and Mayhew (22). The average outlet water temperature is the arithmetic mean of the individual water temperatures leaving each of the elements at the bottom of the pack, and in a similar way the average outlet air conditions can be found from the air leaving each of the horizontal elements.

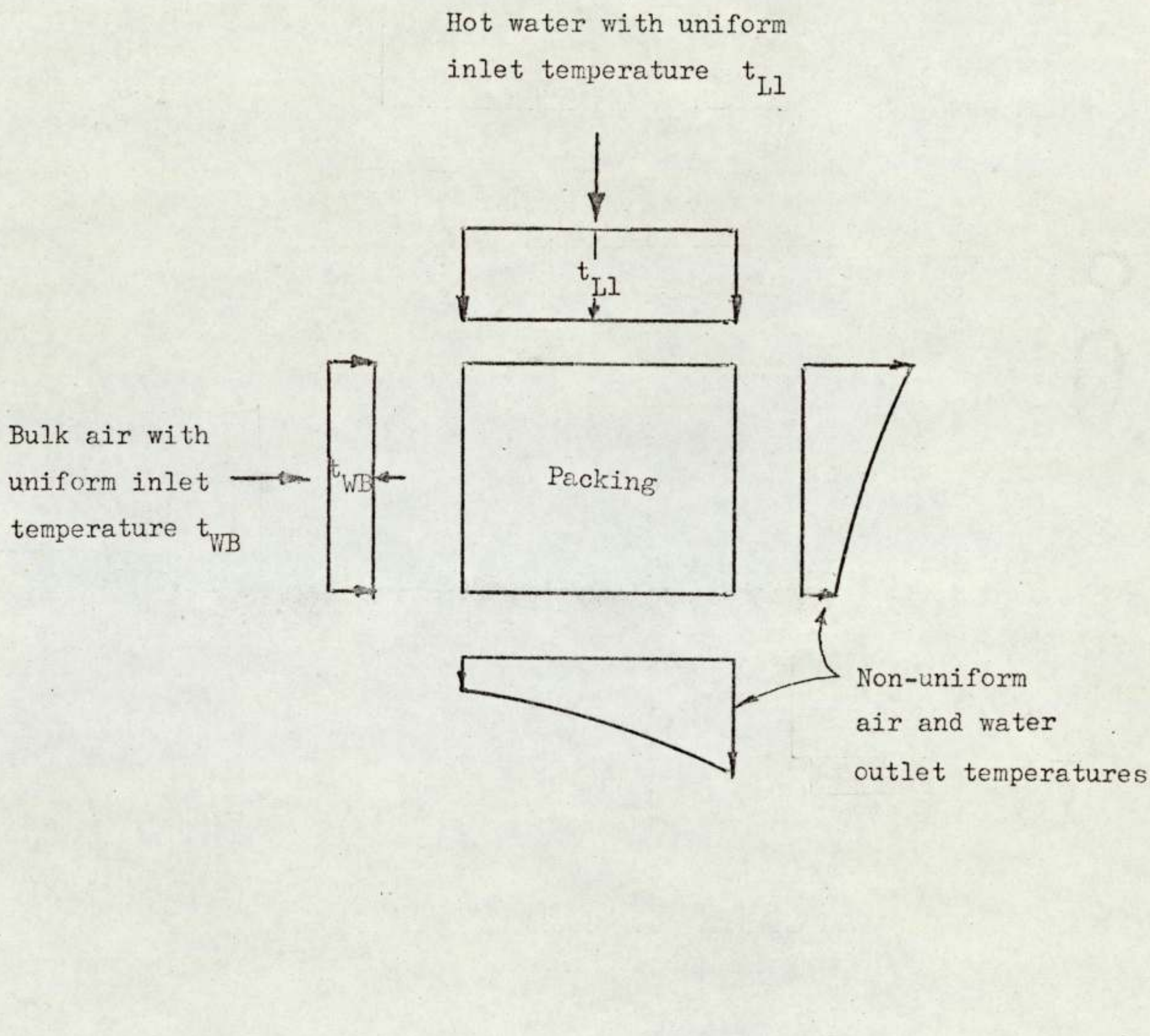
The method of solution for an element can be summarised in the following way :

Consider a section of packing subjected to the hot water temperature $t_{L1} = 90^{o}F$ at inlet, with inlet air at the wet bulb temperature of $t_{WB} = 65^{o}F$, a nominated NDU value of 0.15 for each element and an overall L/G value of 1.25. Starting with the element in the top left hand corner, where the air and water inlet



AIR AND WATER CONDITIONS IN A CROSSFLOW COOLING TOWER

Fig 2.21



GRAPH SHOWING LEAVING AIR AND WATER PROFILES

Fig 2.22

conditions are known, it is necessary to guess a value of the water temperature drop across the element in order to start the iteration. This value is then used to establish the leaving air enthalpy, using equation (2.10) and the driving force at outlet. Equation (2.9) is then solved to obtain a value of the exit water temperature, which is compared with the guessed value of water temperature to determine the completion of the iteration. If not, the guessed water temperature drop is adjusted and the calculation repeated. Further iterations may be necessary to achieve satisfactory agreement. An example of the iterations is shown by Gurney and Cotter (7) as follows :

Starting from the top left hand corner of the section on Fig 2.21 where all the conditions are known the driving force between the water film at 90°F and the inlet air at 65°F WB for $h_{G1} = h_{Gi}$ and

$h_{L1} = h_{Li}$ is :

$h_{Li} - h_{Gi} = 48.6 - 22.6 = 26.0$ Btu/lb

Guess $(t_{Li} - t_{Lo}) = 3.275$ deg F

Hence the water temperature at outlet = 90 - 3.275

i.e. $t_{Lo} = 86.725°F$

The corresponding enthalpy at outlet is :

$h_{Lo} = 44.2$ Btu/lb

From equation (2.10)

$$h_{Gi} - h_{Go} = (L/G)(t_{Li} - t_{Lo}) = 1.25 \times 3.275$$
$$= 4.09 \text{ Btu/lb}$$

The air enthalpy at outlet $h_{Go} = 22.6 + 4.09 = 26.69$ Btu/hr

Hence the corresponding driving force at outlet is

$h_{Lo} - h_{Go} = 44.2 - 26.69 = 17.51$ Btu/lb

Using equation (2.9) to check the temperature drop

$$(t_{Li} - t_{Lo}) = ((17.51 + 26) \times 0.15) /2$$
$$(t_{Li} - t_{Lo}) = 3.27 \text{ deg F}$$

This value of temperature drop (t_{Li} - t_{Lo}) gives satisfactory agreement with the guessed value for a sliderule calculation.

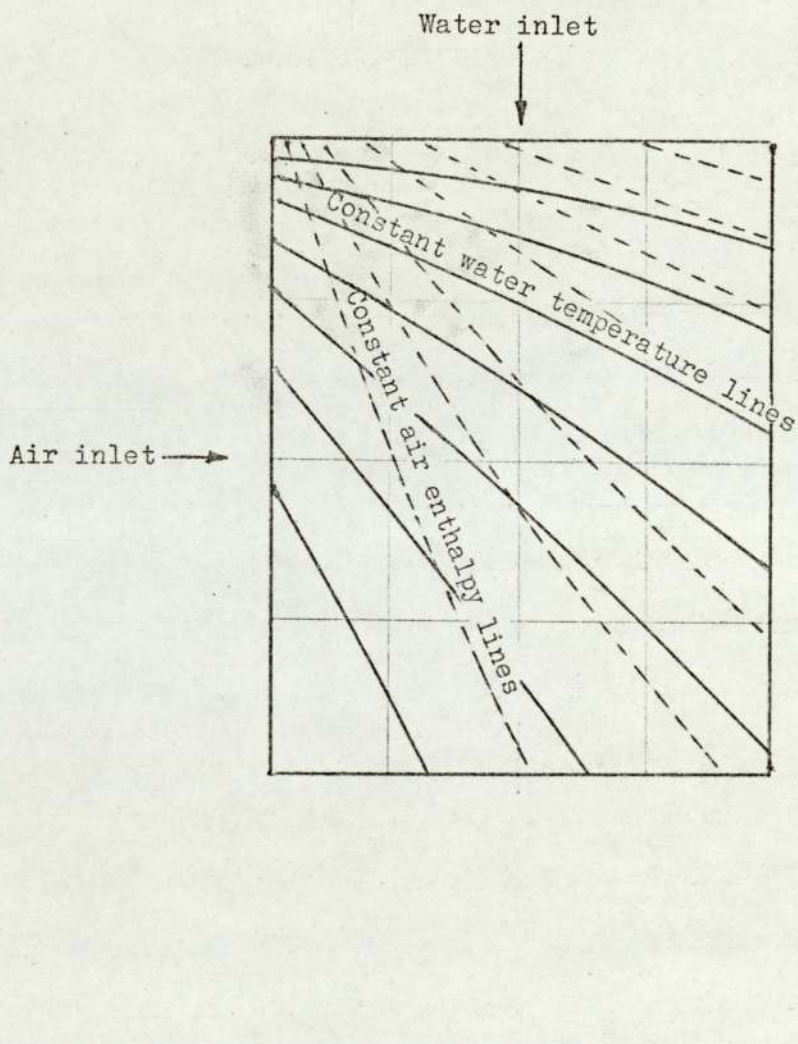
The final leaving conditions are used for calculations on adjacent elements, which enables the evaluation to proceed down and across the columns and rows of elements. A record of the leaving conditions from each of the elements is shown on Fig.2.21.

Zivi and Brand (2), Gurney and Cotter (7), McKelvey and Brooke (10) and the A.S.H.R.A.E. Guide and Data Book (21) use the graphical presentation shown on Fig.2.20 to illustrate for a section of packing the lines of constant water temperature, which are plotted using results obtained from the previous method.

The A.S.H.R.A.E. Guide and Data Book superimposes lines of constant air temperature enthalpy on this graph to give the presentation as shown on Fig 2.23, which is then used to plot the information contained on Fig.2.24. This indicates that the air enthalpy follows the family of curves radiating from the point A, with the air moving across the top of the tower tending to coincide with OA. The air flowing across the bottom of the tower of infinite height will follow a line which tends to coincide with the equilibrium curve AB. The water temperature will follow the family of curves radiating from the point B. The curves will fall between the limits of BO at the air inlet to BA at the air outlet, for a tower of infinite depth.

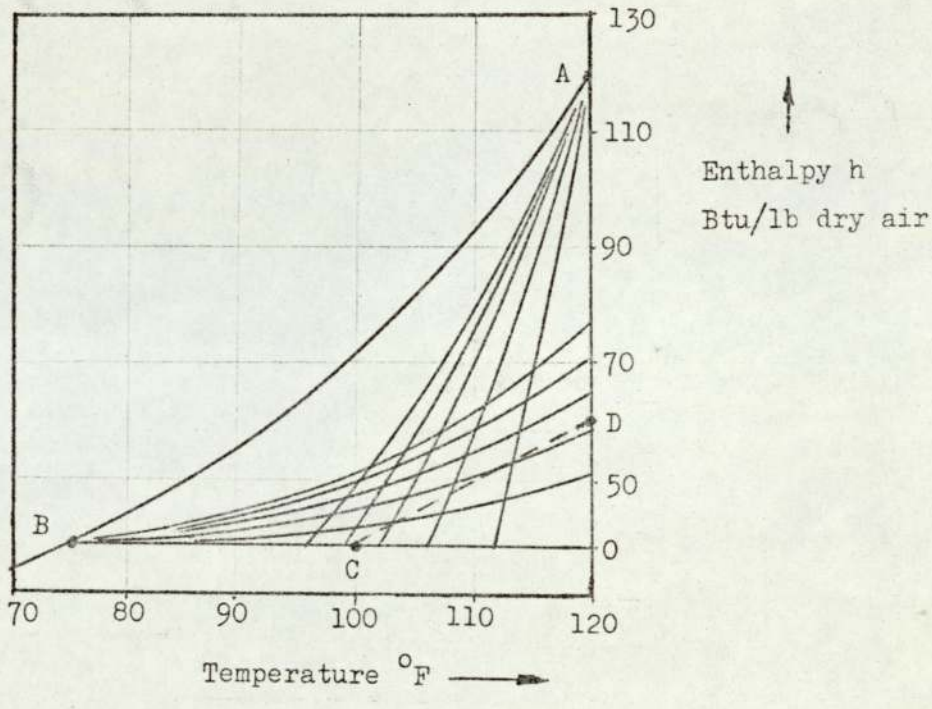
The line CD represents a counterflow tower operating at similar conditions to the crossflow tower. Instead of the single operating line the A.S.H.R.A.E. Guide and Data Book suggests that the equivalent crossflow diagram be considered as an area corresponding to the region of overlap of the two families of curves shown on Fig 2.24.

A slightly different approach is used by Gardner (15)



WATER TEMPERATURE AND AIR ENTHALPY VARIATION THROUGH A CROSSFLOW COOLING TOWER

Fig 2.23



CROSSFLOW COOLING DIAGRAM

Fig 2.24

who develops his reduced equilibrium curve method to accommodate the crossflow process. With water falling in a y direction and air travelling horizontally in an x direction he gives the heat balance as :

$$\partial h_G/\partial x + c_L(L/G)(\partial t_L/\partial y) = 0 \qquad (2.11)$$

for an overall L/G value (L/A)/(G/A'). The transfer rate equation is :

$$G\partial h_G/\partial x = K\,(h_L - h_G) \qquad (2.12)$$

where K is the overall energy transfer coefficient and h_L represents the reduced equilibrium line. Gardner represented h_L by the equation :

$$h_L = E + F\exp(Dt_L) \qquad (2.13)$$

To obtain a useful result he uses the following definitions.

θ the dimensionless temperature as :

$$\theta = D\,(t_L - t_{Lo}) \qquad (2.14)$$

and a dimensionless enthalpy ϕ as :

$$\phi = (h_G - h_{Go})/F\exp(Dt_{Lo}) \qquad (2.15)$$

the subscript o referring to an arbitrary position in the tower. Thus the dimensionless enthalpy difference is :

$$\phi_{Lo} = (h_{Lo} - h_{Go})/F\exp(Dt_{Lo}) \qquad (2.16)$$

Also $X = xK/G$ (2.17)

and $Y = KFD\exp(Dt_{Lo})y/c_L L$ (2.18)

where $h_{Lo} = E + F\exp(Dt_{Lo})$ (2.19)

By substitution the heat balance equation becomes :

$$\partial\phi/\partial X + \partial\theta/\partial y = 0 \qquad (2.20)$$

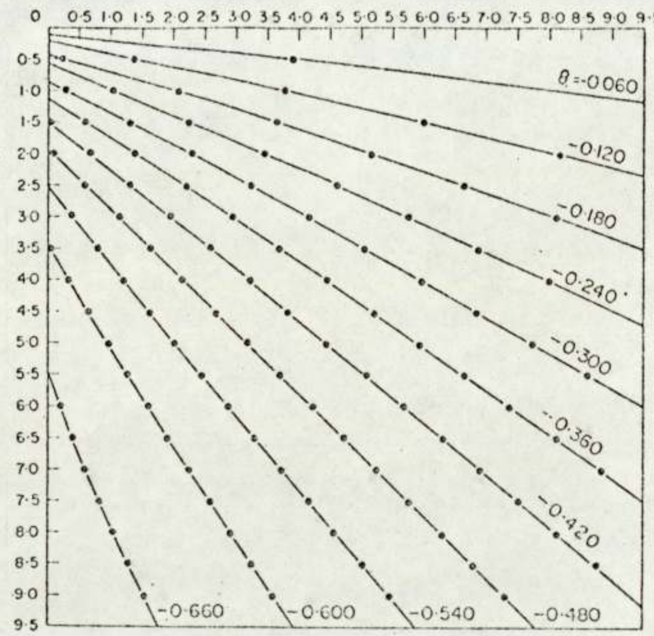
and the transfer equation is

$$\partial\phi/\partial X = \exp(\theta) - \phi + \phi_{Lo} - 1 \qquad (2.21)$$

These equations are solved by computer program. The results are plotted by Gardner to give the chart shown on Fig. 2.25. This chart shows lines of constant reduced mixed water temperature θ_M plotted on X-Y co-ordinates for :

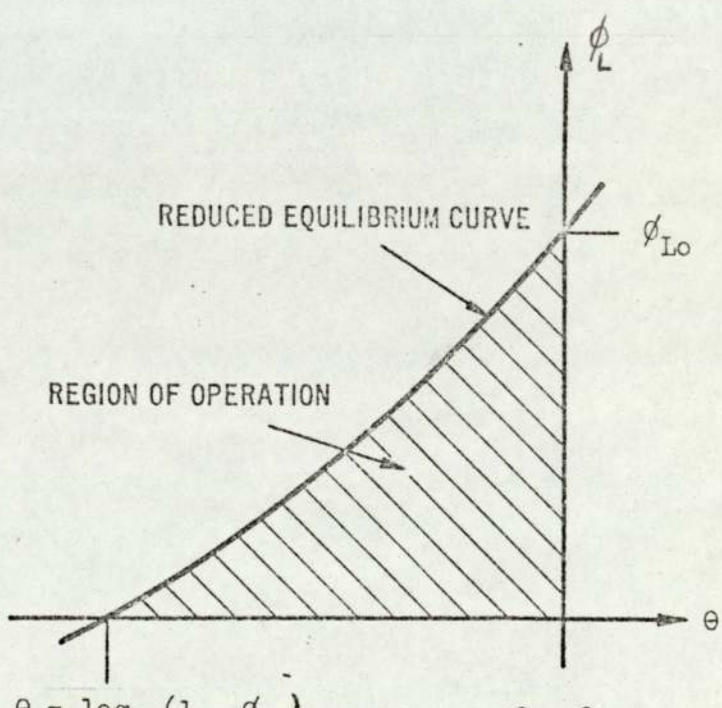
X Direction

Y Direction



CURVES OF CONSTANT REDUCED MIXED WATER TEMPERATURE θ_m IN CROSSFLOW FOR $\phi_{Lo} = 0.5$

Fig 2.25



REGION OF OPERATION IN A CROSSFLOW PROCESS

Fig 2.26

$$\phi_{Lo} = 0.5 \qquad (2.22)$$

where:

$$\theta_M = \left((1/X)\int_0^X \theta \, dX\right)_{Y=\text{Constant}} \qquad (2.23)$$

The number of charts required for cooling tower selection is said to be not more than ten.

The significance of different values of ϕ_{Lo} can be seen from Fig. 2.26 in which the equilibrium curve of equation (2.13) has been put in the reduced form:

$$\phi_L = \phi_{Lo} + \exp(\theta)-1 \qquad (2.24)$$

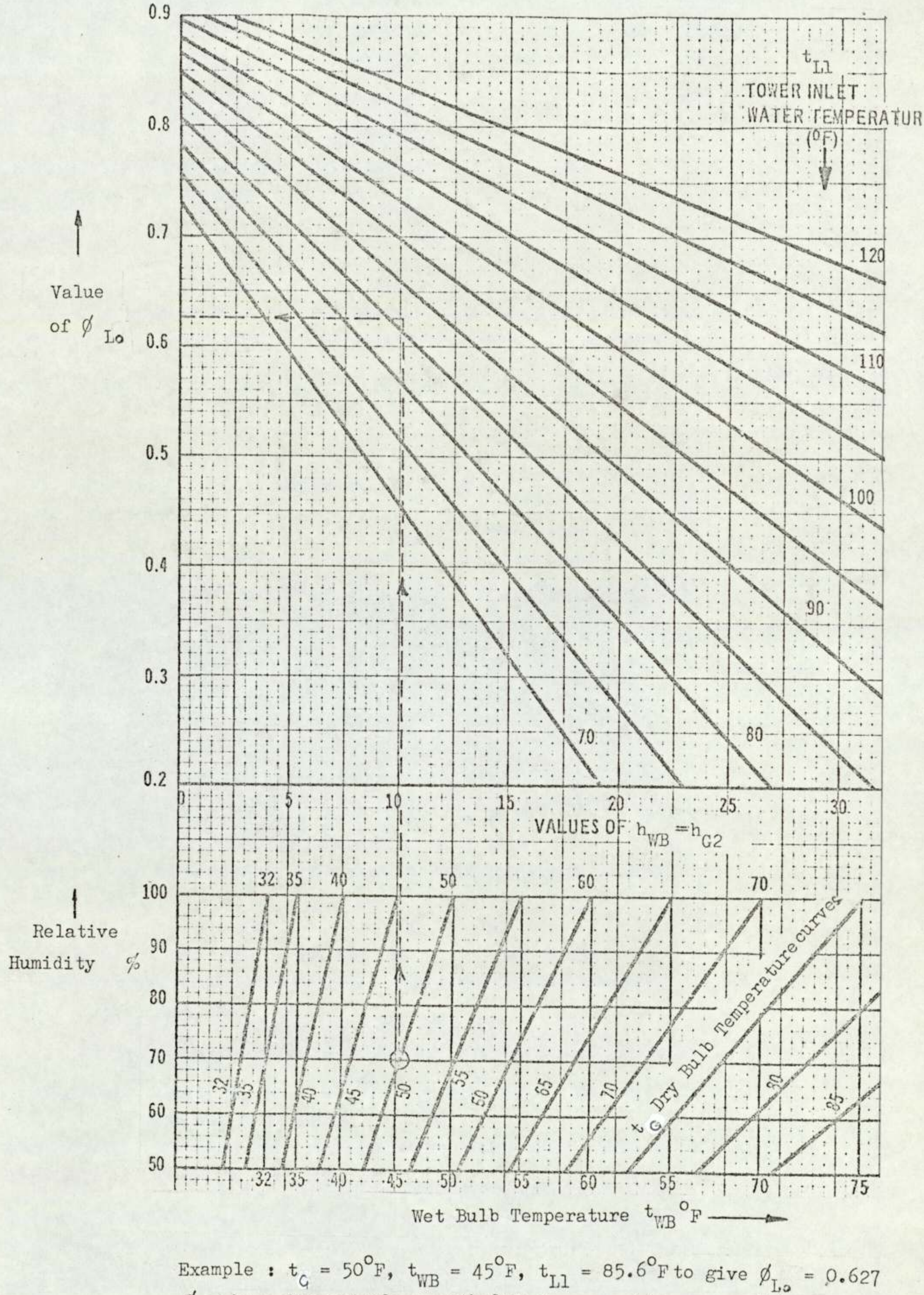
and ϕ_L is plotted versus θ. The origin of the graph represents the boundary condition X = 0 and ϕ_L = 0, Y = 0 and θ = 0. Gardner suggests that conditions inside the tower lie within the shaded area of Fig. 2.26, which is approximately triangular with one curved side. The A.S.H.R.A.E. Guide and Data Book (21) also suggests a similar shaped area of operation as shown on Fig. 2.24.

In a later paper presented jointly by Gardner and MacDonald (31), Gardner's earlier method for estimating the energy transfer in a crossflow cooling tower is considerably simplified. This involved a presentation using three graphs from which the mean mixed water temperature leaving the cross-flow tower can be read directly. These charts are shown on Figs. 2.27 and 2.28.

See P. 59

2.4.2. Summary

By inspection of Fig. 2.21 it can be seen that increasing the pack depth in the direction of air flow, results in a greater volume of water to be cooled whilst the quantity of air remains the same. Furthermore the air becomes progressively warmer as it passes through the increased depth of pack and less effective for cooling the water. A rise in pack height will result in an increased air flow available to cool a constant quantity of water. Thus a situation develops where the physical shape in two

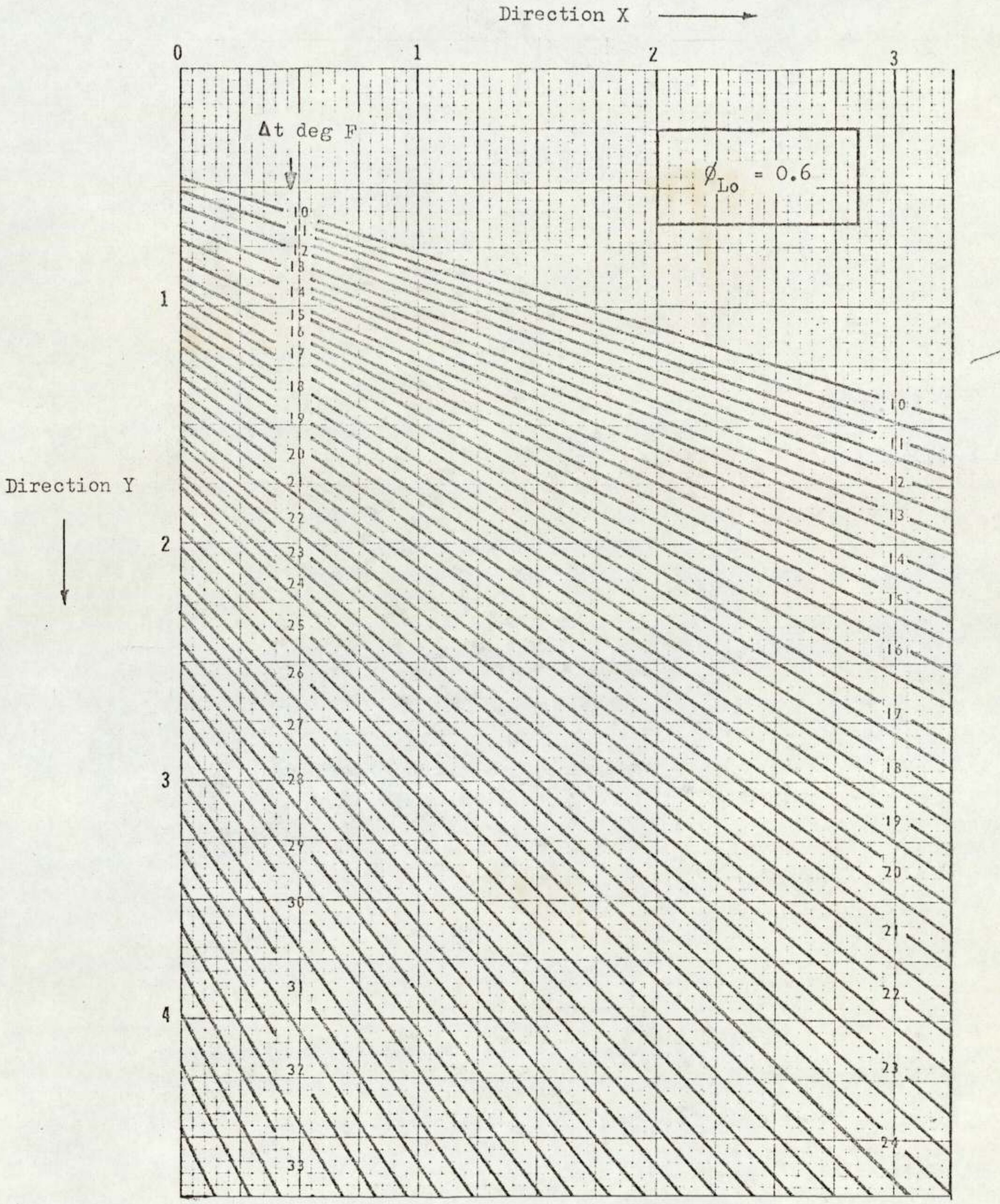


Example : $t_G = 50^oF$, $t_{WB} = 45^oF$, $t_{L1} = 85.6^oF$ to give $\phi_{Lo} = 0.627$

ϕ_{Lo} is a dimensionless enthalpy for an arbituary position.

MASTER CHART FOR DERIVING ϕ_{Lo} FROM THE INLET CONDITIONS

Fig 2.27



Select a chart for ϕ_{Lo} approximately equal to the value found for the example shown on Fig 2.27. In this case the chart corresponds to $\phi_{Lo} = 0.6$. For the known inlet water temperature read a cooling range and hence the proportions of the packing.

Other charts are available for $\phi_{Lo} = 0.4$, 0.5, 0.6 and 0.7 for temperature ranges 3 to 33 deg F covered by two charts for each ϕ_{Lo} value.

CROSSFLOW COOLING TOWER CHART

Fig. 2.28

directions of the pack as well as the pack volume will influence the performance of a crossflow tower.

The results would be applicable to relatively small units and to film flow situations rather than to splash packing.

2.4.1. (Continued):

The crossflow tower equations were put into finite difference form by Schechter and Kang (34) and their solution resulted in the presentation of a set of selection curves which enabled exit water temperature to be obtained for nominated inlet conditions and tower operating parameters. A significant limitation of the method as it stands is that the graphs presented have the ratio of inlet air enthalpy to inlet water enthalpy as a parameter and to cover the full range of possibilities for this ratio, many more graphs would need to be available.

CHAPTER THREE

DISCUSSION OF THE ENTHALPY TEMPERATURE RELATIONSHIP FOR MOIST AIR

3.1 Introduction

The analysis of the energy transfer in a cooling tower can be obtained by evaluation of the Merkel Integral. Solution of this integral requires a suitable enthalpy temperature relationship for moist air, which must be capable of being accurate for the commercially important temperature range 50°F to 135°F. To produce the comprehensive performance data necessary, the enthalpy temperature relationship must be used many times and so computation time can be a significant factor in the choice of relationship.

The following discussion sets out to describe the various types of equation available for use. Since they are all related to a temperature datum for enthalpy this aspect is considered first.

3.2 Temperature Datum for Enthalpy

Two temperature datum values are in common use for calculating the enthalpy of moist air. They are a zero Fahrenheit (0°F) and a zero Centigrade (0°C, 32°F) datum. Zahn (12) like all other American workers uses a zero Fahrenheit temperature datum for enthalpy (and an atmospheric pressure of 1013 millibars) to simplify the calculation of the fundamental temperature enthalpy equation. In Britain and most other countries except the United States of America the zero Centigrade enthalpy datum (0°C, 32°F) and 1000 millibar atmospheric pressure are employed. The Tables of Hygrometric Data in the I.H.V.E. Guide (16) are based on the 32°F datum. The A.S.H.R.A.E. Guide and Data Book (21) uses the zero Fahrenheit enthalpy datum.

At the datum conditions a small quantity of water is present as vapour. This has an enthalpy resulting from the latent heat necessary to produce evaporation. Furthermore the atmospheric pressure value will also be of importance since it will influence the partial pressure of the vapour and hence its mass, which in turn will influence the enthalpy of the vapour present in a given volume of air at the datum condition. In the case of the I.H.V.E. Guide the enthalpy at $32^{\circ}F$ and 1000 millibars is 3.26 Btu/lb. The enthalpy at the datum temperature is referred to as h_D.

3.3 Empirical Equation for Calculating the Enthalpy of Moist Air

3.3.1 Polynomial

The enthalpy temperature relationship can be closely fitted to a second order polynomial equation as used by Zivi and Brand (2), namely :

$$h_L = \varepsilon t_L^{\ 2} + \beta t_L + \gamma \qquad (3.1)$$

for ε, β and γ as constants. Zivi and Brand proposed sets of constants for each of the temperature ranges $83^{\circ}F$ to $95^{\circ}F$ and $90^{\circ}F$ to $115^{\circ}F$ which he states as temperature ranges applicable to most water cooling problems met with in the U.S.A., where high ambients are experienced. In the U.K. lower ambients are the norm and so the scope of the above equation is not sufficient to satisfy selection needs.

First, second and third order polynomial equations were devised for the present work. These equations, shown below, are limited to the temperature range $60^{\circ}F$ to $90^{\circ}F$.

$$h_L = -41.51 + 0.9818 t_L \qquad (3.2)$$

$$h_L = 22.19 - 0.7428 t_L + 0.115\ t_L^{\ 2} \qquad (3.3)$$

$$h_L = -24.33 + 1.152\ t_L - 0.014 t_L^{\ 2} + 0.0001133 t_L^{\ 3} \qquad (3.4)$$

Higher order polynomials could have been found but those of the second order gave adequate accuracy for the temperature range considered. To obtain relationships which embrace a wider range of temperature a higher order polynomial equation in the form of a forward backward difference formula was developed for the present work. The equation was limited to differences up to the fifth order which in turn were calculated from the tabulated hygrometric data published in the I.H.V.E. Guide (16). The equations were suitable for the temperature range 50°F to 170°F. The Appendix 3.1 shows these equations, (3.5) and (3.6), in full detail. They are time consuming to solve by hand calculation but ideally suited for computer evaluation.

3.3.2. Exponential

A straightforward expression for relating saturated air enthalpy to temperature is used by Fuller (1) :

$$h_L = \exp(1.77 + 0.025t_L) \qquad (3.7)$$

who attributed this enthalpy equation to Butcher. The equation holds good for water temperatures in the range 40°F to 130°F. It is based upon an atmospheric pressure of 1013 millibars and a zero Fahrenheit enthalpy datum.

An improved exponential equation is used by Gardner (14) and (15) who expressed the enthalpy as a linear expression plus an exponential function of the form :

$$h_L = (E + Bt_L) + F\exp Dt_L \qquad (3.8)$$

where E, B, F, D are constants.

To simplify the solution of this derived enthalpy equation, Gardner chose a zero value for B and so the other constants then gave :

$$h_L = 10 + \exp(1.954 + 0.0235t_L) \qquad (3.9)$$

This equation applies to a range of water temperature 60 to 90°F. A good fit over a larger range can be obtained if B is chosen to be non-zero as in :

$$h_L = -9.2 + 0.21t_L + \exp(0.9075 + 0.03056t_L) \qquad (3.10)$$

Both these equations are based upon an atmospheric pressure of 1000 millibars and a 32°F enthalpy datum.

3.4 Fundamental Equations for Calculating the Enthalpy of Moist Air

3.4.1. Introduction

Any normal sample of moist air contains two components, a water component and a dry air component, each with a unique enthalpy value. The enthalpy of the moist air is the sum of the enthalpy of dry air (sensible heat) plus the enthalpy of the water vapour present in the dry air, both calculated relative to the same temperature datum t_D. This is a fundamental statement which can be represented by an equation made up of the following parts :

a) Sensible heat of dry air c_{pa} above the datum temperature.

b) Sensible heat of water $c_L \, (t_E - t_D)$ above the datum temperature plus the latent heat of vaporisation ($\mathcal{L}$) and the superheat of the vapour ($c_{pv} \, (t_L - t_E)$)

c) Mass of water vapour (w), normally expressed lb per lb of dry air.

d) Datum, a value chosen for convenience.

Hence the enthalpy of the vapour is given by the equation :

$$h_v = c_L \, (t_E - t_D) + \mathcal{L} + c_{pv} \, (t_L - t_E) \text{ per lb vapour} \qquad (3.11)$$

and the complete fundamental enthalpy equation for moist air becomes :

$$h_L = c_{pa} \, (t_L - t_D) + w(c_L \, (t_E - t_D) + \mathcal{L} + c_{pv} \, (t_L - t_E)) \qquad (3.12)$$

Zahn (12) uses an enthalpy equation based on a zero temperature datum, a known latent heat value, a second order polynomial expression (3.19) for calculating the specific heat of dry air and the Magnus formula (3.29) used to find the mass of water vapour. Ignoring the superheat component (discussed later) his equation becomes :

$$h_L = c_{pa} \, t_L + 0.622 \, p_{vs} \, (c_{pv} \, t_L + 1061)/(P - p_{vs}) \qquad (3.13)$$

Since the constants available for use in the fundamental equation are so numerous, equation (3.12) is often simplified by choosing an appropriate set of constants. Hence the equation can be interpreted in various ways depending on the constants chosen, which in turn will influence the enthalpy value obtained. Some authors might consider constants at $32^{o}F$ while others may choose those for $212^{o}F$ or some other suitable temperature. A better interpretation would be to use the constants at the temperature for which the enthalpy is to be calculated but this is not practical. The Table 3.1 lists some of the constants for various temperatures.

To date a common procedure is not available for selecting the appropriate constants. This has resulted in a liberal interpretation of the constants together with an author's individual assumptions.

The following equation indicates the influence different constants can have on the calculated enthalpy value. Ignoring the effect of superheat the fundamental equation for a $32^{o}F$ enthalpy datum becomes :

$$h_L = c_{pa}\,(t_L - 32) + w\,(c_{pv}\,(t_L - 32) + \mathcal{L}) \qquad (3.14)$$

For those constants corresponding to a temperature of $32^{o}F$ the equation can be written as :

$$h_L^{32} = 0.2402\,(t_L - 32) + w(0.45\,(t_L - 32) + 1075.2) \qquad (3.15)$$

For constants at $150^{o}F$ the equation is :

$$h_L^{150} = 0.2409\,(t_L - 32) + w(0.45\,(t_L - 32) + 1009) \qquad (3.16)$$

The use of these and other derived equations illustrating how the chosen constants affect the calculated enthalpy values are shown in Appendix 3.2.

Often those constants corresponding to the temperature datum for enthalpy are considered satisfactory, as illustrated by equation (3.15) which is identical to that mentioned in the British Standard (5).

Temp ^{o}F	Lewis relationship λ	Specific Heat Btu/lb ^{o}F at constant Pressure		Latent heat of evaporation Btu/lb $\mathcal{L}$	No. of grains at 100% RH	Specific heat of water at one atmosphere.
		Air c_{pa}	Moist Air c_{pm}			
30	-	.2402	.2417	1093	24.4	
32	-	.2402	-	1075.2	26.8	
35	-	.2402	.2420	-	28.0	
40	-	.2402	.2425	1070.6	36.9	
50	0.87	.2402	.2436	1065	54.1	1.0
60	-	.2403	.2452	1059.3	78.4	
70	-	.2403	.2473	1053.7	111.9	
80	-	.2404	.2502	1048.1	157.8	
90	0.88	.2404	.2541	1042.4	220	
100	-	.2405	.2595	1036.7	305	1.005
110	-	.2405	.2666	1031	400	
120	-	.2406	.2762	1025.3	576	
130	0.89	.2407	.2894	1019.5	800	
140	-	.2408	.3075	-	-	
150	-	.2409	.3332	1009	-	1.007
160	0.90	.2409	.3500	-	-	
170	-	.2411	.4285	-	-	
180	-	.2412	.5258	-	-	
190	-	-	-	-	-	1.01
212	-	-	-	-	-	

CONSTANTS FOR USE WITH THE FUNDAMENTAL ENTHALPY EQUATION

Table 3.1

Table 3.1

3.4.2 <u>Sensible Heat of Dry Air (c_{pa})</u>

The specific heat of dry air is a function of the air temperature. That is :

$$c_{pa} = f\ (t) \qquad (3.17)$$

and to be rigorous the equation for the sensible heat of dry air should be set up in differential form and integrated between the appropriate limits giving :

$$h_a = c_{pa} \int dt = \int f(t)dt \qquad (3.18)$$

Zahn (12) uses a second order polynomial equation to express the specific heat of air at any temperature in the range 10^oF to 300^oF. This is :

$$c_{pa} = 0.2401457 + 2.487 \times 10^{-7}t + 2.9907 \times 10^{-8}t^2 \qquad (3.19)$$

In the temperature range applied to cooling towers, namely 50^oF to 135^oF the specific heat variation is small. This fact is shown clearly in Table 3.1. Thus a reasonable approximation for specific heat in this temperature range would be 0.2405. Thus equation (3.18) can be rewritten as :

$$h_a = 0.2405 \int_{32}^{t_L} dt = 0.2405\ (t_L - 32) \qquad (3.20)$$

3.4.3 <u>Enthalpy of Water Vapour</u> (h_v)

The Smithsonian Tables (6) quote a second order polynomial equation :

$$h_v = 638.9 + 0.3745\ (T - 100) - 0.00099\ (T - 100)^2 \ \text{cal/g} \qquad (3.21)$$

for the water temperature expressed in degrees Centigrade. This equation, based upon a zero Centigrade enthalpy datum is used to calculate the enthalpy values shown in the Smithsonian Tables.

The equation (3.11) is the fundamental equation for the enthalpy of water vapour h_v although it is often simplified. Lichtenstein (3) neglects the superheat and the sensible heat of the vapour which he considers small compared with the latent heat required for evaporation. He represents the water vapour enthalpy equation in the form:

$$h_v = w\mathcal{L}$$

which assumes a constant value for the latent heat. The component parts of these equations are discussed in the following subsections.

Latent Heat ($\mathcal{L}$)

McKelvey and Brooke (10) assume that the difference in temperature between the air and water interface is so small that the latent heat value can be considered constant, as assumed by Lichtenstein (3). The value given to the latent heat will depend upon the temperature at which evaporation takes place but often the value at the temperature datum for enthalpy is taken.

The Smithsonian Tables (6) express the latent heat $\mathcal{L}$ as an equation obeying a non-integer power law of the form :

$$\mathcal{L} = 94.210\,(365 - T)^{0.3130} \text{ cal/g} \qquad (3.22)$$

for temperature T expressed in degrees Centigrade.

Specific Heat of Water Vapour (c_{pv})

The value of the specific heat of water vapour is influenced by temperature. Most workers consider it constant for the purposes of cooling tower work since its effect upon the final enthalpy value is small. Zahn (12) uses :

$$c_{pv} = 0.44$$

Gurney and Cotter (7), McKelvey and Brooke, the British Standard (5) and others use :

$$c_{pv} = 0.45$$

Specific Heat of Moist Air (c_{pm})

Often referred to as the humid heat but consisting of the specific heat of dry air plus the specific heat of water vapour described by the equation :

$$c_{pm} = c_{pa} + w\,c_{pv} \qquad (3.23)$$

Zahn (12) analyses the above equation by making use of his quadratic equation (3.19) for calculating the specific heat of air c_{pa}. On the basis of this equation and the value of $c_{pv} = 0.45$, the humid heat can be found but for those temperatures in excess of $135^{o}F$ the equation should be reappraised. Zahn and McKelvey and Brooke quote a similar equation for humid heat, namely :

$$c_{pm} = 0.241 + 0.45\ w \qquad (3.24)$$

where the specific heats are assumed constant for the range of temperature encountered in cooling tower work.

Superheat

The enthalpy of water vapour at a known temperature can be found by considering water heated from $32^{o}F$ to the temperature t and then vaporised at this temperature, shown as route AFED on Fig. 3.1 and represented by the equation :

$$(h_v)^a = w((c_L)^t\ (t - 32) + (\mathcal{L})^t\ ) \qquad (3.25)$$

for constants taken at temperature t. Alternatively the water can be heated to some intermediate temperature t_E, and then vaporised at this temperature. The vapour is then heated from t_E to temperature t i.e. superheated. The final enthalpy of the water vapour is given by :

$$(h_v)^b = w\ (\ (c_L)^{50}\ (50 - 32) + (\mathcal{L})^{50} + (c_{pv})^t\ (t-50)\ ) \qquad (3.26)$$

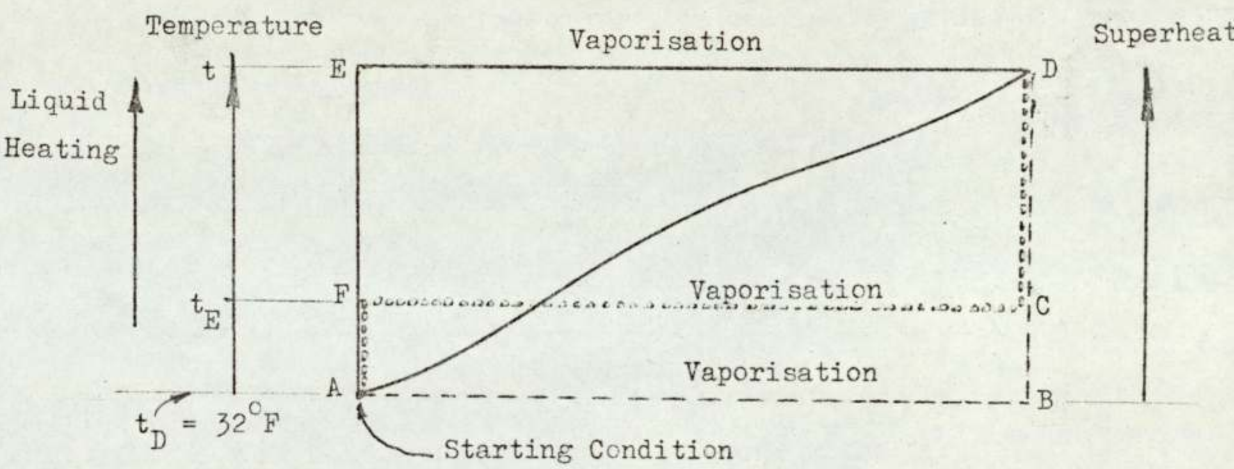
which is shown as route AFCD on Fig. 3.1.

This equation can be simplified by vaporising at the datum temperature $t_D = 32^{o}F$, followed by superheating from the datum to temperature t. This is shown as route ABCD on Fig.3.1 such that :

$$(h_v)^c = w\ (\ (c_{pv})^{32}\ (t-32) + (\mathcal{L})^{32}\ ) \qquad (3.27)$$

In reality the fluid probably follows the route AD.

Using the equations (3.25), (3.26), (3.27) and selecting the appropriate constants from Table 3.1, it is shown in the



——— Route AFED with final enthalpy $(h_v)^a$

ooooooo Route AFCD with final enthalpy $(h_v)^b$

– – – – Route ABCD with final enthalpy $(h_v)^c$

ENTHALPY OF VAPORISATION

Fig 3.1

following example that the choice of route does not have any appreciable effect upon the final enthalpy value calculated.

For the example where the specific heat of water vapour is c_{pv} = 0.45 and a final temperature $t = 150^{o}F$ the above equations for 1 lb of water become:

$$(h_v)^a = 1\ (\ (150-32) + 1009) \qquad = 1127.0\ \text{Btu}$$

$$(h_v)^b = 1\ (\ (50-32) + 1065 + 0.45(150-50)\ ) = 1128.0\ \text{Btu}$$

$$(h_v)^c = 1\ (\ (1075.2) + 0.45\ (150-32)\ ) \qquad = 1128.3\ \text{Btu}$$

These values show very close agreement for all three equations and hence confirm the above comment. The equation (3.27) is utilised in the subsequent computer analysis. This is also the equation used to compile the enthalpy temperature tables in the I.H.V.E. Guide (16).

Mass of Water Vapour w

The specific humidities w are proportional to the partial pressures p_v and the humidity differences are proportional to the partial pressure differences. More particularly the mass of the water vapour is expressed as a function of the vapour pressure of water by the equation for specific humidity (or moisture content) as used by the British Standard (5), Zahn (12) and Mikyaska and Reinisch (13), given by:

$$w = (18\ p_v)\ /\ (28.969\ (P-p_v)\ )$$

In cooling tower work, saturated vapour pressures apply and hence:

$$w = 0.622\ (p_{vs}\ /(P-p_{vs})) \qquad (3.28)$$

where the water vapour and dry air are expressed in the same units, namely pounds of water vapour per pound of dry air or grammes of water vapour per gramme of dry air.

The tables of saturation vapour pressure given in the British Standard (5) have been computed from the formula of Goff and Gratch, and quoted therein. A close approximation to the saturation vapour pressure at a given temperature can be obtained from the Magnus formula of 1844, which is as follows :

$$\log_{10} p_{vs} = JT/(H+T) + I' \text{ millibars} \qquad (3.29)$$

H, I' and J are constants with the values as shown in Appendix 3.1.

Zahn (12) devised a more complicated exponential form of the vapour pressure equation, namely :

$$p_{vs} = \exp(-M-(M^2 - 4NO)^{0.5}/2N) \text{ins Hg} \qquad (3.30)$$

Where O and M are functions of temperature and N is a constant. M, N and O have the values shown in Appendix 3.1. The results from this equation agree with the A.S.H.R.A.E. Guide (21) for temperatures in the range 0°F to 300°F.

An exponential equation is used by Mikyska and Reinisch (13) to indicate the relationship between the pressure of water vapour and its temperature as shown by :

$$p_{vs} = \exp\,(-5656(1/(460 + t) - 1/(460 + 212) + 8.2 \log_e 272/(460+t)$$
$$-0.00138\,(212-t) - 1.139) \text{ psig} \qquad (3.31)$$

It is claimed that this equation is valid for temperatures in the range 32 to 302°F.

An exponential equation using a third order polynomial is used by MacDonald (4) who claims good accuracy for the range of cooling tower application :

$$p_{vs} = \exp(12.054 + 3770.17/t - 6.359\text{x}10^6/t^2 + 995\text{x}10^6/t^3) \qquad (3.32)$$

3.5 Summary

The choice of the 32°F enthalpy datum figure is fortunate in view of the forthcoming change from the Imperial system of units to the International System of units in the early 1970's. A 32°F datum will then become a 0°C datum, and a 1000 millibar atmospheric pressure will be taken as the bar, the standard value for barometric pressure. Thus air enthalpy in Btu units can easily be converted to a Chu unit without any need to consider the datum levels.

The accuracy of the fundamental equation will depend upon the assumptions made in its simplification and the constants chosen for its analysis. If the full equation is used, in conjunction with the appropriate constants and an accurate equation to find the mass of water vapour employed, then a high degree of accuracy can be achieved. Most authors choose to simplify the equation, and in this form, sufficient accuracy can be obtained for cooling tower purposes.

The abbreviated enthalpy equation adopted for use in this paper is the same as that used in the I.H.V.E. Guide (16), namely equation (3.15) which is based upon constants taken at 32°F. This equation is similar to the equation used by MacDonald (4), which is :

$$h = 0.24\ (t-32) + w\ (1061.2 + 0.438t) \qquad (3.33)$$

where w is found from his saturated vapour pressure equation (3.32) and the equation (3.28). If the Magnus formula is used in equation (3.33) to find w then this becomes the revised equation (3.34) as referred to in Chapter Four, Table 4.3 and 4.6.

McKelvey and Brooke (10) use the following equation :

$$h = 0.24\ (t-32) + w\ (1075 + 0.45t) \qquad (3.35)$$

The forward backward difference equations (3.5) and (3.6) are also used but these equations are not suitable for more than one atmospheric pressure without first revising the difference values, whilst the fundamental equation can be readily used for any nominated atmospheric pressure provided equation (3.28) is applied correctly in equation (3.15).

The accuracy of these equations is discussed in Chapter Four.

APPENDIX 3.1

FURTHER DETAILS OF EQUATIONS PRESENTED IN CHAPTER THREE

a) Saturated Vapour Pressure

Equation 3.29

$J = 7.5$

$H = 237.3$

$I' = 0.78571$

$T^{\circ}C = (t - 32)\,(5/9)^{\circ}F$

such that :

$$P_{vs} = \exp\,(9.594\,(t-32)/(237.3 + 0.5555(t-32))+1.8091) \quad \text{millibars.} \quad (3.29)$$

Equation 3.30

$N = -0.07086745$

$M = 0.001626943t - 1.0$

$O = -0.00004286517t^2 + 0.03533457t - 2.519124$

b) Enthalpy

Equation 3.1

Temperature Range.	ε	β	γ
$83^{\circ}F - 95^{\circ}F$	0.01742	-1.7460	71.97
$90^{\circ}F - 115^{\circ}F$	0.02408	-2.9825	129.23

Equations (3.5) and (3.6)

The forward backward difference equation is based upon that given by Bull (28). When reduced to the fifth order the forward difference formula is :

$$h = f_o + z\Delta f_o + z(z-1)\Delta^2 f_o/2! + z(z-1)(z-2)\Delta^3 f_o/3! + z(z-1)(z-2)(z-3)\Delta^4 f_o/4! + z(z-1)(z-2)(z-3)(z-4)\Delta^5 f_o/5!$$

abd the backward formula is :

$$h = f_o + z\nabla f_o + z(z+1)\nabla^2 f_o/2! + z(z+1)(z+2)\nabla^3 f_o/3! + z(z+1)(z+2)(z+3)\nabla^4 f_o/4! + z(z+1)(z+2)(z+3)(z+4)\nabla^5 f_o/5!$$

These equations are reduced to

forward :

$$h = X_o + zX_1 + z(z-1)X_2 + z(z-1)(z-2)X_3 + z(z-1)(z-2)(z-3)X_4$$
$$+ z(z-1)(z-2)(z-3)(z-4)X_5 \qquad (3.5)$$

backward :

$$h = X_o + zX_1 + z(z+1)X_2 + z(z+1)(z+2)X_3 + z(z+1)(z+2)(z+3)X_4$$
$$+ z(z+1)(z+2)(z+3)(z+4)X_5 \qquad (3.6)$$

where the constants z and X in equations (3.5) and (3.6) have the values shown in Table 3.2. These values are obtained from the derived difference table based on the equivalent enthalpies appropriate to a temperature range 50 to 170°F taken at five degree intervals.

Range of Temperature	50 to 90°F		90 to 130°F		130 to 170°F	
	Forward Diff	Backward Diff	Forward Diff	Backward Diff	Forward Diff	Backward Diff
	$50 \leq t \leq 70$	$90 \geqslant t \geqslant 70$	$90 \leqslant t \leqslant 110$	$130 \geqslant t \geqslant 110$	$130 \leqslant t \leqslant 150$	$170 \geqslant t \geqslant 150$
X_0	12.72000	48.60000	48.60000	149.80000	149.80000	522.4
X_1	2.94000	6.50000	7.50000	19.80000	23.00000	82.4
X_2	0.16000	0.35000	0.45000	1.30000	0.80000	5.3
X_3	0.00834	0.00000	0.08333	0.00000	0.56666	-0.33333
X_4	0.00042	-0.07917	-0.02083	-0.02916	0.07083	0.09583
X_5	0.00025v	-0.00333	0.00500	-0.00833	-0.11917	0.17417
z	(t-50)/5	(t-90)/5	(t-90)/5	(t-130)/5	(t-130)/5	(t-170)/5

CONSTANTS FOR USE WITH THE FORWARD BACKWARD DIFFERENCE

ENTHALPY EQUATIONS

TABLE 3.2

TABLE 3.2

APPENDIX 3.2

EFFECT OF CONSTANTS ON THE ENTHALPY VALUE.

From equation (3.15) the enthalpy for a temperature of $132^{o}F$ is :

$$(h_L^{32})_{132} = 0.2402\ (132-32) + (840/7000)(0.45\ (132-32)+1075.2)$$

$$= 158.52 \text{ Btu per lb dry air}$$

From equation (3.16) the enthalpy for $132^{o}F$ is

$$(h_L^{150})_{132} = 0.2409\ (132-32) +(840/7000)(\ 1.0\ (132-32)\ +1009)$$

$$= 157.17 \text{ Btu per lb dry air}$$

where the equivalent moisture content is 840 grains per lb of dry air and 7000 grains of water vapour weighs one pound. The I.H.V.E. Guide (16) gives an enthalpy for a temperature of $132^{o}F$ as :

$$(h_L)_{132} = 158.3 \text{ Btu per lb dry air.}$$

Similarly from equation (3.15) the enthalpies at a temperature of $60^{o}F$ are :

$$(h_L^{32})_{60} = 0.2402\ (60-32)+(78.4/7000)(0.45(60-32)+1075.2)$$

$$= 18.92 \text{ Btu per lb dry air}$$

while equation (3.16) for $60^{o}F$ gives the enthalpy :

$$(h_L^{150})_{60} = 0.2409\ (60-32)+(78.4/7000)(0.45(60-32)+1009)$$

$$= 18.22 \text{ Btu per lb dry air}$$

where the equivalent moisture content is 78.4 grains per lb of dry air at a temperature of $60^{o}F$. The I.H.V.E. Guide gives an enthalpy of :

$$(h_L)_{60} = 18.92 \text{ Btu per lb dry air}$$

For a rise in saturated air temperature from $60^{o}F$ to $132^{o}F$ the change in enthalpy :

a) for constants at $32^{o}F$ is :

$$\triangle h = 158.52 - 18.92$$

$$\underline{\underline{\triangle h = 139.60}} \text{ Btu per lb dry air}$$

b) for constants at $150^{\circ}F$

$$\Delta h = 157.17 - 18.22$$

$$\underline{\Delta h = 138.95} \text{ Btu per lb dry air}$$

From the values tabulated in the I.H.V.E. Guide the enthalpy difference is :

$$\Delta h = 158.3 - 18.92$$

$$\underline{\underline{\Delta h = 139.38}} \text{ Btu per lb dry air}$$

Comparison of these enthalpy differences suggests that the fundamental enthalpy equation as used in the I.H.V.E. Guide is based upon constants taken at a temperature of $32^{\circ}F$, which is in fact the case.

CHAPTER FOUR

COMPUTATION TIME AND ACCURACY OF EQUATIONS

4.1 Calculating Techniques

The equations to be considered are many and of varying complexity. To carry out a satisfactory investigation into their suitability, a reliable method of analysis is needed. This can best be satisfied by using a calculating machine, such as a computer.

4.2 Choice of Computer Language

In the first instance an Elliott 803 machine was used. Of the three languages available (Algol, Machine Code, Autocode) the Autocode language was chosen as the most suitable for the many arithmetic operations since it was easy to use and gave an optimum computer operating time. In consequence this section is based upon the use of this language.

4.3 Computation Time

4.3.1 Introduction

A computer program is written for each of the equations considered. Each program uses the same input data; the programs are arranged to give similar output information. The results of timed computer runs are shown in the following tables (4.1), (4.2) and (4.3). The time required to perform these calculations is directly related to the program length and complexity. The times quoted are not a measure of the true calculation time (since an increment of input and output time is included) but serve only to compare one program operating time with another.

4.3.2 Saturated Vapour Pressure Relationship

Equations (3.29), (3.30) and (3.32) each represent an exponential equation in temperature t for which a computer program is available. Each program takes between 0.78 and 0.90 seconds to obtain a solution for a particular temperature.

Equation	(3.29)	(3.30)	(3.32)
Input Data	From 30^oF by 5 degrees to 180^oF		
Time required to read the program into the computer secs.	10.5	19.5	12
Total time required to read the input data and give the output temperature and its equivalent enthalpy. secs.	25	29	25.5
Average time to Process each piece of data. secs.	0.78	0.9	0.8

$$p_{vs} = \exp\left(9.594(t_L - 32) / (237.3 + 0.555 (t_L - 32)) + 1.8091\right) \quad (3.29)$$

$$p_{vs} = \exp\left(- M - (M^2 - 4NO)^{0.5}/2N\right) \quad (3.30)$$

$$p_{vs} = \exp\left(12.0524 + 3770.17/t_L - 6.359 \times 10^6/t_L^2 + 995 \times 10^6/t_L^3\right) \quad (3.32)$$

TIME REQUIRED TO EVALUATE AND OUTPUT THE VAPOUR PRESSURE

Table 4.1

Table 4.1

Further details are shown in Table 4.1. These same equations are used in the program written for the fundamental enthalpy temperature relationships that follow.

4.3.3 Enthalpy Relationship

Empirical Equation

The program based on equations (3.1), (3.7), (3.9) and (3.10) required between 10 and 17 seconds for entry into the machine. The program for the forward backward difference equations (3.5) and (3.6) is the exception requiring 70 seconds; this is due to the physical length of the program indicating the large number of choices, cycles and arithmetic instructions necessary for its evaluation.

Using a standard piece of data the computing time for each program reveals that the exponential and polynomial equations (3.1), (3.7), (3.9) and (3.10) require about 0.75 seconds to solve each piece of data, whilst the forward backward difference equation (3.5) and (3.6) require 0.5 seconds for a solution to be obtained. See Table 4.2 for further details.

Fundamental Equation

The computer programs corresponding to these equations took between 18 and 25 seconds for the program to be read into the machine. To obtain a solution for one piece of data requires about 1.1 seconds. Details are as shown on Table 4.3.

4.4 Accuracy of Equations

4.4.1 Reference Equations

The expression for the saturated vapour pressure, namely the Magnus formula (3.29) is used as a reference equation. The values obtained from the corresponding computer program are compared with the values published in the I.H.V.E. Guide (16) which themselves are based on the Magnus formula. Deviations between the table values and those obtained from the computer

Equation		(3.1)	(3.5) and (3.6)	(3.7)	(3.9)	(3.10)
Specified Range of Application	^{o}F	-	50 - 170	40-130	60-90	-
Enthalpy Datum	^{o}F	0	32	-	32	32
Input Data		From $30^{o}F$ to $180^{o}F$ in increments of 5^{o} deg. F				
Time required for program to be read into computer	secs.	16	69.5	10	9.5	11.5
Total time required to read all the data, perform the calculation and give an output for temperature and its equivalent enthalpy.	secs.	22.5	15	25	24	24
Average time to process each piece of data.	secs.	0.7	0.47	0.78	0.75	0.75

$h = \varepsilon t_L^{\,2} + \beta t_L + \gamma$ (3.1)

$h = X_o + zX_1 + z(z\pm1)\,X_2 + z(z\pm1)(z\pm2)\,X_3 + \ldots\ldots X_5$ (3.5) and (3.6)

$h = \exp\,(1.77 + 0.025\ t_L)$ (3.7)

$h = -\,10 + \exp\,(1.954 + 0.02352\ t_L)$ (3.9)

$h = -\,9.2 + 0.21t_L + \exp\,(0.9075 + 0.03056t_L)$ (3.10)

TIME REQUIRED TO EVALUATE AND OUTPUT THE EMPIRICAL ENTHALPY

Table 4.2

Table 4.2

Equation	(3.15)	(3.33)	(3.34)	(3.35)
Enthalpy Datum	32	32	32	32
Atmospheric Datum millibars.	1000	1000	1000	1000
Input Data	From $30^{o}F$ to $180^{o}F$ in intervals of 5 deg F			
Time required for program to be read into the computer secs.	18	20	22.5	18
Total time required to read all the data, perform the calculations and give an output for the corresponding temperature, enthalpy and vapour pressure. secs.	35	35	35	35
Average time to process each piece of data. secs.	1.1	1.1	1.1	1.1

$$h = 0.24\ (t_L - 32) + w\ (1075 + 0.45\ (t_L - 32)\ ) \qquad (3.15)$$

$$h = 0.24\ (t_L - 32) + w\ (1061.2 + 0.438\ t_L) \qquad (3.33)$$

h = (equation 3.33) plus use of revised equations for w, (3.29 and 3.28) (3.34)

$$h = 0.24\ (t_L - 32) + w\ (1075 + 0.45\ t_L) \qquad (3.35)$$

TIME REQUIRED TO EVALUATE AND OUTPUT THE FUNDAMENTAL ENTHALPY

Table 4.3

Table 4.3

program are of the order of $\pm$ 0.1%. This small deviation justifies the use of this program to compile a comprehensive list of reference data for use in comparing the accuracy of data from the various saturated vapour pressure relationships.

It is interesting to note that the tabulated saturated vapour pressures published in Perry (19) and the A.S.H.R.A.E. Guide and Data Book (21) show close agreement with the values obtained from the Magnus formula.

The enthalpy tables appearing in Kern (20) are used for comparing the accuracy of the data from the three enthalpy equations (3.1), (3.7) and (3.13), the latter using c_{pv} = 0.44 and c_{pa} = 0.24. These equations are based upon a zero Fahrenheit temperature datum for enthalpy.

The enthalpy relationship based upon a 32^{o}F temperature datum are checked for accuracy by comparing their answers with those of a set of reference values. The reference values are calculated from the enthalpy equation (3.15) together with equations(3.28) and (3.29). The values check closely with the tabulated data in the I.H.V.E. Guide, and hence these equations are used for obtaining the reference values referred to on Table 4.6.

To compare the accuracy of the calculated vapour pressure or enthalpy values with the reference values the following equation is used and the term deviation 'd' adopted, where :

$$d = \frac{\begin{Bmatrix} p_{vs} \\ h_L \end{Bmatrix} \text{reference equation} - \begin{Bmatrix} p_{vs} \\ h_L \end{Bmatrix}}{\begin{Bmatrix} p_{vs} \\ h_L \end{Bmatrix} \text{reference equation.}} \times 100\%$$

4.4.2 <u>Standard Data</u>

To check the accuracies of the pressure and enthalpy expressions a standard piece of data is used. This consists of temperatures in the range 30 to 180°F for intervals of 10 deg. F.

4.4.3 Accuracy of the Saturated Vapour Pressure Relationship Data

The quadratic equation (3.30) prepared by Zahn (12) shows a deviation at 30°F of 2.67% and at 40°F of 1%. From 40°F to 180°F the maximum deviation is ± 0.71%.

MacDonald's (4) saturated vapour pressure equation (3.32) shows negligible deviation (d less than 0.16%) for the temperature range 30 to 110°F. At 120°F the deviation is 0.3% which rises rapidly to 3.25% at 180°F.

Further details are shown on Table 4.4.

4.4.4 Accuracy of the Enthalpy Equation Data

Empirical Relationships 0°F Enthalpy Datum

Fuller (1) found equation (3.7) to be valid for temperatures in the range 40 to 130°F. Outside this range the accuracy of the equation falls off rapidly.

Zivi and Brand (2) use a second order polynomial equation (3.1) for which they specified two sets of constants to cover the temperature range 83 to 115°F. Within this range a deviation of less than 1.5% is achieved, but beyond this range very large deviations occur. See Table 4.5 for details.

Empirical Relationships 32°F Enthalpy Datum

The equations (3.2), (3.3) and (3.4) check to within 0.1 Btu/lb of those values given in the I.H.V.E. Guide (16) for the specified temperature range 60 to 90°F.

The equation (3.9) as used by Gardner (15) expresses the enthalpy values with a deviation less than 0.2% in the temperature range 60 to 90°F. Beyond this range large deviations are experienced. Gardner obtained an acceptable fit over the temperature range 30 to 140°F by designating to B a non-zero value to give equation

Equation		(3.30)	(3.32)
Deviation 'd' expressed as a percentage		d	d
Temperature t ^{o}F	30	2.67	0
	40	1.07	0.12
	50	0	0.16
	60	-0.51	0.11
	70	-0.68	0.08
	80	-0.71	0
	90	-0.60	0
	100	-0.43	0.03
	110	-0.23	0.14
	120	-0.04	0.31
	130	-0.14	0.57
	140	0.29	0.92
	150	0.41	1.36
	160	0.50	1.90
	170	0.56	2.77
	180	0.61	3.25

N.B. Origin of the reference data is the Magnus formula, equation (3.29)

DEVIATION - VAPOUR PRESSURE EQUATIONS

Table 4.4

Table 4.4

(3.10). In the range 30 to 140°F the maximum deviation does not exceed 0.25%. As the temperature increased beyond this top value the deviation rose sharply, such that at 180°F a 21.0% deviation occurred.

The forward backward difference equations (3.5) and (3.6) are developed for a specific range of temperature, namely 50 to 170°F. For ease of use the equation is terminated at the fifth term without jeopardising accuracy, which enables a deviation better than 0.25% within the temperature range 50 to 130°F to be achieved. From 130 to 170°F a maximum deviation of 3.5% is experienced. At 180°F, which is just outside the specified range, the deviation is - 56%.

<u>Fundamental Relationship</u> 0°F Enthalpy Datum

The saturated vapour pressure equation (3.30) is built into the fundamental enthalpy equation (3.13) used by Zahn (12) as well as the second order polynomial equation (3.19) to calculate the specific heat of dry air. With a value of c_{pv} = 0.44 the Zahn equation (3.13) gives deviations less than 2.5% for the temperature range 0 to 200°F when compared with tabulated values shown in Kern (20). See Table 4.5.

<u>Fundamental Relationship</u> 32° Enthalpy Datum

The enthalpy equation (3.33) as used by MacDonald (4) includes his vapour pressure relationship (3.32) as an exponential equation in temperature t. The deviation of values from this enthalpy equation within the temperature range 30 to 120°F is less than 0.25%. For the range 130 to 180°F the deviation rises to 6.25%.

If the Magnus formula is used to calculate the vapour pressure in the MacDonald enthalpy equation (3.33), this gives equation (3.34) which achieves a deviation less than 0.8% for the temperature range 40 to 70°F, whilst the deviation in the

Equations	Empirical / Fundamental	(3.1)	(3.7)	(3.13)+(3.19)+(3.30)
Specified Range of Application	^{o}F	40-130	-	0 - 200
Atmospheric pressure datum	millibars	1013	1013	1013
Deviation 'd' expressed as a percentage		d	d	d
Temperature T ^{o}F	40	-200	-4.45	0.20
	50	-37.7	-0.05	1.30
	60	-12.1	1.46	1.05
	70	-4.66	2.08	1.27
	80	-0.72	1.40	1.02
	90	1.36	1.84	1.53
Note: Reference data is taken from tables published by Kern (20)	100	0.30	2.08	1.65
	110	0.60	2.09	2.03
	120	3.06	2.94	2.26
	130	5.90	4.50	2.40
	140	11.60	6.80	2.35
	150	21.60	12.70	1.25

DEVIATION - ENTHALPY FOR EQUATIONS BASED ON A ZERO FAHRENHEIT TEMPERATURE DATUM

Table 4.5

Table 4.5

range 70 to 170°F does not exceed 2.[illegible]%. Further details are shown on Table 4.6.

4.5 SUMMARY

Time

With the exception of the program representing equations (3.5) and (3.6) which each require 70 seconds, all the other programs considered needed 10 to 25 seconds for computer reading.

The calculation time required for a piece of data depends upon the number and difficulty of the equations to be solved. The fundamental enthalpy equation consists of three separate equations (calculation of vapour pressure, mass of water vapour and enthalpy) which require significantly more computing time than the numerical relationships based on the solution of only one equation. See the results shown in Tables 4.2 and 4.3.

Accuracy

The forward backward difference equation (3.5) and (3.6) and the exponential equation (3.10) for enthalpy give satisfactory accuracy of results with deviations less than 0.25% over the range 40 to 130°F. The former equation gives results with an accuracy of 3.5% for the temperature range 130 to 170°F.

The modified MacDonald equation (3.34) gives satisfactory agreement with a deviation less than 2.0% for the temperature range 30 to 170°F.

Equation	Empirical	(3.5)and(3.6)	(3.9)	(3.10)			
	Fundamental				(3.15)	(3.33)	(3.34)
Specified Range of Application	^{o}F	50 - 170	60-90	32-136	-	-	-
Atmospheric Pressure Datum	millibar	1000	-	-	1000	1000	1000
Deviation 'd' expressed as a percentage		d	d	d	d	d	d
Temperature t ^{o}F	30	-50.00	-30.00	0	-0.30	0.30	1.21
	40	0	- 6.32	-0.13	-0.13	0	0.80
	50	-0.08	- 1.26	-0.08	-0.08	0	0.78
	60	-0.05	- 0.16	-0.05	-0.10	0	0.74
	70	-0.08	- 0.08	0.15	-0.11	0	0.79
	80	-0.25	- 0.16	0.25	-0.08	-0.10	0.83
	90	0	0	0.25	-0.08	-0.08	0.70
	100	0.03	-0.57	0.12	-0.07	-0.08	0.96
Note: Origin of reference data is equation (3.15) in association with equations (3.28) and (3.29).	110	-0.10	1.77	0.04	-0.06	0.06	1.05
	120	-0.04	3.20	0.20	-0.06	0.22	1.33
	130	-0.15	6.30	-0.13	-0.05	-0.50	1.25
	140	2.05	10.00	0.02	-0.04	-1.00	1.35
	150	3.55	14.40	2.05	-0.03	1.60	1.53
	160	1.50	24.40	5.35	-0.03	2.50	1.74
	170	2.10	30.00	11.30	-0.02	4.00	2.02
	180	-56.00	40.60	21.00	-0.01	6.25	2.50

DEVIATION - ENTHALPY FOR EQUATIONS BASED ON A 32^{o}F TEMPERATURE DATUM

Table 4.6

Table 4.6

CHAPTER FIVE

THEORETICAL TREATMENT OF THE COUNTERFLOW COOLING TOWER

5.1 Introduction

In those countries where cooling towers are used the design performance will normally fall into the following categories :

i) A wet bulb temperature in the range 50 to 100°F.

ii) Outlet water temperatures in the band 55 to 135°F.

iii) Approaches greater than 5°F.

iv) A ratio for L/G in the range 0.5 to 3.0.

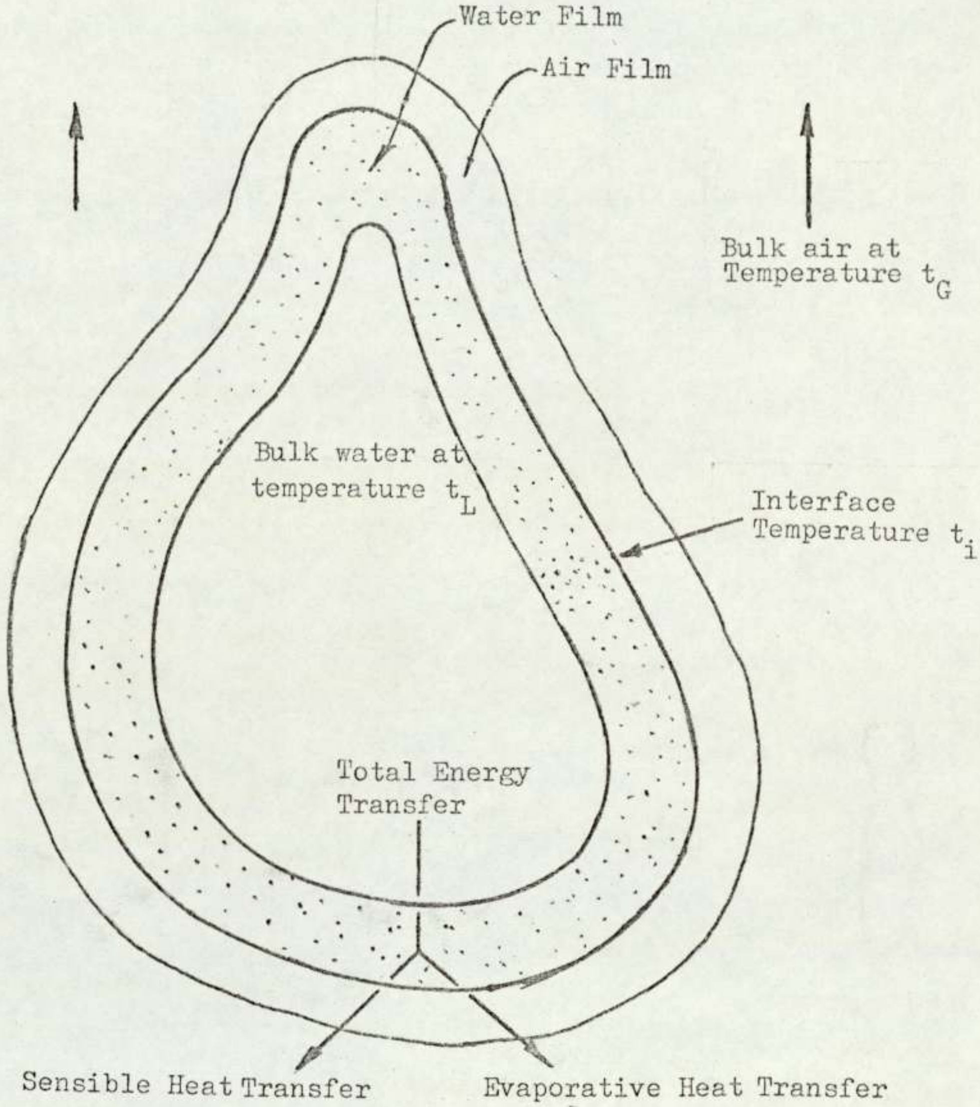
As a result of seasonal weather fluctuations, a cooling tower will probably operate at temperatures below those mentioned in i) and ii) above for most of the time.

5.2 Theory

So that a method of calculating performances can be prepared, it is necessary to consider what happens inside a cooling tower. For this purpose, consider a water droplet with air flowing past as shown on Fig. 5.1.

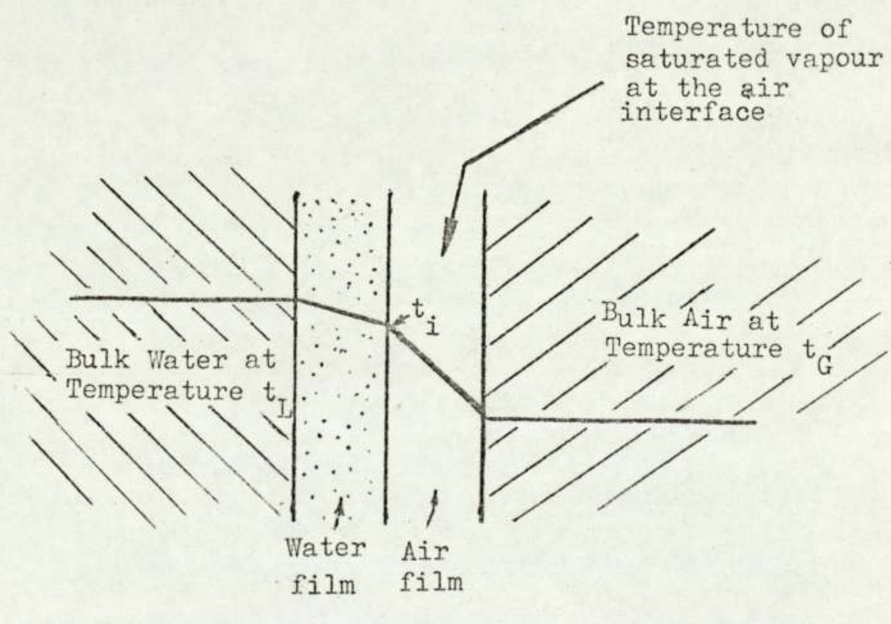
For water cooling to take place the bulk air must have a lower vapour pressure than the bulk water surface. Thus the higher vapour pressure in the water film will cause the transfer of vapour to the air film at the lower vapour pressure. This change in state, converting the liquid to vapour, requires heat (approximately 1000 Btu per lb of water evaporated) which can only come from the main body of water at the expense of reducing its temperature. At the same time, direct heat transfer between the air and water, based on temperature difference, also contributes to the total heat flow.

Between the bulk water and bulk air there is an air - water film, arbitarily separated by an interface. The film is shown on Fig. 5.2. Badger and Banchero (29) consider the interface to be a thin layer of saturated air with a temperature gradient across it. They ultimately delegate this layer to have mean conditions ($t_{\cancel{L}}^{i}$, h_i, w_i,) assuming that no resistance across the interface



For enthalpy: $h_G < h_i < h_L$ For Humidity: $w_G < w_i < w_L$
For a dry bulb temperature: $t_G < t_i < t_L$

AN ENLARGED WATER DROPLET SHOWING CONDITIONS NECESSARY FOR WATER COOLING Fig 5.1



AIR WATER FILM (in a Cooling Process)

Fig 5.2

occurs, and that the two phases are in equilibrium.

Consider a cooling tower having a base area A, where A = 1 square foot, a cooling volume V, containing 'a' square feet of transfer surface per unit volume of packed tower, with water fed to the top of the tower at a rate L_1 and air in at the bottom at a rate G_2, each with the units lb per h ft^2. L and G are the mass flow rates at any point within the tower. Water at temperature t_L is surrounded by an air-water film which is in turn surrounded by air at the dry bulb temperature t_G with an enthalpy h_G and a humidity w_G. The interface is considered as a thin layer of saturated air with an intermediate temperature t_i, an enthalpy h_i, and humidity w_i.

Fig. 2.2 shows a section of cooling tower pack. A mass balance across it gives :

$$dL = Gdw$$

and an energy balance :

$$c_L Ldt = Gdh$$

Assuming a constant mass of water flowing through the tower the total rate of energy transfer from the water to the interfacial layer is :

$$dq_L = c_L Ldt = k_L(adV)(t_L - t_i) \qquad (5.1)$$

and the rate of sensible heat transfer from the interface to the air stream is :

$$dq_s = k_s\,(adV)(t_i - t_G) \qquad (5.2)$$

The mass rate of water vapour transferred by evaporation from the interface to the air is :

$$dm = k_m\,(adV)(w_i - w_G) \qquad (5.3)$$

where w is the mass of vapour present in 1 lb of dry air and hence $(w_i - w_G)$ is a moisture content driving force per unit mass of dry air. Considering the latent heat of evaporation to be a constant $\mathcal{L}$, the corresponding evaporative heat transfer is :

$$dq_E = \mathcal{L}(dm) = \mathcal{L} k_m (adV)(w_i - w_G) \qquad (5.4)$$

i.e.

$$dq_E = k_m (adV)(\mathcal{L} w_i - \mathcal{L} w_G) \qquad (5.5)$$

For the interface to be at equilibrium conditions the total heat gained by the interface is equal to the sum of the sensible and evaporative heats leaving the interface, assuming the radiation and convection effects are negligible. Thus :

$$dq_L = dq_s + dq_E \qquad (5.6)$$

$$dq_L = c_L L dt = k_L (adV)(t_L - t_i) \qquad (5.7)$$

$$dq_L = k_s (adV)(t_i - t_G) + k_m (adV)(\mathcal{L} w_i - \mathcal{L} w_G) \qquad (5.8)$$

Now Merkel utilises the Lewis relationship as mentioned by Gurney and Cotter (7), McKelvey and Brooke (10), Kern (20), and others namely :

$$\lambda = k_s / k_m c_{pm} \qquad (5.9)$$

to give the total energy transfer resulting from the heat and mass transfer processes. Assuming c_{pm} to be constant, a solution based upon enthalpy difference as driving force may be obtained. For temperatures applicable to cooling tower operation, λ has a value in the range 0.87 to 0.90 but for simplification of the above equation a value of unity is adopted for λ, so that :

$$k_s / k_m c_{pm} = 1 \qquad (5.10)$$

Thus :

$$c_L L dt = c_{pm} k_m (adV)(t_i - t_G) + k_m (adV)(\mathcal{L} w_i - \mathcal{L} w_G) \qquad (5.11)$$

$$c_L L dt = k_m (adV)(\mathcal{L} w_i + c_{pm}(t_i - t_D) - \mathcal{L} w_G - c_{pm}(t_G - t_D)) \qquad (5.12)$$

where t_D is a datum temperature given the value of $32^\circ F$, with the enthalpy of the operating line defined by :

$$h = \mathcal{L} w + c_{pm} (t - 32) \qquad (5.13)$$

Simplification gives :

$$c_L L dt = k_m (adV)(h_i - h_G) \qquad (5.14)$$

The equation (5.14) considers the transfer from the interface to the air stream. The interface conditions are difficult to determine. This difficulty is overcome by assuming the water film and interfacial layer to have the same temperature as the bulk water. Since the film resistance to mass transfer is small, then :

$$t_i = t_L$$

An overall system coefficient K, can then be considered to include these assumptions. This enables the driving force to be related to the enthalpy h_L at the bulk water temperature t_L. Thus equation (5.14) becomes :

$$c_L L dt = K\,(adV)(h_L - h_G)$$

Integrating and rearranging gives the Merkel Integral

$$KaV/L = \int dt/(h_L - h_G) \times c_L \qquad (5.15)$$

Since the energy balance is :

$$L\ dt = G\ dh \quad \text{for } c_L = 1$$

then substitution in equation (5.15) gives :

$$KaV/G = \int dh/(h_L - h_G) \qquad (5.16)$$

For a known water temperature range defined by t_{L1} and t_{L2} the first integral equation (5.15) can be written as :

$$KaV/L = \int_{t_{L2}}^{t_{L1}} dt/(h_L - h_G) \qquad (5.17)$$

If V, the pack volume is the product of the plan area A of packing and the height of packing y i.e.

$$V = Ay \qquad (5.18)$$

the above equation can be rearranged to yield :

$$Ka\ (Ay)/L = \int_{t_{L2}}^{t_{L1}} dt/(h_L - h_G) \qquad (5.19)$$

so that :

$$y = (L/KaA) \int_{t_{L2}}^{t_{L1}} dt/(h_L - h_G) \qquad (5.20)$$

The numerical value of the integral on the right of equation (5.20) is the Number of Diffusion Units (NDU) characteristic of the process and is determined wholly by the set of operating conditions. Thus such an integral can be calculated independently of the nature and height of the tower that is to perform the process. However, it appears from equation (5.20) that, whatever the tower may be its height is proportional to the NDU. Inspection of the equation reveals two distinct groups identified as :

i) $L/KaA = (1/Ka)(L/A)$ = Height of a Diffusion Unit (HDU) (5.21)

where L/A is the water mass flow rate per unit area,

and ii) $\int_{t_{L2}}^{t_{L1}} dt/(h_L - h_G)$ = Number of Diffusion Units (NDU) otherwise called the Merkel Integral (5.22)

The term Diffusion Unit is used by Kern (20)

Thus the pack height is given by :

$$y = (HDU)\,(NDU) \tag{5.23}$$

The NDU integral shown above as equation (5.22) is non-dimensional and hence the group L/KaA has the dimension of height to satisfy the equation (5.20) and (5.23).

The NDU equation (5.22) described above cannot be solved simply but it can be considered to have an approximate solution represented by the equation :

$$NDU = (t_{L1} - t_{L2})/f(h_{LM} - h_{GM}) \tag{5.24}$$

Where the enthalpy correction factor 'f' is found from the Steven's Chart shown by Stanford and Hill (23); the mean driving force is represented by $f(h_{LM} - h_{GM})$.

Since steady state conditions prevail at any point inside the tower packing, the heat lost by the water equals the heat gained by the air. Thus the energy balance is :

$$Gdh = c_L\, L\, dt$$

For c_L equal to unity the above equation can be integrated from the water outlet condition to an intermediate condition to give :

$$G\,(h_G - h_{G2}) = L(t_L - t_{L2}) \tag{5.25}$$

and this equation rearranged to yield :

$$h_G = h_{G2} + (L/G)(t_L - t_{L2}) \tag{5.26}$$

This is a straight line passing through the points (t_L, h_G), (t_{L2}, h_{G2}) and with a slope L/G as shown on Fig. 2.2. The line represents the conditions through which the air passes during its rise through the cooling tower. The line is often referred to as the operating line.

The less useful term Transfer Unit can be obtained from a development of equation (5.16). Substituting

$$V = Ay$$

the equation can be rearranged to give :

$$Ka\,(Ay)\,/\,G = \int dh/(h_L - h_G)$$

so that :

$$y = (G/KaA) \int dh/(h_L - h_G)$$

Inspection of this equation again reveals two distinct groups identified as :

i) $G/KaA = (1/Ka)(G/A)$ = Height of a Transfer Unit (HTU)

where G/A is the mass flow rate of air per unit area and

ii) $\int dh/(h_L - h_G)$ = Number of Transfer Units (NTU)

such that the pack height is given by :

$$y = (HTU)(NTU)$$

Furthermore the relationship between the Number of Diffusion Units and Number of Transfer Units is :

$$NTU = (L/G)\,NDU$$

The NTU equation is not very convenient to use in cooling tower calculations where the principal interest lies in the temperature of the water at inlet and outlet to the tower. Hence the NDU integral is used exclusively in the subsequent work.

5.3 Presentation of Equation (5.20)

Because of complications with the solution of the NDU integral in equation (5.20) it is necessary to consider the equation in two parts identified by the terms HDU and NDU as defined by equations (5.21) and (5.22). These equations will require separate analysis.

The presentation of satisfactory performance data will depend upon the analysis of these two equations and the method employed to record the data. In this instance the use of two charts is considered necessary, one to represent each of the equations, so that all possible selection alternatives are adequately recorded.

The solution of the integral in equation (5.20) is complicated by the presence of two variables, enthalpy and temperature which are not easily related. To define the integral completely the ratio L/G, t_{WB}, t_{L1} and t_{L2} must be known. The various methods for solving the integral are listed in Chapter Two; the chosen method of analysis will be the subject of Chapter Six.

5.4 Assumptions.

In developing the theory the main short cuts and simplifications included the following :

a) Specific heat of water is taken as unity.

b) Latent heat of evaporation is taken as constant.

c) The specific heat of moist air is taken as constant.

d) The Lewis relationship is taken as unity.

e) The liquid and gas mass flow rates vary slightly as a result of evaporation, resulting in a varying value for the L/G ratio. This ratio is assumed constant.

f) The driving force is a vertical distance on the

Process Diagram Fig. 2.2.

Where considered necessary the above remarks have been elaborated in Appendix 5.1.

APPENDIX 5.1

JUSTIFICATION OF THE ASSUMPTIONS MADE IN CHAPTER FIVE

a) Specific Heat

From Table 3.1 of Chapter Three it can be seen that the variation in the specific heat of water at atmospheric pressure is so small it can be ignored and a constant value used.

b) Latent Heat of Evaporation

Table 3.1 shows a small variation in latent heat for those temperatures up to 130°F. This justifies the use of a constant value. For further evidence, see the calculations performed in Appendix 3.2.

c) Specific Heat of Moist Air

For the industrially useful range 50°F to 135°F, the variation in specific heat of moist air (see Table 3.1) is considered small enough to be ignored and so a constant value is adopted.

d) Lewis Relationship

$$k_s / k_m c_{pm} = \lambda \qquad (5.10)$$

Experimentation over a wide range of temperature has contributed to the accurate analysis of the Lewis Relationship, whose value for the air-water system is unity at 570°F, 0.9 at 160°F and 0.86 at 0°F. Further values can be extracted from the graph shown by Kern (20). For the purposes of simplification only, a value of unity is used for λ in the above equation so that the energy transfer equations can be readily solved.

If for the sake of argument, the mean specific heat of moist air c_{pm} is 0.25 Btu/lb °F, then the heat transfer rate k_s will be one quarter of the mass transfer rate k_m on the basis of the above assumption.

Kern discusses a method of solving the performance equations for a Lewis Number not equal to unity particularly relevent for the diffusion of gases (H_2 & CO_2) into water.

e) <u>To show the Effect of Evaporation upon the L/G Value</u>

Consider a section of packing having a base area of one square foot. For the section of packing shown on Fig. 2.2 a mass balance gives :

$$dL = dG \tag{5.27}$$

integrating between the limits 1 and 2 gives :

$$\int_2^1 dL = \int_2^1 dG \tag{5.28}$$

and $L_1 - L_2 = G_1 - G_2$ (5.29)

Where L_1 and G_2 are the known conditions.

If the mass flow rate of dry air passing through the tower is G_D, then the mass flow rate of wet air at outlet is :

$$G_1 = G_D\,(1 + w_1) \tag{5.30}$$

and the mass of wet air at inlet is :

$$G_2 = G_D\,(1 + w_2) \tag{5.31}$$

Combining equations (5.30) and (5.31)

$$G_1 = G_2\,(1 + w_1)\ /\ (1 + w_2) \tag{5.32}$$

The change in the mass of air passing through the packing is :

$$L_1 - L_2 = G_1 - G_2 = G_D\,(w_1 - w_2) \tag{5.34}$$

$$\therefore G_1 - G_2 = G_2\,(w_1 - w_2)\ /\ (1 + w_2) \tag{5.35}$$

The ratio of water to air on top of the pack, at position (1) on Fig. 2.2 is :

$$R_1 = L_1/G_1 = (L_1\ /G_2\ (1 + w_1)\ /\ (1 + w_2)\ ) \tag{5.36}$$

$$R_1 = L_1\ (1 + w_2)\ /G_2\ (1 + w_1) \tag{5.37}$$

The ratio of water to air at the bottom of the pack, or position (2) on Fig. 2.2 is :

$$R_2 = L_2\ /G_2 = (L_1 - (G_1 - G_2)\ )\ /\ G_2 \tag{5.38}$$

$$R_2 = (L_1 - G_2 (w_1 - w_2) / (1 + w_2)) / G_2 \qquad (5.39)$$

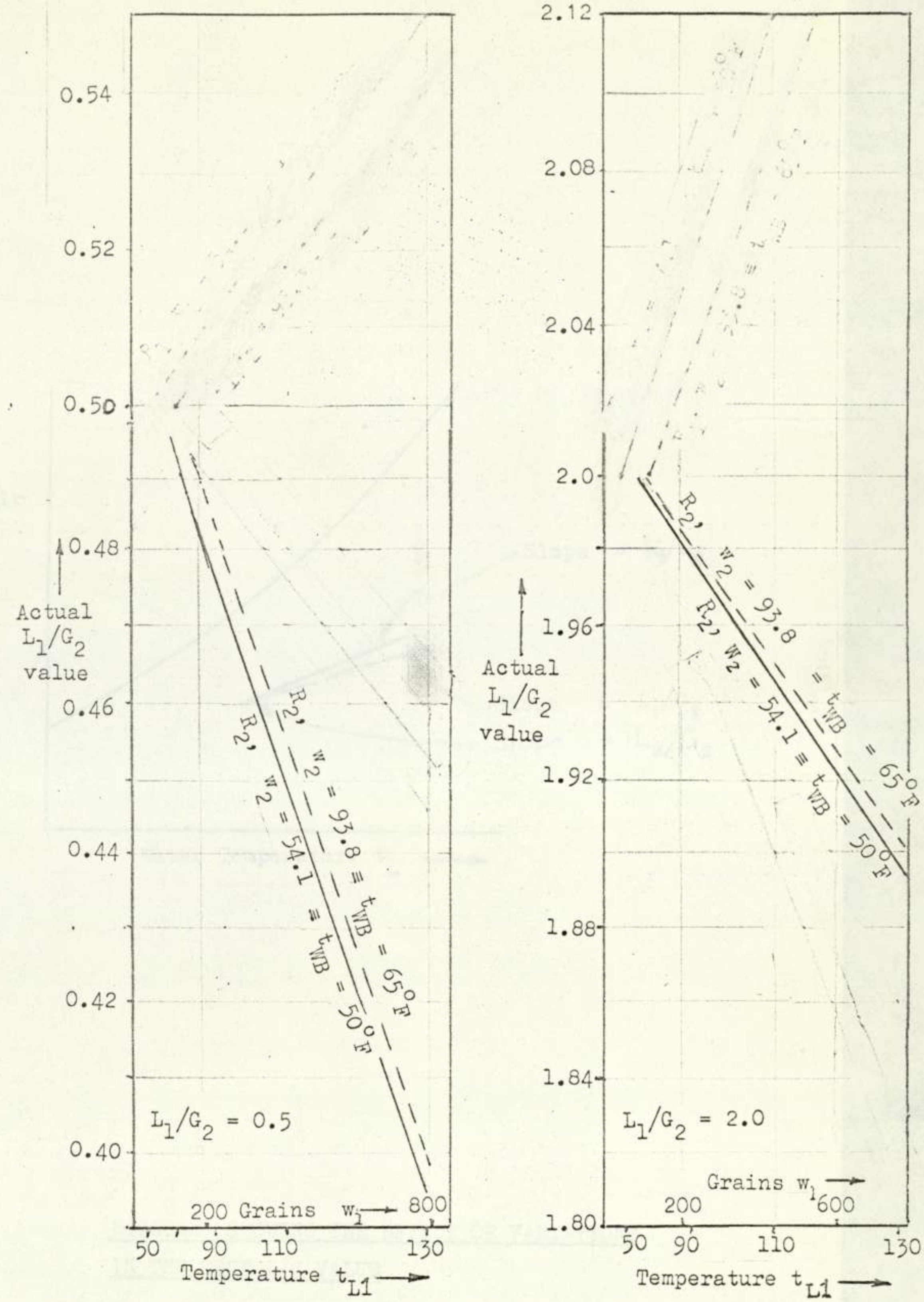
$$R_2 = (L_1 / G_2) - (w_1 - w_2) / (1 + w_2) \qquad (5.40)$$

The ratio L_1/G_2 is the liquid to gas ratio based upon the inlet gas rate G_2 at the bottom of the tower and the inlet liquid rate L_1 at the top of the tower. This value of the ratio is used for the solution of all the appropriate equations mentioned in this thesis.

The results obtained for various values of the equations R_1, (5.37) and R_2, (5.40) are shown on Fig 5.3. From these graphs it can be seen that the phenomenon of mass transfer by evaporation within the system gives mass values of the liquid and gas phases which are continually changing as the air condition through the tower changes but the total mass of the system will always remain at steady state for all points within the system. In consequence, the true L/G value at the top of the tower will be less than the L_1/G_2 value; and at the bottom of the tower the true value will be less than the L_1/G_2 value, thus :

$$L_1/G_1 < L_1/G_2 \text{ and } L_2/G_2 < L_1/G_2 \qquad (5.41)$$

Furthermore, the driving force value becomes progressively influenced at the higher water inlet temperature due to curvature in the operating line arising from variations in the L_1/G_2 values. Even so, a low L_1/G_2 value will reduce the temperature at which this effect becomes significant,



CHARTS SHOWING VARIATION OF L_1/G_2

VALUE FOR VARIOUS CONDITIONS

Fig 5.3

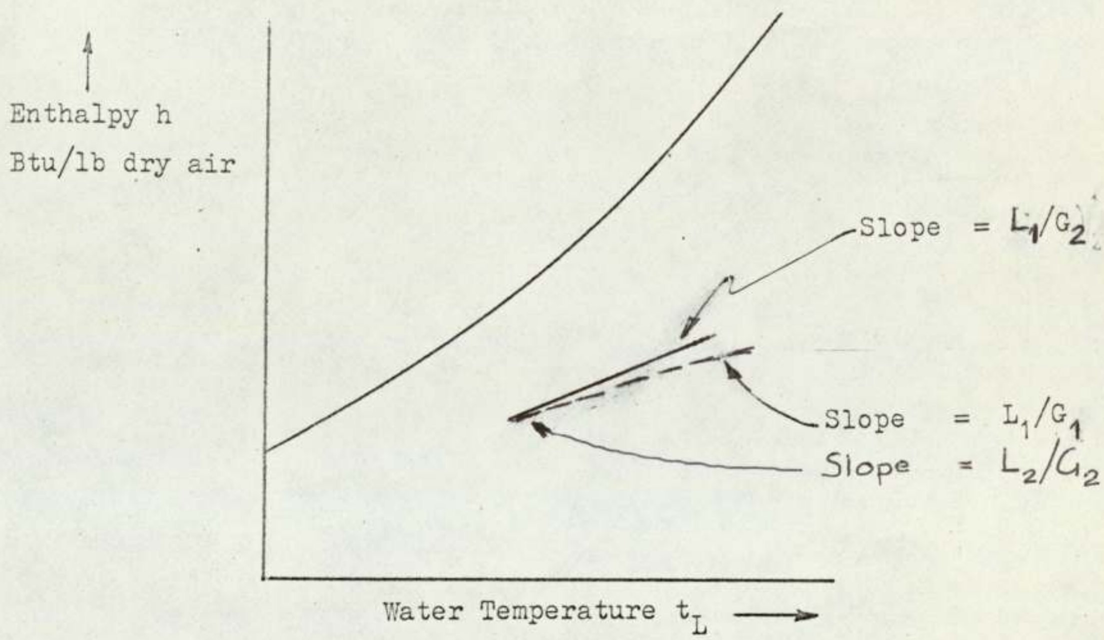


DIAGRAM SHOWING THE EFFECT OF VARIATIONS IN THE TRUE L/G VALUE

Fig 5.4

particularly for those values of L_1/G_2 less than unity.
These comments are summarised on Fig. 5.4.

The use of the L_1/G_2 value instead of the true value, see equation (5.41) tends to reduce the driving force and hence give a slightly pessimistic result which leads to limited oversizing of cooling towers. This effect is generally small, and within the context of cooling tower sizing selection it can be ignored.

See P. 109

f) To show that the Driving Force is a Vertical Difference on Fig. 5.5

The overall energy transfer equation for the liquid phase is :

$$L dt = k_L \, (a dV) \, (t_L - t_i) \tag{5.1}$$

and for the gas phase :

$$L dt = k_m \, (a dV) \, (h_i - h_G) \tag{5.14}$$

Combining these equations gives :

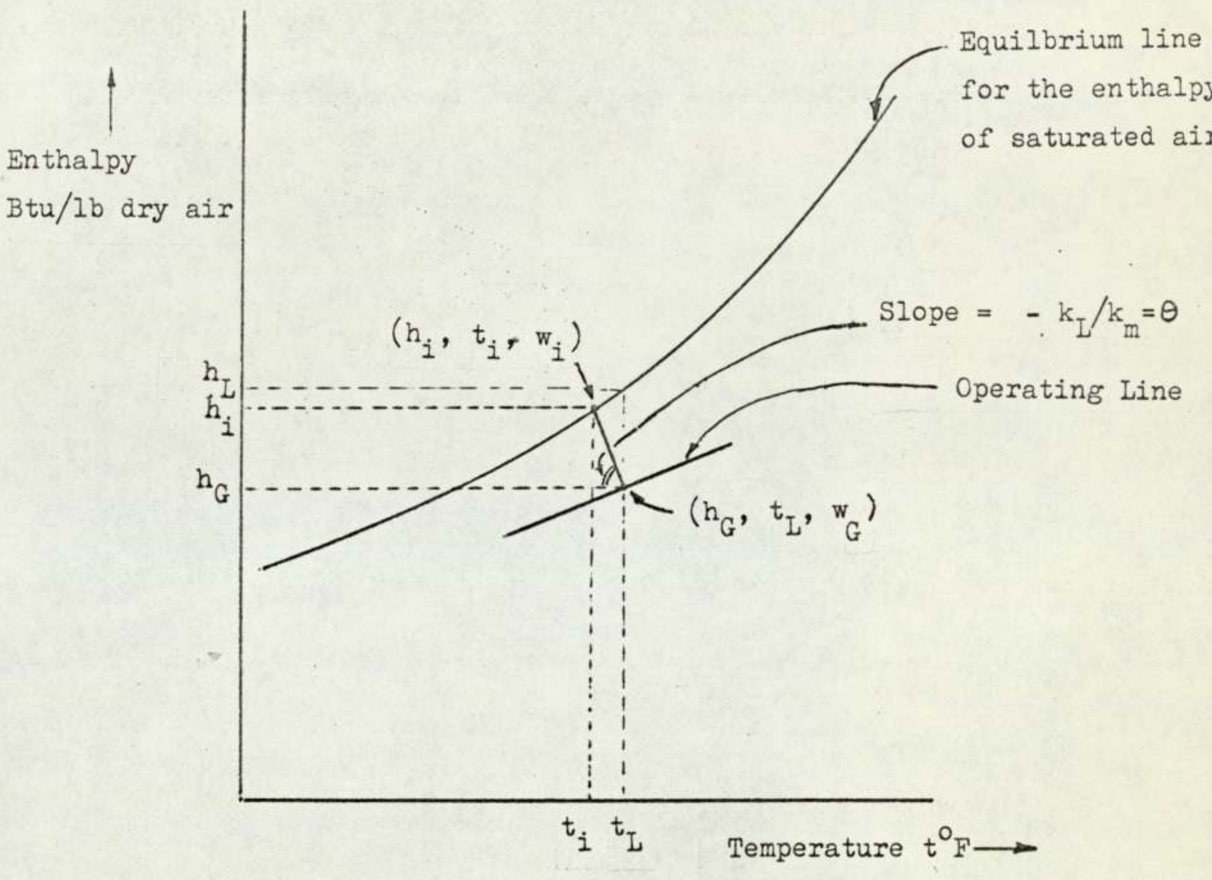
$$k_L/k_m = (h_i - h_G) \,/\, (t_L - t_i) = - (h_i - h_G) \,/\, (t_i - t_L)$$

$$\text{i.e. } - k_L/k_m = (h_i - h_G) \,/\, (t_i - t_L) \tag{5.42}$$

Fig. 5.5 shows a curve of h_i as a function of t_i representing the enthalpy of a saturated air-vapour mixture. The equation for the operating line relates h_G and t_L. The point (h_i, t_i) lies on the equilibrium curve, and the point (h_G, t_L) represents a point on the operating line. A straight line linking (h_i, t_i) with (h_G, t_L) is represented by the equation (5.42) shown above. This line has the slope θ where :

$$\theta = - k_L \,/\, k_m \tag{5.43}$$

In the absence of information for the values of the coefficients k_L, k_m since it is known that the cooling process is controlled by the gas phase, and since, it is also generally accepted that the heat energy transfer rate in the liquid phase is much greater than the mass transfer rate in the gas phase, a



TEMPERATURE ENTHALPY DIAGRAM FOR
AN AIR WATER PROCESS

Fig 5.5

first approximation is for the ratio to be infinite such that :

$$k_L / k_m = \infty = \tan \theta$$

Hence :

$$\theta = 90^{\circ}$$

and t_i is equal to t_L, that is the temperature drop through the liquid phase is assumed negligible; hence $h_i = h_L$. In such a case, a point on the operating line has a corresponding point on the equilibrium curve directly above it. This is shown on Fig. 5.5 which implies that the driving force is represented by the vertical distance $(h_L - h_G)$.

e) (Continued):

The extent of variation of the L/G ratio throughout the tower has been developed above. It must be emphasised, however, that such considerations would be necessary in the analysis only if point or differential mass transfer data were to be used. In fact the data as normally presented is obtained from integral results which will have involved the relevant L/G variation in the determining tests.

CHAPTER SIX

EVALUATION OF THE NDU INTEGRAL FOR THE COUNTERFLOW PROCESS

6.1 Introduction

Figure 2.2 illustrates the principles and shows the relationships involved in the evaporative cooling process. It also shows the air water enthalpy temperature equilibrium line KL, the cooling tower inlet and outlet air and water conditions, and the operating line EF, with slope L/G.

The operating line EF can be assumed to be described completely by a straight line originating at the point (h_{G2}, t_{L2}) with slope L/G and terminated by the inlet water temperature t_{L1}. The temperature t_{WB} fixes h_{G2} since the enthalpy of moist air is a function of the wet bulb temperature.

The conditions at the top and bottom of the tower are completely represented by the points (h_{G1}, t_{L1}) and (h_{G2}, t_{L2}) the end conditions of the operating line. The driving force at the top of the tower is shown as the difference between h_{L1} and h_{G1}, and at any other level between the top and the bottom of the tower as the difference between h_L and h_G.

The integral $\int_{t_{L2}}^{t_{L1}} dt/\ (h_L - h_G)$ would then be described by the area under the curve of $1\ /\ (h_L - h_G)$ versus temperature, between the limits t_{L1} and t_{L2}. This area ABCD is identified with cross hatching on Fig. 2.2. The area may be evaluated by any one of the methods described earlier.

All of these methods when evaluated by hand techniques are laborious and time consuming. A quick and accurate method for solving the integral is desirable and essential for design and selection of evaporative cooling equipment.

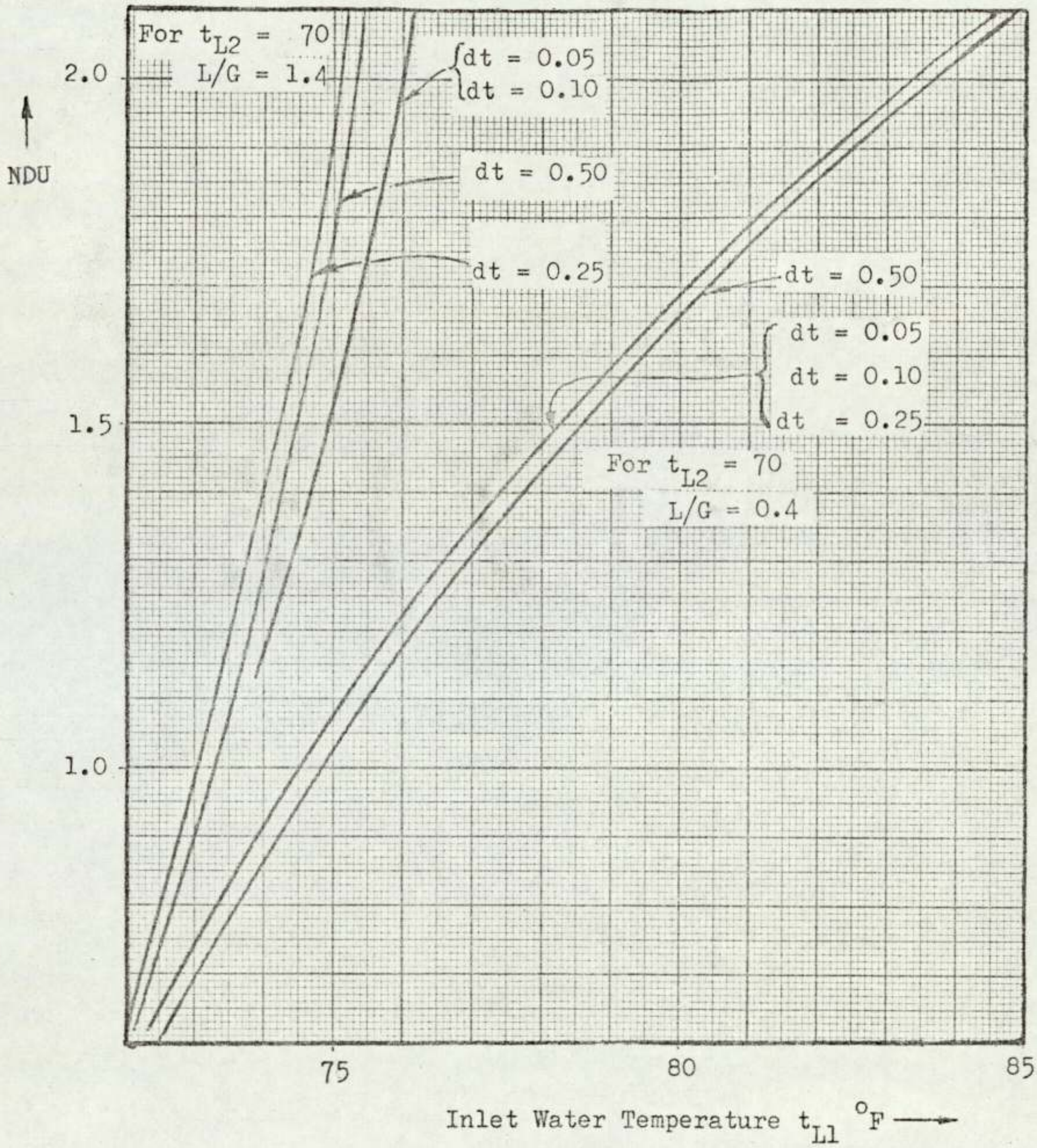
Since the integral cannot be solved directly, an approximate method is necessary. This will involve the use of iterative techniques which make it ideally suited for computer analysis.

The Trapezoidal Rule is used for the approximate evaluation of the integral. Although less accurate than quadrature formulae, this shortcoming can be reduced by using very small temperature increments. The small temperature increment also lends itself to the type of calculation envisaged, where advancing by small intervals the temperature increased until it satisfied a nominated NDU value. Mikyska and Reinisch (13) and Fuller (1) use the Simpson Rule with the temperature range of the integral broken down into 30 and 10 intervals respectively, to give the required accuracy.

In applying the Trapezoidal Rule to this work it is necessary to consider the smallest commercially practical cooling range, namely 5 deg F and then perform an exercise to decide on the number of intervals necessary to give the desired accuracy. By this means the number of intervals and hence their size can be selected to give the minimum tolerable deviation in the calculated temperature value for the 5 deg F range.

The accuracy required of the integral is dependent upon reading from the performance charts an acceptable inlet temperature to the tower. If the temperature can be read from the chart to $\pm$ 0.5 deg F then satisfactory accuracy can be achieved for normal commercial usage.

To achieve an accuracy of this order it is necessary to plot Fig. 6.1 from the performance results discussed in Chapter Eight, using data based on temperature increments of 0.05, 0.1, 0.25 and 0.5. The Figure shows that increments of 0.05, 0.1, 0.25 and 0.5 give the desired accuracy, and hence to keep the iterations to a minimum, an increment of 0.5 is chosen for use in the following work.



All temperatures in
degrees Fahrenheit

CHART SHOWING THE INFLUENCE OF THE INTEGRATING INTERVAL

Fig 6.1

6.2 Method

Knowing the conditions which define the starting point of the operating line namely h_{G2}, t_{L2}, and a nominated slope L/G, and using a known incremental value dt = 0.5 the area EFGH on Fig. 6.2 is fully defined.

Considering the small area ECMH, the enthalpies h_{L2}, h_L and h_{G2} are computed from the enthalpy temperature equation, whilst the enthalpy h_G is calculated from the definition of the operating line given by equation (5.26) as :

$$h_G = h_{G2} + dt\ L/G \tag{6.1}$$

The driving forces are calculated and their reciprocals found to give the values $1/(h_{L2} - h_{G2})$ and $1/(h_L - h_G)$. Using the Trapezoidal Rule, the first incremental area ANPD is calculated from :

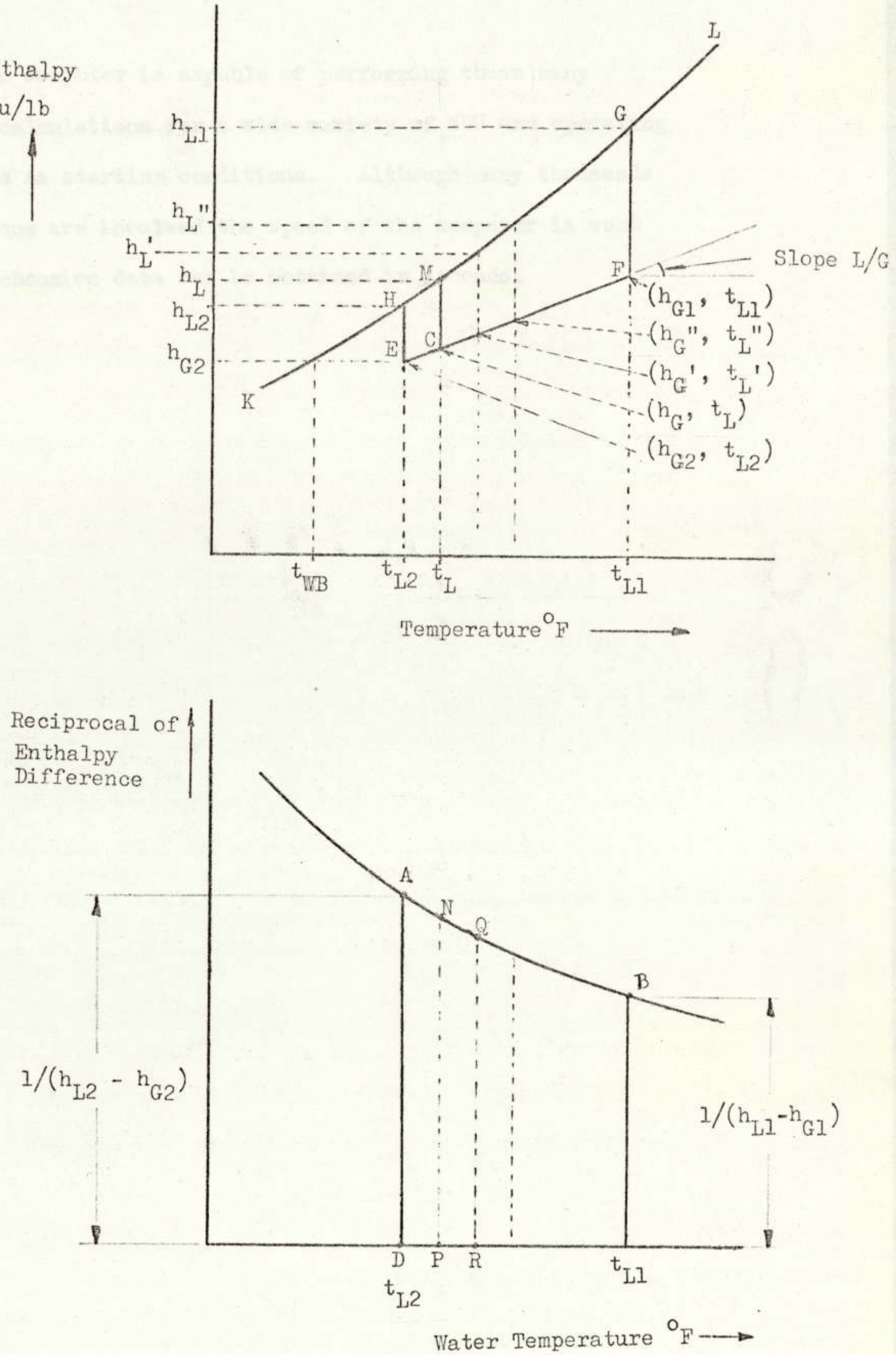
$$d\ (NDU) = dt\ (1/(h_{L2} - h_{G2}) + 1/(h_L - h_G))/2 \tag{6.2}$$

The integral value for the adjacent area is found by calculating the new enthalpy values h_L' and h_G' from the enthalpy temperature relationship and the equation for the operating line :

$$h_G' = h_G + (L/G)dt = h_{G2} + 2\ (L/G)\ dt \tag{6.3}$$

From the driving force $(h_L' - h_G')$, the reciprocal value $1/(h_L' - h_G')$ can be found, and knowing the earlier $1/(h_L - h_G)$ reciprocal the integral value equivalent to the area NQRP can be computed. The second integral value is added to the first.

This technique is continued along the operating line until the summation of all incremental integral values becomes equal to a nominated NDU value. Summation of the corresponding temperature increments establishes the final inlet water temperature t_{L1} equivalent to the summed NDU value. The nominated NDU value is used to control the integration advance; it is also an important feature of the performance charts presented in Chapter Eight.



COUNTERFLOW PROCESS DIAGRAM

Fig 6.2

The computer is capable of performing these many iterative calculations for a wide variety of NDU and operating line values as starting conditions. Although many thousands of iterations are involved the speed of the computer is such that comprehensive data can be obtained in seconds.

CHAPTER SEVEN

COMPUTER PROGRAM FOR COUNTERFLOW OPERATION

7.1 Description

The iterative method for solving the integral is laborious by hand calculation, whilst it is of ideal form for computer solution. Knowing the problem to be solved as outlined in Chapter Six, the task of writing a computer program can be facilitated by the preparation of a Flow Diagram, which summarises systematically in block and line form the sequence of computer operations.

An example of such a Flow Diagram is shown on Fig.7.1 The steps are represented by boxes with pseudo Autocode descriptions written into them to describe each step.

A further elaboration of the program is as follows. The sequence of instructions is :

Start

Read the wet bulb temperature T (0)

Print the program title and headings

Enter subroutine 8 to find the enthalpy H(0) corresponding to the wet bulb temperature T(0).

Calculate the starting temperature at outlet T(1) = T(0) + 5

Enter cycle C to find the first T(1) value C=T(1):5:90

Enter subroutine 8 to find the outlet water enthalpy H(1) corresponding to temperature T(1)

Enter cycle D to find the first L/G value D=0.5:0.5:2

Set N(2), the summation of NDU, to zero N(2)=0

Find the inlet water temperature T(2)=T(1)+dt

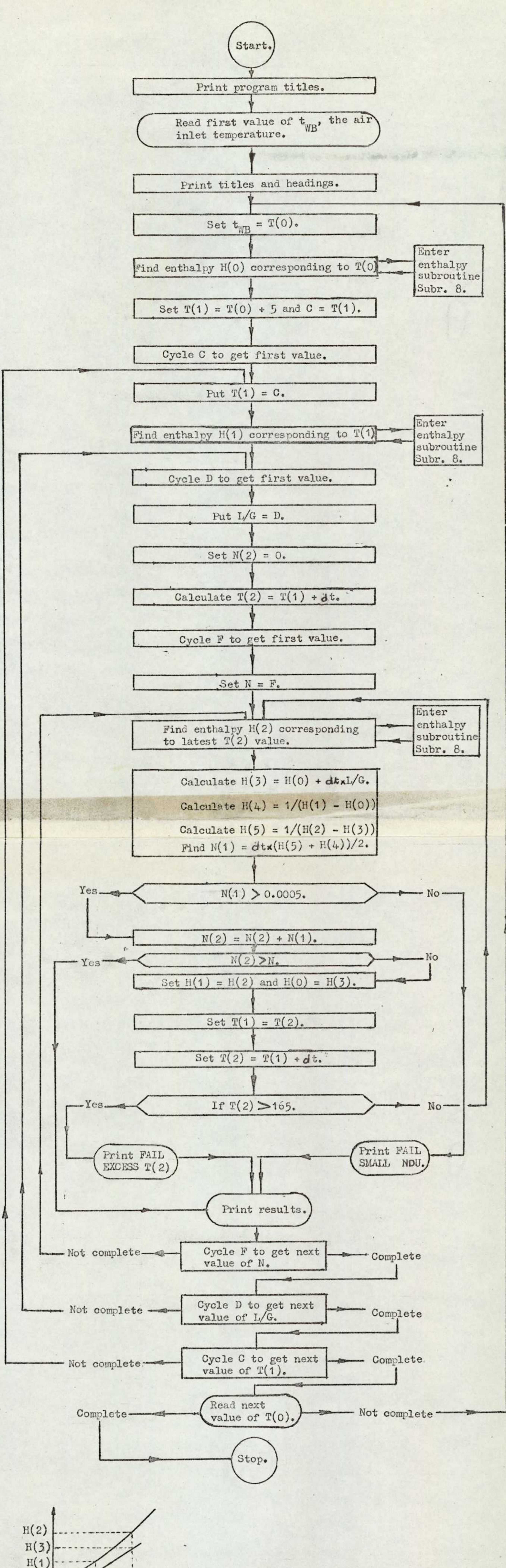
where dt the incremental temperature has the value 0.5.

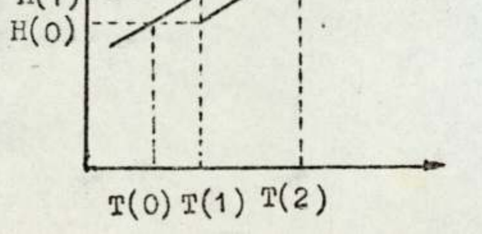
Enter cycle F to find the first value of N F=0.5:0.5:2

Enter subroutine 8 to find the inlet water enthalpy H(2) corresponding to the inlet water temperature T(2)

The air outlet enthalpy is calculated from H(3)=H(0)+dt L/G

Calculate the driving forces (H(2)-H(3))and(H(1)-H(0))





FLOW DIAGRAM FOR THE COUNTERFLOW COOLING TOWER - AUTOCODE PROGRAM.

Fig. 7.1

Calculate the reciprocals of the driving forces

Calculate the incremental NDU value $N(1)$

Conditional statement jump if $N1 < 0.0005$

to print the title "FAIL N1 = " as well as the corresponding results, and then continue the F cycle instruction to select the next value of N. Otherwise add this NDU value to the summation of all previous NDU values $N(2)=N(2)+N(1)$

Conditional statement, Jump if $N(2) > N$ to print the corresponding results and then continue F cycle instruction to get next value of N.

Otherwise update the enthalpy and temperature conditions so that :

$$H(1) = H(2)$$
$$H(0) = H(3)$$
$$T(1) = T(2)$$

Add a further increment of temperature to the inlet water temperature which becomes : $T(2)=T(1) + dt.$

Check to see if this value is less than 165, before finding the enthalpy $H(2)$ corresponding to $T(2)$. If greater than 165 go to next cycle value.

Calculate the reciprocals of driving force, the incremental NDU value and hence the cummulative NDU. Repeat this cycling process until a value of $T(2)$ satisfies the selected N value.

The results are printed and then a new value of N is found from the cycle F, and the sequence repeated.

When the N values are exhausted, a new value of L/G is found from cycle D. This sequence continues until the L/G cycle is completed whereupon the final cycle C is entered to find a new start condition, the outlet water temperature $T(1)$.

When the $T(1)$ values have been exhausted, the machine returns to read a new value of the input data, namely wet bulb temperature T(0), otherwise it will obey the stop instruction.

7.2 Computer Language

More recently the Elliott Machine has been phased out and replaced by an International Computer's (I.C.L.) Machine. This has necessitated a change in language since the Autocode facility was no longer available. In consequence the original Autocode cooling tower performance program has been re-written in the Fortran language.

The I.C.L. Machine computes the equivalent Fortran performance program in about one percent of the time required by the Elliott computer for the Autocode program.

7.3 Autocode Program

This is a program written in the Autocode language to solve the integral using the Trapezoidal Rule, iterative techniques and the enthalpy temperature relationship. Analysis of run time and accuracy, as discussed in Chapter Four, showed that the enthalpy temperature relationship in the form of the forward backward difference equation (based on P = 1000 millibar) should be employed with the Autocode cooling tower performance program.

The program is prepared as a five hole punched tape with a printout which can be broken down into its component parts as follows :

a) Introduction indicating the facilities which the machine will employ during the program analysis.

b) Subroutine 8 which embraces all the equations, constants and conditions necessary to solve the forward backward difference enthalpy temperature equations.

c) Titles, for identifying the columns of printout.

d) Cycle instructions to obtain successive values for a particular variable.

e) Arithmetic operations to find the inlet water temperature for a nominated NDU value and known atmospheric conditions.

f) Printout of the results.

Examples of the computer program and results are shown on Figs 7.2 and 7.3.

7.4 Fortran Program.

This program is the Fortran equivalent of the previous Autocode cooling tower performance program. The program is produced on IBM punched cards.

The printout of the Fortran program is shown on Fig 7.4. Down the right hand side of the printout a list of numbers is recorded. The list shows the identity of the line, for each line of instruction has a separate card and number. A breakdown of the program is similar to that describing the Autocode program.

To satisfy the present needs of world industry, performance charts at various heights above sea level would be a significant contribution to the correct cooling tower selection. These can be provided by introducing the parameter atmospheric pressure into the enthalpy temperature relationship contained in the performance program described earlier. This requirement involves replacing the forward backward difference equations (3.5 and 3.6) with a formula containing a term in atmospheric pressure, such as the fundamental enthalpy equation (3.15).

From experiments undertaken with the original Autocode program the use of the fundamental enthalpy equation requires additional calculation time to obtain an enthalpy value. This fact is not significant, when using this equation as part of the Fortran program which is run on the fast I.C.L. Machine. Thus a further program is made differing only from the previous

INTRODUCTION

```
SETS I
SETV A(6)B(6)CDE(1)FH(5)L(1)M(4)N(2)P(1)
Q(1)R(1)S(6)T(2)U(1)V(1)W(6)X(3)Y(5)Z(3)
SETR 28
```

ENTHALPY TEMPERATURE SUBROUTINE 8

```
8)JUMP IF P≥150@9
JUMP IF P≥130@7
JUMP IF P≥117@4
JUMP IF P≥90@3
JUMP IF P≥80@2
5)P=P/5
P=P-10
CHECK P
S(0)=1
R=S(0)*W(0)
VARY I=1:1:6
V=STAND I
U=P-V
U=U+1
S(I)=S(I-1)*U
Q=S(I)*W(I)
R=R+Q
REPEAT I
E=R
CHECK E
JUMP @10
2)P=P/5
P=P-16
CHECK P
S(0)=1
R=S(0)*X(0)
CHECK R
VARY I=1:1:3
V=STAND I
U=P+V
U=U-1
CHECK U
S(I)=S(I-1)*U
Q=S(I)*X(I)
R=R+Q
REPEAT I
E=R
JUMP @10
3)P=P/5
P=P-18
CHECK P
R=S(0)*Y(0)
VARY I=1:1:5
V=STAND I
U=P-V
U=U+1
S(I)=S(I-1)*U
Q=S(I)*Y(I)
R=R+Q
CHECK R
REPEAT I
E=R
JUMP @10
4)P=P/5
P=P-26
CHECK P
S(0)=1
R=S(0)*Z(0)
VARY I=1:1:3
V=STAND I
U=P+V
U=U-1
S(I)=S(I-1)*U
Q=S(I)*Z(I)
R=R+Q
CHECK R
REPEAT I
E=R
JUMP @10
7)P=P/5
P=P-26
CHECK P
S(0)=1
R=S(0)*A(0)
VARY I=1:1:6
V=STAND I
U=P-V
U=U+1
CHECK U
S(I)=S(I-1)*U
Q=S(I)*A(I)
R=R+Q
REPEAT I
E=R
CHECK E
JUMP @10
9)P=P/5
P=P-34
CHECK P
S(0)=1
R=S(0)*B(0)
CHECK R
VARY I=1:1:6
V=STAND I
U=P+V
U=U-1
CHECK U
S(I)=S(I-1)*U
Q=S(I)*B(I)
R=R+Q
REPEAT I
E=R
CHECK E
10)EXIT
1)S(0)=1.0
W(0)=12.72
W(1)=2.94
W(2)=0.16
W(3)=0.00834
W(4)=0.000417
W(5)=0.000025
W(6)=0.0000264
X(0)=48.6
X(1)=6.5
X(2)=0.35
X(3)=0.0001667
Y(0)=48.6
Y(1)=7.5
Y(2)=0.45
Y(3)=0.0834
Y(4)=-0.0208
Y(5)=0.005
Z(0)=149.8
Z(1)=19.8
Z(2)=1.3
Z(3)=0.0005667
A(0)=149.8
A(1)=23
A(2)=0.8
A(3)=0.564
A(4)=0.0707
A(5)=-0.121
A(6)=0.0682
B(0)=522.4
B(1)=62.4
B(2)=5.3
B(3)=-0.333
B(4)=-0.0958
B(5)=0.1758
B(6)=0.085
```

COUNTERFLOW COMPUTER PROGRAM - AUTOCODE

Fig 7.2

Continued

TITLE INSTRUCTIONS

```
READ T(0)
LINES 3
SPACES 12
TITLE TAPE 127(22)     INTEGRATING INTERVAL=0.5DEG F
LINES 2
SPACES 2
TITLE T.WB
SPACES 2
PRINT T(0),3:2
LINES 2
SPACES 19
TITLE T.OFF
SPACES 4
TITLE T.ON
SPACES 6
TITLE APP
SPACES 6
TITLE RANGE
SPACES 5
TITLE PR
SPACES 10
TITLE A/R
SPACES 3
TITLE A/PR
SPACES 3
TITLE L/G
SPACES 5
TITLE N-DEL
SPACES 2
TITLE N-IT
LINE
JUMP @20
23)LINE
TITLE FAIL  N1=
PRINT N1,2:3
EXIT
20)P=T(0)
SUBR 8
H(0)=E
```

CYCLE INSTRUCTIONS

```
T(1)=T(0)÷5
CYCLE C=T(1):5:90
T(1)=C
CHECK T(1)
P=T(1)
SUBR 8
H(1)=E
19)CYCLE D=0.2:0.2:1.2
```

ARITHMETIC INSTRUCTIONS

```
L=D
CHECK L
L(1)=0.5*L
N(2)=0
CYCLE F=0.5:0.5:2
N=F
CHECK N
T(2)=T(1)+0.5
12)P=T(2)
SUBR 8
H(2)=E
H(3)=H(0)+L(1)
H(4)=H(1)-H(0)
H(4)=1/H(4)
H(5)=H(2)-H(3)
H(5)=1/H(5)
N(1)=H(4)+H(5)
N(1)=0.25*N(1)
CHECK N(1)
JUMP IF N(1)$0.0005@21
N(2)=N(2)+N(1)
CHECK N(2)
JUMP IF N(2)%N@13
H(0)=H(3)
H(1)=H(2)
T(2)=T(2)÷0.5
CHECK T(2)
JUMP IF T(2)%165@24
JUMP @12
21)SUBR 23
JUMP @25
13)M(0)=T(1)-T(0)
CHECK M(0)
M(1)=T(2)-T(1)
CHECK M(1)
JUMP IF M(1)&5@22
M(2)=T(2)-T(0)
CHECK M(2)
M(3)=M(0)/M(1)
CHECK M(3)
M(4)=M(0)/M(2)
CHECK M(4)
LINE
SPACES 17
JUMP @25
```

COUNTERFLOW COMPUTER PROGRAM - AUTOCODE

Fig 7.2

Continued

PRINTOUT INSTRUCTIONS

```
24)LINE
TITLE FAIL EXCESS T.ON
25)PRINT T(1),3:1
SPACES 1
PRINT T(2),3:2
SPACES 1
PRINT M(0),2:2
SPACES 1
PRINT M(1),2:2
SPACES 1
PRINT M(2),2:2
SPACES 1
PRINT M(3),3:3
SPACES 1
PRINT M(4),3:3
SPACES 1
PRINT L,1:2
SPACES 1
PRINT N,1:2
SPACES 2
PRINT N(2),2:3
REPEAT F
22)REPEAT D
REPEAT C
STOP
START 1
```

COUNTERFLOW COMPUTER PROGRAM - AUTOCODE

Fig 7.2

TAPE 127(22)   INTEGRATING INTERVAL=0.5DEG F

T.WB   65.00   P = 1000

	T.OFF	T.ON	APP	RANGE	PR	A/R	A/PR	L/G	N-DEL	N-IT
	70.0	72.50	5.00	2.50	7.50	2.000	0.667	0.20	0.50	0.515
	70.0	75.50	5.00	5.50	10.50	0.909	0.476	0.20	1.00	1.045
	70.0	79.50	5.00	9.50	14.50	0.526	0.345	0.20	1.50	1.528
	70.0	85.50	5.00	15.50	20.50	0.323	0.244	0.20	2.00	2.004
	70.0	72.50	5.00	2.50	7.50	2.000	0.667	0.40	0.50	0.540
	70.0	75.00	5.00	5.00	10.00	1.000	0.500	0.40	1.00	1.060
	70.0	78.00	5.00	8.00	13.00	0.625	0.385	0.40	1.50	1.539
	70.0	82.00	5.00	12.00	17.00	0.417	0.294	0.40	2.00	2.011
	70.0	72.50	5.00	2.50	7.50	2.000	0.667	0.60	0.50	0.569
	70.0	74.50	5.00	4.50	9.50	1.111	0.526	0.60	1.00	1.068
	70.0	76.50	5.00	6.50	11.50	0.769	0.435	0.60	1.50	1.506
	70.0	79.50	5.00	9.50	14.50	0.526	0.345	0.60	2.00	2.027
	70.0	72.50	5.00	2.50	7.50	2.000	0.667	0.80	0.50	0.602
	70.0	74.00	5.00	4.00	9.00	1.250	0.556	0.80	1.00	1.063
	70.0	75.50	5.00	5.50	10.50	0.909	0.476	0.80	1.50	1.503
	70.0	77.50	5.00	7.50	12.50	0.667	0.400	0.80	2.00	2.018
	70.0	72.00	5.00	2.00	7.00	2.500	0.714	1.00	0.50	0.509
	70.0	73.50	5.00	3.50	8.50	1.429	0.588	1.00	1.00	1.038
	70.0	75.00	5.00	5.00	10.00	1.000	0.500	1.00	1.50	1.583
	70.0	76.50	5.00	6.50	11.50	0.769	0.435	1.00	2.00	2.136
	70.0	72.00	5.00	2.00	7.00	2.500	0.714	1.20	0.50	0.537
	70.0	73.50	5.00	3.50	8.50	1.429	0.588	1.20	1.00	1.149
	70.0	74.50	5.00	4.50	9.50	1.111	0.526	1.20	1.50	1.661
	70.0	75.00	5.00	5.00	10.00	1.000	0.500	1.20	2.00	2.026
	75.0	80.50	10.00	5.50	15.50	1.818	0.645	0.20	0.50	0.514
FAIL	75.0	88.00	10.00	13.00	23.00	0.769	0.435	0.20	1.00	1.005
	75.0	123.00	10.00	48.00	58.00	0.208	0.172	0.20	1.50	1.502
EXCESS T.ON	75.0	155.50	10.00	48.00 ?	58.00 ?	0.208	0.172	0.20	2.00	1.722

PRINTOUT OF TYPICAL COUNTERFLOW RESULTS OBTAINED FROM THE AUTOCODE COMPUTER PROGRAM

Fig 7.3

Fig 7.3

INTRODUCTION

```
      LIST (LP)
      MAP                                                         COU  100
      PROGRAM(COUNTERFCT)                                         COU  200
      INPUT 1=CR0                                                 COU  300
      OUTPUT 2=LP0                                                COU  400
      TRACE 2                                                     COU  500
      END                                                         COU  600
```

TITLES, HEADINGS AND ARITHMETIC INSTRUCTIONS

```
      MASTER COUNTERFCT                                           COU  700
      COMMON E,ATM                                                COU  750
      ATM=1000                                                    COU  775
      ITWB=65                                                     COU  802
      SUMINT=0.05                                                 COU  902
   11 FORMAT(24H PROG 127(24) INTEG INT 1F4.2,5H DEG F)           COU 1000
      WRITE(2,11)SUMINT                                           COU 1100
      WRITE(2,12)ITWB,ATM                                         COU 1200
   12 FORMAT(6H T.WB=,I2,5H ATM=,F7.1)                            COU 1300
   14 FORMAT(116H                         T.OFF   T.ON   APP   RANGE COU 1600
    1   PR        A/R        A/PR   L/G   N-DEL   N-1T   NDU)     COU 1700
      WRITE(2,14)                                                 COU 1800
      Q=ITWB                                                      COU 1850
      CALLTTOH(Q)                                                 COU 1900
      HSIX=E                                                      COU 2000
      ITONE=70                                                    COU 2101
   27 DO 53 K=ITONE,ITONE+20,5                                    COU 2202
      TONEE=K                                                     COU 2300
   25 CALLTTOH(TONEE)                                             COU 2400
      HSEVEN=E                                                    COU 2500
```

COUNTERFLOW COMPUTER PROGRAM - FORTRAN

Continued Fig 7.4

Fig 7.4

CYCLE AND ARITHMETIC INSTRUCTIONS

```
      DO 52  I=2,30,2                                                   COU 2601
      FZERO=I/10.0                                                      COU 2700
      GTWO=0                                                            COU 2800
      ZLONE=SUMINT*FZERO                                                COU 2900
      HZERO=HSIX                                                        COU 3000
      HONE=HSEVEN                                                       COU 3100
      TTWO=TONEE+SUMINT                                                 COU 3200
      DO 51  J=5,30,5                                                   COU 3301
      ZERO=J/10.0                                                       COU 3400
    9 CALLTTOH(TTWO)                                                    COU 3500
      HTWO=E                                                            COU 3600
      GONE=0.5*SUMINT*(1/(HTWO-HZERO-ZLONE)+1/(HONE-HZERO))             COU 3700
      GTWO=GTWO+GONE                                                    COU 3900
      IF(ZERO.LT.GTWO)GOTO41                                            COU 4000
      HZERO=HZERO+ZLONE                                                 COU 4100
      HONE=HTWO                                                         COU 4200
      TTWO=TTWO+SUMINT                                                  COU 4300
      IF(TTWO.GT.165)GOTO44                                             COU 4400
      GOTO9                                                             COU 4500
   41 QZERO=TONEE-Q                                                     COU 4800
      QONE=TTWO-TONEE                                                   COU 4900
      IF(QONE.LT.2)GOTO43                                               COU 5000
      QTWO=TTWO-Q                                                       COU 5100
      QTHREE=QZERO/QONE                                                 COU 5200
      QFOUR=QZERO/QTWO                                                  COU 5300
      GTWLG=GTWO*FZERO                                                  COU 5350
      PN=ZERO*FZERO                                                     COU 5375
      GOTO50                                                            COU 5400
   43 TTWO=TTWO+0.5                                                     COU 5500
      GOTO 51                                                           COU 5600
   42 FORMAT(14H FAIL XS  T.ON)                                         COU 5700
   44 WRITE(2,42)                                                       COU 5800
      GOTO60                                                            COU 5850
   20 FORMAT(1H,22X,1I2,2X,F7.2,2X,F6.2,4X,F6.2,4X,F6.2,4X,F8.3,2X,F8.3,COU 5900
     12X,F5.2,3X,F5.2,3X,F7.3,2X,F7.3)                                  COU 5950
```

COUNTERFLOW COMPUTER PROGRAM - FORTRAN

Continued Fig 7.4

Fig 7.4

Sheet 3 of 3

ARITHMETIC AND CYCLE INSTRUCTIONS (CONTINUED)

```
50 WRITE(2,20)K,TTWO,QZERO,QONE,QTWO,QTHREE,QFOUR,FZERO,ZERO,GTWLG,PNCOU 6000
   HZERO=HZERO+ZLONE                                                 COU 6002
   HONE=HTWO                                                         COU 6004
   TTWO=TTWO+SUMINT                                                  COU 6006
   IF(TTWO.GT.165)GOTO44                                             COU 6008
   GOTO51                                                            COU 6009
60 WRITE(2,20)K,TTWO,QZERO,QONE,QTWO,QTHREE,QFOUR,FZERO,ZERO,GTWLG,PNCOU 6010
51 CONTINUE                                                          COU 6015
52 CONTINUE                                                          COU 6020
53 CONTINUE                                                          COU 6030
   STOP                                                              COU 6050
   END                                                               COU 6051
```

ENTHALPY TEMPERATURE SUBROUTINE

```
SUBROUTINE TTOH(T)                                                   COU 6625
COMMON E,ATM                                                         COU 6650
P=EXP(1.8091+9.594*(T-32)/(237.3+0.5555*(T-32)))                     COU 6751
G3=4354*P/(ATM-P)                                                    COU 6752
G1=0.45*G3/7000                                                      COU 6753
G2=1075*G3/7000                                                      COU 6754
E=(0.241+G1)*(T-32)+G2                                               COU 6755
RETURN                                                               COU 9600
STOP                                                                 COU 9800
END                                                                  COU 9900
```

COUNTERFLOW COMPUTER PROGRAM - FORTRAN

Fig 7.4

program in its utilisation of a subroutine for the fundamental enthalpy temperature relationship. A copy of this subroutine is shown on sheet 3 of Fig 7.4. Fig 7.5 illustrates a typical Fortran printout.

7.5 Program Operating Times.

With the Autocode cooling tower performance program the choice of enthalpy temperature relationship had a significant effect upon the computing time required for its evaluation. With the Fortran program, the choice of equation has no significance. Thus, to contain the time necessary to analyse the Autocode program, the forward backward difference equation was used.

The final Fortran program utilises the fundamental enthalpy equation. This latter equation gives greater scope since results can be obtained for more than one atmospheric pressure and hence altitude. Furthermore, its use had little effect upon the computing time.

Experience with the Fortran program shows that 400 data points can be calculated in 1 to 2 minutes whereas the equivalent Autodode program took about 100 to 200 minutes, dependent upon the enthalpy equation employed.

Fig 7.6 shows a Fortran computer program for the forward backward enthalpy temperature relationship.

PROG 127(24) INTEG INT 0.05 DEG
T.WB=65 ATM= 1000.0

T.OFF	T.ON	APP	RANGE	PR	A/R	A/PR	L/G	N-DEL	N-1T	NDU
70	72.45	5.00	2.45	7.45	2.041	0.671	0.20	0.50	0.101	0.100
70	75.85	5.00	5.85	10.85	0.855	0.461	0.20	1.00	0.200	0.200
70	81.00	5.00	11.00	16.00	0.455	0.313	0.20	1.50	0.301	0.300
70	89.10	5.00	19.10	24.10	0.262	0.207	0.20	2.00	0.400	0.400
70	104.05	5.00	34.05	39.05	0.147	0.128	0.20	2.50	0.500	0.500
70	148.90	5.00	78.90	83.90	0.063	0.060	0.20	3.00	0.600	0.600
70	72.30	5.00	2.30	7.30	2.174	0.685	0.40	0.50	0.201	0.200
70	75.25	5.00	5.25	10.25	0.952	0.488	0.40	1.00	0.401	0.400
70	79.20	5.00	9.20	14.20	0.543	0.352	0.40	1.50	0.602	0.600
70	84.70	5.00	14.70	19.70	0.340	0.254	0.40	2.00	0.801	0.800
70	93.05	5.00	23.05	28.05	0.217	0.178	0.40	2.50	1.001	1.000
70	108.00	5.00	38.00	43.00	0.132	0.116	0.40	3.00	1.200	1.200
70	72.20	5.00	2.20	7.20	2.273	0.694	0.60	0.50	0.304	0.300
70	74.70	5.00	4.70	9.70	1.064	0.515	0.60	1.00	0.600	0.600
70	77.75	5.00	7.75	12.75	0.645	0.392	0.60	1.50	0.902	0.900
70	81.50	5.00	11.50	16.50	0.435	0.303	0.60	2.00	1.201	1.200
70	86.40	5.00	16.40	21.40	0.305	0.234	0.60	2.50	1.501	1.500
70	93.35	5.00	23.35	28.35	0.214	0.176	0.60	3.00	1.802	1.800
70	72.10	5.00	2.10	7.10	2.381	0.704	0.80	0.50	0.407	0.400
70	74.25	5.00	4.25	9.25	1.176	0.541	0.80	1.00	0.803	0.800
70	76.60	5.00	6.60	11.60	0.758	0.431	0.80	1.50	1.205	1.200
70	79.20	5.00	9.20	14.20	0.543	0.352	0.80	2.00	1.606	1.600
70	82.15	5.00	12.15	17.15	0.412	0.292	0.80	2.50	2.005	2.000
70	85.65	5.00	15.65	20.65	0.319	0.242	0.80	3.00	2.402	2.400
70	74.55	5.00	4.55	9.55	1.099	0.524	1.00	1.00	1.012	1.000
70	76.55	5.00	6.55	11.55	0.763	0.433	1.00	1.50	1.501	1.500
70	78.60	5.00	8.60	13.60	0.581	0.368	1.00	2.00	2.002	2.000
70	80.70	5.00	10.70	15.70	0.467	0.318	1.00	2.50	2.502	2.500
70	82.95	5.00	12.95	17.95	0.386	0.279	1.00	3.00	3.009	3.000

PRINTOUT OF TYPICAL COUNTERFLOW RESULTS
OBTAINED FROM THE FORTRAN COMPUTER PROGRAM

Fig 7.5

Fig 7.5

```
      SUBROUTINE TTOH(P)                                                COU 6600
      COMMON F                                                          COU 6650
      IF(150.LE.P)GOTO1                                                 COU 6700
      IF(130.LE.P)GOTO2                                                 COU 6800
      IF(117.LE.P)GOTO3                                                 COU 6900
      IF(90.LE.P)GOTO4                                                  COU 7000
      IF(80.LE.P)GOTO5                                                  COU 7100
      R=P/5-10                                                          COU 7200
      F=12.62+2.94*R+0.16*R*(R-1)+0.00834*R*(R-1)*(R-2)+0.00042*R*(R-1)* OU 7400
     1(R-2)*(R-3)+0.00025*R*(R-1)*(R-2)*(R-3)*(R-4)-0.00026*R*(R-1)*(R-2 OU 7500
     2)*(R-3)*(R-4)*(R-5)                                               COU 7600
      RETURN                                                            COU 7700
    1 R=P/5-34                                                          COU 7800
      F=522.4+82.4*R+5.3*R*(R+1)-0.33333*R*(R+1)*(R+2)+0.09583*R*(R+1)*( OU 7900
     1R+2)*(R+3)+0.17417*R*(R+1)*(R+2)*(R+3)*(R+4)+0.8513*R*(R+1)*(R+2)* OU 8000
     2(R+3)*(R+4)*(R+5)                                                 COU 8001
      RETURN                                                            COU 8100
    2 R=P/5-26                                                          COU 8200
      F=149.8+23.0*R+0.6*R*(R-1)+0.56666*R*(R-1)*(R-2)+0.07083*R*(R-1)*( OU 8300
     1R-2)*(R-3)-0.11917*R*(R-1)*(R-2)*(R-3)*(R-4)+0.06764*R*(R-1)*(R-2) OU 8400
     2*(R-3)*(R-4)*(R-5)                                                COU 8500
      RETURN                                                            COU 8600
    3 R=P/5-26                                                          COU 8700
      F=149.8+19.8*R+1.3*R*(R+1)-0.02916*R*(R+1)*(R+2)*(R+3)-0.00833*R*( OU 8800
     1R+1)*(R+2)*(R+3)*(R+4)-0.00138*R*(R+1)*(R+2)*(R+3)*(R+4)*(R+5)    COU 8801
      RETURN                                                            COU 8900
    4 R=P/5-18                                                          COU 9000
      F=48.6+7.5*R+0.45*R*(R-1)+0.08333*R*(R-1)*(R-2)-0.02083*R*(R-1)*(R OU 9100
     1-2)*(R-3)+0.005*R*(R-1)*(R-2)*(R-3)*(R-4)-0.00055*R*(R-1)*(R-2)*(R OU 9200
     2-3)*(R-4)*(R-5)                                                   COU 9201
      RETURN                                                            COU 9300
    5 R=P/5-18                                                          COU 9400
      F=48.6+6.5*R+0.35*R*(R+1)-0.07917*R*(R+1)*(R+2)*(R+3)-0.00333*R*(R OU 9500
     1+1)*(R+2)*(R+3)*(R+4)-0.00101*R*(R+1)*(R+2)*(R+3)*(R+4)*(R+5)     COU 9501
      RETURN                                                            COU 9600
      STOP                                                              COU 9800
      END                                                               COU 9900
```

FORTRAN COMPUTER PROGRAM FOR THE FORWARD

BACKWARD DIFFERENCE ENTHALPY TEMPERATURE RELATIONSHIP

Fig 7.6

Fig 7.6

CHAPTER EIGHT

PRESENTATION OF PERFORMANCE DATA FOR COUNTERFLOW OPERATION

8.1 Selection Technique

The presentation must have a wide range of application, be easy to apply and accurate in use. The many Lichtenstein (3) charts satisfy these requirements but do not compare with the compactness of Merkel's less accurate method using a single Cooling Diagram. A technique based upon Lichtenstein's method but requiring fewer charts, would be a significant improvement. It is also desirable that the chosen technique be applicable for demonstrating the performance of crossflow cooling towers. These requirements suggest the type of presentation to be achieved.

The main variables to be accommodated on a chart are those listed by Jackson (9) in Chapter Two.

Considering atmospheric pressure P as constant, the remaining variables six in all, are difficult to handle. Their number can be reduced by relating KaA/L and y to the single variable NDU, the Number of Diffusion Units as given by equation (5.22). This reduced the number of variables to five, which are t_{L1}, t_{L2}, t_{WB}, L/G and NDU. The variables KaA/L and y will be reidentified in a later analysis of the Selection Technique.

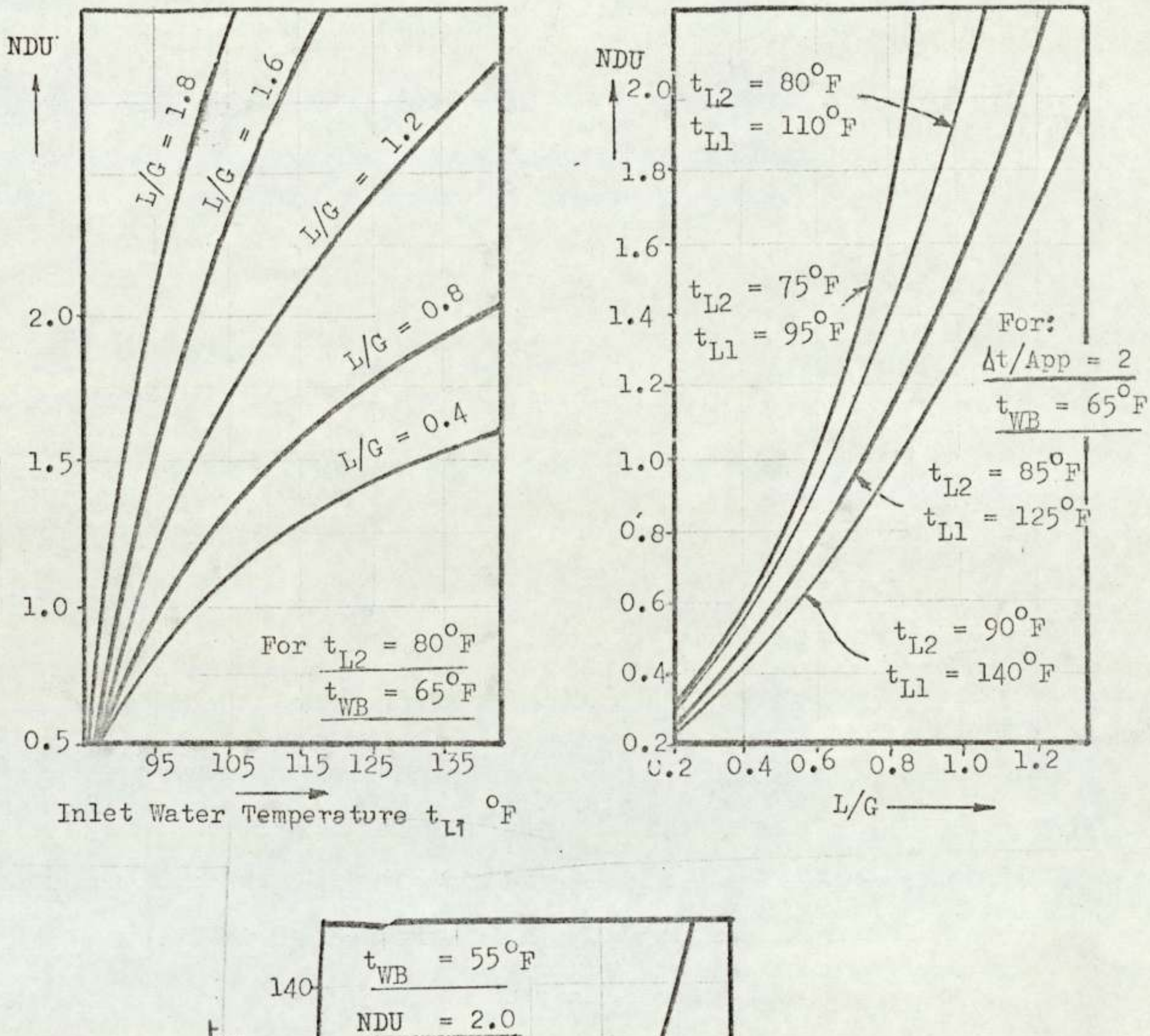
Preliminary plots of these variables were not successful because of the large number of curves required, their steepness and spacing. Fig 8.1 illustrates some of these comments in more detail. This led to the development of alternative parameters, in the form of non-dimensional groups, namely ratios of approach/range and approach/potential range where :

$$\text{Potential Range} \quad PR = t_{L1} - t_{WB}$$

$$\text{Range} \quad \Delta t = t_{L1} - t_{L2}$$

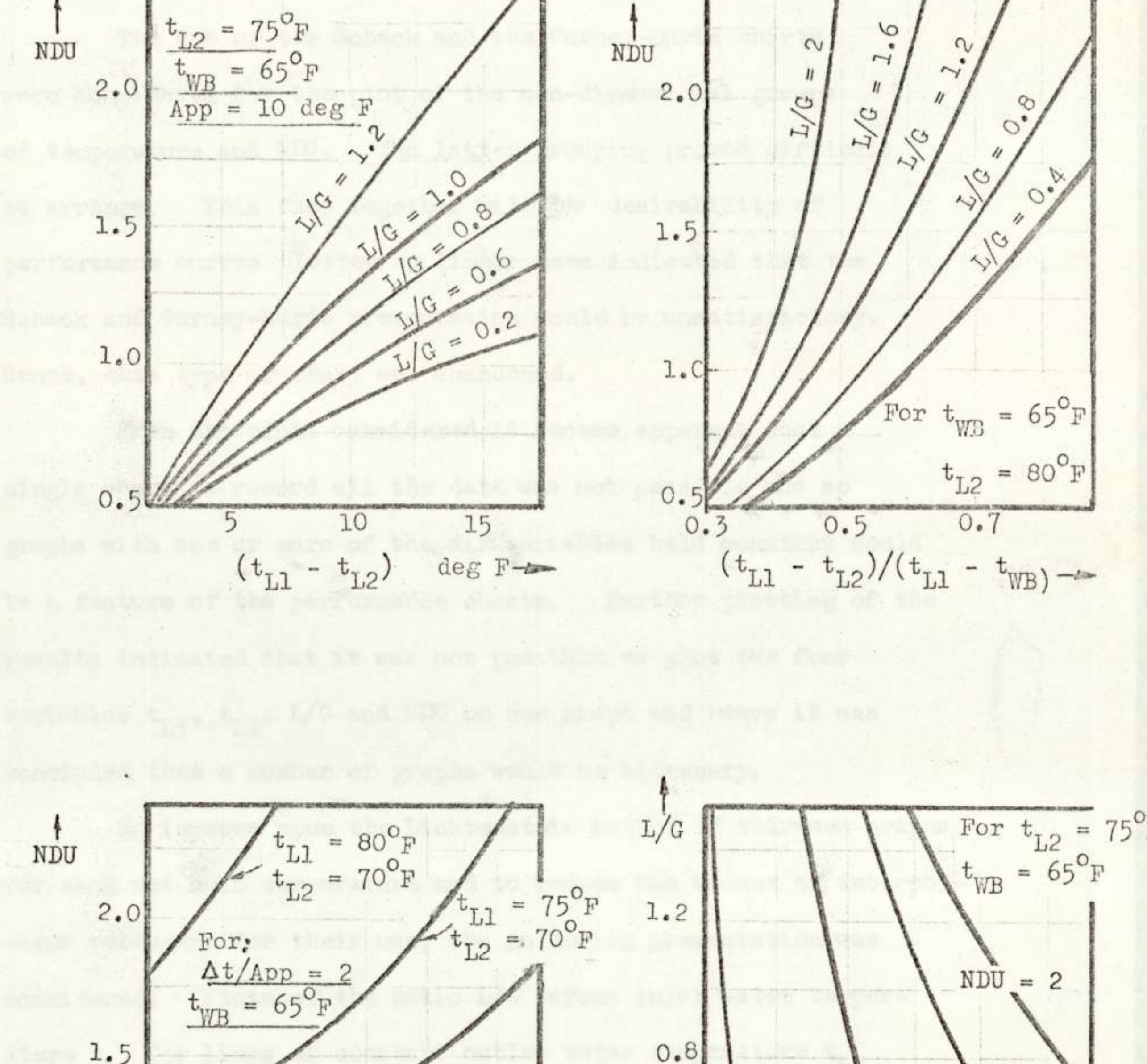
$$\text{Approach} \quad App = t_{L2} - t_{WB}$$

Examples of these plots are shown on Fig 8.2. It can be seen that this technique does not improve the presentation.



VARIOUS TYPES OF PERFORMANCE CHART

Fig 8.1



VARIOUS TYPES OF PERFORMANCE CHART

Fig. 8.2

The use of the Schack and the Gurney-Lurie charts were considered for the plot of the non-dimensional groups of temperature and NDU. The latter grouping proved difficult to arrange. This fact together with the desirability of performance curves plotted on linear axes indicated that the Schack and Gurney-Lurie presentation would be unsatisfactory. Hence, this type of chart was abandoned.

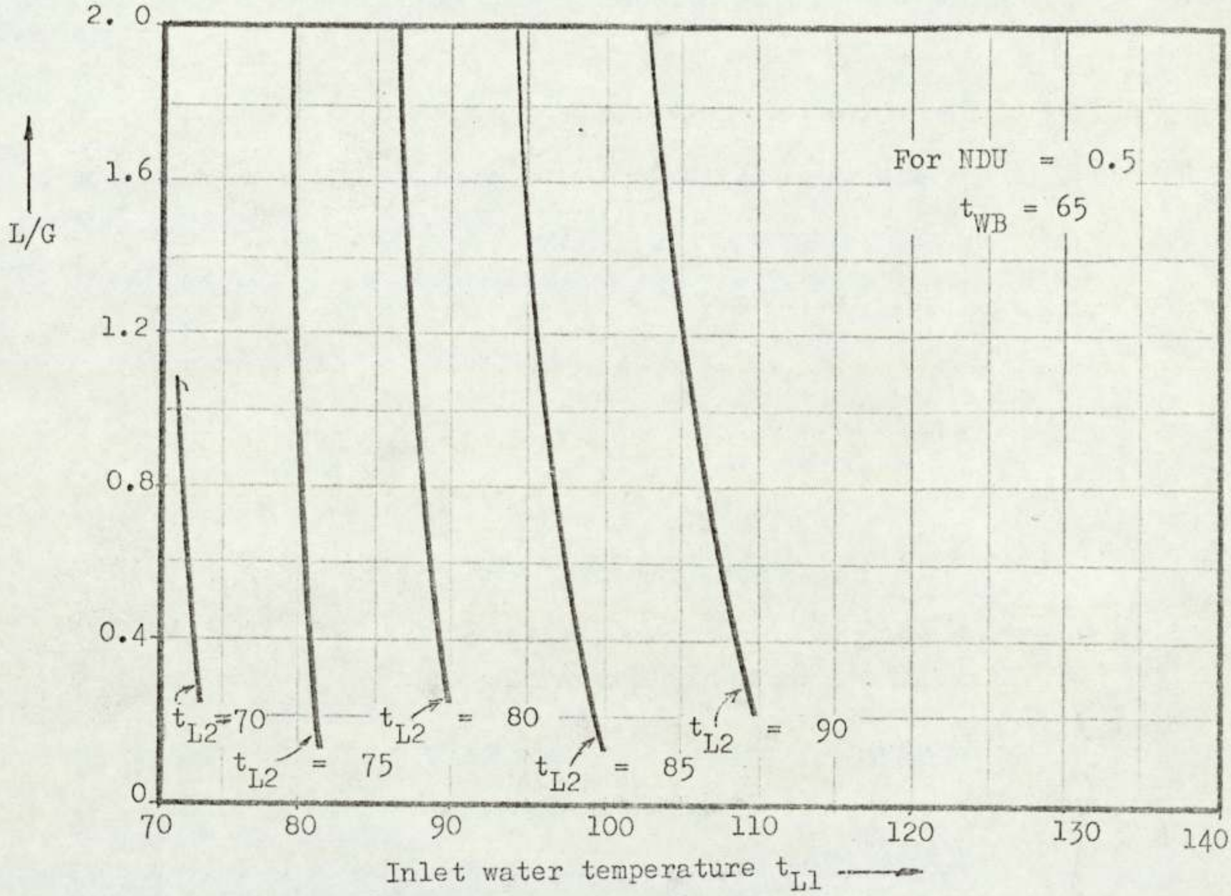
From the plots considered it became apparent that a single chart to record all the data was not possible and so graphs with one or more of the six variables held constant would be a feature of the performance charts. Further plotting of the results indicated that it was not possible to plot the four variables t_{L1}, t_{L2}, L/G and NDU on one graph and hence it was concluded that a number of graphs would be necessary.

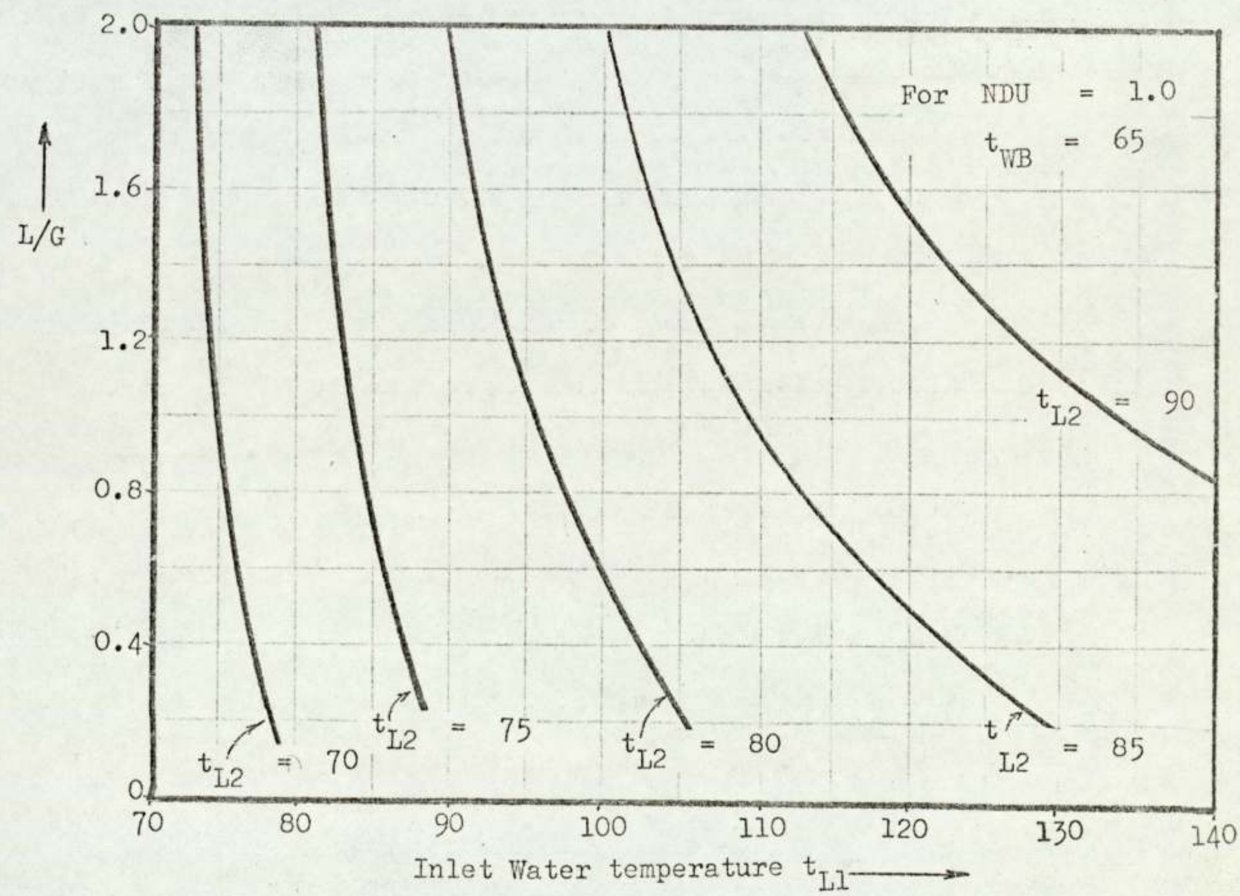
To improve upon the Lichtenstein method of thirteen graphs for each wet bulb temperature and to reduce the amount of interpolation necessary for their use, the following presentation was considered. Plots of the ratio L/G versus inlet water temperature t_{L1} for lines of constant outlet water temperature t_{L2} resulted in the graphs shown on Fig 8.3 where the wet bulb temperature and NDU value were held constant. This presentation looked promising.

Further development revealed that four graphs per wet bulb temperature were necessary, each for a different NDU value. In this form the Performance Charts were not of direct application since to achieve their full utilisation, an auxilliary graph called the Performance Diagram has to be prepared.

8.2 Use of the Counterflow Performance Charts

The Performance Diagram is obtained by selecting a wet bulb temperature, a cooling duty and from the corresponding set of Performance Charts Fig 8.3 a value of L/G is read from each



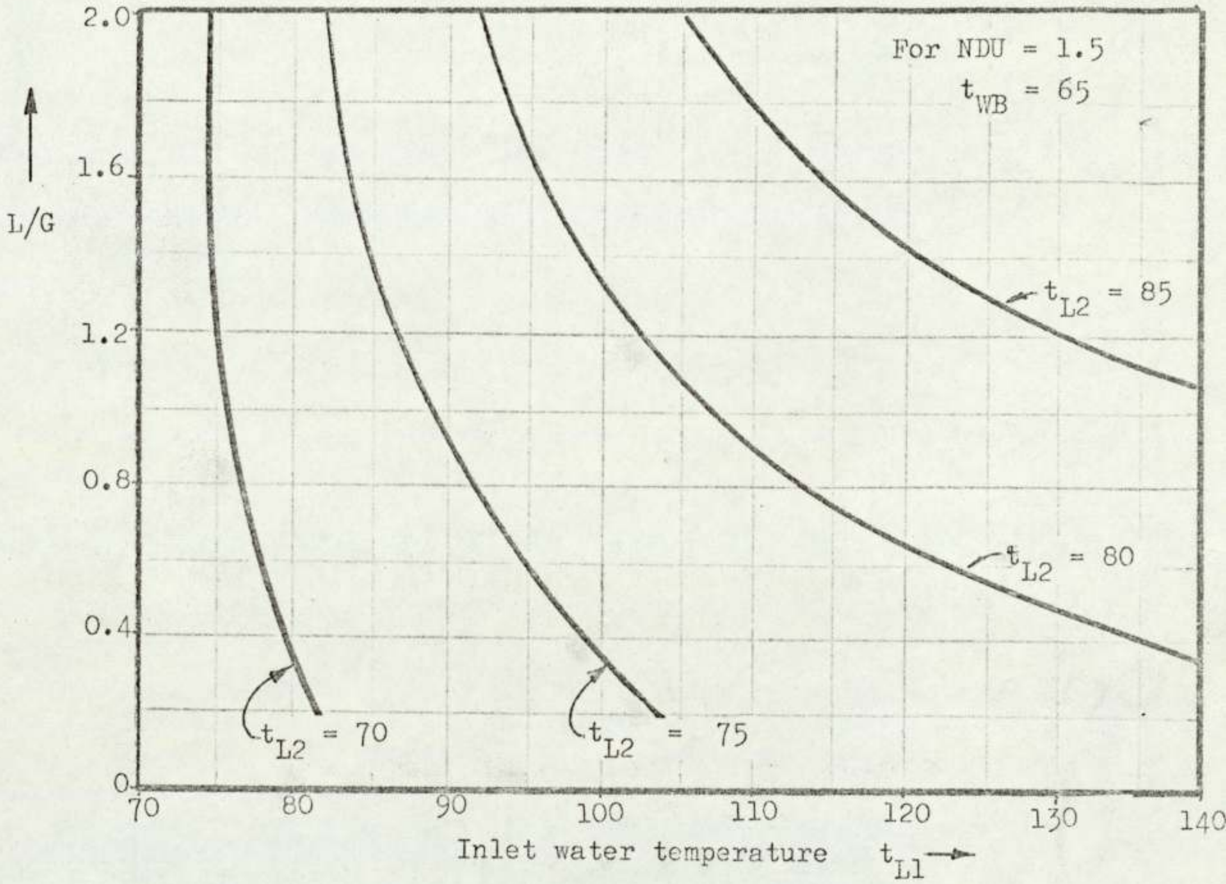


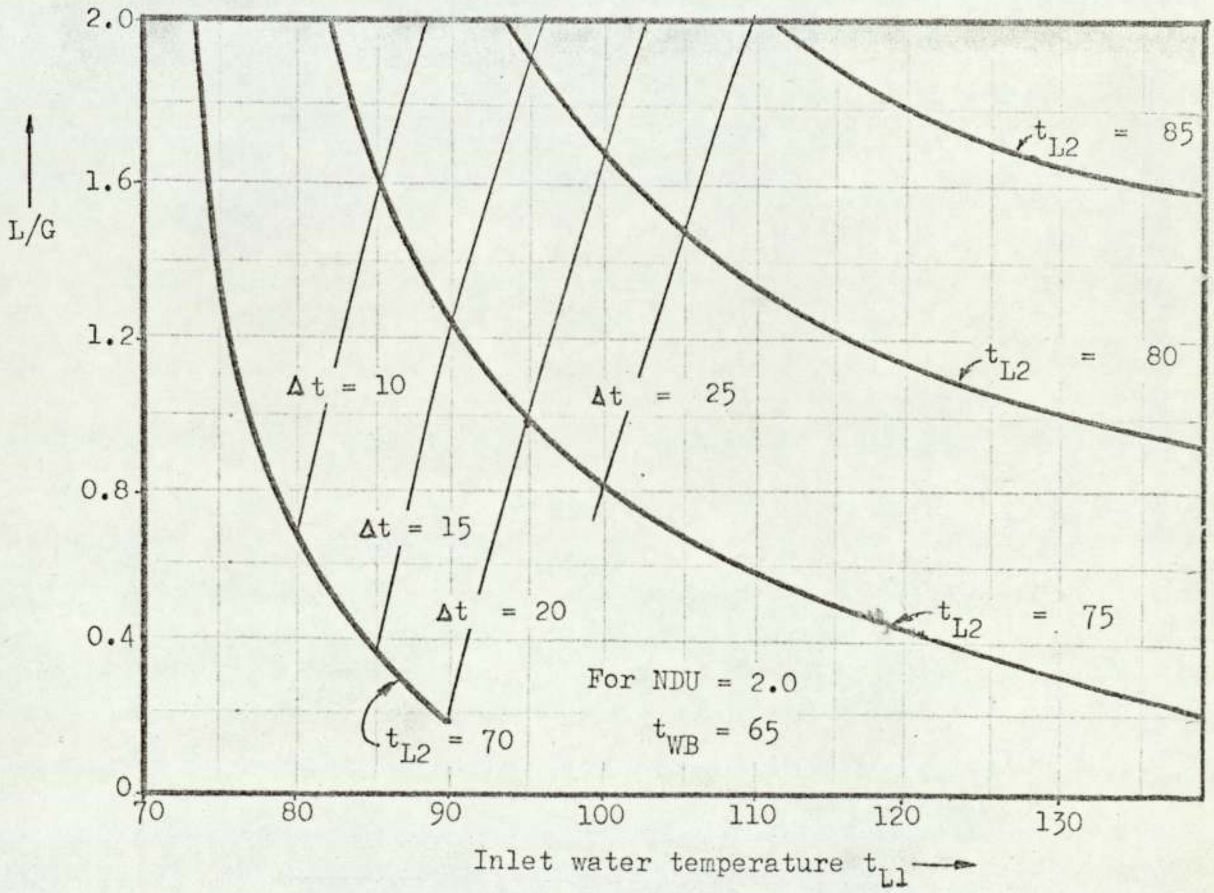
NOTE All temperatures are in degrees Fahrenheit

COUNTERFLOW PERFORMANCE CHARTS

Fig 8.3

Continued





NOTE: All temperatures are in degrees Fahrenheit.

COUNTERFLOW PERFORMANCE CHARTS

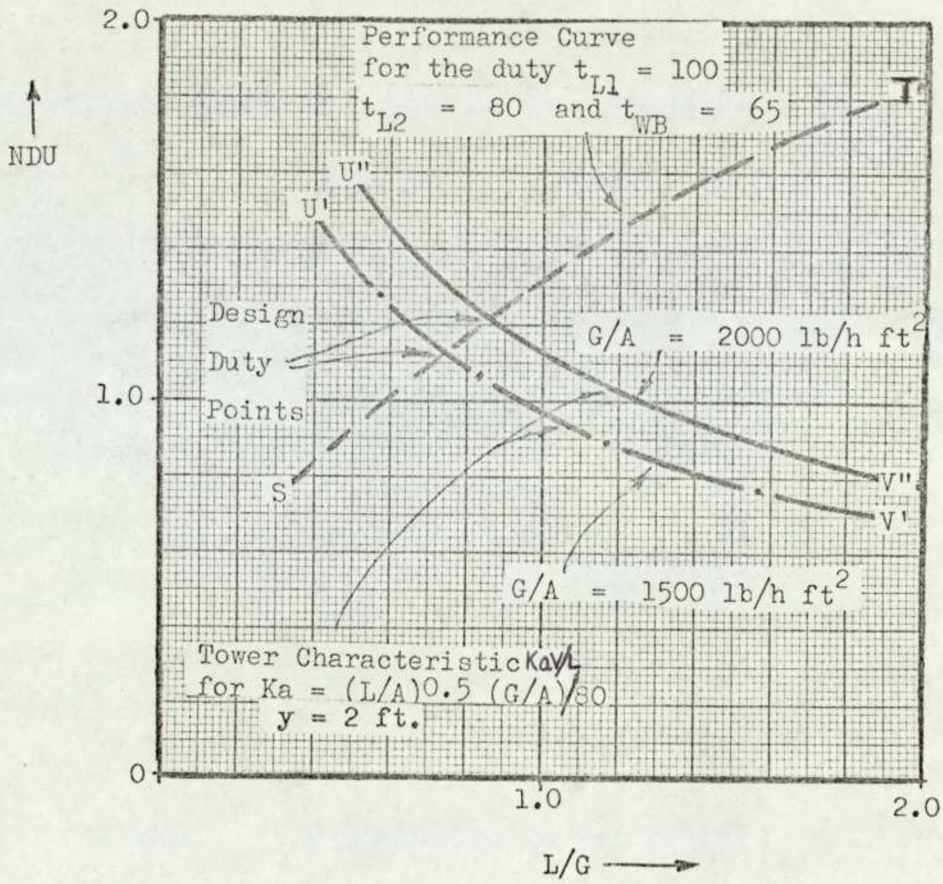
Fig 8.3

of the four NDU graphs. These results are plotted on the Performance Diagram, see Fig 8.4 as a performance curve represented by the line ST. The line UV which describes the tower characteristic KaV/L for the specified operating conditions and packing arrangements is also plotted on the Diagram. The intersection of the tower characteristic UV with the performance curve ST gives the corresponding cooling tower operating condition, called the Design Duty Point.

The Performance Diagram can be extended to accommodate other cooling duties and tower characteristics to give a series of operating conditions for a particular selection. The ability of the Diagram to accommodate a variety of curves and hence a series of operating conditions makes this method of significant use despite the necessity of an indirect plot to get the desired result. In this form the Diagram acts as a permanent, concise record of the final cooling tower selection. Each selection should be prepared on a new Performance Diagram.

This Selection Technique serves as a useful extension to the method proposed by Lichtenstein. Since the Performance Charts are unique, only one set per wet bulb temperature is required to represent all selection possibilities for a large variety of counterflow pack and tower arrangements. One set of Performance Charts consists of four graphs compared to the thirteen graphs used by Lichtenstein.

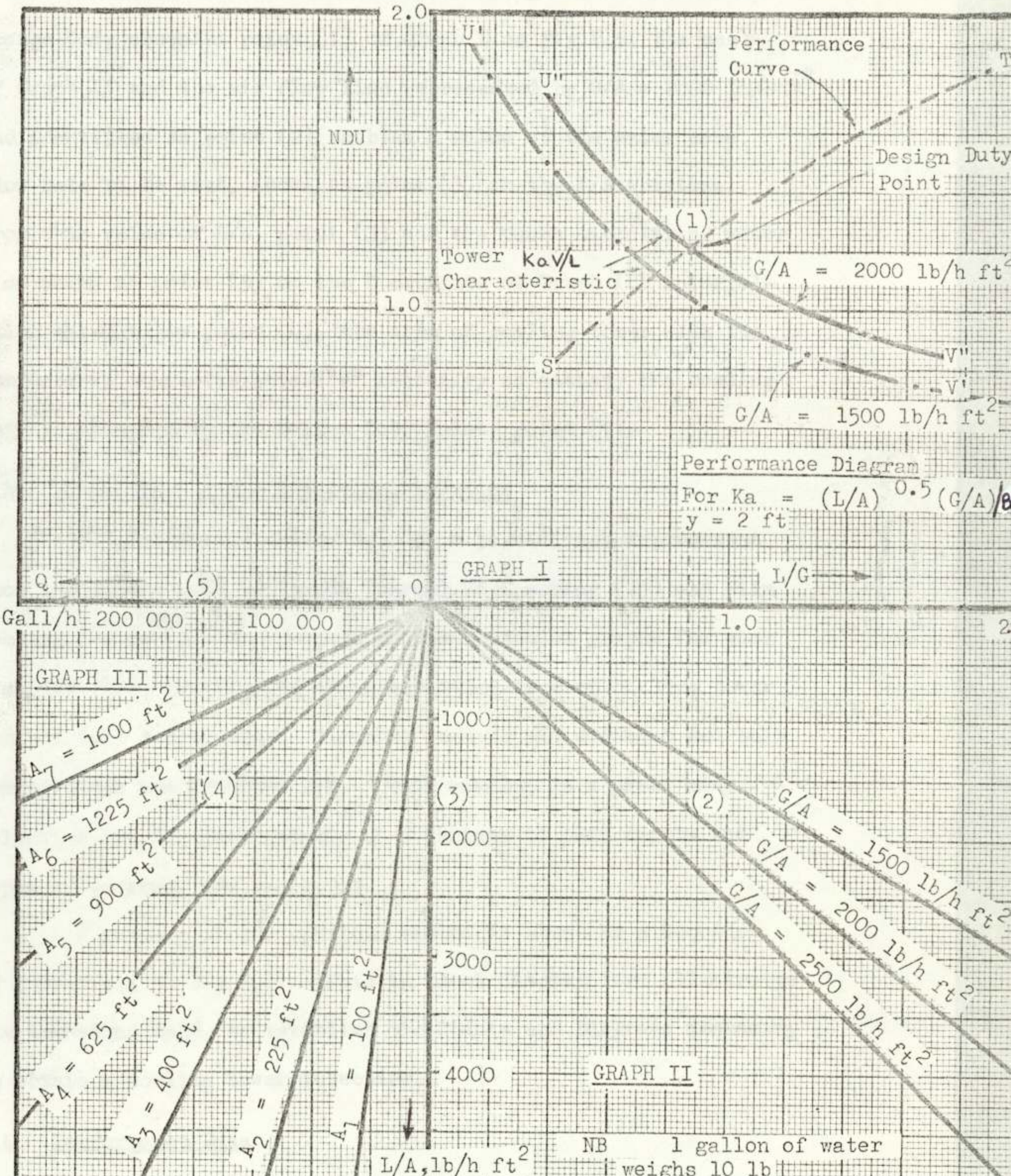
A further development of the Performance Diagram is shown as Graph I on Fig 8.5 where the ordinate L/G is used as the axis of a secondary graph (Graph II). The secondary graph leads into a further graph (Graph III) for relating tower base areas. The Graphs, I, II and III relate cooling duty, circulating water flow rate Q, and the L/G value to cooling towers of particular base areas and performance characteristic.



NOTE All temperatures in degrees Fahrenheit.

COUNTERFLOW PERFORMANCE DIAGRAM

Fig 8.4



NOTE All temperatures in degrees Fahrenheit

EXPANDED COUNTERFLOW PERFORMANCE DIAGRAM

Fig 8.5

From the established Performance Diagram and the chosen Design Duty shown as point (1) of Graph I on Fig 8.5, the use of Graph II and the air flow line represented by $G/A = 2000$ lb/hft^2 and identified as point (2) enables a value of the water mass flow rate to be read, shown as point (3) on the L/A abscissa. From this value of L/A, point (3) and the known circulating water flow shown as point (5) on the Q ordinate of Graph III interpolation establishes point (4), the area of cooling tower with the performance characteristic U"V" necessary to satisfy the cooling duty represented by the Design Duty Point, point (1).

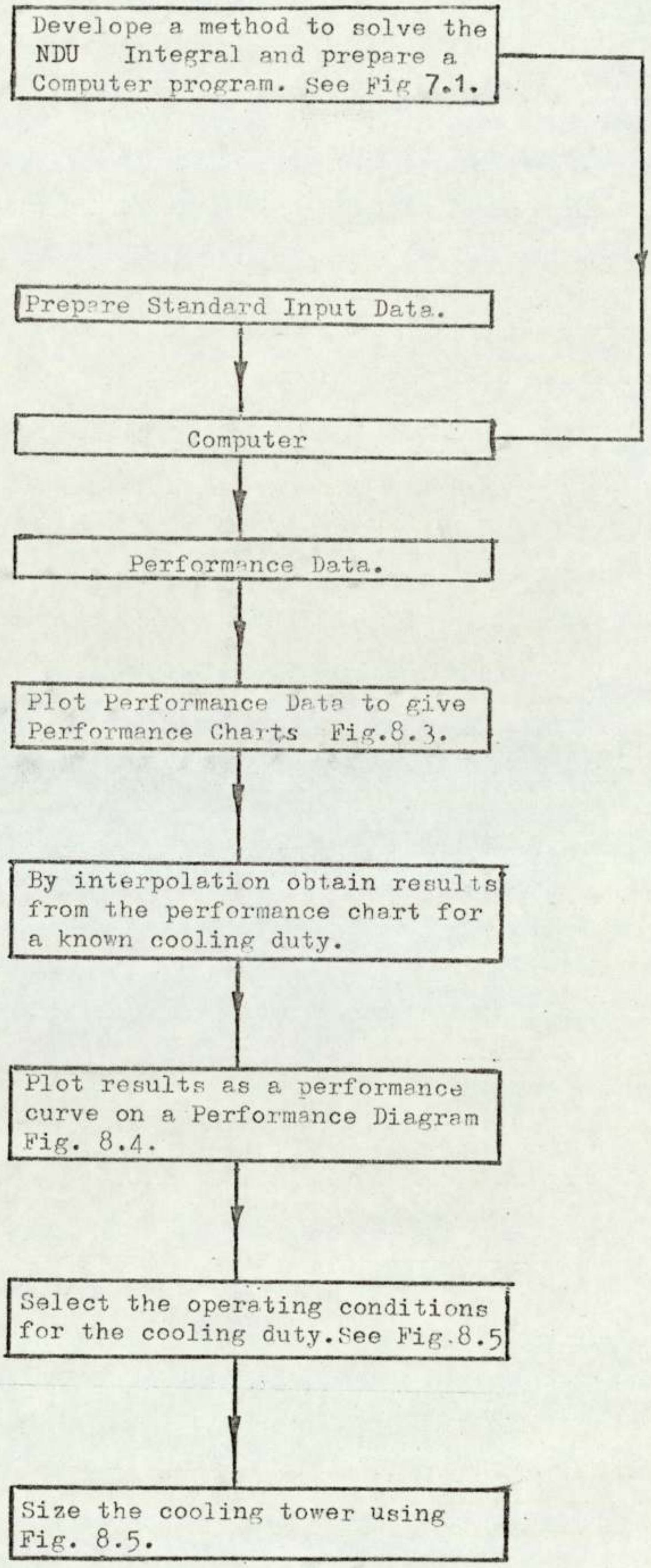
8.3 Comparison of the Selection Technique

This selection method is also a significant improvement upon the method used by Carter Thermal Engineering (23) and Film Cooling Towers (26) who prepare unique selection graphs for each air flow rate, performance characteristic, type of cooling tower and wet bulb temperature. Usually four to six different wet bulb temperatures are provided for. This results in a great many selection graphs, particularly if a variety of pack depths and types of packings are available.

A summary of the previous method is shown as a Sequence of Operations on Fig 8.6. This in block diagram form illustrates the technique used, to establish the Charts and Diagrams necessary to obtain a cooling tower selection.

8.4 Performance Data

During the development of the work necessary to arrive at the Selection Technique described earlier, many sets of Performance Data were obtained (each capable of giving a unique set of Performance Charts). These include data at the following conditions where the air inlet wet bulb temperature t_{WB}, incremental temperature dt and atmospheric pressure P were the parameters considered.



SEQUENCE OF OPERATIONS NECESSARY TO EVOLVE THE SELECTION TECHNIQUE

Fig 8.6

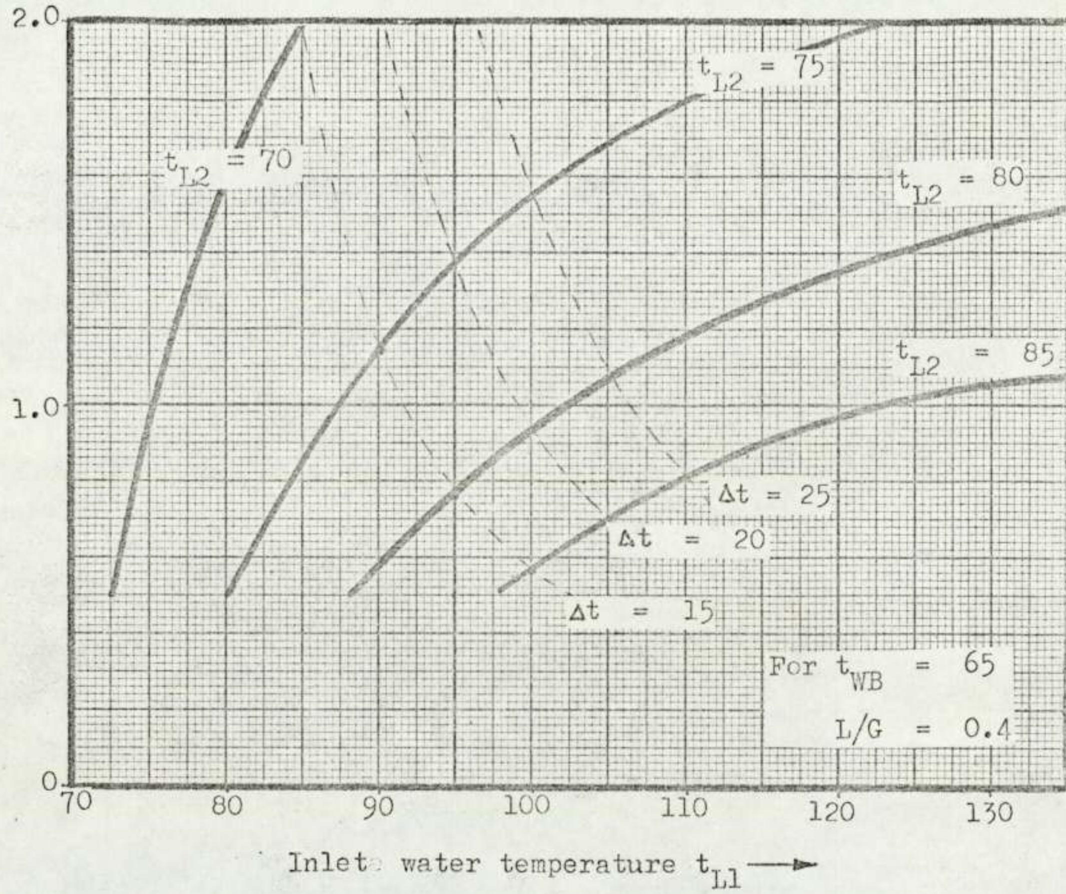
	t_{WB} °F	dt °F	P millibars	
a)	65	0.05	1000	(Sea Level)
b)	65	0.1	1000	"
c)	65	0.25	1000	"
d)	65	0.5	1000	"
e)	65	1.0	1000	"
f)	68	0.5	1000	"
g)	62	0.5	1000	"
h)	60	0.5	1000	"
i)	65	0.5	847	(5000' above sea level).

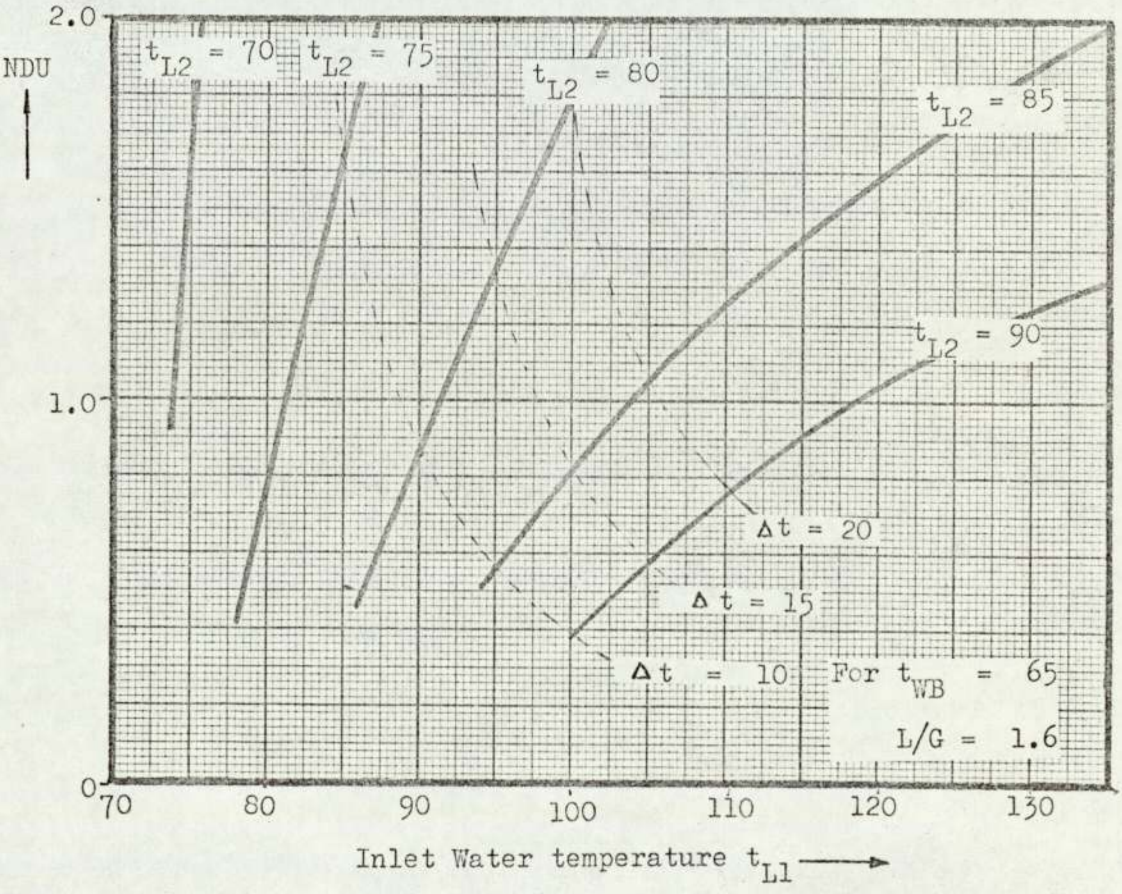
The results used to plot the Intermediate Charts shown on Fig 8.7 were taken from the tabulated data listed on Fig 8.8. From the intermediate graphs, crossplotting enables the Performance Charts as shown on Fig 8.3 to be prepared. The data listed on Fig 8.8 constitutes a typical table of Performance Data based on the parameters identified as a) above.

It is also a simple task to obtain numerical data for a variety of operating conditions other than those mentioned above. The scope of the computer prográm could also be expanded or altered to accommodate additional parameters, or extend the range of the existing parameters.

Variations to the computer program are achieved by replacing the appropriate IBM card. This is a simple operation enabling a vast amount of data to be calculated.

The results and charts should be used with caution, if at all, in the case of large units because the air distribution in particular would deviate from true counterflow. Furthermore the cross-flow of air in the upper region of the tower would distort the water flow pattern and give an uneven distribution.





NOTE All temperatures in degrees Fahrenheit

INTERMEDIATE PERFORMANCE CHARTS

Fig 8.7

PROG 127(24)	T.OFF	T.ON	APP	RANGE	PR	A/R	A/PR	L/G	N-DEL	N-1T	NDU
INTEG. INT=0.05DEG	70	72.45	5.00	2.45	7.45	2.041	0.671	0.20	0.50	0.101	0.100
T.WB=65	70	75.85	5.00	5.85	10.85	0.855	0.461	0.20	1.00	0.200	0.200
ATM= 1000.0	70	81.00	5.00	11.00	16.00	0.455	0.313	0.20	1.50	0.301	0.300
	70	89.10	5.00	19.10	24.10	0.262	0.207	0.20	2.00	0.400	0.400
	70	104.05	5.00	34.05	39.05	0.147	0.128	0.20	2.50	0.500	0.500
	70	148.90	5.00	78.90	83.90	0.063	0.060	0.20	3.00	0.600	0.600
	70	72.30	5.00	2.30	7.30	2.174	0.685	0.40	0.50	0.201	0.200
	70	75.25	5.00	5.25	10.25	0.952	0.488	0.40	1.00	0.401	0.400
	70	79.20	5.00	9.20	14.20	0.543	0.352	0.40	1.50	0.602	0.600
	70	84.70	5.00	14.70	19.70	0.340	0.254	0.40	2.00	0.801	0.800
	70	93.05	5.00	23.05	28.05	0.217	0.178	0.40	2.50	1.001	1.000
	70	108.00	5.00	38.00	43.00	0.132	0.116	0.40	3.00	1.200	1.200
	70	72.20	5.00	2.20	7.20	2.273	0.694	0.60	0.50	0.304	0.300
	70	74.70	5.00	4.70	9.70	1.064	0.515	0.60	1.00	0.600	0.600
	70	77.75	5.00	7.75	12.75	0.645	0.392	0.60	1.50	0.902	0.900
	70	81.50	5.00	11.50	16.50	0.435	0.303	0.60	2.00	1.201	1.200
	70	86.40	5.00	16.40	21.40	0.305	0.234	0.60	2.50	1.501	1.500
	70	93.35	5.00	23.35	28.35	0.214	0.176	0.60	3.00	1.802	1.800
	70	72.10	5.00	2.10	7.10	2.381	0.704	0.80	0.50	0.407	0.400
	70	74.25	5.00	4.25	9.25	1.176	0.541	0.80	1.00	0.803	0.800
	70	76.60	5.00	6.60	11.60	0.758	0.431	0.80	1.50	1.205	1.200
	70	79.20	5.00	9.20	14.20	0.543	0.352	0.80	2.00	1.606	1.600
	70	82.15	5.00	12.15	17.15	0.412	0.292	0.80	2.50	2.005	2.000
	70	85.65	5.00	15.65	20.65	0.319	0.242	0.80	3.00	2.402	2.400
	70	74.55	5.00	4.55	9.55	1.099	0.524	1.00	1.00	1.012	1.000
	70	76.55	5.00	6.55	11.55	0.763	0.433	1.00	1.50	1.501	1.500
	70	78.60	5.00	8.60	13.60	0.581	0.368	1.00	2.00	2.002	2.000
	70	80.70	5.00	10.70	15.70	0.467	0.318	1.00	2.50	2.502	2.500
	70	82.95	5.00	12.95	17.95	0.386	0.279	1.00	3.00	3.009	3.000
	70	74.15	5.00	4.15	9.15	1.205	0.546	1.20	1.00	1.204	1.200
	70	75.75	5.00	5.75	10.75	0.870	0.465	1.20	1.50	1.801	1.800
	70	77.20	5.00	7.20	12.20	0.694	0.410	1.20	2.00	2.405	2.400
	70	78.55	5.00	8.55	13.55	0.585	0.369	1.20	2.50	3.023	3.000
	70	79.75	5.00	9.75	14.75	0.513	0.339	1.20	3.00	3.615	3.600
	70	73.85	5.00	3.85	8.85	1.299	0.565	1.40	1.00	1.414	1.400
	70	75.10	5.00	5.10	10.10	0.980	0.495	1.40	1.50	2.106	2.100
	70	76.15	5.00	6.15	11.15	0.813	0.448	1.40	2.00	2.831	2.800
	70	76.95	5.00	6.95	11.95	0.719	0.418	1.40	2.50	3.509	3.500

TYPICAL PERFORMANCE DATA OBTAINED FROM THE FORTRAN PROGRAM   Fig 8.8

Continued

	70	77.65	5.00	7.65	12.65	0.654	0.395	1.40	3.00	4.230	4.200
	70	73.55	5.00	3.55	8.55	1.408	0.585	1.60	1.00	1.604	1.600
	70	74.55	5.00	4.55	9.55	1.099	0.524	1.60	1.50	2.404	2.400
	70	75.30	5.00	5.30	10.30	0.943	0.485	1.60	2.00	3.238	3.200
	70	75.85	5.00	5.85	10.85	0.855	0.461	1.60	2.50	4.090	4.000
	70	76.20	5.00	6.20	11.20	0.806	0.446	1.60	3.00	4.828	4.800
	70	73.30	5.00	3.30	8.30	1.515	0.602	1.80	1.00	1.806	1.800
	70	74.10	5.00	4.10	9.10	1.220	0.549	1.80	1.50	2.714	2.700
	70	74.65	5.00	4.65	9.65	1.075	0.518	1.80	2.00	3.692	3.600
	70	74.95	5.00	4.95	9.95	1.010	0.503	1.80	2.50	4.507	4.500
	70	75.20	5.00	5.20	10.20	0.962	0.490	1.80	3.00	5.549	5.400
	70	73.10	5.00	3.10	8.10	1.613	0.617	2.00	1.00	2.026	2.000
	70	73.75	5.00	3.75	8.75	1.333	0.571	2.00	1.50	3.083	3.000
	70	74.10	5.00	4.10	9.10	1.220	0.549	2.00	2.00	4.064	4.000
	70	74.35	5.00	4.35	9.35	1.149	0.535	2.00	2.50	5.322	5.000
	70	74.45	5.00	4.45	9.45	1.124	0.529	2.00	3.00	6.220	6.000
	70	72.90	5.00	2.90	7.90	1.724	0.633	2.20	1.00	2.229	2.200
	70	73.40	5.00	3.40	8.40	1.471	0.595	2.20	1.50	3.336	3.300
	70	73.70	5.00	3.70	8.70	1.351	0.575	2.20	2.00	4.673	4.400
	70	73.80	5.00	3.80	8.80	1.316	0.568	2.20	2.50	5.526	5.500
	70	73.90	5.00	3.90	8.90	1.282	0.562	2.20	3.00	7.304	6.600
	70	72.75	5.00	2.75	7.75	1.818	0.645	2.40	1.00	2.491	2.400
	70	73.15	5.00	3.15	8.15	1.587	0.613	2.40	1.50	3.764	3.600
	70	73.35	5.00	3.35	8.35	1.493	0.599	2.40	2.00	5.238	4.800
	70	73.45	5.00	3.45	8.45	1.449	0.592	2.40	2.50	7.532	6.000
FAIL XS T.ON	70	165.00	5.00	3.45	8.45	1.449	0.592	2.40	3.00	7.532	6.000
	70	72.60	5.00	2.60	7.60	1.923	0.658	2.60	1.00	2.735	2.600
	70	72.90	5.00	2.90	7.90	1.724	0.633	2.60	1.50	4.027	3.900
	70	73.05	5.00	3.05	8.05	1.639	0.621	2.60	2.00	5.692	5.200
	70	73.10	5.00	3.10	8.10	1.613	0.617	2.60	2.50	7.421	6.500
FAIL XS T.ON	70	165.00	5.00	3.10	8.10	1.613	0.617	2.60	3.00	7.421	6.500
	70	72.45	5.00	2.45	7.45	2.041	0.671	2.80	1.00	2.907	2.800
	70	72.70	5.00	2.70	7.70	1.852	0.649	2.80	1.50	4.367	4.200
	70	72.80	5.00	2.80	7.80	1.786	0.641	2.80	2.00	6.074	5.600
	70	72.85	5.00	2.85	7.85	1.754	0.637	2.80	2.50	39.653	7.000
	70	72.90	5.00	2.90	7.90	1.724	0.633	2.80	3.00	71.744	8.400
	70	72.30	5.00	2.30	7.30	2.174	0.685	3.00	1.00	3.018	3.000

TYPICAL PERFORMANCE DATA OBTAINED FROM THE FORTRAN PROGRAM Fig 8.8 Continued

	70	72.55	5.00	2.55	7.55	1.961	0.662	3.00	1.50	5.160	4.500
	70	72.60	5.00	2.60	7.60	1.923	0.658	3.00	2.00	7.032	6.000
	70	148.15	5.00	78.15	83.15	0.064	0.060	3.00	2.50	24.647	7.500
	70	148.20	5.00	78.20	83.20	0.064	0.060	3.00	3.00	59.429	9.000
	75	80.30	10.00	5.30	15.30	1.887	0.654	0.20	0.50	0.100	0.100
	75	88.65	10.00	13.65	23.65	0.733	0.423	0.20	1.00	0.200	0.200
	75	103.85	10.00	28.85	38.85	0.347	0.257	0.20	1.50	0.300	0.300
	75	149.60	10.00	74.60	84.60	0.134	0.118	0.20	2.00	0.400	0.400
FAIL XS T.ON	75	165.00	10.00	74.60	84.60	0.134	0.118	0.20	2.50	0.400	0.400
FAIL XS T.ON	75	165.05	10.00	74.60	84.60	0.134	0.118	0.20	3.00	0.400	0.400
	75	80.05	10.00	5.05	15.05	1.980	0.664	0.40	0.50	0.201	0.200
	75	87.15	10.00	12.15	22.15	0.823	0.451	0.40	1.00	0.400	0.400
	75	98.50	10.00	23.50	33.50	0.426	0.299	0.40	1.50	0.601	0.600
	75	122.00	10.00	47.00	57.00	0.213	0.175	0.40	2.00	0.800	0.800
FAIL XS T.ON	75	165.00	10.00	47.00	57.00	0.213	0.175	0.40	2.50	0.800	0.800
FAIL XS T.ON	75	165.05	10.00	47.00	57.00	0.213	0.175	0.40	3.00	0.800	0.800
	75	79.80	10.00	4.80	14.80	2.083	0.676	0.60	0.50	0.302	0.300
	75	85.85	10.00	10.85	20.85	0.922	0.480	0.60	1.00	0.600	0.600
	75	94.40	10.00	19.40	29.40	0.515	0.340	0.60	1.50	0.901	0.900
	75	108.50	10.00	33.50	43.50	0.299	0.230	0.60	2.00	1.201	1.200
	75	147.65	10.00	72.65	82.65	0.138	0.121	0.60	2.50	1.500	1.500
FAIL XS T.ON	75	165.00	10.00	72.65	82.65	0.138	0.121	0.60	3.00	1.500	1.500
	75	79.55	10.00	4.55	14.55	2.198	0.687	0.80	0.50	0.402	0.400
	75	84.75	10.00	9.75	19.75	1.026	0.506	0.80	1.00	0.802	0.800
	75	91.20	10.00	16.20	26.20	0.617	0.382	0.80	1.50	1.201	1.200
	75	100.15	10.00	25.15	35.15	0.398	0.284	0.80	2.00	1.601	1.600
	75	115.60	10.00	40.60	50.60	0.246	0.198	0.80	2.50	2.001	2.000
FAIL XS T.ON	75	165.00	10.00	40.60	50.60	0.246	0.198	0.80	3.00	2.001	2.000
	75	79.30	10.00	4.30	14.30	2.326	0.699	1.00	0.50	0.500	0.500
	75	83.80	10.00	8.80	18.80	1.136	0.532	1.00	1.00	1.005	1.000
	75	88.70	10.00	13.70	23.70	0.730	0.422	1.00	1.50	1.503	1.500
	75	94.55	10.00	19.55	29.55	0.512	0.338	1.00	2.00	2.001	2.000
	75	102.45	10.00	27.45	37.45	0.364	0.267	1.00	2.50	2.502	2.500

TYPICAL PERFORMANCE DATA OBTAINED FROM THE FORTRAN PROGRAM   Fig 8.8   Continued

Fig 8.8

75	115.35	10.00	40.35	50.35	0.248	0.199	1.00	3.00	3.001	3.000
75	79.10	10.00	4.10	14.10	2.439	0.709	1.20	0.50	0.601	0.600
75	82.95	10.00	7.95	17.95	1.258	0.557	1.20	1.00	1.206	1.200
75	86.70	10.00	11.70	21.70	0.855	0.461	1.20	1.50	1.805	1.800
75	90.60	10.00	15.60	25.60	0.641	0.391	1.20	2.00	2.401	2.400
75	95.00	10.00	20.00	30.00	0.500	0.333	1.20	2.50	3.003	3.000
75	100.35	10.00	25.35	35.35	0.394	0.283	1.20	3.00	3.604	3.600
75	78.90	10.00	3.90	13.90	2.564	0.719	1.40	0.50	0.701	0.700
75	82.20	10.00	7.20	17.20	1.389	0.581	1.40	1.00	1.406	1.400
75	85.10	10.00	10.10	20.10	0.990	0.498	1.40	1.50	2.109	2.100
75	87.75	10.00	12.75	22.75	0.784	0.440	1.40	2.00	2.808	2.800
75	90.30	10.00	15.30	25.30	0.654	0.395	1.40	2.50	3.509	3.500
75	92.85	10.00	17.85	27.85	0.560	0.359	1.40	3.00	4.208	4.200
75	78.75	10.00	3.75	13.75	2.667	0.727	1.60	0.50	0.810	0.800
75	81.55	10.00	6.55	16.55	1.527	0.604	1.60	1.00	1.606	1.600
75	83.80	10.00	8.80	18.80	1.136	0.532	1.60	1.50	2.417	2.400
75	85.60	10.00	10.60	20.60	0.943	0.485	1.60	2.00	3.214	3.200
75	87.10	10.00	12.10	22.10	0.826	0.452	1.60	2.50	4.005	4.000
75	88.40	10.00	13.40	23.40	0.746	0.427	1.60	3.00	4.802	4.800
75	78.55	10.00	3.55	13.55	2.817	0.738	1.80	0.50	0.903	0.900
75	81.00	10.00	6.00	16.00	1.667	0.625	1.80	1.00	1.814	1.800
75	82.70	10.00	7.70	17.70	1.299	0.565	1.80	1.50	2.708	2.700
75	83.95	10.00	8.95	18.95	1.117	0.528	1.80	2.00	3.612	3.600
75	84.90	10.00	9.90	19.90	1.010	0.503	1.80	2.50	4.547	4.500
75	85.60	10.00	10.60	20.60	0.943	0.485	1.80	3.00	5.475	5.400
75	78.40	10.00	3.40	13.40	2.941	0.746	2.00	0.50	1.008	1.000
75	80.50	10.00	5.50	15.50	1.818	0.645	2.00	1.00	2.017	2.000
75	81.85	10.00	6.85	16.85	1.460	0.593	2.00	1.50	3.045	3.000
75	82.70	10.00	7.70	17.70	1.299	0.565	2.00	2.00	4.049	4.000
75	83.25	10.00	8.25	18.25	1.212	0.548	2.00	2.50	5.035	5.000
75	83.65	10.00	8.65	18.65	1.156	0.536	2.00	3.00	6.137	6.000
75	78.25	10.00	3.25	13.25	3.077	0.755	2.20	0.50	1.110	1.100
75	80.05	10.00	5.05	15.05	1.980	0.664	2.20	1.00	2.214	2.200
75	81.10	10.00	6.10	16.10	1.639	0.621	2.20	1.50	3.348	3.300
75	81.70	10.00	6.70	16.70	1.493	0.599	2.20	2.00	4.482	4.400
75	82.05	10.00	7.05	17.05	1.418	0.587	2.20	2.50	5.635	5.500
75	82.25	10.00	7.25	17.25	1.379	0.580	2.20	3.00	6.780	6.600
75	78.10	10.00	3.10	13.10	3.226	0.763	2.40	0.50	1.207	1.200
75	79.65	10.00	4.65	14.65	2.151	0.683	2.40	1.00	2.405	2.400

TYPICAL PERFORMANCE DATA OBTAINED FROM THE FORTRAN PROGRAM Fig 8.8

APPENDIX 8.1

EVALUATION OF THE COUNTERFLOW TOWER CHARACTERISTIC

The tower characteristic Ka (mass transfer coefficient X area) defines the simultaneous heat and mass transfer taking place within the confines of a cooling tower. The characteristic can be represented by an empirical equation expressed as a function of L/A and G/A as an equation of the form :

$$Ka = C(L/A)^{q}(G/A)^{n} \qquad (8.1)$$

which represents a forced draught cooling tower, where typical values of the constants are :

$q = 0.5 \quad n = 0.8 \quad C = 1/80$

For a particular pack type and height a value of $n = 1.0$ is used in the calculations to facilitate evaluation of the equation (8.1) when establishing the Performance Diagrams Fig 8.4 and Fig 8.5. In order to plot the characteristic on the Diagram a value for the mass rate of air flowing per unit area, G/A has to be nominated. To aid calculation the values used are :

$G/A = 1500$ lb/h ft^2 and $G/A = 2000$ lb/h ft^2

In pratice a much used air velocity through a mechanical draught tower is 500 FPM. This gives a corresponding value for the mass flow rate of air as :

$G/A = 2200$ lb/ h ft^2

whose actual value is dependent upon the air conditions chosen. With a known value of G/A and a nominated value of L/G the corresponding L/A value can be obtained. The overall transfer coefficient can then be calculated for these values using equation (8.1). The results are plotted as the Tower Characteristic, lines U'V', and U"V" on Fig 8.4 and Fig 8.5.

CHAPTER NINE

THEORETICAL TREATMENT OF THE CROSSFLOW PROCESS.

From the examination of a crossflow cooling tower it is found that the temperature conditions vary in two directions (unlike the one direction for the counterflow tower) as the air passes through the packing, hence the driving force becomes progressively less. This results in the isothermals (solid) and lines of constant enthalpy (dotted) as shown on Fig 2.23.

With the air and water conditions varying vertically and horizontally the pack section has to be divided into many elemental volumes to enable a solution to be developed. The section is divided into elemental volumes δV of unit width, where δy and δx are the vertical and horizontal dimensions of the element. See Fig 9.1.

Referring to Fig 9.1 it is necessary to analyse completely the elemental volume ABCD within the section of packing. This element has the air and water conditions at inlet of t_{Ly}, h_{Gx} and at outlet $(t_{Ly} + (\partial t_{Ly}/\partial y)\delta y)$, $(h_{Gx} + (\partial h_{Gx}/\partial x)\delta x)$. For air and water flow rates through the element represented by $\delta y G/Y$ and $\delta x L/X$, the energy balance becomes :

$$\delta y(G/Y)h_{Gx} + c_L(\delta x\, L/X)t_{Ly}$$

$$= \delta y(G/Y)(h_{Gx} + (\partial h_G/\partial x)\delta x) + c_L\delta x(L/X)(t_{Ly} + (\partial t_{Ly}/\partial y)\delta y) \qquad (9.1)$$

which reduced to a partial differential equation relating the rate of water temperature decrease with the rate of air enthalpy increase, namely :

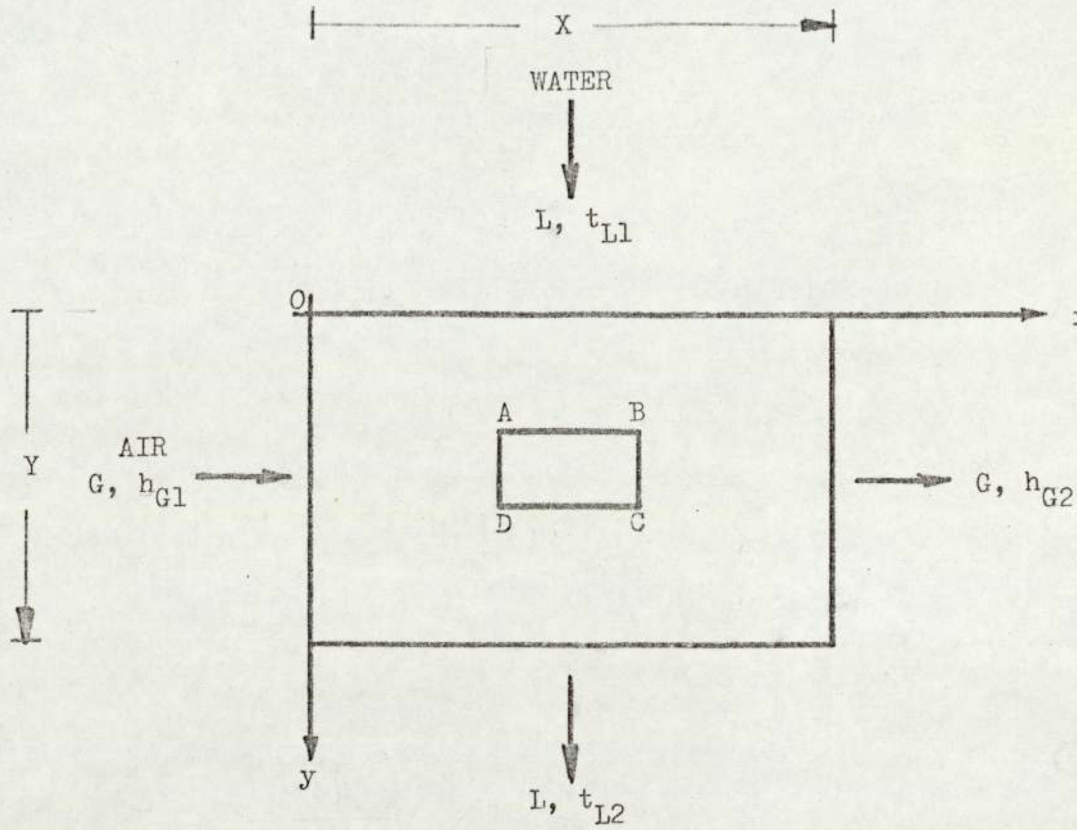
$$(\delta y\ G/Y)(\partial h_{Gx}/\partial x)\delta x = -c_L(\delta x L/X)(\partial t_{Ly}/\partial y)\delta y \qquad (9.2)$$

The corresponding rate equation for energy transfer is :

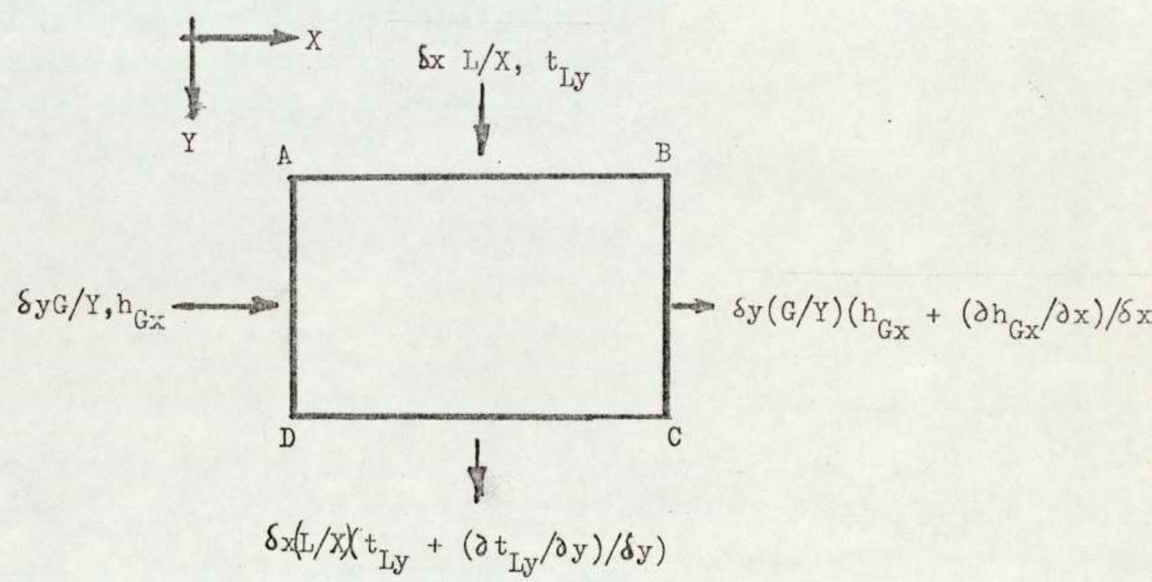
$$(\delta y\ G/Y)(\partial h_{Gx}/\partial x)\delta x = Ka\,\delta x\delta y\,(h_L - h_G) \qquad (9.3)$$

where K is the overall energy transfer coefficient and $(h_L - h_G)$ is the average driving force within the element.

Referring to Fig 9.2 in which the vertical section is divided into distance increments Δy; let $t_{L(n,m)}$ represent the

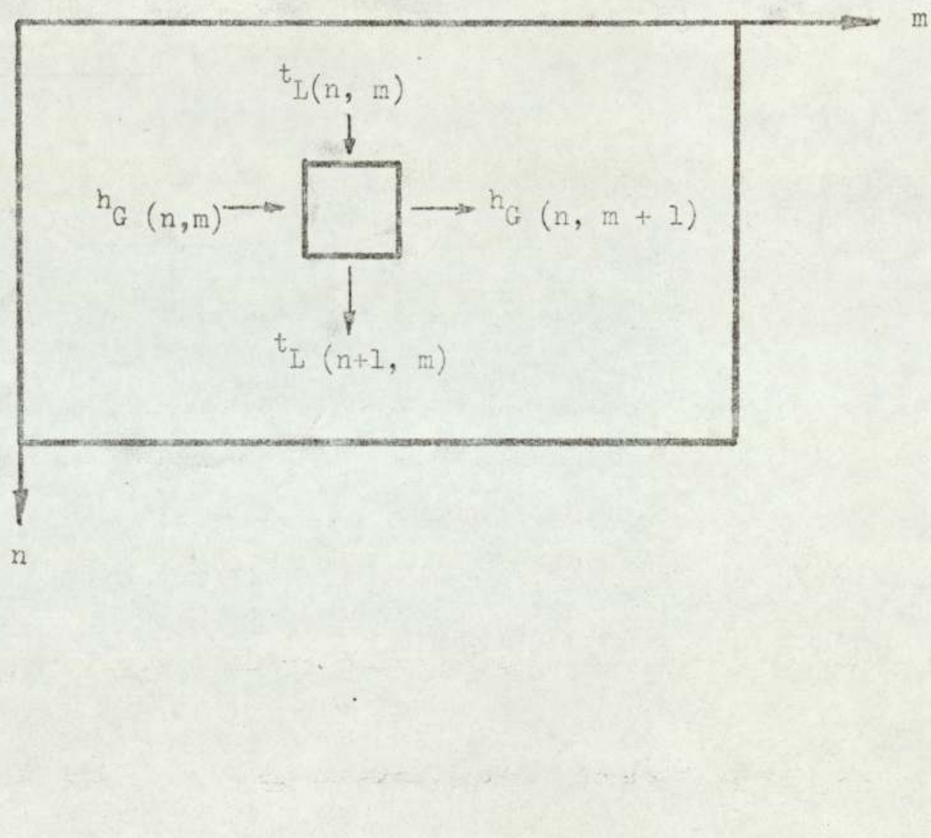


SKETCH SHOWING CONDITIONS AROUND A SECTION OF CROSSFLOW PACKING



ENLARGEMENT OF SMALL ELEMENT ABCD FROM A SECTION OF CROSSFLOW PACKING

Fig 9.1



FINITE DIFFERENCE NOTATION

Fig 9.2

water temperature at n distance increments from the origin and $t_{L(n+1,m)}$ at (n + 1) distance increments from the origin. For the section divided into horizontal distance increments Δx let $h_{G(n,m)}$ represent the air enthalpy at m distance increments from the origin and $h_{G(n,m+1)}$ at (m + 1) distance increments from the origin.

Expressing the partial derivatives in finite difference form gives :

$$\partial h_{Gx} / \partial x = (h_{G(n,m+1)} - h_{G(n,m)}) / \Delta x \tag{9.4}$$

$$\partial t_{Ly} / \partial y = (t_{L(n+1,m)} - t_{L(n,m)}) / \Delta y \tag{9.5}$$

By substitution the equations (9.2) and (9.3) can be written in finite difference form as :

a) $$\Delta y (G/Y)(h_{G(n,m+1)} - h_{G(n,m)}) \Delta x / \Delta x = c_L \Delta x (L/X)(t_{L(n+1,m)} - t_{L(n,m)}) \Delta y / \Delta y .$$

b) $$\Delta y (G/Y)(h_{G(n,m+1)} - h_{G(n,m)}) \Delta x / \Delta x = Ka \Delta x \Delta y \; \left((h_{L(n+1,m)} + h_{L(n,m)})/2 - (h_{G(n,m+1)} + h_{G(n,m)})/2 \right).$$

c) $$c_L \Delta x (L/X)(t_{L(n+1,m)} - t_{L(n,m)}) \Delta y / \Delta y = - Ka \Delta x \Delta y \; \left((h_{L(n+1,m)} + h_{L(n,m)})/2 - (h_{G(n,m+1)} + h_{G(n,m)})/2 \right)$$

For an element arranged to have $\Delta x = \Delta y$ the above equations can be simplified to give :

$$(G/Y) \, dh_{Gx} = -c_L \, (L/X) dt_{Ly} \tag{9.6}$$

$$(G/Y) \, dh_{Gx} = Kadx(h_L - h_G)_{average} \tag{9.7}$$

$$c_L (L/X) dt_{L_y} = -Kady(h_L - h_G)_{average} \tag{9.8}$$

Solving the equations for the element with the limits 'in' and 'out' identified with the suffix 'i' and 'o' gives :

$$(G/Y)(h_{Gxi} - h_{Gxo}) = -c_L (L/X)(t_{Lyi} - t_{Lyo}) \tag{9.9}$$

$$\frac{KadxY}{G} = \frac{(h_{Gxi} - h_{Gxo})}{\left(\frac{h_{Li} + h_{Lo}}{2}\right) - \left(\frac{h_{Gi} + h_{Go}}{2}\right)} \tag{9.10}$$

$$\frac{KadyX}{c_L L} = - \frac{c_L\,(t_{Lyi} - t_{Lyo})}{\frac{(h_{Li} + h_{Lo})}{2} - \frac{h_{Gi} + h_{Go})}{2}} \tag{9.11}$$

Rearranging the equation (9.11) ignoring the sign and putting $c_L = 1$ gives :

$$dy = \frac{L}{KaX}\,\frac{(t_{Lyi} - t_{Lyo})}{\frac{(h_{Li} + h_{Lo})}{2} - \frac{(h_{Gi} + h_{Go})}{2}} \tag{9.12}$$

Since the Number of Diffusion Units characteristic of a process is determined wholly by the operating conditions, then the Number of Diffusion Units can be calculated independently of the height, depth and type of pack that is to perform the process. Thus comparing equation (9.12) with equations (5.20) and (5.22), the Number of Diffusion Units (NDU) within the element of height dy is represented by :

$$NDU = \frac{(t_{Lyin} - t_{Lyout})}{\frac{(h_{Lin} + h_{Lout})}{2} - \frac{(h_{Gin} + h_{Gout})}{2}} \tag{9.13}$$

such that the element's height dy for known values of Ka the tower characteristic and L/X the water flow rate per unit area is given by :

$$dy = (L/KaX)(NDU) \tag{9.14}$$

The NDU equation (9.13) shown above is non-dimensional and hence the group L/KaX has the dimensions of height, to satisfy the equation (9.14), for an element of unit width.

CHAPTER TEN

EVALUATION OF THE NDU INTEGRAL FOR THE CROSSFLOW PROCESS

The method used to analyse the crossflow cooling tower involves the use of equations (9.9) and (9.13) to evaluate the outlet conditions for each elemental volume within a cross-section of packing. The first element to be solved will be that in the top corner of the packing where it is exposed to air at the inlet wet bulb temperature t_{WB}, and water at the inlet temperature t_{L1}. The outlet conditions obtained are then used as the inlet conditions for evaluation of adjacent elements. This procedure results in a step-wise calculation down and across the packing.

In order to evaluate equations (9.9) and (9.13) for a set of operating conditions at inlet to a section of crossflow packing, a method similar to that discussed in Chapter Two, Section 2.4 is employed. In the first instance this involves the use of a guessed value for the outlet water temperature t_{Lo} from the element. This value is then used to find the leaving air enthalpy h_{Go} by using the equation (9.9). With all the known and calculated values substituted into equation (9.13) the NDU value equivalent to the nominated outlet temperature can be obtained. This NDU value is compared with the nominated NDU value for the element and the temperature drop adjusted to achieve better agreement between the calculated and nominated NDU values. Successive adjustments to the estimate of the temperature drop are made using the following relationship :

$$(t_{Li} - t_{Lo})_{\text{new value}} = (1+(NDU - NDUC)/NDU)(t_{Li} - t_{Lo})_{\substack{\text{previous}\\ \text{value}}} \tag{10.1}$$

for the nominated value NDU, and the calculated value NDUC. The equation (10.1) gives a new value for the outlet water temperature t_{Lo}. A satisfactory exit water temperature is obtained when the ratio :

$$(NDU - NDUC)/NDU < 0.01 \qquad (10.2)$$

is satisfied. The value 0.01 for termination of the iteration was selected after investigation using the ratio values 0.005, 0.01, 0.05 and 0.1 to give low computer usage consistent with one decimal place accuracy, considered adequate for commercial application. A list of results is shown on Table 10.1.

For the purpose of evaluation, the cross section of packing is considered to be a box containing 64 elements as shown on Fig 10.1. Element No. 1 is the first to be solved since all the conditions at inlet to it are known, namely t_{WB}, $t_{L1(in)}$, L/X, G/Y and the nominated NDU value. The calculated outlet conditions are used to solve elements No.2 and No.9. This enables successive evaluation down and across the box to be performed. The overall outlet water temperature conditions t_{L2} for various box arrangements are calculated as shown below. For a box made up of elements No. 1 to 16 inclusive and identified as 1Hx4V on Fig 10.2, the mean outlet water temperature is :

$$T.OFF1 = t_{L2} = ((t_{Lo})_8 + (t_{Lo})_{16}) /2 \qquad (10.1)$$

with the outlet water temperatures identified by the appropriate element number. For the pack arrangement 2Hx4V as shown on Fig 10.2 which includes elements No. 1 to No.32 inclusive, the mean outlet water temperature from the pack section is given by :

$$T.OFF2 = t_{L2} = ((t_{Lo})_8 + (t_{Lo})_{16} + (t_{Lo})_{24} + (t_{Lo})_{32})/4 \qquad (10.2)$$

For the pack arrangement 3Hx4V as shown on Fig 10.2 which includes elements No. 1 to No.48 inclusive, the mean outlet water temperature from the pack section is given by :

$$T.OFF3 = t_{L2} = ((t_{Lo})_8+(t_{Lo})_{16}+(t_{Lo})_{24}+(t_{Lo})_{32} + (t_{Lo})_{40}+(t_{Lo})_{48}) /6 \qquad (10.3)$$

Cooling Duty: $L/G = 0.5$, $t_{WB} = 65^{o}F$, $t_{L1} = 80^{o}F$ $P = 1000mb$ $NDU_N = 0.05$.				
	Water temperature at outlet, t_{L2} ^{o}F			
Ratio.	1Hx4V	2Hx4V	3Hx4V	4Hx4V
0.100	69.47	69.96	70.41	70.83
0.050	69.37	69.85	70.31	70.75
0.010	69.37	69.83	70.27	70.69
0.005	69.36	69.83	70.27	70.69

Where Ratio = $(NDU_N - NDU_C)/NDU_N$, suffix N refers to the nominated NDU value, suffix C refers to the calculated NDU value.

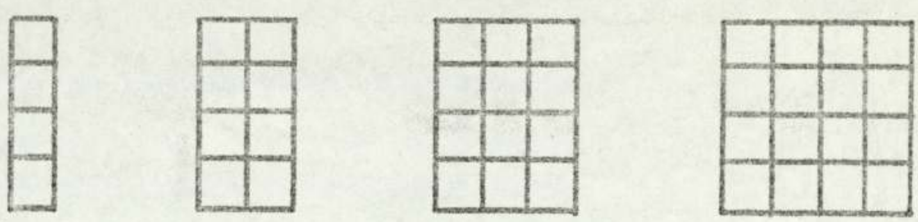
TABLE SHOWING THE EFFECT OF NDU AGREEMENT ON THE OUTLET WATER TEMPERATURE

Table 10.1.

1	9	17	25	33	41	49	57
2	10	18	26	34	42	50	58
3	11	19	27	35	43	51	59
4	12	20	28	36	44	52	60
5	13	21	29	37	45	53	61
6	14	22	30	38	46	54	62
7	15	23	31	39	47	55	63
8	16	24	32	40	48	56	64

CROSS SECTION OF PACKING SHOWING IT DIVIDED INTO SIXTY FOUR ELEMENTS

Fig 10.1



1H x 4V   2H x 4V   3H x 4V   4H x 4V

CROSSFLOW PACK ARRANGEMENTS

Fig 10.2

For the pack arrangement 4Hx4V as shown on Fig 10.2 which includes elements No. 1 to No.64 inclusive, the mean outlet water temperature from the pack section is given by :

$$T.OFF4 = t_{L2} = ((t_{Lo})_8 + (t_{Lo})_{16} + (t_{Lo})_{24} + (t_{Lo})_{32} + (t_{Lo})_{40} + (t_{Lo})_{48} + (t_{Lo})_{56} + (t_{Lo})_{64})/8 \qquad (10.4)$$

CHAPTER ELEVEN

COMPUTER PROGRAM FOR THE CROSSFLOW PROCESS

11.1 Description

From the method described in the previous Chapter the evaluation of the crossflow cooling tower requires a solution for all the elements in the pack, each element solved by a series of iterations. Since 64 elements have to be computed for the analysis of the pack at any one set of conditions, it can be seen that such an evaluation is long and tedious.

To enable a computer program to be written a Flow Diagram for the sequence of operations must be prepared. Such a diagram for the crossflow cooling tower is shown on Fig 11.1. To keep the Diagram relatively simple the element counting routine is not shown. In detail, this routine consists of three parts, considered in the following order, and as shown on Fig 11.2.

a) Count for the vertical elements 1 to 8 inclusive.

b) Count for the horizontal elements 9, 17, 25, 33, 41, 49 and 57.

c) Count for the remaining elements, working down the columns headed by elements 10, 18, 26, 34, 42, 50 and 58.

An elaboration of the program is as follows:

Read the wet bulb temperature TWB (ie t_{WB})

Set the wet bulb temperature, the inlet water temperature, water to air ratio L/G and the nominated NDU value.

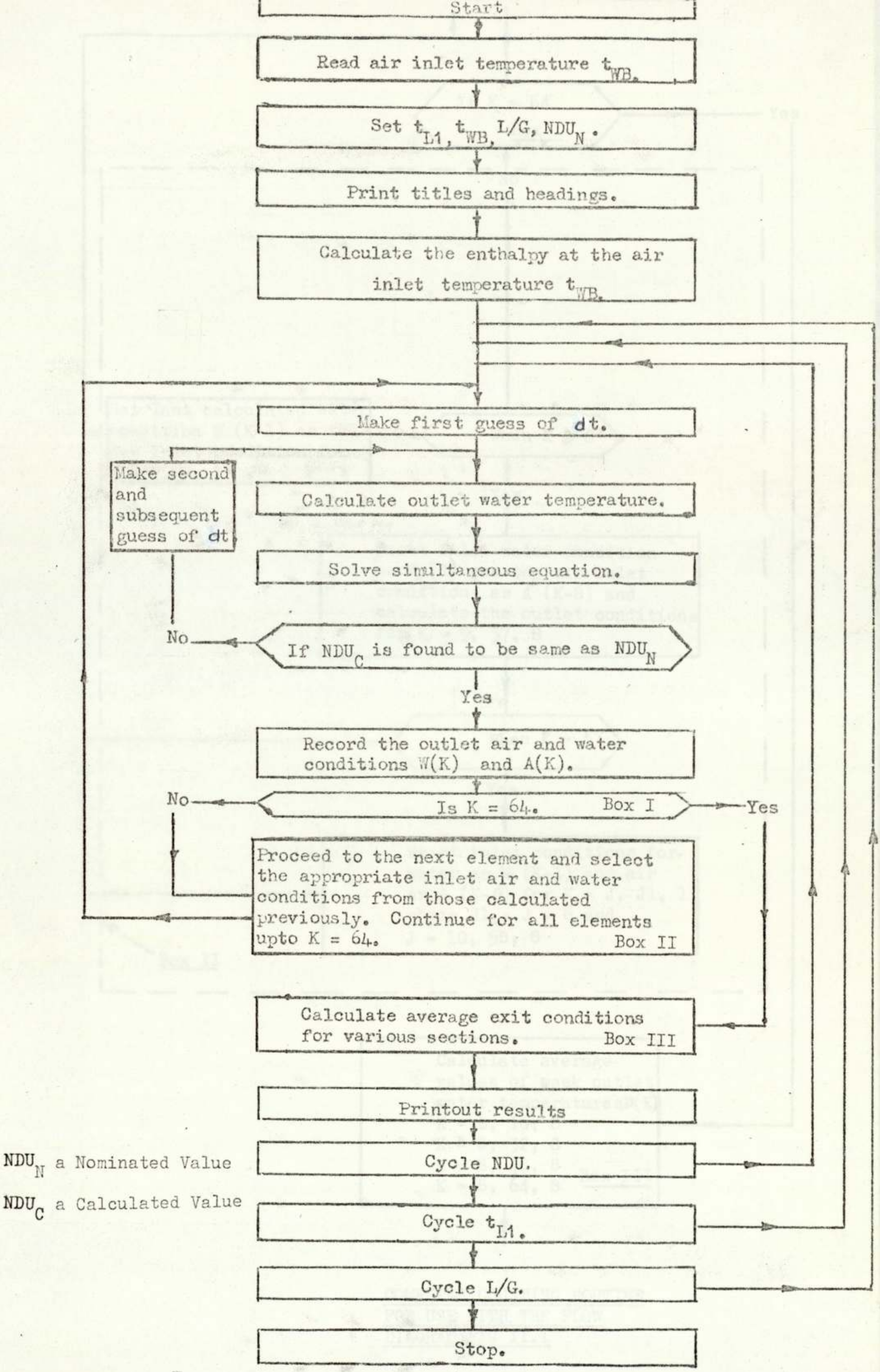
Print the program headings.

Calculate the air inlet enthalpy from the subroutine TTOH.

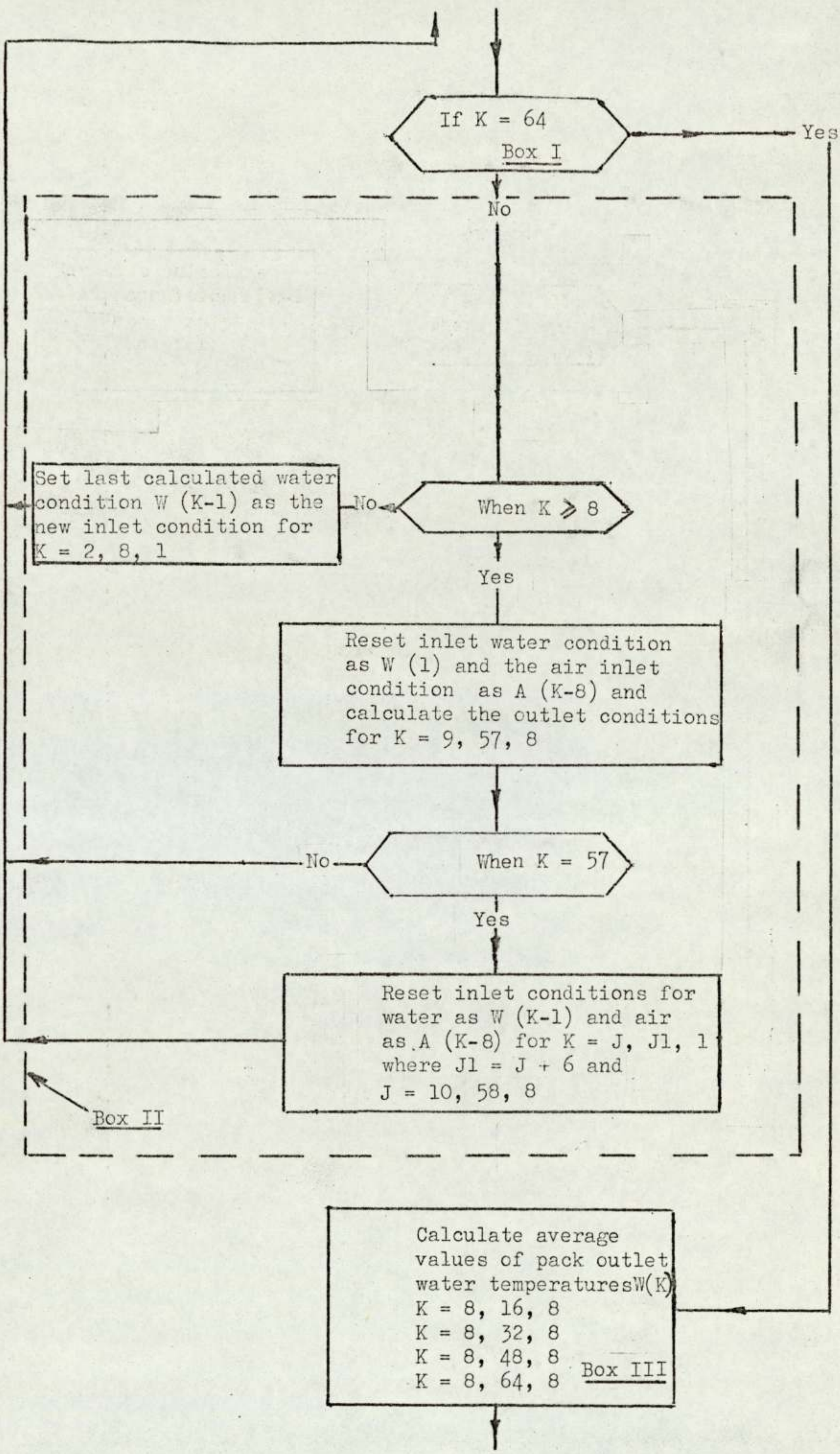
Enter the subroutine and carry out the following instructions.

Make first guess of outlet water temperature and calculate its equivalent enthalpy HWOFF using subroutine TTOH.

From the energy balance equation (9.9) find the air condition HAOFF at outlet from the element.



FLOW DIAGRAM FOR THE CROSSFLOW COOLING TOWER   Fig.11.1



COMPUTER COUNTING ROUTINE FOR USE WITH THE FLOW DIAGRAM FIG 11.1

Fig 11.2

Use the known and calculated conditions to evaluate the NDU value shown by equation (9.13).

Compare the evaluated NDU (DUNC) with the nominated NDU value (DUN), adjust the outlet water temperature to suit, and calculate its equivalent enthalpy HWOFF using subroutine TTOH.

Repeat the above instructions until NDU agreement is achieved, whereupon the appropriate outlet air and water conditions are stored.

Leave the subroutine SOLUTION.

Proceed through the cycle instructions to find the outlet air and water conditions for each of the sixty four elements.

Calculate the average outlet water conditions at the bottom of the pack T.OFF1, T.OFF2, T.OFF3 and T.OFF4.

Print the average water outlet temperatures T.OFF1, T.OFF2, T.OFF3 and T.OFF4.

Cycle to the next nominated NDU value.

Cycle to the next inlet water temperature TWON

Cycle L/G

Stop.

Note. The L/G ratio used in crossflow is obtained from $(L/A)/(G/A')$.

11.2 Computer Language

The Fortran language was used for writing the program represented by the Flow Diagram shown on Fig 11.1. It consists of four parts, the introduction, the main program 'Master Crossflow' and two subroutines, one for the calculation of the exit conditions from each of the elements, called SOLUTION and the other called TTOH for calculating the enthalpy of saturated air using the fundamental equation (3.15).

11.3 Example of the Fortran program.

Before the main program was finalised a subsidiary program was required to give a printout of the outlet conditions from each of the 64 elements, for one set of inlet conditions. The program consists of four parts, namely an introduction, a subsidiary program and two subroutines all of which are shown on Fig 11.3. The subsidiary program was arranged to give a printout of all the exit water temperatures and air enthalpies leaving each element, see Fig 11.4. The results obtained were plotted on a grid, see Fig 11.5, to illustrate the outlet conditions from each of the sixty four elements.

The program operation was tested by carrying out a heat balance check on selected elements taken from the results shown on Fig 11.4. They were found to give heat balance agreement to better than 0.5%.

The main program, see Fig 11.6 sheet 1, consists of the subsidiary program suitably modified to incorporate three cycle procedures. The cycles provide a series of operating conditions based on different inlet water temperatures t_{L1}, liquid to gas ratios L/G and nominated NDU values. To accommodate the main program, the subroutine Solution required modification to that shown on sheet 2 of Fig 11.6.

11.4 Results

Fig 11.7 shows a typical printout of the results obtained from the main program mentioned above.

11.5 Computer Operating Time

The computer required 105 seconds to produce the 112 results shown on Fig 11.7 which is a typical computer printout for this program.

INTRODUCTION

```
LIST (LP)$                                  CRO  100
MAP                                         CRO  200
PROGRAM(CROSSFLOWCT)                        CRO  300
INPUT 1=CRO                                 CRO  400
OUTPUT 2=LPO                                CRO  500
TRACE 2                                     CRO  600
END                                         CRO  700
```

TITLES, HEADINGS AND ARITHMETIC INSTRUCTIONS

(SUBSIDIARY PROGRAM)

```
      MASTER CROSSFLOW                          CRO  800
      COMMON E,ATM,W(64),A(64)                  CRO  900
      ATM=1000                                  CRO 1000
      TWB=65                                    CRO 1100
   11 FORMAT(13H PROG 130(01))                  CRO 1200
      WRITE(2,11)                               CRO 1300
   12 FORMAT(6H T.WB=,F5.1,5H ATM=,F7.1)        CRO 1400
      WRITE(2,12)TWB,ATM                        CRO 1500
      CALLTTOH(TWB)                             CRO 1900
      HAWB=E                                    CRO 2000
      R=0.5                                     CRO 2102
      V1=80                                     CRO 2302
      TUN=0.15                                  CRO 2502
      WON=V1                                    CRO 2600
      K=1                                       CRO 2700
      V21=HAWB                                  CRO 2800
      CALL SOLUTION(WON,TUN,V21,R,K)            CRO 2900
      DO 22 K=2,8,1                             CRO 3000
      WON=W(K-1)                                CRO 3100
      CALL SOLUTION(WON,TUN,V21,R,K)            CRO 3200
   22 CONTINUE                                  CRO 3300
      DO 23 K=9,57,8                            CRO 3400
      WON=V1                                    CRO 3500
      V21=A(K-8)                                CRO 3600
      CALL SOLUTION(WON,TUN,V21,R,K)            CRO 3700
   23 CONTINUE                                  CRO 3800
      DO 24 J=10,58,8                           CRO 3900
      J1=J+6                                    CRO 4000
      DO 25 K=J,J1,1                            CRO 4100
      WON=W(K-1)                                CRO 4200
      V21=A(K-8)                                CRO 4300
      CALL SOLUTION(WON,TUN,V21,R,K)            CRO 4400
   25 CONTINUE                                  CRO 5100
   24 CONTINUE                                  CRO 5200
      STOP                                      CRO 5600
      END                                       CRO 5700
```

CROSSFLOW COMPUTER PROGRAM (SUBSIDIARY) - FORTRAN

Fig 11.3

Continued

ENTHALPY TEMPERATURE SUBROUTINE TTOH

```
      SUBROUTINE TTOH(T)                                        CRO 5800
      COMMON E,ATM,W(64),A(64)                                  CRO 5900
      P=EXP(1.8091+9.594*(T-32)/(23(.3+0.5555*(T-32)))          CRO 6000
      G3=4354*P/(ATM-P)                                         CRO 6100
      G1=0.45*G3/7000                                           CRO 6200
      G2=1075*G3/7000                                           CRO 6300
      E=(0.241+G1)*(T-32)+G2                                    CRO 6400
      RETURN                                                    CRO 6500
      STOP                                                      CRO 6600
      END                                                       CRO 6700
```

SUBROUTINE SOLUTION (SUBSIDIARY PROGRAM)

```
      SUBROUTINE SOLUTION(WON,TUN,V21,R,K)                      CRO 6800
      COMMON E,ATM,W(64),A(64)                                  CRO 6900
      DELTAT=2                                                  CRO 7000
      WOFF=WON-DELTAT                                           CRO 7100
   50 CALLTTOH(WOFF)                                            CRO 7200
      HWOFF=E                                                   CRO 7300
      HAOFF=V21+R*(WON-WOFF)                                    CRO 7400
      CALLTTOH(WON)                                             CRO 7500
      HWON=E                                                    CRO 7600
      TUNC=(WON-WOFF)/((HWON+HWOFF)-(V21+HAOFF))*0.5            CRO 7700
      RATIO=(TUN-TUNC)/TUN                                      CRO 7800
      RAT=ABS(RATIO)                                            CRO 7900
      IF(RAT.LT.0.01)GOTO51                                     CRO 7905
      IF(RAT.LT.1.0)GOTO53                                      CRO 7910
      DELTAT=DELTAT*0.5                                         CRO 7915
      GOTO54                                                    CRO 7920
   53 DELTAT=DELTAT*(1+RATIO)                                   CRO 7925
   54 WOFF=WON-DELTAT                                           CRO 8000
      GOTO50                                                    CRO 8100
   51 W(K)=WOFF                                                 CRO 8200
      A(K)=HAOFF                                                CRO 8300
   52 FORMAT(I5,2(5X,F7.2))                                     CRO 8302
      WRITE(2,52)K,W(K),A(K)                                    CRO 8303
      RETURN                                                    CRO 8400
      STOP                                                      CRO 8500
      END                                                       CRO 8600
```

CROSSFLOW COMPUTER PROGRAM (SUBSIDIARY) - FORTRAN

Fig 11.3

PROG 130(01)
T.WB= 65.0 ATM= 1000.0

Element No	t_{Lo} ^oF	h_{Go} Btu/lb
1	74.31	25.38
2	70.95	24.22
3	68.88	23.57
4	67.54	23.21
5	66.68	22.97
6	66.11	22.82
7	65.73	22.73
8	65.49	22.66
9	75.50	27.63
17	76.44	29.41
25	77.21	30.81
33	77.76	31.93
41	78.22	32.82
49	78.59	33.53
57	78.88	34.09
10	72.39	25.78
11	70.21	24.66
12	68.68	23.97
13	67.60	23.51
14	66.84	23.20
15	66.30	23.00
16	65.92	22.85
18	73.63	27.18
19	71.44	25.76
20	69.80	24.79
21	68.56	24.13
22	67.63	23.67
23	66.93	23.35
24	66.41	23.11
26	74.64	28.47
27	72.52	26.82
28	70.81	25.65
29	69.47	24.80
30	68.42	24.19
31	67.60	23.76
32	66.94	23.43
34	75.50	29.59
35	73.50	27.82
36	71.78	26.51
37	70.36	25.51
38	69.20	24.77
39	68.28	24.22
40	67.54	23.80
42	76.24	30.58
43	74.36	28.76
44	72.67	27.36
45	71.22	26.24
46	70.00	25.38
47	68.98	24.73
48	68.16	24.21
50	76.86	31.44
51	75.10	29.64
52	73.47	28.17
53	72.02	26.96
54	70.75	26.02
55	69.67	25.26
56	68.77	24.66
58	77.39	32.19
59	75.78	30.44
60	74.22	28.95
61	72.78	27.68
62	71.49	26.66
63	70.37	25.83
64	69.41	25.14

PRINTOUT OF CROSSFLOW RESULTS OBTAINED FROM THE FORTRAN COMPUTER PROGRAM (SUBSIDIARY)

Fig 11.4

INLET WATER TEMPERATURE = $80^{\circ}F$

INLET AIR TEMPERATURE = $65^{\circ}FWB$

Overall L/G = 0.5 Nominated NDU = 0.15 PER ELEMENT.

80	80	80	80	80	80	80	80
25.38	27.63	29.41	30.81	31.93	32.82	33.53	34.09
1	9	17	25	33	41	49	57
74.31	75.55	76.44	77.21	77.76	78.22	78.59	78.88
24.32	25.78	27.18	28.47	29.59	30.58	31.44	32.19
2	10	18	26	34	42	56	58
70.95	72.39	73.63	74.64	75.50	76.24	76.86	77.39
23.57	24.66	25.76	26.82	27.82	28.75	29.64	30.44
3	11	19	27	35	43	51	59
68.88	70.21	71.44	72.52	73.50	74.36	75.10	75.78
23.21	23.97	24.79	25.65	26.51.	27.36	28.17	28.95
4	12	20	28	36	44	52	60
67.54	68.68	69.80	70.81	71.78	72.76	73.47	74.22
22.97	23.51	24.13	24.80	25.51	26.24	26.96	27.68
5	13	21	29	37	45	53	61
66.68	67.6	68.56	69.47	70.36	71.22	72.02	72.78
27.82	23.20	23.67	24.19	24.77	25.38	26.02	26.66
6	14	22	30	38	46	54	62
66.11	66.84	67.63	68.42	69.20	70.00	70.75	71.49
22.73	23.0	23.35	23.76	24.22	24.73	25.26	25.83
7	15	23	31	39	47	55	63
65.73	66.3	66.93	67.60	68.28	68.98	69.67	70.37
22.66	22.85	23.11	23.43	23.80	24.21	24.66	25.14
8	16	24	32	40	48	56	64
65.49	65.92	66.41	66.96	67.54	68.16	68.77	69.41

t_{L2} = T OFF 1 = $65.7^{\circ}F$

t_{L2} = T OFF 2 = $66.19^{\circ}F$

t_{L2} = T OFF3= $66.75^{\circ}F$

t_{L2} = T OFF 4 = $67.33^{\circ}F$

Results taken from Fig 11.4

GRID SHOWING THE EXIT CONDITIONS FOR AIR AND WATER LEAVING EACH ELEMENT. Fig 11.5

TITLES, HEADINGS AND ARITHMETIC

INSTRUCTIONS (MAIN PROGRAM)

```
      MASTER CROSSFLOW                                             CRO  800
      COMMON E,ATM,W(64),A(64)                                     CRO  900
      ATM=1000                                                     CRO 1000
      TWB=65                                                       CRO 1100
   11 FORMAT(13H PROG 130(01))                                     CRO 1200
      WRITE(2,11)                                                  CRO 1300
   12 FORMAT(6H T.WB=,F5.1,5H ATM=,F7.1)                           CRO 1400
      WRITE(2,12)TWB,ATM                                           CRO 1500
   13 FORMAT(57H     L/G     TOH      NDU     TOFF1      TOFF2     CRO 1600
     14)TOFF3     TOFF4                                            CRO 1700
      WRITE(2,13)                                                  CRO 1800
      CALLTTOH(TWB)                                                CRO 1900
      HAWB=E                                                       CRO 2000
      DO 19 LLG=5,20,5                                             CRO 2101
      R=LLG/10.0                                                   CRO 2200
      DO 20 NV1=80,140,10                                          CRO 2300
      V1=NV1                                                       CRO 2301
      TUN=0.0065                                                   CRO 2509
      WON=V1                                                       CRO 2600
      K=1                                                          CRO 2700
      V21=HAWB                                                     CRO 2800
      CALL SOLUTION(WON,TUN,V21,R,K)                               CRO 2900
      DO 22 K=2,8,1                                                CRO 3000
      WON=W(K-1)                                                   CRO 3100
      CALL SOLUTION(WON,TUN,V21,R,K)                               CRO 3200
   22 CONTINUE                                                     CRO 3300
      DO 23 K=9,57,8                                               CRO 3400
      WON=V1                                                       CRO 3500
      V21=A(K-8)                                                   CRO 3600
      CALL SOLUTION(WON,TUN,V21,R,K)                               CRO 3700
   23 CONTINUE                                                     CRO 3800
      DO 24 J=10,58,8                                              CRO 3900
      J1=J+6                                                       CRO 4000
      DO 25 K=J,J1,1                                               CRO 4100
      WON=W(K-1)                                                   CRO 4200
      V21=A(K-8)                                                   CRO 4300
      CALL SOLUTION(WON,TUN,V21,R,K)                               CRO 4400
   25 CONTINUE                                                     CRO 4401
   24 CONTINUE                                                     CRO 4402
      TOFF1=(W(8)+W(16))/2                                         CRO 4500
      TOFF2=(2*TOFF1+W(24)+W(32))/4                                CRO 4600
      TOFF3=(4*TOFF2+W(40)+W(48))/6                                CRO 4700
      TOFF4=(6*TOFF3+W(56)+W(64))/8                                CRO 4800
   14 FORMAT(F4.1,F6.1,F6.3,4F7.2)                                 CRO 4900
      WRITE(2,14)R,V1,TUN,TOFF1,TOFF2,TOFF3,TOFF4                  CRO 5000
   20 CONTINUE                                                     CRO 5400
   19 CONTINUE                                                     CRO 5500
      STOP                                                         CRO 5600
      END                                                          CRO 5700
```

CROSSFLOW COMPUTER PROGRAM (MAIN) - FORTRAN

Fig 11.6

Continued

SUBROUTINE SOLUTION (MAIN PROGRAM)

```
      SUBROUTINE SOLUTION(WON,TUN,V21,R,K)                      CRO 6800
      COMMON E,ATH,W(64),A(64)                                  CRO 6900
      DELTAT=2                                                  CRO 7000
      WOFF=WON-DELTAT                                           CRO 7100
   50 CALLTTOH(WOFF)                                            CRO 7200
      HWOFF=E                                                   CRO 7300
      HAOFF=V21+R*(WON-WOFF)                                    CRO 7400
      CALLTTOH(WON)                                             CRO 7500
      HWON=E                                                    CRO 7600
      TUNC=(WON-WOFF)/((HWON+HWOFF)-(V21+HAOFF))*0.5            CRO 7700
      RATIO=(TUN-TUNC)/TUN                                      CRO 7800
      RAT=ABS(RATIO)                                            CRO 7900
      IF(RAT.LT.0.01)GOTO51                                     CRO 7905
      IF(RAT.LT.1.0)GOTO53                                      CRO 7910
      DELTAT=DELTAT*0.5                                         CRO 7915
      GOTO54                                                    CRO 7920
   53 DELTAT=DELTAT*(1+RATIO)                                   CRO 7925
   54 WOFF=WON-DELTAT                                           CRO 8000
      GOTO50                                                    CRO 8100
   51 W(K)=WOFF                                                 CRO 8200
      A(K)=HAOFF                                                CRO 8300
      RETURN                                                    CRO 8400
      STOP                                                      CRO 8500
      END                                                       CRO 8600
```

CROSSFLOW COMPUTER PROGRAM (MAIN)- FORTRAN

Fig 11.6

PROG 130(01)
T.WB= 65.0 ATM= 1000.0

L/G	TON	NDU	TOFF1	TOFF2	TOFF3	TOFF4
0.5	80.0	0.0065	77.47	77.50	77.53	77.56
0.5	90.0	0.0065	85.33	85.38	85.43	85.48
0.5	100.0	0.0065	92.73	92.81	92.89	92.97
0.5	110.0	0.0065	99.59	99.70	99.81	99.01
0.5	120.0	0.0065	105.85	105.99	106.12	106.26
0.5	130.0	0.0065	111.42	111.58	111.75	111.92
0.5	140.0	0.0065	116.23	116.43	116.63	116.84
1.0	80.0	0.0065	77.50	77.56	77.61	77.67
1.0	90.0	0.0065	85.38	85.48	85.58	85.68
1.0	100.0	0.0065	92.81	92.96	93.11	93.26
1.0	110.0	0.0065	99.70	99.91	100.12	100.32
1.0	120.0	0.0065	105.99	106.26	106.53	106.79
1.0	130.0	0.0065	111.58	111.92	112.25	112.57
1.0	140.0	0.0065	116.42	116.83	117.23	117.62
1.5	80.0	0.0065	77.53	77.61	77.69	77.77
1.5	90.0	0.0065	85.44	85.58	85.73	85.86
1.5	100.0	0.0065	92.88	93.11	93.33	93.54
1.5	110.0	0.0065	99.81	100.12	100.42	100.70
1.5	120.0	0.0065	106.11	106.52	106.91	107.28
1.5	130.0	0.0065	111.76	112.25	112.74	113.20
1.5	140.0	0.0065	116.63	117.23	117.81	118.36
2.0	80.0	0.0065	77.56	77.67	77.77	77.86
2.0	90.0	0.0065	85.46	85.68	85.86	86.04
2.0	100.0	0.0065	92.96	93.25	93.53	93.80
2.0	110.0	0.0065	99.92	100.32	100.70	101.07
2.0	120.0	0.0065	106.25	106.78	107.28	107.76
2.0	130.0	0.0065	111.91	112.57	113.19	113.78
2.0	140.0	0.0065	116.83	117.61	118.36	119.07

TYPICAL RESULTS FROM THE FORTRAN COMPUTER PROGRAM (MAIN)

Fig. 11.7

CHAPTER TWELVE

PRESENTATION OF PERFORMANCE DATA FOR THE CROSSFLOW COOLING TOWER

12.1 Introduction

The technique should be easy to use and consist of a minimum number of performance charts compatible with the required accuracy.

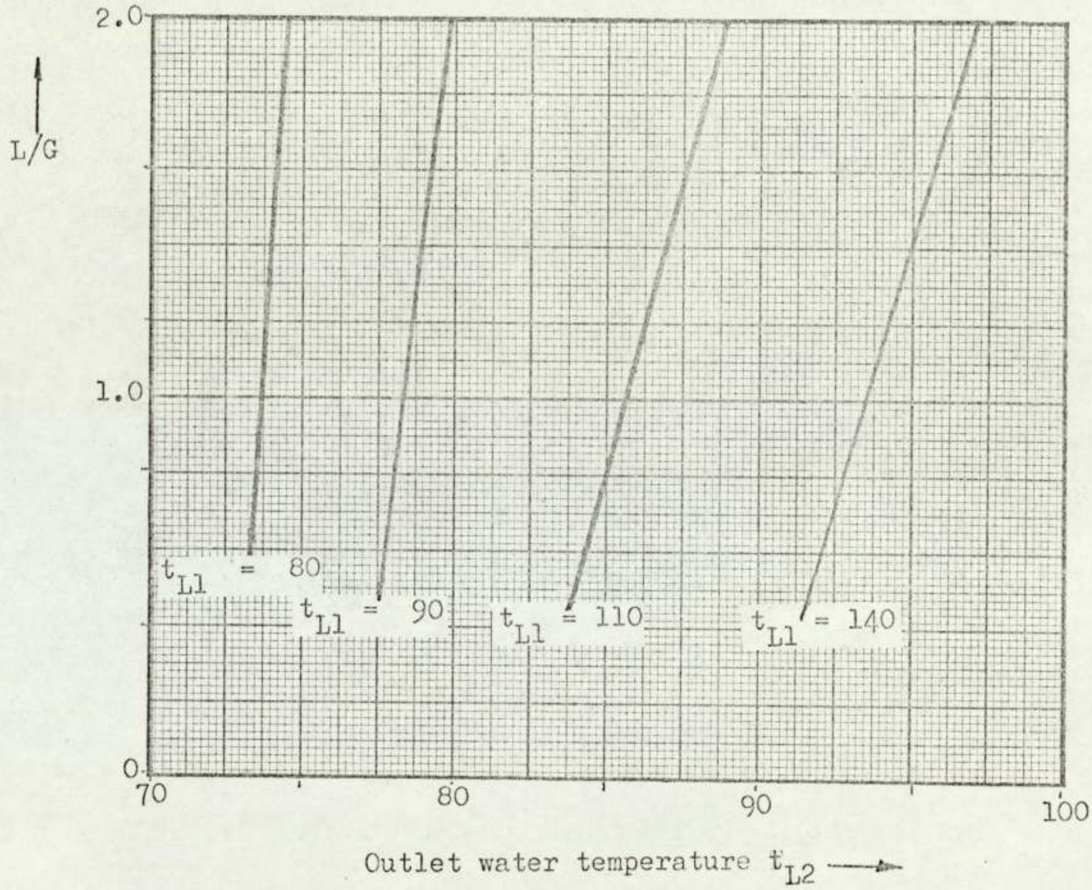
The variables to be accommodated by the method are as outlined in Chapter Two. To achieve a presentation similar to the counterflow one requires the wet bulb temperature and the atmospheric pressure to be kept constant leaving four variables, namely t_{L1}, t_{L2}, L/G and NDU(d_yKaX/L) to be accommodated on a chart similar to the Counterflow Performance Charts shown on Fig 8.3.

12.2 Preparation of the Crossflow Performance Chart

The computer method used to analyse the crossflow equations to obtain performance data requires a constant inlet water temperature t_{L1} to enable the corresponding outlet water temperature t_{L2} to be calculated for a known NDU value. In counterflow the method of calculating temperature is reversed such that a varying inlet water temperature t_{L1} is calculated for a constant outlet water temperature t_{L2} for a known NDU value.

Tables of results were obtained for individual elements with the NDU values 0.0065, 0.0075, 0.0100, 0.0175, 0.0250, 0.0375, 0.0500 and 0.0650 of which Fig 11.7 is a typical example. The 2Hx4V configuration was chosen for analysis and plotting since it would be of commercial interest.

The first attempt to plot a chart, see Fig 12.1 similar to that for the counterflow performance charts was not successful since initial plots of the L/G ratio versus outlet water temperature t_{L2} for curves of constant inlet water temperature t_{L1} gave nearly vertical straight lines. A plot



Crossflow pack configuration is 2H x 4V

NDU = 0.025 P= 1000 millibars $t_{WB} = 65$

All temperatures in degrees Fahrenheit

Dimensions of element 0.5 ft x 0.5 ft x 1.0 ft wide

NOTE The L/G ratio used in Crossflow is obtained from (L/A)/(G/A')

PRELIMINARY PERFORMANCE CHART

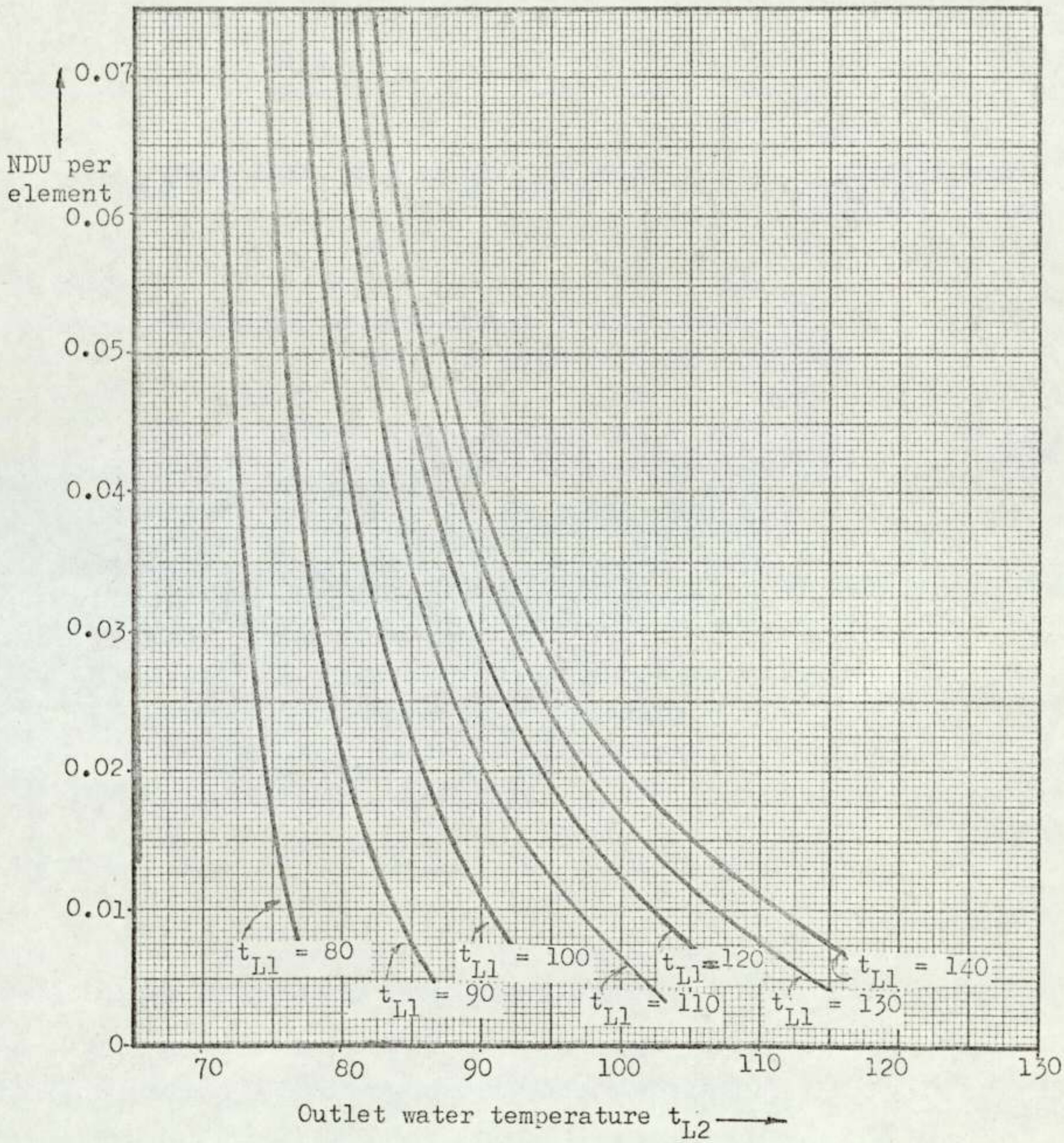
Fig 12.1

of NDU and outlet water temperature t_{L2} gave an acceptable chart, see Fig 12.2 called the Intermediate Crossflow Performance Chart. A better presentation was obtained by interpolation of Fig 12.2 to give a chart of NDU per element versus inlet water temperature t_{L1} for lines of constant outlet water temperature t_{L2}. See Fig 12.3. These Performance Charts give the better presentation, besides being suitable for superimposing lines of constant cooling range Δt. See sheet 3 of Fig 12.3. A similarity between this chart and the counterflow equivalent on Fig 8.3 can be seen. Although the charts have different abscissae Fig 12.3 is perfectly suitable as a Performance Chart for interpolation to obtain the Performance Curve.

Four Intermediate Performance Charts, of which Fig 12.2 is a typical example are necessary for plotting the four (L/G = 0.5, 1.0, 1.5 and 2.0) Performance Charts shown on Fig 12.3. Additional Performance Charts can be obtained for other wet bulb temperatures, pack configuration and atmospheric pressures.

With the conditions already nominated as constant the Performance Diagram is obtained by reading from the Performance Charts the NDU and L/G values satisfying the nominated cooling duty. These results are plotted on the Performance Diagram see Fig 12.4 as a Performance Curve represented by the line ST. The tower characteristic $(KaX/L$ versus L/G is also plotted on the Diagram as the line UV. Intersection of the Performance Curve with the Tower Characteristic gives the Design Duty Point which is the corresponding cooling tower operating condition. Other cooling duties for the same parameters can be plotted on this Diagram to obtain alternative operating conditions.

As in the counterflow method a useful extension to the Performance Diagram can be obtained by using the secondary Graphs II and III to relate cooling duty, circulating waterflow



NOTE The L/G ratio used in Crossflow is obtained from (L/A)/(G/A')

Crossflow pack configuration is 2H x 4V

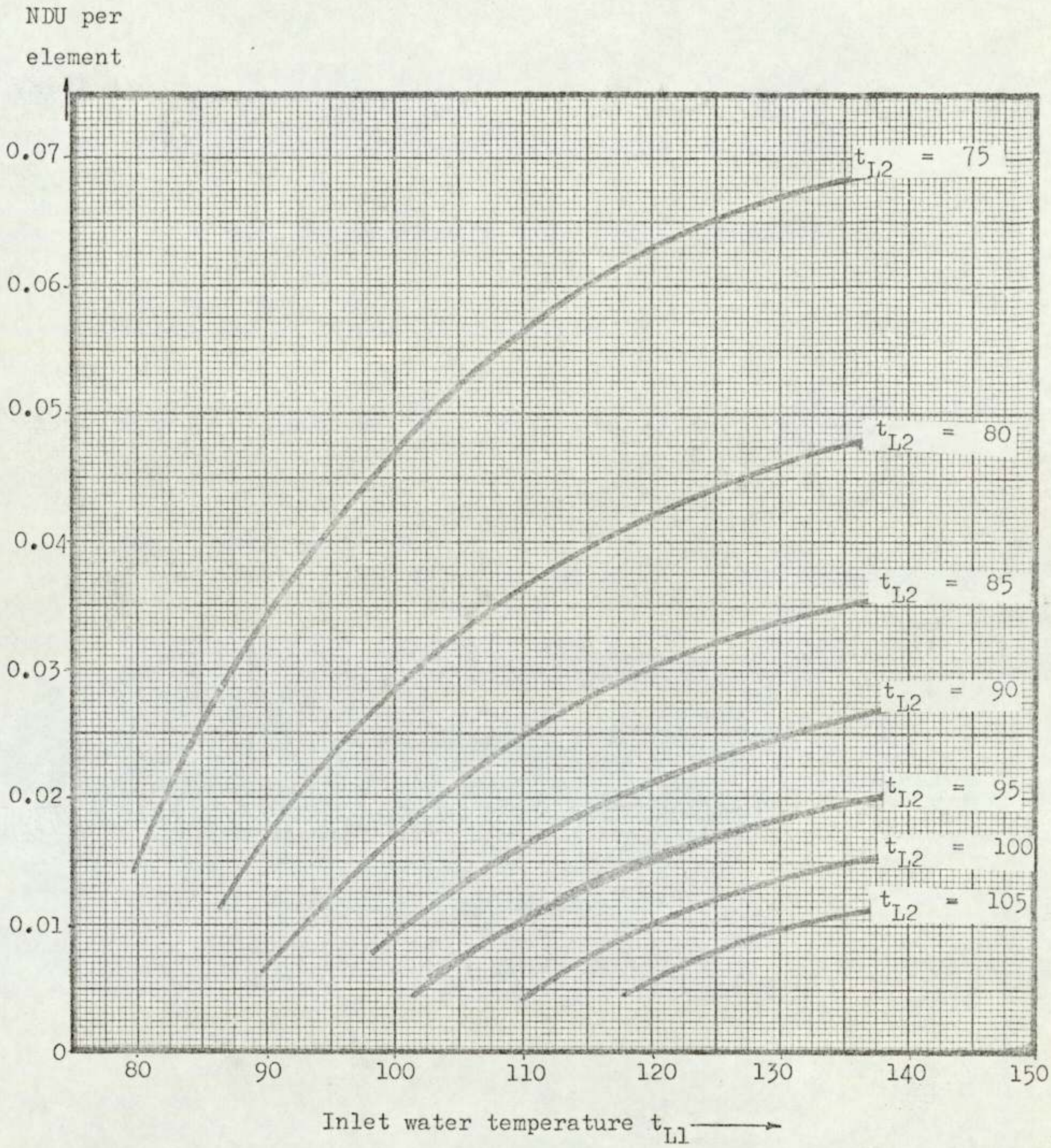
L/G = 2.0 P = 1000 millibars t_{WB} = 65

All temperatures in degrees Fahrenheit

Dimensions of element 0.5 ft x 0.5 ft x 1.0 ft wide

TYPICAL INTERMEDIATE CROSSFLOW PERFORMANCE CHART

Fig 12.2



Pack configuration 2H x 4V

$L/G = 0.5$ $P = 1000$ millibars $t_{WB} = 65^{o}F$

All temperatures in degrees Fahrenheit

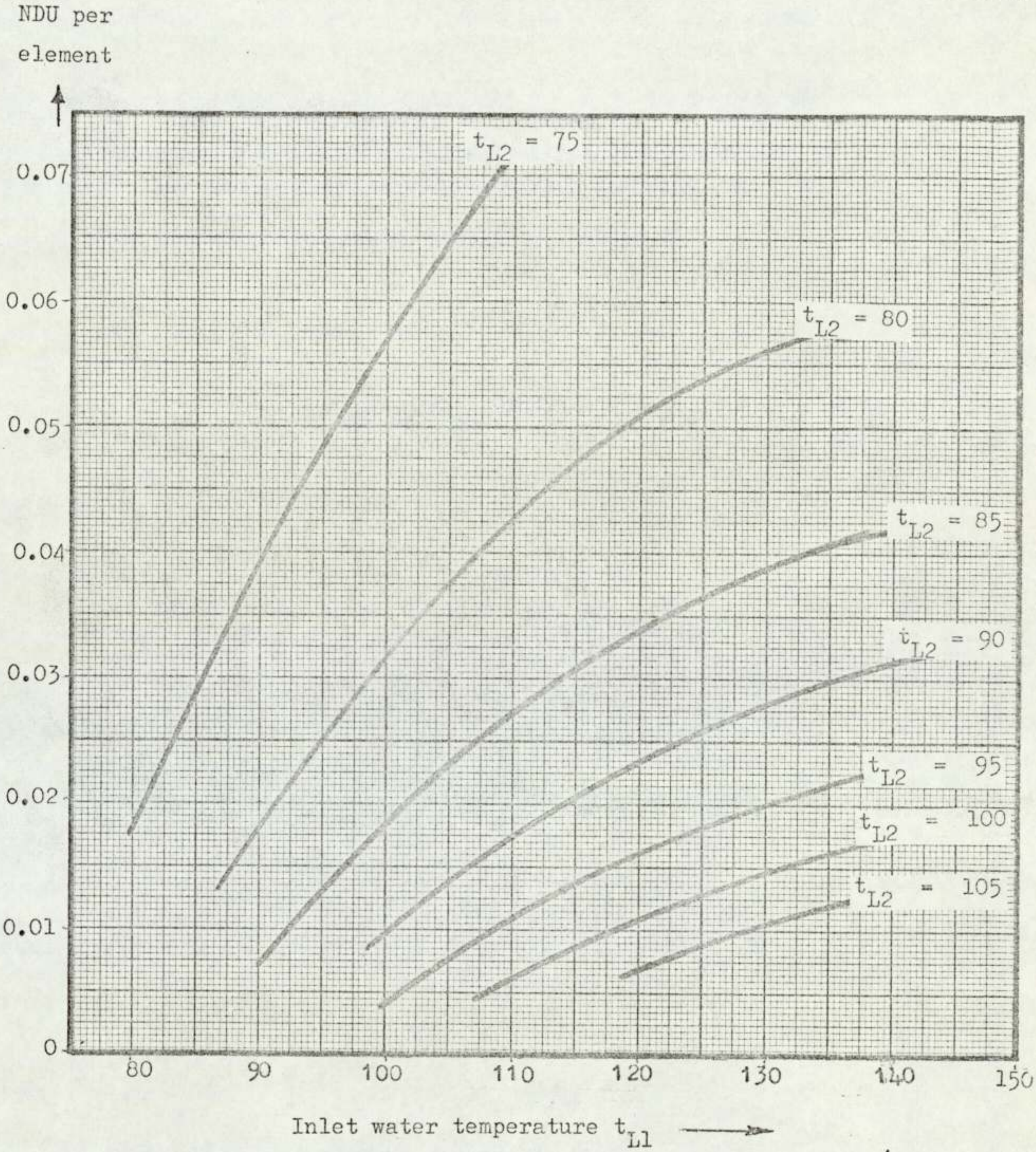
Dimensions of element 0.5 ft x 0.5 ft x 1.0 ft wide

NOTE The L/G ratio used in Crossflow is obtained from $(L/A)/(G/A')$

CROSSFLOW PERFORMANCE CHARTS

Fig 12.3

Continued



NOTE The L/G ratio used in Crossflow is obtained from (L/A)/(G/A')

Pack configuration 2H x 4V

L/G = 1.0 P = 1000 millibars $t_{WB} = 65^{\circ}F$

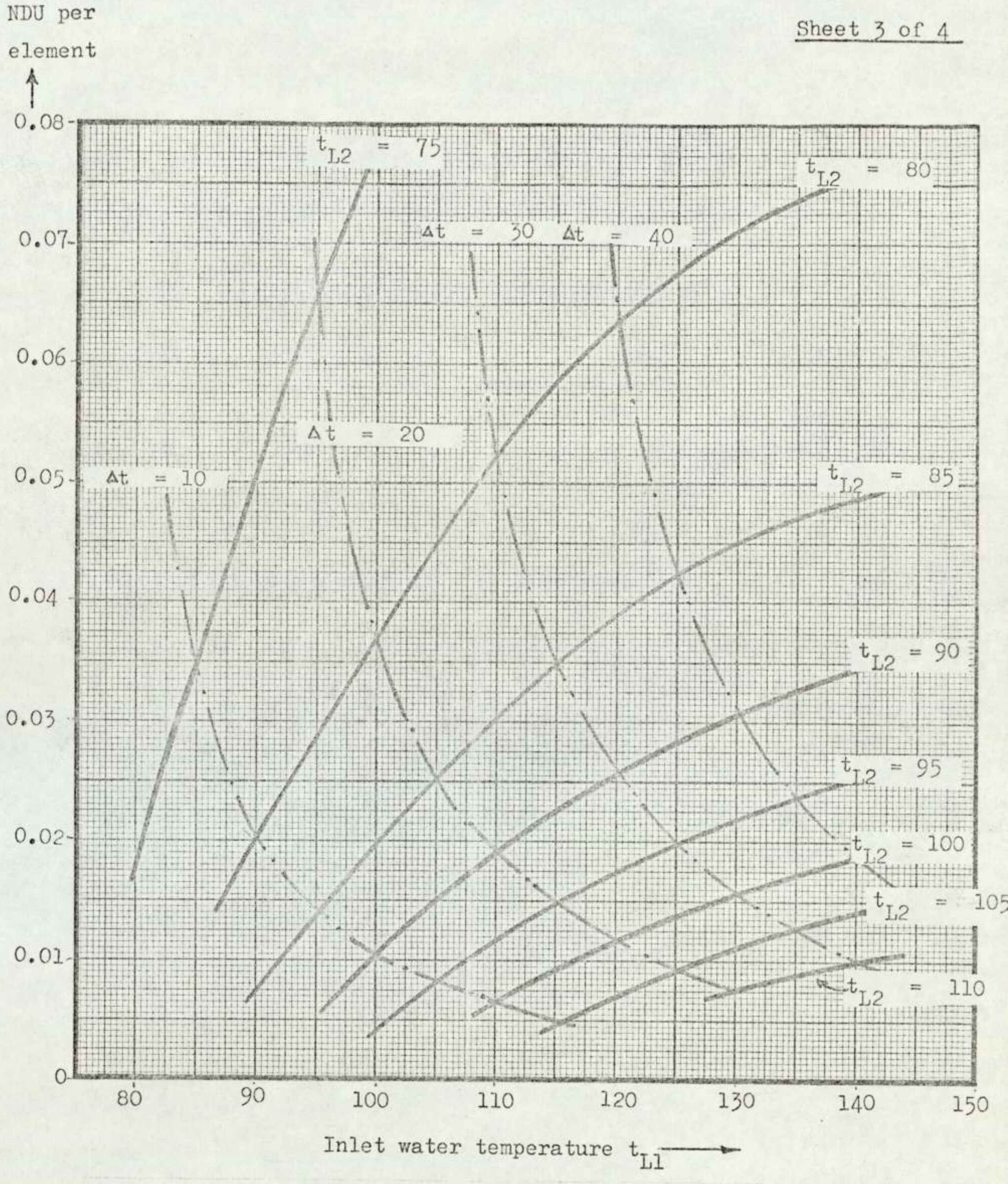
All temperatures in degrees Fahrenheit

Dimensions of element 0.5 ft x 0.5 ft x 1.0 ft wide

CROSSFLOW PERFORMANCE CHARTS

Fig 12.3

 Continued



NOTE The L/G ratio used in Crossflow is obtained from (L/A)/(G/A')

Pack configuration 2H x 4V

L/G = 1.5 P = 1000 millibars $t_{WB} = 65^{o}F$

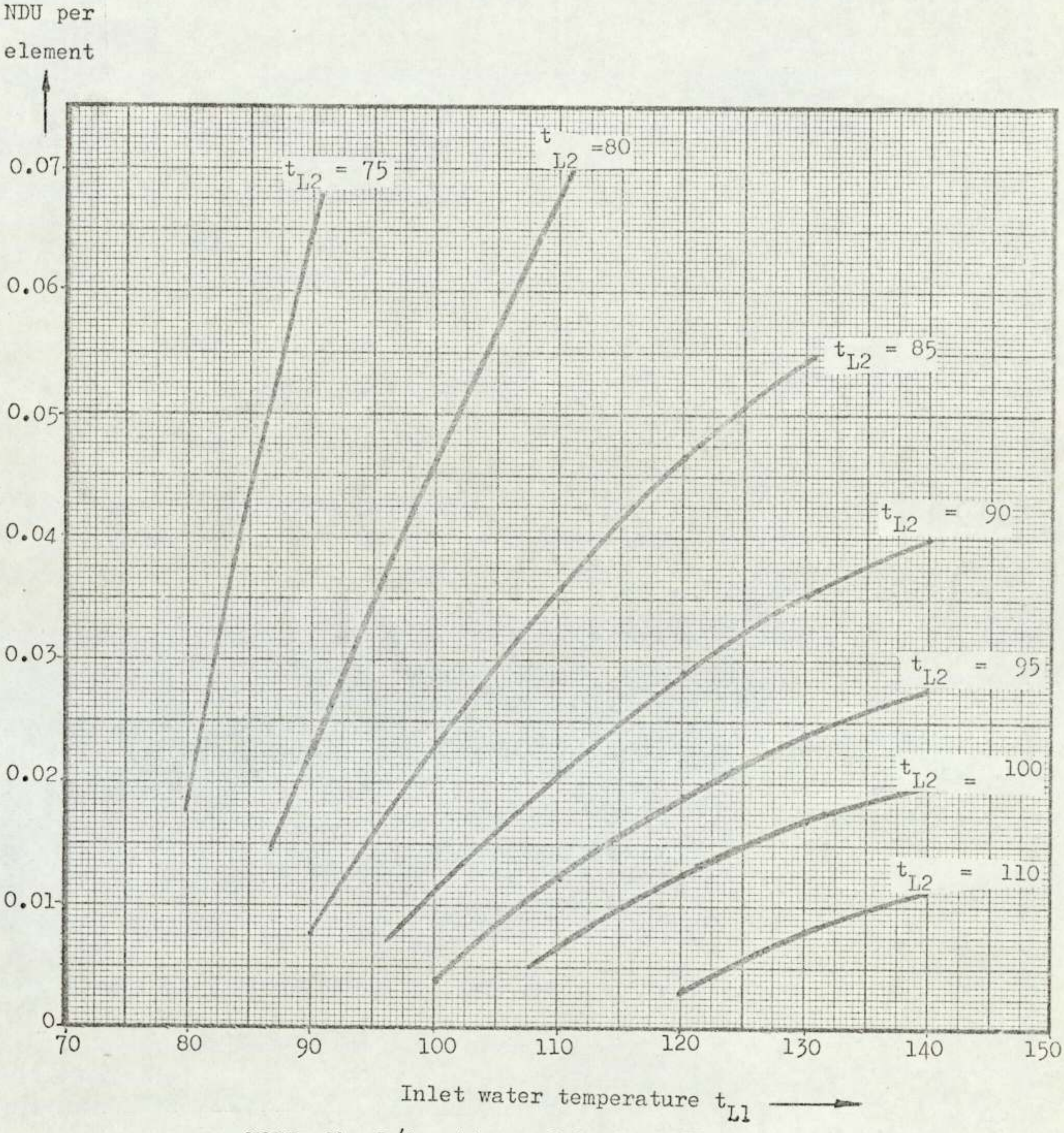
All temperatures in degrees Fahrenheit

Dimensions of element 0.5 ft x 0.5 ft x 1.0 ft wide

CROSSFLOW PERFORMANCE CHARTS

Fig 12.3

Continued



NOTE The L/G ratio used in Crossflow is obtained from (L/A)/(G/A')

Pack configuration 2H x 4V

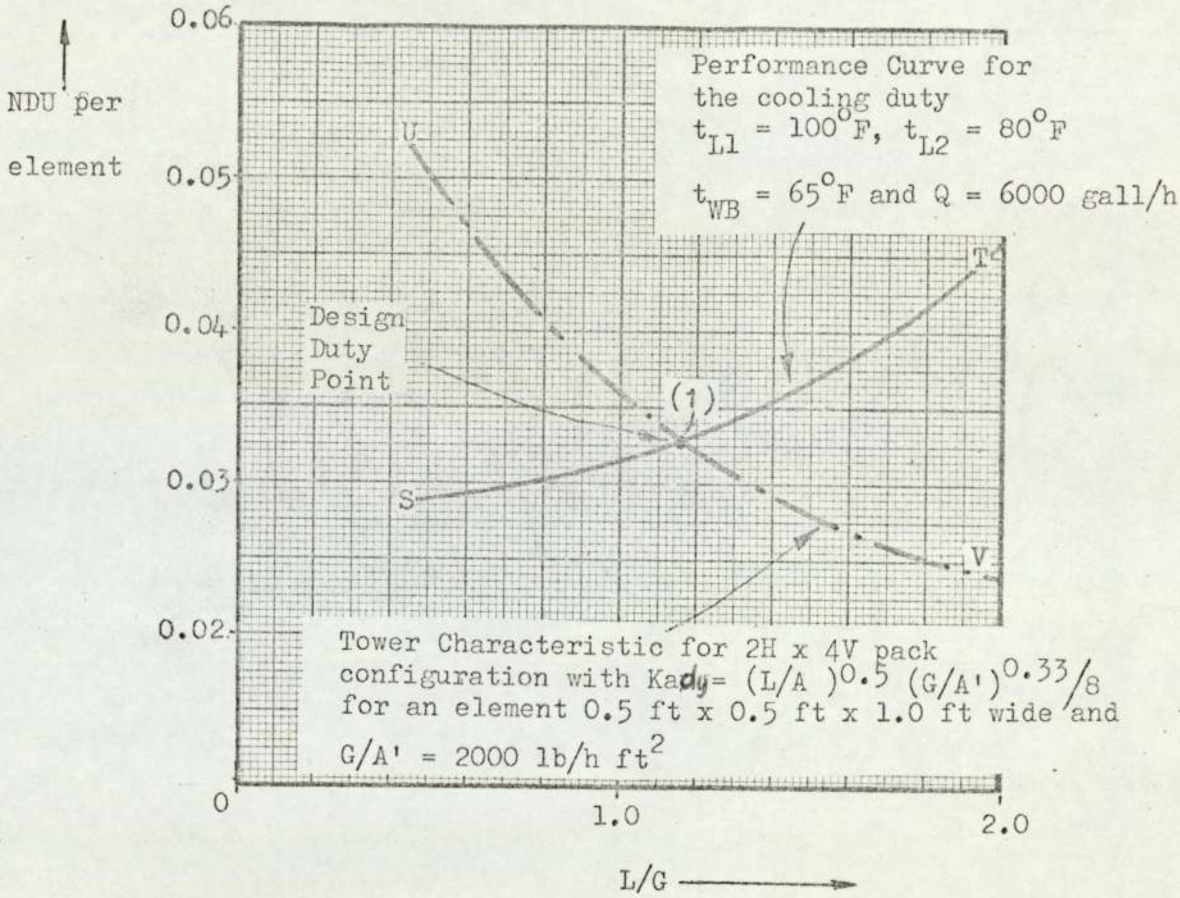
L/G = 2.0 P = 1000 millibar $t_{WB} = 65^{o}F$

All temperatures in degrees Fahrenheit

Dimensions of element 0.5 ft x 0.5 ft x 1.0 ft wide

CROSSFLOW PERFORMANCE CHART

Fig 12.3



NOTE The L/G ratio used in Crossflow is obtained from (L/A)/(G/A')

NOTE NDU = Kady(X/L)

Where A = X × width of pack
and A' = Y × width of pack

CROSSFLOW PERFORMANCE DIAGRAM

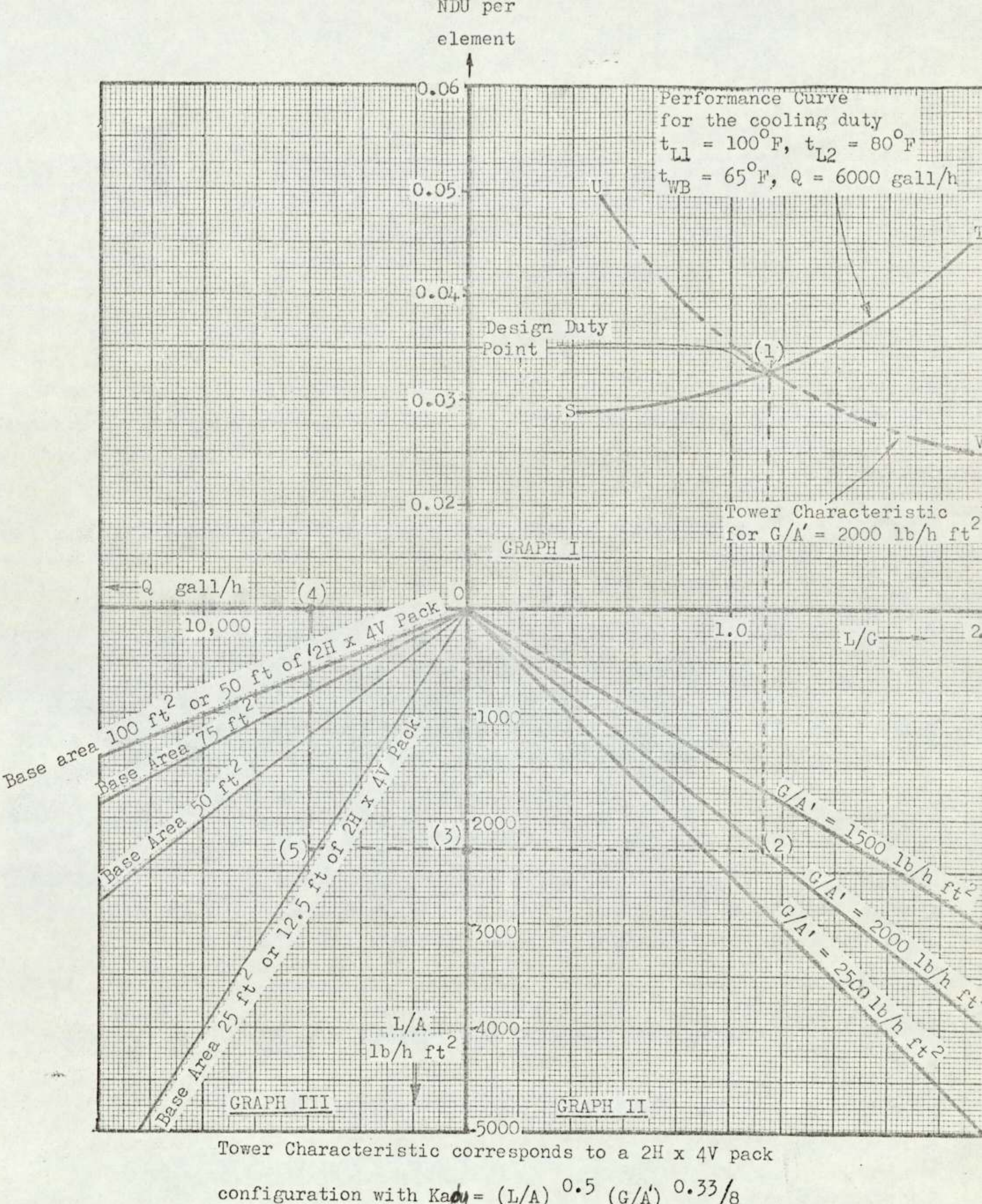
Fig 12.4

rate Q, and the overall L/G value to cooling towers of particular base areas and performance characteristics as shown on Fig 12.5. The base area is a function of the pack vertical cross section which is fixed by the notation 2Hx4V, and so the area of tower will be dependent upon the width of packing (since 2H deep to air flow) required to satisfy the area indicated by the chart.

12.3 Use of the Crossflow Performance Charts

An example using the Expanded Performance Diagram is shown on Fig 12.5. For the duty $t_{L1} = 100^{o}F$, $t_{L2} = 80^{o}F$, $t_{WB} = 65^{o}F$ and Q = 6000 gall/h i.e. L = 60,000 lb/h the performance curve ST is plotted for the appropriate results read from the Performance Charts on Fig 12.3. The tower characteristic UV for the 2Hx4V pack configuration where 2H = 2 ft, 4V = 4 ft is also shown calculated for G/A = 2000 lb/h ft^2. The point (1) is the Design Duty giving the required L/G value which is then converted to an L/A value, point (3) on Graph II. Graph III allows the tower width, point (5) to be established for the known liquid loading Q, point (4) since the depth to air flow is 2H = 2 ft.

This Selection Technique is quite unique for crossflow application. In addition it is identical to the counterflow method. A typical crossflow tower characteristic is analysed in Appendix 12.1.



Tower Characteristic corresponds to a 2H x 4V pack configuration with $Ka\cancel{dy} = (L/A)^{0.5} (G/A')^{0.33}/8$ for an element 0.5 ft x 0.5 ft x 1.0 ft wide and $G/A' = 2000$ lb/h ft^2

NOTE $\quad NDU = Ka\cancel{dy}/(L/\cancel{X})$

Q gall/h = (L lb/h) /10 for water

EXPANDED CROSSFLOW PERFORMANCE DIAGRAM

Fig 12.5

APPENDIX 12.1

EVALUATION OF THE CROSSFLOW TOWER CHARACTERISTIC.

A typical crossflow performance characteristic is represented by an empirical equation expressed as a function of L/A and G/A' giving a relationship of the form :

$$Kady = C\ (L/A)^q\ (G/A')^n \qquad (12.1)$$

where typical values of the constants for a 2Hx4V pack configuration and a unit element with the dimensions 0.5 ft x 0.5 ft are :

$C = 1/8 \quad q = 0.5 \quad n = 0.33$

With $G/A' = 2000$ lb/h ft^2 and using L/G values in the range 0.5 to 2.0 enables equation (12.1) to be solved to give a selection of Kady values. These results are plotted on the Performance Diagram to give the tower characteristic UV. Characteristics for other G/A' values can be plotted on the Diagram to give alternative Duty Points.

NOTE: Area A' = y x Width of packing exposed to air flow (where y = 4V)

= Vertical area of cooling tower in direction of horizontal air flow.

Area A = 2H x Width of packing exposed to air flow

= Base area of cooling tower

The overall L/G ratio used in crossflow is obtained from $(L/A)/(G/A')$

In a counterflow process air and water conditions vary in one direction (height) only and hence a heat transfer coefficient per unit height can be obtained, namely Ka where

$$Ka = (L/A)(\Delta t/y\Delta h) = C'(L/A)^q(G/A)^n$$

In a crossflow process water and air conditions vary in two directions (height and depth) and so the measured heat transfer coefficient is applicable to that particular section of packing considered. Thus, to draw significance to this fact the relationship is written as :

$$Kay = (L/A)(\Delta t/\Delta h) = C\ (L/A)^{q}(G/A')^{n}$$

CHAPTER THIRTEEN

CONCLUSIONS

13.1 Literature Survey

Almost without exception all workers use the Method first recorded by Merkel for counterflow analysis. Many of the early counterflow selection charts are in fact a development of the Merkel Cooling Diagram shown on Fig 2.3 in which all possible combinations of cooling duties are recorded on one chart. In the early days of cooling tower selections these charts were adequate but for present day requirements they have the limitations of difficulty in use, restricted scope and general inaccuracy. To improve upon the accuracy of these charts most workers developed a series of counterflow performance charts or crossflow temperature contours for a nondimensional height and depth of pack to cover operating conditions. However, the large number of charts required makes them inconvenient for tower selection.

The conclusion drawn from the Literature Survey suggests that a single chart to represent all the many combinations of operating condition was not compatible with commercial accuracy; and of the many selection methods surveyed none satisfied all the requirements outlined in the Introduction.

It is worth recording that the Gardner Method was the only technique reviewed which could be applied to both counterflow and crossflow operation but as a method it is not easy to apply.

In contrast, the Lichtenstein method whilst applying only to the counterflow process was very easy to use even though an auxilliary graph was necessary to obtain the final selection. The writer considered this method provided scope for improvement, including application to the crossflow process. It is of interest to record that the British Standard BS.4485 (25) published in 1969, suggests the use of the Lichtenstein method for presenting cooling tower test results. This reinforced the writer's decision to investigate, apply and extend this method.

The Lichtenstein Method was not only applied successfully to both processes but the number of Performance Charts required for counterflow selection was reduced from thirteen to four per wet bulb temperature, whilst only four Performance Charts were required for crossflow operation. In addition the Performance Diagrams (Figs 8.4 and 12.4) were incorporated into Expanded Diagrams (Figs 8.5 and 12.5) from which the tower base area can be established for a known cooling water load Q. The Expanded Performance Diagram gives a complete history for the entire selection procedure and because of its versatility can be used for a variety of alternative performance curves and tower characteristics so enabling an optimised tower base area to be obtained and permanently recorded.

13.2 Datum Temperature for Enthalpy Calculation

Of the two enthalpy temperature datums considered, the $32^{\circ}F$ datum enables a straight conversion from Btu's to Chu's to be performed without the need to consider datum levels. Also, this datum level is generally accepted in Britain and on the Continent. For these two reasons the work is based upon the use of a $32^{\circ}F$ enthalpy temperature datum.

13.3 Enthalpy Equation for Saturated Air

The analysis of equations to represent the equilibrium line for the enthalpy of saturated air finally led to the use of the fundamental enthalpy temperature relationship. Although requiring a greater computation time than the other relationships considered the use of the new I.C.L. 1905 computer eliminated computation time as the problem it had been when using the Elliott 803 computer. In addition, this relationship was used by the I.H.V.E. Guide (16) and other well established British organisations, namely the C.E.G.B. and B.S.I. The equation can also accommodate changes in atmospheric pressure which enables performance charts to be obtained for any elevation above sea level. This is considered an important feature of the Selection Technique.

13.4 Counterflow Performance Charts

The Counterflow Performance Charts Fig 8.3 were easy to use although they were found to have their limitations. This was particularly true for selections involving short approaches (less than 15^{o}F) to the wet bulb temperature. This difficulty can be overcome by plotting two additional Performance Charts, for NDU = 2.5 and NDU = 3.0 so increasing their number from four to six. The curves of constant outlet water temperature t_{L2} should also be extended to give cooling curves up to $t_{L2} = 100^{o}F$. This can be achieved by extending the computer program to give results for additional L/G values and plotting the corresponding intermediate charts to obtain the desired extension to the Performance Charts. A maximum L/G value of 3.0 is considered adequate since the air and water mass flow rates will normally lie in the ranges :

$$1000 < L/A < 5000 \quad lb/h\ ft^2 \text{ and } 1500 < G/A < 2500\ lb/h\ ft^2.$$

The Performance Charts can be made easier to use by incorporating additional outlet water temperature curves for intervals of 2 deg F, and adding lines of constant cooling range at intervals of 5 deg F from 5 to 50 deg F. Interpolation to $\pm$ 0.5^{o}F accuracy should then be possible.

For certain conditions the Intermediate Performance Charts, Fig 8.7 would appear to be more useful but only exhaustive practical application would establish which is the better presentation.

For the selection of smaller towers Graph III of the Expanded Performance Diagram Fig 8.5 should be extended or duplicated to accommodate towers in the range of base areas 5 to 200 ft^2. A log-linear plot may be suitable for Graph III such that it can be applied to the areas in the range 5 to 1600 ft^2 or more.

13.5 Crossflow Performance Charts

Observation of the crossflow computer results for a complete cross section of packing, Fig 11.5 shows that the differential temperature drop for the various pack sections does not reduce significantly in the direction of increasing pack depth. This is due to choosing a large NDU value, 0.15 for each element. A more realistic value would be in the range 0.005 to 0.05. This also accounts for the very small approach temperature (between 0.5 and 4.5 deg F) achieved for the water at outlet from the pack.

When using the Crossflow Performance Chart Fig 12.3 it is important to remember the area A' in the direction of air flow lies in the vertical plane and the area A in the direction of water flow lies in the horizontal plane and so the overall L/G ratio based on L/A and G/A' should be calculated with care such that :

$$\text{Overall } L/G = (L/A)(G/A')$$

Errors in some of the plotted points did occur but they were relatively small when related to the projected curve which always followed the expected trend. The errors were eliminated by more careful plotting of the results.

As with the counterflow method a useful extension to the Crossflow Performance Charts is the addition of extra constant outlet water temperature curves, say at 2 deg F intervals and the plotting of lines showing the cooling range at 5 deg F intervals from 5 to 50 deg F as shown on sheet 3 of Fig 12.3.

The temperature ranges indicated on the Counterflow and Crossflow Performance Charts are applicable to wet bulb temperatures in the range 60 to 70°F. For other wet bulb temperatures an approach temperature of 5 to 10 deg F should be established and all other temperatures set accordingly.

A similar technique can be used for plotting Counterflow and Crossflow Performance Charts based on the Number of Transfer Units where

$NTU = (L/G)\ NDU.$

For general commercial application it would be necessary to carry out development work on the Counterflow and Crossflow Performance Charts to identify their usefulness and overcome any limitations.

TABLE OF SYMBOLS AND UNITS

SYMBOL	QUANTITY	UNITS
a	Area of transfer surface per unit of tower packed volume	ft^2/ft^3
V	Effective packed volume	ft^3
A	Base area of cooling tower	ft^2
A'	Vertical cross-sectional area of cooling tower in direction of horizontal air flow	ft^2
y	Height of packing	ft
L	Mass rate of water flowing	lb/h
G	Mass rate of air flowing	lb/h
L/G	Water to air ratio giving the slope of the operating line	Dimensionless
G_D	Mass of dry air flowing	lb/h
R	Water to air ratio at either inlet or outlet conditions	Dimensionless
h_L	Enthalpy of saturated air in contact with and at the temperature of the water passing through the packing (i.e. at interface).	Btu/lb
h_{LM}	Enthalpy of saturated air at the mean water temperature t_{LM}	Btu/lb
h_v	Enthalpy of Water Vapour	Btu/lb
h_G	Enthalpy of air water vapour mixture passing through the pack.	Btu/lb
h_{GM}	Enthalpy of air water vapour mixture at the mean water temperature t_{LM}.	Btu/lb
Δh	Change in air enthalpy i.e. driving force.	Btu/lb
h_a	Enthalpy of dry air	Btu/lb
h_D	Enthalpy at the datum temperature	Btu/lb
q_L	Rate of heat transfer, bulk water to interface	Btu/h
X	Base area of crossflow tower per unit width	ft^2/ft
Y	Vertical cross-sectional area of crossflow tower per unit width in direction of horizontal air flow	ft^2/ft

q_s	Rate of sensible heat transfer, interface to the air stream.	Btu/h
q_E	Evaporative heat transfer	Btu/h
k_L	Film coefficient, bulk water to interface.	Btu/h ft^2 deg F
k_s	Film coefficient for sensible heat transfer between the interface and the main air stream	Btu/h ft^2 deg F
k_m	Mass transfer coefficient, interface to the air stream	lb/h ft^2 lb water vapour/lb dry air
K	Overall mass transfer coefficient	lb/h ft^2 lb water vapour/lb dry air
NTU	Number of Transfer Units	Dimensionless
NDU	Number of Diffusion Units otherwise called the Merkel Integral	Dimensionless
HTU	Height of a Transfer Unit	ft
HDU	Height of a Diffusion Unit	ft
α	Merkel Cooling Factor	Dimensionless
f	Enthalpy correction factor	Dimensionless
T	Refers to any Centigrade temperature	^oC
t	Refers to any Fahrenheit temperature	^oF
t_{WB}	Bulk air at the wet bulb temperature	^oF
t_G	Bulk air at the dry bulb temperature	^oF
t_i	Saturated air temperature at the interface.	^oF
t_L	Bulk water temperature	^oF
t_E	Temperature at which evaporation takes place	^oF
t_D'	Datum temperature of air	^oF
t_{L1}	Hot water temperature at inlet	^oF
t_{LM}	Mean water temperature	^oF
t_{L2}	Water temperature at outlet	^oF
t_o	Temperature at an arbitrary position in the tower.	^oF

t_f	Freezing point of water	^{o}F
Δt	Cooling range $(t_{L1} - t_{L2})$	deg F
dt	Incremental interval of temperature	deg F
App	Approach $(t_{L2} - t_{WB})$	deg F
PR	Potential range $(t_{L1} - t_{WB})$	deg F
g	Grains of water vapour present in 1 lb dry air	grains/lb dry air
w	Mass of water vapour present in 1 lb dry air	lb/lb dry air
m	Mass of vapour transfer, interface to air stream	lb/h
Q	Circulating water flow	gall/h
c_L	Specific heat of water	Btu/lb deg F
c_{pa}	Specific heat of dry air at constant pressure	Btu/lb deg F
c_{pv}	Specific heat of water vapour at constant press	Btu/lb deg F
c_{pm}	Specific heat of humid air at constant pressure	Btu/lb deg F
$\mathcal{L}$	Latent heat of evaporation at temperature t_E	Btu/lb
P	Atmospheric pressure	millibars
p_v	Partial pressure of water vapour in the air	millibars
p_{vs}	Partial pressure of saturated vapour at the dry bulb temperature	millibars
ρ	Density of air	lb/ft^3
d	Pencentage deviation	
O	See equation defined in text	
I	NTU Integral	
H_{oo}	See equation defined in text	
R'	See equation defined in text	

C	Constant defined in the text
n	Constant defined in the text
q	Constant defined in the text
B	Constant defined in the text
D	Constant defined in the text
E	Constant defined in the text
F	Constant defined in the text
J	Constant defined in the text
H	Constant defined in the text
I'	Constant defined in the text
N	Constant defined in the text
ε	Constant defined in the text
β	Constant defined in the text
γ	Constant defined in the text

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