THE DESIGN AND PERFORMANCE OF EVAPORATIVE CONDENSER

THESIS
Submitted in partial fulfilment of the requirements for the degree of
MASTER OF ENGINEERING
in
APPLIED THERMODYNAMICS
(Refrigeration & Airconditioning)

By
R. K. Goel

DEPARTMENT OF MECHANICAL ENGINEERING
UNIVERSITY OF ROORKEE
Roorkee (INDIA)

Feb. 1964

C E R T I F I C A T E

CERTIFIED that the thesis entitled " The Design and Performance of Evaporative Condenser" which is being submitted by Shri R. K.Goel, in partial fulfilment for the award of the degree of Master of Engineering in Applied Thermodynamics (Refrigeration and Air Conditioning) of the University of Roorkee, is a record of student's own work carried out by him under my supervision and guidance. The matter embodied in this thesis has not been submitted for the award of any other Degree or Diploma.

This is further to certify that he has worked for a period of 4 ½ months for preparing this thesis for Master of Engineering Degree at the University.

Roorkee,
Dated Feb; 6 , 1964.

(Rajendra Prakash)
Reader in Mech. Engineering
University of Roorkee
Roorkee.

ACKNOWLEDGEMENTS

The author with a deep sense of gratitude wishes to express his sincere thanks to Shri Rajendra Prakash, Reader in Mechanical Engineering Department, University of Roorkee , for his indispensible guidance, constant encouragement and keen interest throughout the course of work.

The author is also indebted to Prof. M.V.Kamlani, Head of Mechanical Engineering Department for giving encouragement to carry out the work.

The author wishes to acknowledge the kind help received from Shri S.C.Pruthi of Voltas Ltd., and Shri N.R. Taliwal of Atlas Engineering Corporation, Bombay, in connection of this work.

Free use of the published literature has been made on this subject, the bibliography of which is given in the end. The author wishes to acknowledge the valuable assistance received from these sources in compiling this work .

R.K.Goel.

SYNOPSIS

Evaporative condensers now so widely used, are comparatively simple mechanism devices, yet their heat transfer relations are decidedly complex. The test data are not readily reduced to simple coefficients as in the case wit-h many other heat transfer processes.

The mechanism of heat transfer has been analysed by James,(12) Thomson,(22) Wiles(23) Goodman,(8) yet their mathematical treatment varies widely. A comparative study of analyses of each of the investigations is attempted.

The determination of various heat transfer coefficients involved in the mechanism and the effect of varying the air and water flow rate on these coefficients have been discussed extensively. Rating curves are reviewed and prepared to predict the performance of the condenser at changing operating conditions.

Further there is an excellent graphical method suggested by Thomson for analysing the performance of evaporative condensers based on various intervening resistances involved in the heat transfer process. The problem selected for the design purposes has been analysed by this method to illustrate its validity.

A chart based on Goodman's analysis for determining spray water temperature is prepared, which will be found very useful in the design work. An illustrative design of evaporative condenser has been made on the basis of economical combination of air flow rate and surface area.

The various factors which contribute to fouling in an evaporative condenser are elaberately dealt. The effective water treatment and chemical cleaning of the condensers are also discussed.

Finally the economy of using an evaporative condenser is discussed in comparision with other condensing methods such as cooling tower and condenser combination and " Once through " water condenser. The factors which influence the selection of the best all round condensing method for specific application are also dealt with.

C O N T E N T S

CHAPTER I

INTRODUCTION

The mechanism of evaporative condenser is discussed.

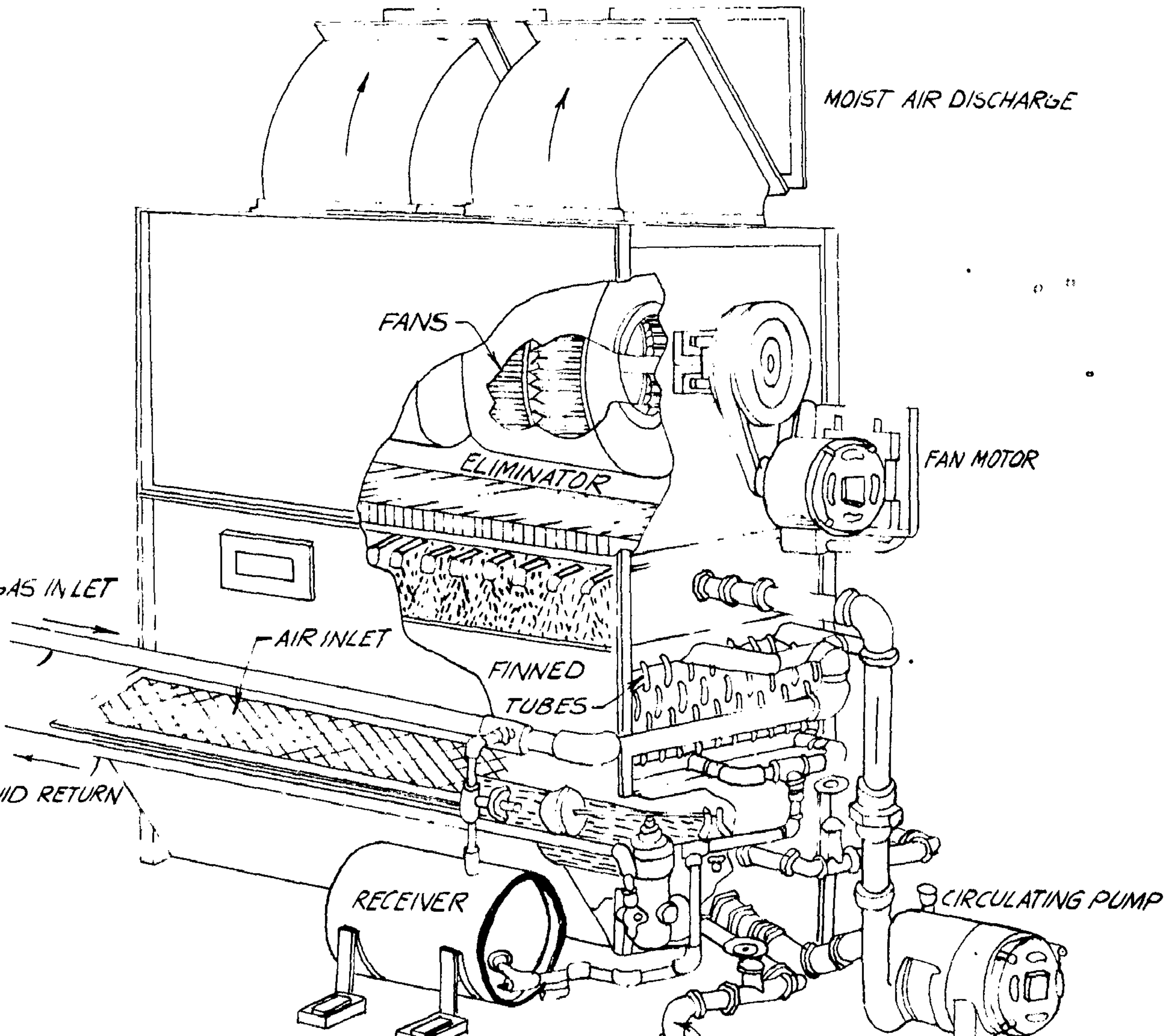
CHAPTER I

INTRODUCTION

Evaporative cooling was practised thousands of years ago by Egyptions in cooling water in porous earthenware pots. It may came as a surprise to many, who have looked upon the evaporative condenser as a recent invention to learn that at least as far back as 1902 patents had been granted for this device.

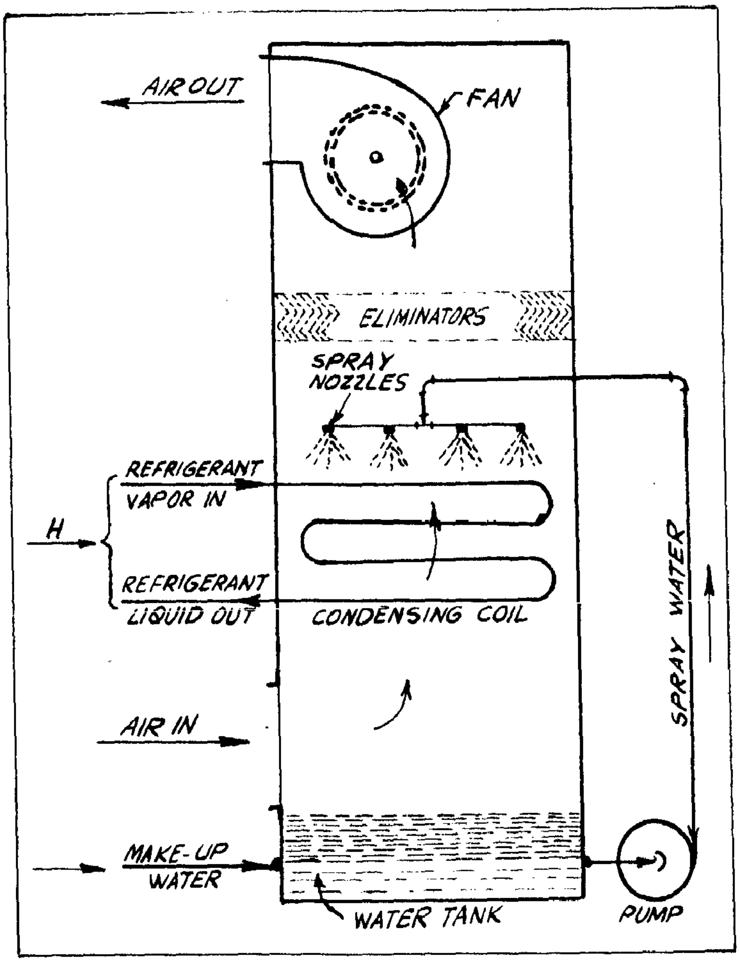
In early days of air conditioning there were few restrictions, if any, on the use of water for refrigeration purposes in most localities. Water was usually available in abundance and at low cost. The situation has now reversed itself, however, and very few cities in India and abroad are free from water problem. For this reason it became necessary to have a clear understanding of the several types of refrigerant condensing methods and water saving devices which may be used.

The evaporative condenser combines in one piece of apparatus two of the oldest elements in refrigeration : the condenser and the cooling tower. Unlike a seperate condenser or cooling tower the condensing coil of the evaporative condenser is directly in the spray of water and the air is drawn through both the condensing coil and the spray water as shown in Fig. 1.1.



EVAPORATIVE CONDENSER

FIG. 1.1



SCHEMATIC DIAGRAM OF AN EVAPORATIVE CONDENSER

The refrigerant is condensed inside the coil, and water is sprayed continuously over the outside of the coil. The spray water is collected in a tank at the bottom and is again delivered to the spray nozzles at the top by means of a pump. The same water is circulated continuously, and is neither heated nor cooled during its passage through the pipes connecting the tank, pump and spray nozzles. The spray water wets the outside surface of the coil, the heat is transferred through the wall of the coil to the water on its outer surface. But, the water as fast as it receives this heat transfers it in turn to the air flowing over the coil.

The temperature of the spray water must always be above both the initial and final wet bulb temperature of the air or heat could not be transferred from water to the air.

Evaporation of water takes place into air resulting an increase in the moisture content of t-he air passing through the condenser. The total heat surrendered by the condensing refrigerant is ultimately transferred to the air and results in an increase in its enthalpy.

The water consumption of an evaporative condenser is of the order of 2 to 2.5 gallons/hr/ton of referigeration, which amounts to approximately 10% of the consumption in "once through" water cooled condensers. This low water consumption is possible since most evaporative condenser designs include efficient eliminaters which prevent any carry over of free moisture,or the usual windage losses such as obtained for cooling tower operation. In addition to this, evaporative condenser handling outdoor air produces an additional saving by operating without water as an air cooled condenser during comparatively cold weather. Compactness of this equipment makes it suitable for installation in crowded cities where space is a costly factor as it can be easily located on the roofs of the buildings.

CHAPTER II

THEORY OF HEAT TRANSFER AS APPLIED TO EVAPORATIVE CONDENSERS.

Heat transfer phase of Evaporative condenser has been analysed by various investigators. A comparative study of analyses of different investigators has been made.

SYMBOLS

A_o	Outside surface area of the coil in sq.ft.
A_i	Inside surface area of the coil in Sq.ft.
F_r	Coefficient of heat transfer of the refrigrant film Btu/ °F / hr/ sq.ft. of internal surface.
l	Thickness of pipe in inches.
f_m	Coefficient of heat transfer through the metal wall of the pipe Btu / °F/hr/sq.ft./ inch of thickness.
f_e	External film coefficient of heat transfer Btu/hr/°F/ sq.ft. of external surface.
x	Fin surface efficiency. as defined by equation 2.1.
Wet bulbk	Overall coefficient of heat transfer from the refrigerant film to the outside air (including the film coefficient from water to air) in Btu/°F/ hr/ sq.ft. (Same as U_1)
Wet bulb m.t.d.	Lograthimic mean temperature difference of the spray water temperature and the entering air wet bulb temperature.
f_w	Coefficient of heat transfer of the water film from the metal wall to the wetted surface Btu/hr/°F / sq.ft.
f_s	Coefficient for the sensible and latent heat transfer from water surface to the outside air Btu/hr/°F/ sq.ft.
R_r	Resistance of the condensing film on inner tube wall
R_m	Resistance of metal tube wall.
R_w	Resistance of the water film on the outside of the tube.
r	Latent heat of evaporation, Btu/lb.
f_d	Convective mass transfer coefficient between water and air lb/ hr. °F (lb./lb.).

W_w , W	Moisture content of dry air at spray water air interface and main air stream flowing past section ~~dA_o~~ dA_o in lbs/lb.
f_c	Convective coefficient of heat transfer between water and air Btu/hr / ^oF / sq.ft.
t_w	Temperature of the surface of spray water film on condenser coils, also spray water temperature leaving.
h_w	Enthalpy of the air saturated at the constant water surface temperature t_w .
t	Dry bulb temperature of the air passing section dA_o of condenser.
t_1	Wet bulb temperature of air passing through the section dA_o of condenser.
h	Enthalpy of air passing through the section dA_o of condenser.
f_1	Equivalent heat transfer coefficient for latent and sensible heat transfer from wetted surface of the tube to air stream Btu/hr/^oF / sq.ft. as defined by equation 2.18.
F	Equivalent heat transfer coefficient for latent and sensible heat transfer from wetted surface of the tube to air stream Btu/hr/^oF/sq.ft. as defined by equation 2.25.
cp	Specific heat of air Btu/lb.
G	Weight of air passing through the condenser lbs/hr.
q	Total heat transfer from the wetted surface to air Btu/hr.
U_1	Overall coefficient of heat transfer from the condensing refrigerant to the outside air stream.
R_f	Resistance for sensible and latent heat transfer from the wetted surface to air stream.
h_1	Ethalpy of the entering air in Btu/lb.
h_2	Ethalpy of the leaving air in Btu/lb.

f_{s_1}	Coefficient of heat transfer for sensible and latent heat from wetted surface to air Btu/hr/sq.ft./Btu.defined by equation (2.27).
h_{w_1}, h_{w_2}	Enthalpies of air saturated at entering and leaving surface ~~and~~ temperatures respectively Btu/~~hr~~ lb.
hm	Logarithimic mean enthalpy difference between the wetted surface and air, Btu.
U	Coefficient of heat transfer from refrigerant vapour to wetted surface Btu/hr/deg.F/sq.ft.
Z	By pass factor defined by the equation 2.36.
R_{f_1}	Resistance for sensible and latent heat transfer from the wetted surface to air stream defined by equation 2.22.
R_{t_1}	Resistance (total) for heat flow from the refrigerant to the air stream defined by equation 2.21.

CHAPTER II

THEORY OF HEAT TRANSFER AS APPLIED TO EVAPORATIVE CONDENSERS

2.1. The evaporative condenser is a heat transfer equipment in which the refrigerant coils are wetted by the spray nozzles, the air is either sucked in or forced over the coils, according to the induced or forced draft air supply. A thin film of water is formed on the coils. Water is evaporated from the wetted surface to the moving air. The evaporated water gets the latent heat of evaporation from the tubes, and the tubes in turn from the refrigerant. There is a sensible heat transfer also from the spray water to the air, if the spray water temperature is higher than the dry bulb temperature of the entering air. This follows that there is both sensible and latent heat transfer from water film to air.

The mechanism of heat transfer in an evaporative condenser has been analysed by James[12], Goodman[8], Thomson[22] and Wile[23] yet there is insufficient data to permit the prediction of an evaporative condenser's performance with any reasonable accuracy. We find substantial agreement among investigations in approach, although the mathematical treatment varies widely.

2.2. HEAT TRANSFER THEORY - JAMES.

James[12] presented his analysis of heat transfer in

FIG. 2.1

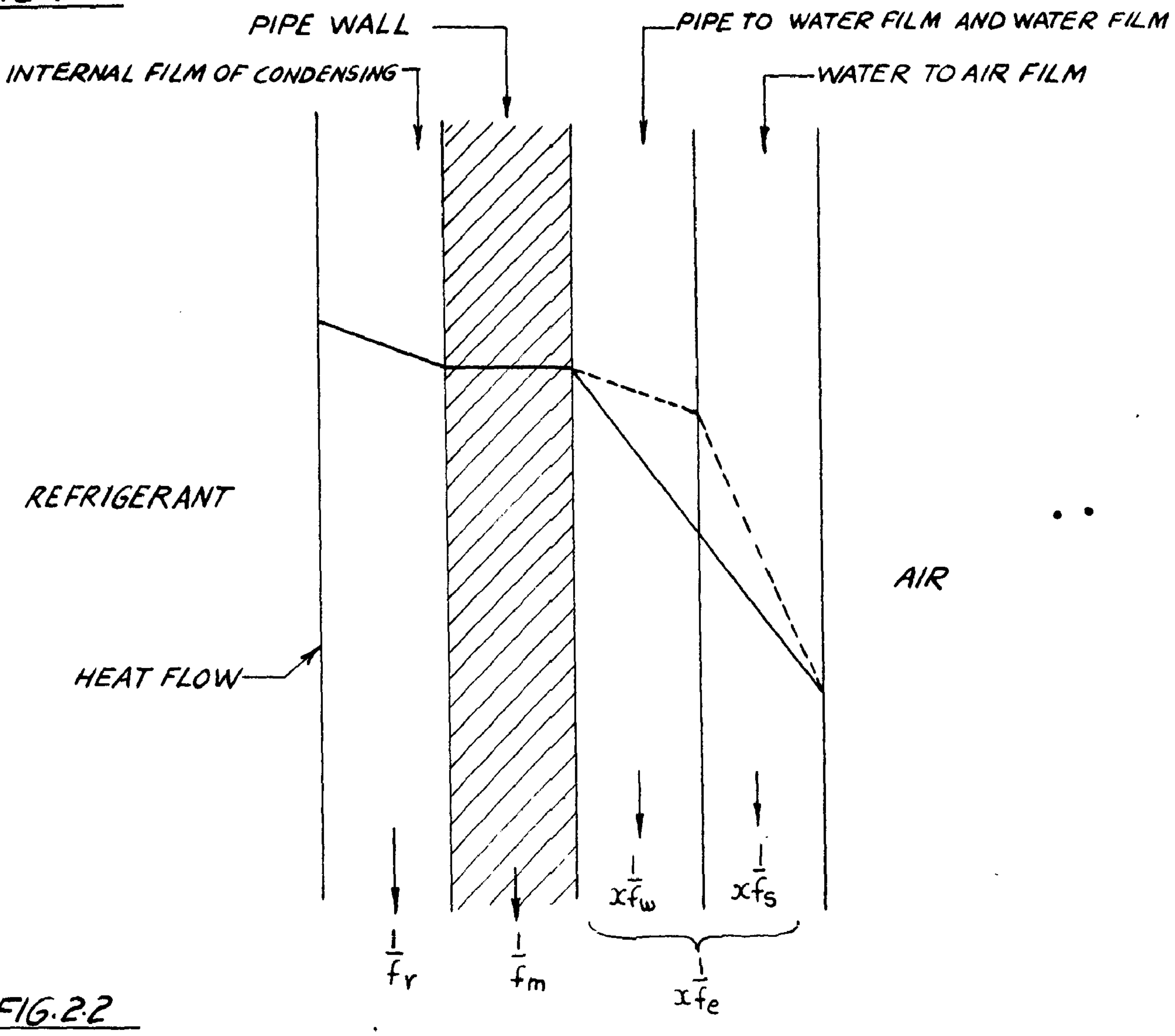
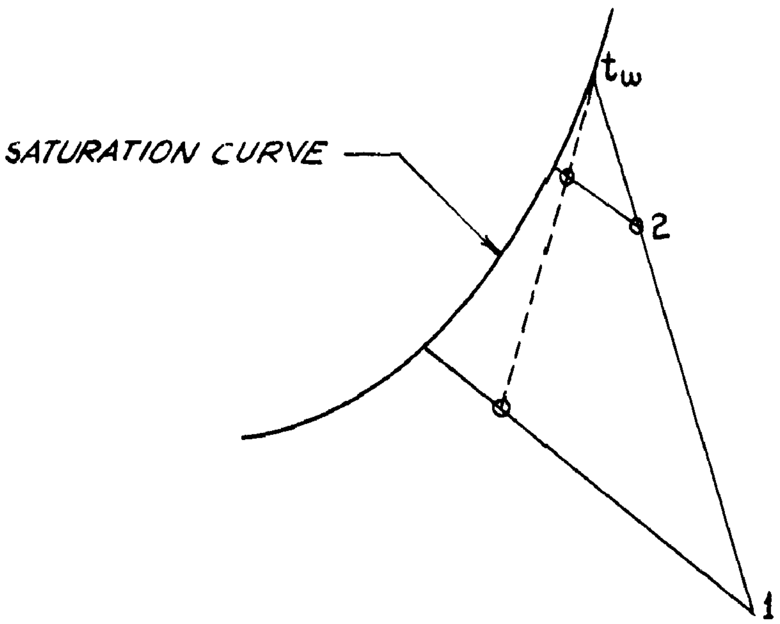

PIPE WALL
INTERNAL FILM OF CONDENSING
PIPE TO WATER FILM AND WATER FILM
WATER TO AIR FILM
REFRIGERANT
AIR
HEAT FLOW
$\frac{1}{f_r}$
$\frac{1}{f_m}$
$\frac{1}{xf_w}$
$\frac{1}{xf_s}$
$\frac{1}{xf_e}$

FIG. 2.2


t_w
SATURATION CURVE
2
1

evaporative condenser at the annual meeting of ASRE 1936.

The ultimate heat transfer from refrigerant to air involves three different resistances in between, starting from inside of the coil to the outside of the coil, as shown in fig 2.1

(i) That offered by the condensate film inside the coil.

$$A_o / A_i . f_r \quad ..$$

(ii) That offered by the pipe wall.

$$1/f_m$$

(iii) That offered by the moisture and by the moisture and air films on the outside surface of the pipe

$$= 1 / x.f_e$$

Where ,

x = fin surface efficiency

$$x = \frac{\text{Temperature difference between the air and the effective fin temperature.}}{\text{Temperature difference between the air and fin temperature at its base.}}$$

= 1.0 for bare pipe.

$$\text{"Wet bulk K"} = \frac{1}{\frac{A_o}{A_i f_r} + \frac{1}{f_m} + \frac{1}{xf_e}} \quad ....(2.2)$$

2.2.1. The internal coefficient f_r has wide range of values ranging from 160 to 300.

The following are the factors, have a modifying effect on f_r :

(a) Refrigerant velocity should be as high as possible, but care should be taken of the increasing pressure loss.

(b) Hydraulic radius of the tube should be low.

(c) Roughness of the internal surface is important in view of the greater turbulence obtained by increasing internal roughness. But pressure loss will also be greater.

(d) Mean temperature difference is increased with increase of m.t.d.

Presence of noncondensible gases and other impurities also effect f_r but this is common to all condensers.

2.2.2. $1/f_m$, the wall resistance in the pipes is so insignificant that it can be easily neglected, without the sacrifice of any appreciable accuracy. Since f_m is very large, 360 for iron pipes and 2500 for copper tubing and 1 is also small, the factor $1/f_m$ will be very small . Say for Copper tube 1 = 0.05 inch; and f_m = 2500. It needs no actual calculations to show how insignificant will be the value of $1/f_m$.

2.2.3. The external resistance $\dfrac{1}{x\,f_e}$ can be broken into two parts, viz $1/xf_w$ and $1/xf_s$

The resistance $1/xf_w$ may be treated as the resistance for the water film and $1/xf_s$ as the resistance for the water film to air. It is this resistance which has the major share in the total resistance. If any one of the two resistances $1/xf_w$ and $1/xf_s$ is lowered, a considerable change in the heat transfer may result.

The water to pipe coefficient, f_w will range normally from 250 to 1000. It depends upon the following factors :-

(a) Water quantity.

(b) Efficiency of wetted surface, which in turn depends on water quality and method of water distribution.

The water to air coefficient , f_s will range from 70 to 100 in a well designed unit. It will depend upon following factors :

(a) Vapour pressure difference.

(b) Relative air and water velocities and air turbulence.

(c) Water quantity.

The external coefficient f_e a combination of f_w and and f_s , will range from 60 to 90. It will be more effected by the factors controlling f_s , than that by f_w .

2.2.4. James[12] has considered a single resistance for the both sensible and latent heat transfer with a driving potential as the logarithimic mean temperature difference of the spray water temperature and air wet bulb temperature. Note that the wet bulb temperature rises logarithimically as the air travels through the unit. He termed logarithimic mean temperature difference as "wet bulb m.t.d." and the heat transfer as " wet bulb k" to stress on the importance of wet bulb temperature.

Goodman[8] emphasized on the point that the heat and mass transfer are two distinct phenomenon and cannot be combined. Latent heat transfer caused by evaporation of moisture, that is a mass transfer, has a driving potential which is the difference of the partial vapour pressures between the air saturated at the spray water temperature and the air of the main stream. On the otherhand the sensible heat transfer depends on the difference of the spray water temperature and the dry bulb temperature of air. Goodman in his writt-en discussion stressed that these two different mechanisms cannot be replaced with a single resistance and a driving potential.

However Thomson E.J.[22] has theoretically proved that a concept of a single resistance can be formed with a driving potential as the difference of the spray water temperature and average air wet bulb temperature. His analysis is dealt next in the Chapter.

2.3. HEAT TRANSFER THEORY - THOMSON.

Thomson E.J.[22] gave an excellent analysis of heat transfer applied to evaporative condensers. He explains several mechanisms involved in the heat transfer process from refrigeratnt to air.

Considering for the clean surface of the coil

(a) Condensation of wapour on the inner tube surface.

(b) Conduction through tube wall or extended surface.

(c) Conduction and Convection from outer tube surface through the water film.

(d) Simultaneous latent heat transfer, sensible heat transfer and radiation from the wetted surface to the air stream.

The resistances involved in the first three mechanisms be,

R_r = Resistance of condensing film on inner tube wall.

R_m = Resistance of metal tube wall.

R_w = Resistance of water film on the outside of the tube.

A temperature difference exists to overcome these resistances. It is assumed that the water temperature remains constant throughout. This follows that the heat transferred to the water film is immediately dissipated to the air film. This heat transfer from water film to air will be according to as item (d). Neglecting heat transfer by radiation, for all practical purposes, it is fairly accurate to consider sensible and latent heat transfer only.

2.3.1. When air wet bulb and dry bulb temperatures are less than the spray water temperature, both sensible and latent heat transfer will take place from water to air. If only air wet bulb temperature is less and dry bulb temperature is higher than the spray water temperature, latent heat will flow from water to air and sensible heat from air to water, i.e. the latent heat to be transferred will be the sum of the sensible heat of air plus the heat of condensing vapour.

The latent heat transfer is through the evaporation of water from the wetted surface to the air film. The driving force for this evaporation is the partial vapour pressure difference between the air saturated at the spray water temperature and the air in the main air stream. It is assumed that the air just near the water film will be saturated at the spray water temperature. Therefore partial vapour pressure may be replaced by the difference in the moisture content in lbs./lb. of dry air.

Thomson[22] has considered a thin air boundary layer in which air is essentially in laminar motion around the wetted surface. The whole resistance to evaporation is concentrated in this film of boundary layer. The water vapour is first diffused through the boundary layer and then carried to the main air stream by convection. Since the boundary layer is affected by both the shape of the surface and the air velocity, the water vapour transfer rate is also affected likewise.

Similar to that of the mass transfer is the sensible heat transfer from wetted surface to air in the case when spray water temperature is more than the dry bulb temperature of the air. A concept of boundary layer can be formed here also, so that the heat transfer is by conduction first and is followed by convection after the boundry layer to main air stream.

In analyzing the effectiveness of the heat transfer equipment, the various intervening resistances will have to be determined. It is quite simple comparatively where the driving potential is homogeneous throughout like temperature. But in this case of evaporative condenser there is both heat and mass transfer. The latent heat transfer caused by the evaporation from the wetted surface has a driving potential as the difference between the partial vapour pressures, while for sensible heat transfer from wetted surface to air the driving potential is the temperature difference between the spray water and db. of air.

James[12] has tried to analyse by formulating a concept of a single resistance for heat transfer between the wetted surface and air stream and considering a driving potential as the logarithimic mean temperature difference of the spray water temperature and the air wet bulb temperature. The details of the James analysis has already been discussed. Goodman who has done excellent analysis of heat transfer has criticised James approach to the problem.

2.3.2. Thomson[22] has , theoretically proved that a concept of a single resistance can be taken, provided the driving potential for heat transfer may be the difference between the spray water temperature and average air wet bulb temperature. This equivalent resistance is given a symbol R_f.

The rate of heat transfer for wetted surface of infinitesimal area dA_o is the sum of latent heat and sensible heat transferred per unit time. This may be expressed in the following equation:

$$dq = \left\{ rfd\ (W_w - W) + f_c\ (t_w - t) \right\} dA_o \quad \ldots\ldots.(2.3)$$

in which $rfd\ (W_w - W)\ dA_o$ represents the latent heat transfer and $f_c\ (t_w - t)\ dA_o$ represents sensible heat transfer.

London, Manson and Boelter have shown for water and air mixtures Raynolds modulus

$$F_d\ C_p / F_c = 1 \qquad ..$$

$$f_d = f_c\ / C_p \qquad ....(2.4)$$

Substituting (2.4) in equation (2.3.)

$$d = f_c\ /C_p\ (\ r\ (W_w - W) + C_p\ (t_w - t)\ dA_o \qquad .....(2.5)$$

In as much as the enthalpy or the heat content of the dry air with moisture associated with it is given by

$$h = rw + c_p t. \qquad .....(2.6)$$

Equation (2.5) may be reduced to the following form

$$dq = f_c / C_p \left\{ h_w - h \right\} dA_o \qquad ....(2.7)$$

This is the equation used by Goodman in his analysis of heat transfer in evaporative condenser.

The heat flowing from the infinitesimal area causes a change of enthalpy of the air flowing past it .

$$dq = Gdh = f_c / C_p \ (h_w - h)\, dA_o \qquad \ldots.(2.8)$$

Rearranging equation (2.8)

$$\frac{dh}{h_w - h} = \frac{f_c}{C_p \ G}\, dA_o \qquad \ldots.(2.9)$$

Assuming that the spray water temperature remains. . constant and therefore h_w is constant equation(2.9) can be integrated

$$\int_{h^1}^{h^2} \frac{dh}{h_w - h} = \int_0^A \frac{f_c}{C_p \ G}\, dA_o \qquad \ldots(2.10)$$

or , $$\log \frac{(h_w - h_1)}{(h_w - h_2)} = A_o \frac{f_c}{C_p \ G}$$

or $$\frac{h_w - h_1}{h_w - h_2} = e^{A_o \ f_c / C_p \ G}$$

$$\frac{h_w - h_1}{H_w - h_2} = e^{M} \qquad \ldots.(2.11)$$

Where $M = A_o \dfrac{f_c}{C_p \ G}$. But the total rate of change in Enthalpy of the air flowing through the condenser is given by the following equation,

$$- q = G (h_1 - h_2) \qquad \ldots(2.12)$$

$$h_2 = q/G + h_1$$

Substituting h_2 in equation (2.11)

$$(h_w - h_1) = e^{M} (h_w - q/G - h_1)$$

or $$(h_w - h_1) (1 - e^{M}) = - e^{M} (q/G)$$

Therefore, $$q = - G (h_w - h_1) (1 - e^{M}) e^{-M}$$

$$= - G (h_w - h_1) (e^{-M} - 1)$$

$$= G (h_w - h_1) (1 - e^{-M}) \qquad \ldots(2.13)$$

For the limited range of w.b.t. (55% to 100%) the enthalpy of the air and associated moisture may be expressed as a function of the air wet bulb temperature. This can be represented as

$$h = \frac{(t_1)^2}{145} \qquad \ldots(2.14)$$

Substituting in the above equation (2.13)

$$q = \frac{G}{145} (tw^2 - t_1^2) (1 - e^{-M}) \qquad \ldots(2.15)$$

$$q = \frac{G}{145} (t_w + t_1) (t_w - t_1) (1 - e^{-M}) \quad ......(2.16)$$

Assuming that the equation (2.16) may be written in the following simplified form.

$$q = F_1 A_o (t_w - t_1) \quad .....(2.17)$$

Then F_1, the latent and sensible heat transfer coefficient may be written as

$$F_1 = \frac{G}{145 A_o} (t_w + t_1) (1 - e^{-M}) \quad .....(2.18)$$

In as much as the heat flow/unit area from the condensing refrigerant to the air stream is constant, the following equations may be written for heat flow through the wetted tube.

$$q/A_o = F_1 (t_w - t_1) = f_w (t_o - t_w)$$

$$= f_m / l (t_i - t_o)$$

$$= \frac{A_i f_r}{A_o} (t_r - t_i)$$

$$= U_1 (t_r - t_1) \quad ...(2.19)$$

Solving for $1/U_1$ in equation 2.19

$$\frac{1}{U_1} = \frac{1}{F_1} + \frac{1}{f_w} + \frac{l}{f_m} + \frac{A_o}{A_i f_r} \quad ......(2.20)$$

Since the conductivities may be expressed as the reciprocal of the resistances per sq.ft. of outside surface A_o

Therefore ;

$$R_{t_1} = R_{F_1} + R_w + R_m + R_r \qquad \ldots\ldots(2.21)$$

where

$$\frac{1}{U_1} = R_{t_1}, \quad \frac{1}{F_1} = R_{f_1},$$

$$1/f_w = R_w, \quad 1/f_m = R_m$$

$$\text{and} \quad \frac{A_o}{A_1 f_r} = R_r \qquad \ldots\ldots\ldots(2.22)$$

The resistances here are considered as resistances to heat flow/ sq.ft. of outside surface.

The total heat transferred from an evaporative condenser tube will be given by :-

$$q = U_1 A_o (t_r - t_1) = \frac{A_o}{R_{t_1}} (t_r - t_1) \quad \ldots\ldots(2.23)$$

It must be emphasized that the driving potential in equation (2.23) is the difference in refrigerant temperature and incoming air wet bulb temperature.

This equation can be successfully used for calculating the heat transfer for condenser.

The equation (2.17) can also be used for the above purpose. The driving potential in this equation is the difference in spray water temperature and incoming air wet bulb temperature. The effect of changing operating conditions or design on F_1 may be easily misinterpretted. Therefore, equation (2.17) is redefined.

$$q = FA_o \left(t_w - t_{1\,ave} \right) \qquad \ldots.(2.24)$$

where t_1 - average, is the average wet bulb temperature between incoming and outgoing air,

Comparing equation (2.24) with (2.16)

$$F = \frac{G \left(t_w + t_1 \right) \left(t_w - t_1 \right) \left(1 - e^{-M} \right)}{145\ A_o \left(t_w - t_{1\,ave.} \right)} \qquad \ldots(2.25)$$

It may be stated that F & F_1 are related by

$$\frac{F}{F} = \frac{R_F}{R_{f_1}} = \frac{t_w - t_{1\,ave.}}{t_w - t_1} \qquad \ldots..(2.26)$$

2.4. HEAT TRANSFER THEORY - WILE

Earlier investigators James[12], Thomson[22] used the combination of temperature potential and vapour pressure related by a common coefficient, but the important transition to enthalpy as the combination of these two forces seems to have occurred as a logical sequence without specific record in the literature. However, it was Wile[23] who used enthalpy as the driving potential in the analysis of heat transfer in evaporative condenser.

2.4.1. The heat transfer in evaporative condenser has two distinct mechanisms.

(a) The heat transfer from the wetted surface of the tubes to air. The driving potential is composed of both temperature and the vapour pressure difference. This can be expressed by the difference of the enthalpies of the air saturated at the surface temperature and that of the air in the main stream.

(b) The heat transfer from the condensing refrigerant to the wetted surface has the temperature driving potential. This is the difference in temperature of the condensing refrigerant and that of the wetted surface. The heat transfer for the refrigerant through the tube to the wetted surface can be expressed as the function of this temperature difference. The problem now remains in determining the theoretical wetted surface temperature t_w, which forms a separating line between the temperature driving potential process and the enthalpy driving potential process. Fortunately, this temperature t_w, may not be determined precisely. A small error in determining t_w may not cause an appreciable error, since this will have an increasing effect on the heat transfer on one side of the separating line and a decreasing effect on the other side. They may not balance exactly each other, but however their effect is compensating.

2.4.2. Similar to the heat transfer equation with temperature driving potential, is the following equation used for the heat transfer from the wetted surface to air using enthalpy driving potential.

$$q = f_{s_1} \quad A_o \quad \Delta h_m \qquad \text{....(2.27)}$$

$$\Delta h_m = \frac{(h_{w1} - h_1) - (h_{w2} - h_2)}{\log \dfrac{(h_w - h_1)}{(h_{w_2} - h_2)}} \qquad \text{....(2.28)}$$

$$q = f_{s_1} \cdot A_o \frac{(h_{w_1} - h_1) - (h_{w_2} - h_2)}{\log \dfrac{(h_{w_1} - h_1)}{(h_{w_2} - h_2)}} \qquad \text{......(2.29)}$$

Assuming that the wetted surface temperature is constant -

$$t_{w_2} = t_{w_1} = t_w \qquad \text{...(2.30)}$$

Therefore , $h_{w_2} = h_{w_1} = h_w$ ...(2.31)

Where h_w corresponds to a theoretical constant surface temperature t_w which will produce the same effect as the variable surface temperature actually existing in the condenser.

$$q = f_{s_1} A \frac{(h_2 - h_1)}{\log \dfrac{h_w - h_1}{h_w - h_2}} \qquad \text{....(2.32)}$$

$$\log \frac{h_w - h_1}{h_w - h_2} = \frac{f_{s_1} A}{q} (h_2 - h_1)$$

$$\frac{h_w - h_1}{h_w - h_2} = e^{\frac{f_{s_1} A}{q} (h_2 - h_1)}$$

$$\text{OR} \quad \frac{h_w - h_2}{h_w - h_1} = e^{-f_{s_1} A / q \cdot (h_2 - h_1)} \qquad \text{.....(2.33)}$$

Now since the heat absorbed by air

$$q = (h_2 - h_1) G \qquad \text{.....(2.34)}$$

Therefore

$$\frac{h_w - h_2}{h_w - h_1} = e^{-fs_1A/q \cdot q/G} = e^{-fs_1A/G}$$

Let $$\frac{h_w - h_2}{h_w - h_1} = Z = 1 - \frac{h_2 - h_1}{h_w - h_1} \quad ......(2.35)$$

and $$z = e^{-(fs_1A/G)} \quad .....(2.36)$$

Z is the bypass factor, represents that portion of the total air passing through the wett-ed surface, which passed unchanged and remaining is supposed to be in intimate contact and becomes saturated at the surface temperature tw. The inlet and the outlet conditions can be plotted on the chart. The intersection of the saturation curve and the process line joining the inlet and outlet conditions gives the constant wetted surface temperature tw. The corresponding enthalpy hw can be readily determined now. Knowing hw , Z can be calculated. Z depends upon fs, A and G . For the accuracy in determining t_w , the inlet and outlet conditions should be such that the intersection between the process line and the saturation curve is well defined. The dotted line in the Figure 2.2 represents test points (1) and (2) but have such high relative humidity that the intersection of the process line and the saturation line is poorly defined.

U, the overall coefficient of heat transfer from refrigerant vapour through the tube wall and water film to wetted surface can be determined now.

$$U = \frac{q}{A_o(t_r - t_w)} \qquad \text{........(2.37)}$$

CHAPTER III

THE PERFORMANCE OF EVAPORATIVE CONDENSERS

The effect of varying water flow and air flow rates on heat transfer coefficients is discussed extensively. In addition, coefficients of heat transfer, applicable to evaporative condensers for bare pipe and finned surfaces have been compiled in tabular form from published literature. A graphical method suggested by Thomson for analysing the performance of evaporative condensers is discussed.

SYMBOLS

A_o	Outside surface area of the coil in sq.ft.
A_i	Inside surface area of the coil in sq.ft.
F_r	Coefficient of heat transfer of the refrigerant film Btu/deg.F/hr/sq.ft. of internal surface.
f_m	Coefficient of heat transfer through the metal wall of the pipe Btu/deg.F/hr/sq.ft./inch of thickness.
f_w	Coefficient of heat transfer of the water film from the metal wall to the wetted surface Btu/hr/deg.F/sq.ft.
B	Ratio of the outside to inside surface of the coil.
q	Total heat surrendered by the condensing refrigerant that is, the total heat capacity of the condenser in Btu/hr.
f_c	Convective Coefficient of heat transfer between air and water(Air film coefficient) Btu/hr/sq.ft./deg.F.
f_e	External film coefficient of heat transfer Btu/hr/deg.F/sq.ft. of external surface.
f_s	Coefficient for the sensible and latent heat transfer from water surface to the outside air Btu/hr/deg.F/sq.ft.
U_1	Overall coefficient of heat transfer from refrigerant vapour to outside air Btu/hr/deg.F/sq.ft.
W'	the rate of water flow per sq.ft of the projected area in gpm.
W	rate of water flow per hour
L	Length of pipe in feet.
R	Total resistance to heat flow from inner tube surface to spray water. defined by eqn 2.3
a	Cross sectional area for conduction.
δ	Thickness of fins in ft.
S	fin spacing in ft.
l_1	Length of fins in ft.

P	Average perimeter of finned surface
A_f	Surface of bare pipe to which fins are attached, sq.ft.
t_{s1}	Temperature of inside surface of metal tube deg.F.
t_w	Temperature of spray water in deg.F.
R_p	Resistance per unit outside area.
G	Weight of air circulated through the condenser lbs/hr.
c_p	Specific heat of air Btu/lb.
μ_f	Viscosity of air lbs/hr/ft.
$G_{max.}$	Air rate in lbs/hr/sq.ft. of minimum free section.
Z_p	2 x heated length of fin over which air travels.
f_1	Equivalent heat transfer coefficient for latent and sensible heat transfer from wetted surface of the tube to air stream Btu/hr/deg.F/sq.ft. as defined by equation 2.18.
F	Equivalent heat transfer coefficient for latent and sensible heat transfer from wetted surface of the tube to air stream Btu/hr/Deg.F/sq.ft. as defined by equation 2.25.
R_r	Resistance of the condensing film on inner tube wall.
R_m	Resistance of metal tube wall.
R_w	Resistance of the water film on the outside of the tube.
h_1, h_2	Ethalpy of the entering and leaving air in Btu/lb. res.
R_f	Resistance for sensible and latent heat transfer from the wetted surface to air stream.
R_{f_1}	Resistance for sensible and latent heat transfer from the wetted surface to air stream defined by equation 2.22.
R_{t_1}	Resistance (total) for heat flow from the refrigerant to the air stream defined by equation 2.21.
R_t	Total resistance for heat flow from the refrigerant to the air stream defined by equation $R_t = R_f + f_w + R_m + R_r$.

t_1	Wet bulb temperature of the entering air Deg. F.
U	Overall coefficient of heat transfer from refrigerant vapour to wetted surface Btu/hr/sq.ft./deg.F.
Z	Bypass factor defined by the equation 2.36.
h_w	Enthalpy of the saturate air at the spray water temp. Btu/lb.
t_r	Temperature of the condensing refrigerant deg. F.
f_{s_1}	Coefficient of heat transfer for sensible and latent heat from wetted surface to the air Btu/hr/sq.ft/Btu. defined by equation 2.27.
m	quantity equal to $(f_c \ P/f_m \cdot a)^{\frac{1}{2}}$.
Z'	Adjusted bypass factor defined by the equation
U'	Adjusted overall coefficient of heat transfer from refrigerant vapour to wetted surface Btu/hr/sq.ft/deg.F.

CHAPTER III

THE PERFORMANCE OF EVAPORATIVE CONDENSERS :

3.1. In order to predict the performance of an evaporative condenser, it is useful to study first the effect of the variables such as air flow rate, water flow rate and the condensing temperature on the various heat transfer coefficients involved in the heat flow process. Once this has been studied, the performance characteristics could be drawn based on the test results.

3.2. JAMES'S ANALYSIS:

In the analysis of heat transfer (discussed in Chapter II), James considered fe, the external coefficient of heat transfer as composed of both the water film coefficient f_W and the water to air film coefficient fs (Fig. 2.1)

The external film coefficient $h_2$fe will be affected by the change in the air quantity or the water flow rate.

3.2.1. Water flow as affecting fe, the external coefficient:

Water quantity will affect both the fw and fs. Increased quantity of water will tend to increase f_W, since the higher flow rate over the tubes will be obtained. But as f_W is already 250 or more, the effect of increasing the water quantity above a certain point will be negligible in respect to overall performance. At that certain point, the amount of the water needed to wet the outer surface of the tubes completely into a thin film of water, is obviously, the minimum quantity that would be required. Increased water quantity will tend to increase f_s, the water to air coefficient, but as in the case of f_w, beyond a certain limit its effect on f_s will not be pronounced. At higher water flow rate higher relative air and water velocities will be obtained and more water surface will be offered as diffusion surface. Test data in table 3.1. on the experimental bare pipe unit illustrate this point.

According to test results not more than 10 gpm of water would have been necessary to secure practically optimum performance. However, it is desirable to circulate a little excess water so as to keep the coils clean and wash all deposits down into the drain pan. Observing the water flow in run no. 3 Table 3.1 under glass it was seen that all coils were completely wetted but there certainly, was not excess of water being circulated.

While carrying out these tests, James noticed that it was not necessary to spray water over coils. Distributing water through holes drilled in the pipe gave excellent result and to some extent avoided the trouble of nozzle clogging.

TABLE 3.1.

CHANGE IN HEAT TRANSFER WITH WATER QUANTITY.

Run No.	Entering db °F	Entering wb °F	Air quantity cfm	Condensing Temp. °F	Water circulated gpm.	% of U_1 value.
1	89.6	77.5	4303	103.0	2.5	75.5
2	91.8	78.1	4273	103.1	4.95	84.4
3	91.0	78.0	4456	103.1	10.05	98.2
4	90.2	78.3	4280	103.1	14.25	100.0

The amount of the make up water required varies with the operating conditions. About 1.75 gallons per hr. per ton will usually be sufficient but ordinarily some excess water is supplied to carry off deposits on the coils.

3.2.2. Air Flow as Affecting f_e, the external Coefficient:

With the increasing air flow rates, f_e the external film coefficient increases. When the air flow rate is increased higher rubbing velocities of water and air are obtained. The cross flow type of condensers in which the air and water flow are opposing each other, greater turbulence and increased diffusion.

surface is created and this tends to increase f_s and finally ~~the~~ U_1 , the overall heat transfer coefficient. U_1 will increase with increasing air velocity and the rate of change depending on the type of surface. For a bare pipe with opposing air and water flow, U_1 increases as approximately 0.475 power of velocity Table 3.2. contains the test data obtained by James on an experimental bare pipe unit, illustrates this trend.

TABLE 3.2.

CHANGE IN HEAT TRANSFER WITH AIR QUANTITY

Run No.	Entering db °F	Entering wb °F	Condensing Temp °F	Water re-circulated gpm	Air quantity cfm	% of U_1 value
1	84.4	78.15	110.3	17.8	1901	71.8
2	81.15	78.3	110.2	17.8	2780	85.4
3	80.75	77.85	110.2	17.8	3830	100.0

Starting with a higher value of cfm per ton and decreasing this quantity in regular increments, it will be found that the rate of decrease of lograthimic ~~mtd~~ m.t.d. is fairly uniform down to the point where wb. temperature rise is approximately one half of the difference between condensing temperature and entering air wet bulb temperature. Any further decrease will result in a sudden fall of m.t.d.

With assumed designed conditions of 110 °F condensing and 78 degrees F. entering wet bulb temperature, the wet bulb temperature rise to satisfy the above point should not be above 16 deg. F. With 16 Deg. F wet bulb temperature change through the unit, approximately 190 cfm of air per ton is required ; However it must be pointed out that either higher of lower values may prove out to be economical for a particular combination of coil and fan.

A balance between the cost due to fan size and Horse Power and the effect of air quantity on heat transfer should be maintained.

3.2.3. Condensing Temperature Effect on f_e , the external Coefficient :

A change in the condensing temperature , or temperature level has a marked effect on coefficient f_s , since it directly affects the vapour pressure difference between the wet bulb temperature and the temperature of the wetted surface. Heat transfer decreases with decreasing condensing temperatures. The vapour pressure difference is an important factor in determining the magnitude f_s which might more correctly be called a coefficient of vapour diffusion.

Suppose that the following two sets of conditions existed in two identical units.

1. 110 ^oF condensing temperature 85 ^oF wet bulb temperature.
2. 90 ^oF condensing temperature 65 ^oF wet bulb temperature

When all other variables such as air velocity, water quantity etc., are same, unit 1 will have the higher heat transfer value.

In both cases the mean temperature difference is the same but analysis shows that in unit 1 there exists a greater vapour pressure difference between water to air than exists in unit 2. Assuming a drop of 10 ^oF between the condensing temperature and the temperature of the outer surface of the wall, the conditions should be written as follows :

1. Surface water temperature = 100 ^oF with 85 ^oF Av. w.b.
2. Surface water temperature = 80 ^oF with 65 ^oF Av. w.b.

Unit 1 should have a vapour pressure difference of 1.916 - 1.201 = 0.715 inches of Hg, while unit 2 would have a vapour pressure difference of 1.022 - 0.616 = 0.406 in Hg. Since vapour pressure difference being the driving force, unit 1 would obviously have a faster rate of evaporation, and therefore a higher value of film coefficient.

Test data are shown in Table 3.3. to illustrate the effect of condenser temperature on heat transfer.

FIG. 3.1.

RATING CURVES

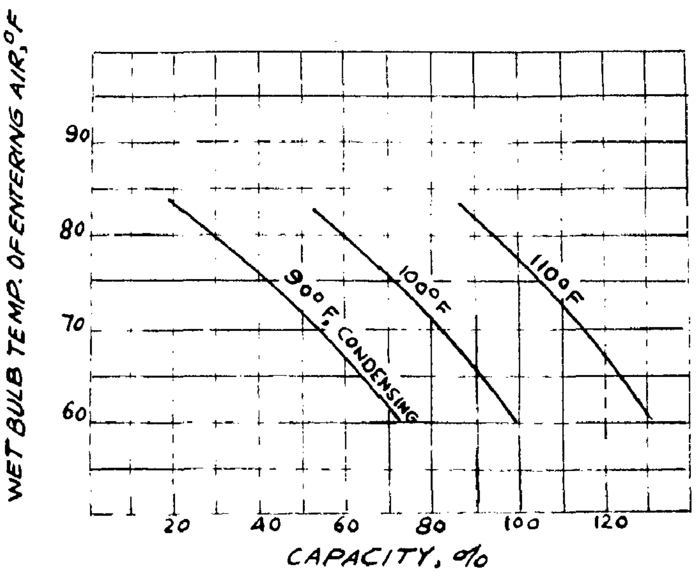


FIG. 3.2

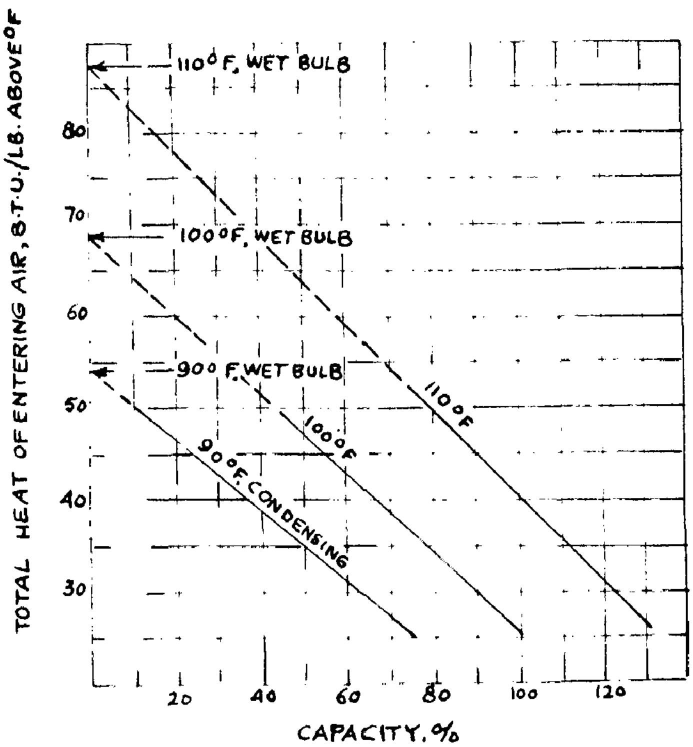


TABLE 3.3.

CHANGE IN HEAT TRANSFER WITH CHANGING CONDENSING TEMPERATURE

Run No.	Entering d.b. °F	Entering w.b. °F	Air quantity cfm	water recirculated gpm	Condensing temp. °F	Percentage of U_1 value
1	97.1	78.05	3570	17.4	90.3	88.3
2	79.95	77.85	3730	17.5	100.0	94.4
3	80.75	78.22	3836	17.8	110.0	100.0

3.2.4. Rating Curves:

By selecting a set of different wet bulb temperatures the corresponding total heat capacity can be computed with the help of the wet bulb m..t.d. and U_1 , keeping the condensing temperature constant. Plotting wet bulb temperatures versus heat capacity in number of curves could be obtained for different constant condensing temperatures. These curves are shown in Fig. 3.1.

An improvement in this usual type of rating curves is suggested by James. Plotting capacity versus total heat of entering air tends to straighten out the curved line relationship obtained in the Fig. 3.1. Such type of rating curves are shown in fig. 3.2.

A series of tests have indicated that a straight line could be drawn through the test points at any fixed condensing temperature when the total heat of the air is plotted against the capacity. Such a straight line intersects the zero capacity ordinate at a point where the entering air total heat corresponds to a wet bulb air temperature equal to condensing temperature.

3.3. THOMASON'S ANALY-SIS:

3.3.1. Thermal Coefficient of heat Transfer:

The overall coefficient of heat transfer U_1, depends upon the various intervening resistances in the path of heat transfer, such as those due to the refrigerant film , metal wall, water film and air film.

For R_r the resistance of the refrigerant film only enough data are available. Kratz with experiments on horizontal shell and tube type evaporators found out the average film conductance f_r [21] for condensing Ammonia to be 1635 Btu/hr/^oF /sq.ft. Sherwood also with shell and tube type of ammonia condensers for condensing and desuperheating found a value of F_r = 950.

Sherwood and Holland with double pipe ammonia condensers including desuperheating and condensing obtained experimental data which were converted by a graphical analysis by Thomason to film conductance from which f_r = 600 was obtained.

In the absence of more specific data it seems desirable to use some average value for calculation purposes. Sherwood's value of F_r = 950 might be considered suitable for this purpose.

Table 3.4 shows the values obtained by different investigators for F_r.

TABLE 3.4.

APPROXIMATE HEAT TRANSFER COEFFICIENT F_r BETWEEN CONDENSING REFRIGERANT AND TUBE WALL.

Condensing Reft.	F_r, Btu/hr/^oF/ sq.ft.	SOURCE
Ammonia	950	Sherwood (21)
Freon 12	250 160-300	Hull (11) James
Carbon Tetrachloride	300	Hull
Dichloro ethylene	350	Hull

The thermal conductivities for materials used in condensers are so large that the resulting resistances become negligible and may be neglected in most cases. Thermal conductivities for steel and copper are approximately 360 and 2500 Btu/hr/^oF / sq.ft./in respectively.

The film conductance from the tube surface through the water film appears to be influenced by the rate of evaporation , air rate, water film thickness, or rate of spray water Table 3.5. shows the results ,Thomson has obtained from an evaporative condenser.

Table 3.5. Effect of Water Film Thickness and Air Velocity upon the Water film heat transfer Coefficient f_w

Water rate, gal./min /sq.ft. of projected area.	Film thickness	Air Velocity , ft/ min. 100	200	300	400
		f_w - Btu/hr/°F/sq.ft.			
7.45	0.015	870	950	---	----
3.33	0.0118	640	700	760	820
1.52	0.0091	450	500	530	580
0.62	0.0067	300	350	390	430

Thomson also noticed that although heat transfer rate increased as the water film thickness increases while the rate of evaporation decreased with the increasing film thickness.

The data obtained for f_w from the 'trickle' type of coolers can be successfully used for the evaporative condensers. The water is supplied by perforated water distributing pipes placed above the banks of the coil. McAdams[15] gives the emperical equation for the film conductance for trickle coolers.

$$f_w = 65 \left\{ \frac{W}{2 D L_p} \right\}^{1/3} \quad .....(3.1)$$

The equation given by McAdams can be reduced to the following form :

$$f_w = 408 \ (W')^{\frac{1}{3}} \qquad \ldots.(3.2)$$

Where W' is the rate of water flow/sq.ft of the projected area in Gallons per minute.

Film conductances f_w for various water rates have been computed by means of the equation(3.2.)and are given in Table 3.6 .

TABLE 3.6.

HEAT TRANSFER COEFFICIENTS BETWEEN PIPE SURFACE AND WATER FILM FOR VARIOUS FLOW RATES.

W'	f_w	W'	f_w	W'	f_w
0.5	325	2.5	555	4.5	675
1.0	407	3.0	591	5.0	700
1.5	470	3.5	622	5.5	722
2.0	516	4.0	650	6	743

It may be noted that the values of f_w given in Table 3.6 agree closely with the obtained ~~values~~ by Thomson for low air rates, for high air rates the values of f_w found out by Thomson are higher than those shown in Table 3.6. This may be explained by greater turbulence obtained with higher air flow rates.

If the water is made to flow turbulently over the coils a higher value of f_w may be expected then those computed from McAdams equation. This may be accomplished by spraying the water from a nozzle rather than water drip.

In case of the finned surfaces the resistance to heat flow offered, by the tube wall and water film wetting the surface can be expressed approximately by a single equation as given by Ashve guide[2] .

$$R = \frac{S + \delta}{f_c \left(2/m \cdot \tanh(ml_1) + S \right) A_f} \qquad \ldots(3.3)$$

And the total rate of heat transfer in Btu/hr in terms of R is expressed by the following equation

$$q = \frac{(t_{si} - t_w)}{R} \qquad \ldots.(3.4)$$

Equation (3.3.) can be simplified by the following considerations :

1. The numerical magnitude of the product ml_1 for evaporative condensers is greater than 2.3, therefore $\tanh ml_1 = ml_1$

2. The thickness of the fins is small as compared with the spacing , thus $S + \delta = S$

3. The surface area A_f is approximately the inner pipe area A_i .

With these approximations equation 3.3. may be simplified

to ,

$$R = \frac{S}{f_w \left(2\, l_1 + S \right) A_1} \qquad \ldots.(3.5)$$

We have been always considering the resistances per sq.ft. of the ~~xxx~~ outside area, the equation (3.5)may be reduced to:

$$R_p = \frac{S}{f_w \left(2\, l_1 + S \right) \frac{A_1}{Ao}} \qquad \ldots(3.6)$$

Where R_p is the resistance per unit outside area.

The values $R_p\, f_w$, based on equation (3.6) are given in Table 3.7 for several finned surfaces. In order to find R_p , divide values of $R_p\, f_w$ from table 3.7 by f_w, as selected from table 3.6.

TABLE 3.7

VALUES OF THE DIMENSIONLESS MODULUS $R_p\, f_w$ FOR SEVERAL FINNED SURRACES OF AN EVAPORATIVE CONDENSER$

Nominal dia. of standard pipe, in.	Spacing of fins, in.	Length of fins, in ¼	½	3/4	1
3/8	1/4	1.67	2.18	2.56	3.16
	1/2	1.62	2.05	2.42	2.99
1/2	1/4	1.61	1.99	2.38	2.77
	1/2	1.55	1.88	2.25	2.64

The convective coefficient of heat transfer f_c , is a function of both the outside diameter of the condenser pipes and the air rate. Table 3.8 was computed from the equation 3.7 given by McAdams for air flow past 10 rows of staggered pipes.

$$f_c = 0.133\ c_p\ G_{max.}^{0.6} / D^{0.4} \qquad .....(3.7)$$

TABLE 3.8

VARIATION OF THE CONVECTIVE HEAT TRANSFER FACTOR f_c for BARE DRY PIPE, STAGGARD 10 ROWS DEEP WITH AIR RATE AND PIPE DIAMETER

Air rate, lbs/hr / sq.ft. of min. free crossection of condenser.	Pipe diameter, inches.			
	3/8	1/2	3/4	1
1000				4.7
1500			6.8	6.2
2000		9.2	8.0	7.4
2500	11.0	10.0	9.1	8.4
3000	12.1	11.0	10.2	9.4
3500	13.5	12.3	11.3	10.4
4000	14.5	13.2	12.1	11.4

For the other arrangements of staggared condenser piping, the values of Table 3.8 may be multiplied by the coefficients shown in Table 3.9.

TABLE 3.9

COEFFICIENTS TO BE USED WITH VALUES OF TABLE (3.8)

Rows deep	2	3	4	5	6	7	8	9	10
Coefficients	0.7	0.82	0.87	0.92	0.94	0.96	0.97	0.99	1.00

The convective heat transfer factor f_c for dry finned surfaces are slightly lower than those for bare pipes . McAdams gives an emperical equation for several types of finned surface namely

$$f_c = 1/3.3 \left(\frac{\mu_f \, G_{max}^m}{Z_p} \right)^{1/2} \qquad ....(3.8)$$

Table 3.10 has been prepared by the above equation for several finned surfaces;

TABLE 3.10

CONVECTIVE COEFFICIENT OF HEAT TRANSFER f_c , FOR SEVERAL NON CRIMPED DRY FINNED SURFACES, (Values of f_c are given in Btu/hr/°F/ sq.ft. for air at 90°F. Fin spacing is 1/4 inch.)

Nominal dia. of standard pipe, in.	Fin length inches	Air rate in lbs/hr/ft² of minimum free area.		
		2000	3000	4000
3/8	0.5	8.3	10.0	11.6
	1.0	6.4	7.9	9.1
1/2	0.5	8.1	9.8	11.4
	1.0	6.3	7.7	8.9
3/4	0.5	7.8	9.4	10.9
	1.0	6.1	7.5	8.9

f_c , for the above table are based on dry surfaces, but it is believed that they will hold approximately also for a wetted surface.

3.3.2. <u>Graphical Analysis of Evaporative Condenser Performance:</u>

The graphical method has been found to be very useful and simple in determining the performance of an evaporative condenser.

The calculated resistances are plotted on the abcissa and the temperatures at various points through the heat exchanger on the ordinate. For illustrating this method the data of the condenser designed in Chapter IV are taken.

:Condenser Data:

Compressor rating to which condenser is to be connected	75 Tons
Condensing Temperature	103 °F
Suction Temperature	40 °F
Wet Bulb temperature of air	80 °F
Spray water temperature	97 °F
Air handled per hour	83,500 lbs.
Surface Area	1283 Sq.ft.
Pipe size for the coils	3/8 inches std. nominal size.
Rate of water flow	0.7 gpm/sq.ft. of projected area.
f_c value taken	12 Btu/hr/sq.ft./°F
U , overall coeff. of heat transfer	133 -do-

f_r = 300 Btu/hr/°F/sq.ft. f_w = 364 Btu/hr/°F/sq.ft. & 1 =0.091" inch.

SOLUTION:

$$F_1 = \frac{G \quad (t_w + t_1)(1 - e^{-M})}{145 A_o} \qquad \ldots(2.18)$$

$$M = \frac{f_c A_o}{0.24 G}$$

substituting the value of f_c , A_o and G

$$M = \frac{12 \times 1}{65 \times 0.24} = 0.77$$

t_w , the spray water temperature as determined = 97 °F

substituting the values of G , A_o t_w and t_1 in the equation(2.18) we get ,

$$F_1 = \frac{83500 \quad (97 + 80)}{145 \times 1285} (1 - e^{-M})$$

put M = 0.77

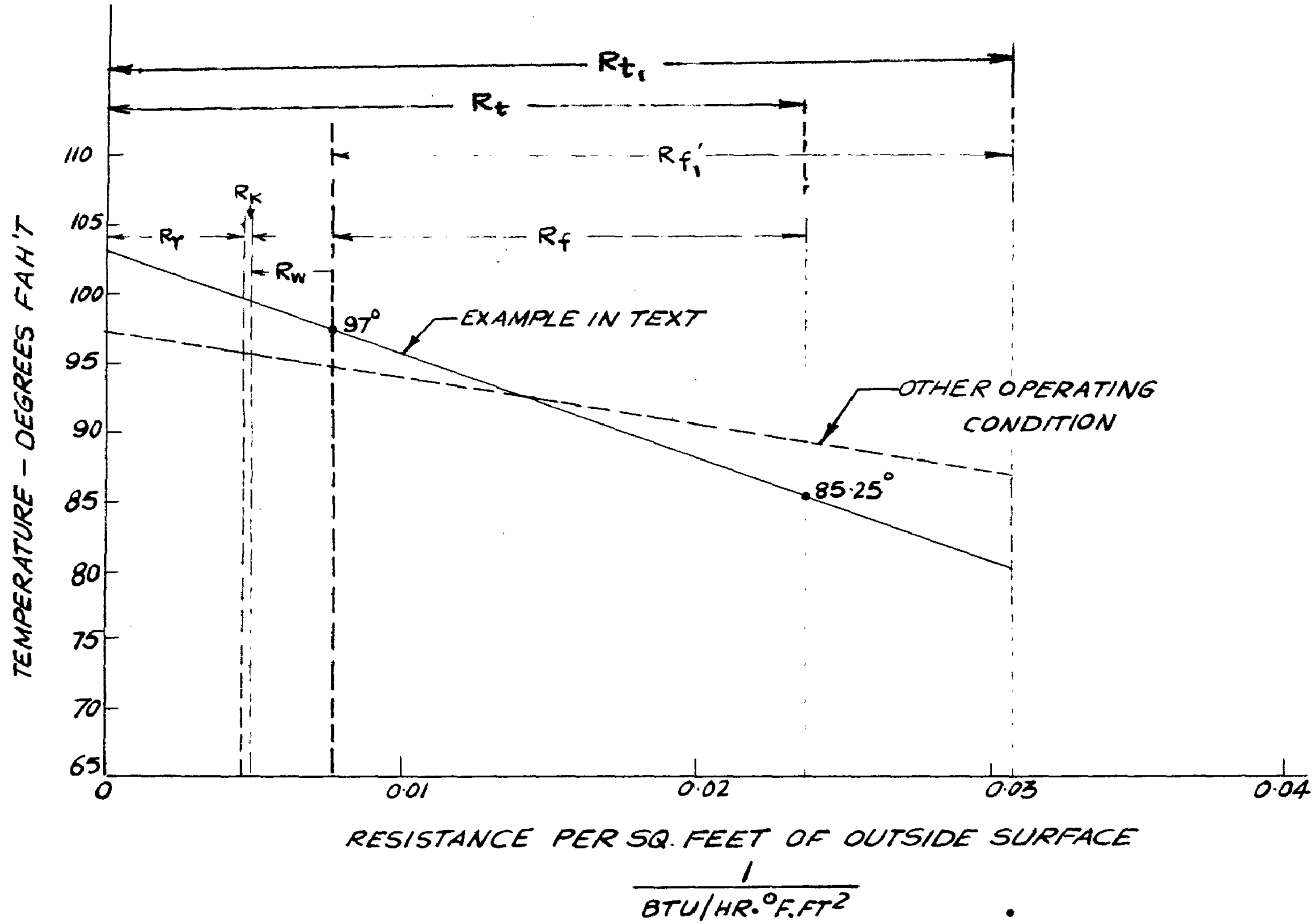
$$= \frac{83500 \quad (97 + 80)(1 - e^{-0.77})}{145 \times 1285}$$

$$= 42.7$$

Therefore $R_{f_1} = \frac{1}{F_1} = \frac{1}{42.7}$

$$R_{f_1} = 0.0234$$

FIG.3.3



Rt1
Rt
Rf1'
Rk
Rr
Rf
Rw
97°
EXAMPLE IN TEXT
OTHER OPERATING CONDITION
85.25°
TEMPERATURE - DEGREES FAH'T
110
105
100
95
90
85
80
75
70
65
0
0.01
0.02
0.03
0.04
RESISTANCE PER SQ. FEET OF OUTSIDE SURFACE
1 / BTU/HR.°F.FT2

The various resistances may therefore be calculated.

$$\text{Now} \quad R_r = \frac{A_o}{A_i f_r} = \frac{1.37}{300} = 0.00457$$

$$R_m = \frac{1}{f_r} = \frac{0.091}{360} = 0.00025$$

$$R_w = \frac{1}{f_w} = \frac{1}{364} = 0.00275$$

$$R_{f_1} = \frac{1}{f_1} = \frac{1}{42.7} = 0.0234$$

R_{t_1}, the total resistance for the condensing refrigerant to the air flowing outside, is the sum of all the resistances, calculated above

$$= 0.03097$$

The total heat flow therefore will be,

$$q = \frac{1285\ (103 - 80)}{0.03097} = 960,000 \text{ Btu/hr.}$$

Fig. 3.3. shows the graph of the magnitude of the resistances and the temperature gradient. The temperature gradient is obtained by drawing a straight line from a point corresponding to the condensing refrigerant temperature (103 ^{o}F), to the point of intersection of the entering air wet bulb temperature (80 ^{o}F) and the total resistant R_{t_1}. The temperature of

the spray water can be read from the graph directly and is 97 °F as given in the problem.

It may be noted here that the resistance from the refrigerant film to the water film is smaller than that of the resistance from the water film to air. These greatly unbalanced resistances which appear to exist lead to erroneous conclusions, however, since the air wet bulb temperature changes in its path through the condenser. Another equation (2.25) may therefore be used for mass and heat transfer factor. The driving force for the total heat transfer from condenser wetted surface is taken as the difference between the spray water temperature and the average air wet bulb temperature. The leaving air wet bulb temperature may be determined corresponging to h_2 computed from the following equation used :

$$q = G\,(h_2 - h_1) \qquad ..(2.34)$$

$$\frac{q}{G} + h_1 = h_2$$

The leaving air wet bulb temperature is 90.5 °F, an-d the average wet bulb temperature of entering and leaving air will be 85.25 °F.

Substituting the known quantities in the equation

$$F = \frac{G}{145\,A_o} \quad \frac{(t_w + t_1)\;(t_w - t_1)}{(t_w - t_{1\,ave.})}\,(1 - e^{-M}) \qquad ..(2.25)$$

$$F = \frac{65}{145} \quad \frac{(97 + 80)(97 - 80)}{(97 - 85.25)} \quad (0.538)$$

$$= 62.$$

Therefore $R_f = 1/F = 1/62 = 0.01615$

The various resistances known are :

$R_r = 0.00475$

$R_m = 0.00025$

$R_w = 0.00275$

$R_f = 0.01615$

The total resistance $R_t = 0.02372$. U, the overall coefficient of heat transfer from refrigerant to air stream

$$= 1/R_t = 1/0.02372 = 42.2 \text{ Btu/hr/sq.ft/}^oF$$

$$U = 42.2 \text{ Btu/hr/sq.ft./}^oF$$

Therefore, the total heat transfer from the condenser will be given by

$$q = UA_o (t_r - t_{1\ ave.})$$

$$= 42.2 (103 - 85.2) \times 1285$$

$$= 965,000 \text{ Btu/hr.}$$

This result checks closely with that previously obtained.

If a straight line is drawn joining the condensing temperature to the intersection of the average wet bulb temperature and the equivalent resistance R_t ; It will be noticed that this line lies on the previously drawn temperature gradient line. It may also be seen from Fig. (3.3) that the apparant unbalanced condition for heat transfer has been altered by using R_f instead of R_{f_1} for the resistance to mass and heat transfer.

The various resistances for an evaporative condenser, established once, will not change with the changing operating conditions such as load or entering air wet bulb temperature, provided the water flow and air flow rates remain constant. In order to determine the performance of the evaporative condenser for any changed conditions of the entering air wet bulb temperature and the refrigerant temperature, another straight line should be drawn for the new refrigerant temperature to the intersection of previously drawn total resistance line R_{t_1} with the new wet bulb temperature of entering air, Such a line is shown in dotted in the fig. (3.3)

The variables such as the spray water temperature, average air wet bulb temperature and the pipe wall temperature can be obtained from the graph immediately by noting the intersection of the temperature gradient line with the various resistances; The total heat from the condensers may also be computed, from the known total resistance and the difference •

between entering air wet bulb temperature and refrigerant temperature.

3.4. WILE'S ANALYSIS:

Wile has presented a method of analysing the test data so as to obtain two empirical constants U and K, from which the rating data can be obtained for any desired operating conditions. The method of analysis also provides design data that are of value in selecting the most economical combination of air flow, tube surface area and water flow.

The test apparatus used by Wile was of conventional type except that a unique method of controlling heat input "load" and permitted variation of superheat of the refrigerant vapour. The main feature of the apparatus was "steam to refrigerant" heat exchanger.

3.4.1. Determination of Empirical Constants:

The readings noted from the test were as follows :

1. Condensing refrigerant temperature.
2. Entering wet bulb temperature.
3. Heat input to condenser.
4. Entering dry bulb temperature.
5. Leaving wet bulb temperature.
6. Leaving dry bulb temperature.
7. Air flow rate (for confirming purposes)

FIG. 3.4

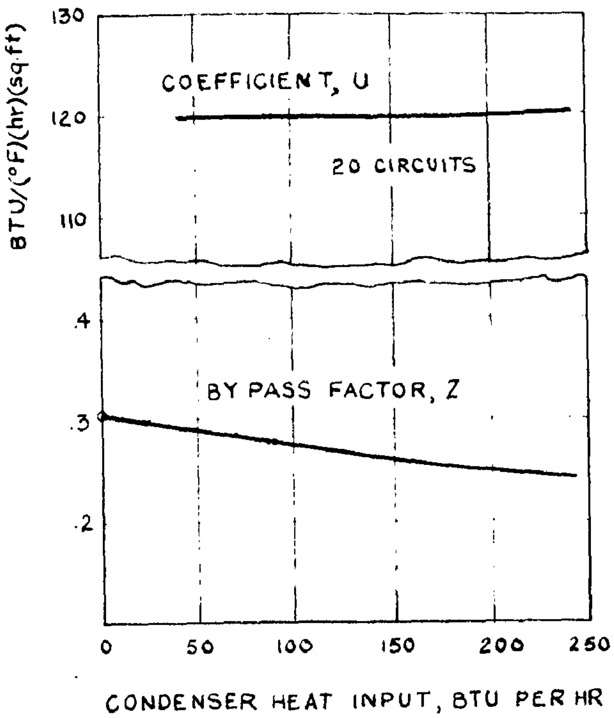
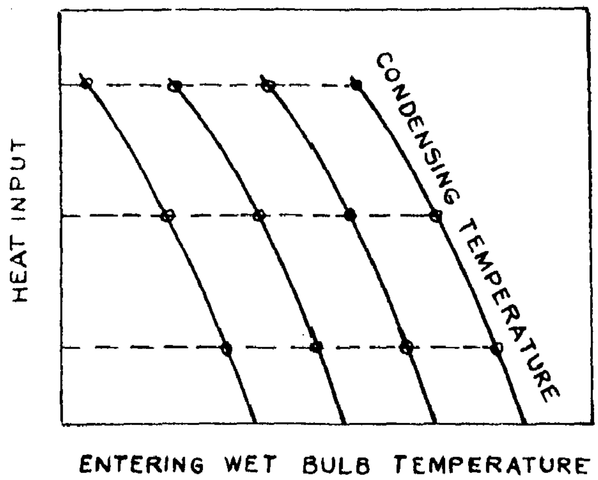

130
120
110
BTU/(°F)(hr)(sq.ft)
COEFFICIENT, U
20 CIRCUITS
.4
.3
.2
BY PASS FACTOR, Z
0
50
100
150
200
250
CONDENSER HEAT INPUT, BTU PER HR

FIG. 3.5


HEAT INPUT
CONDENSING TEMPERATURE
ENTERING WET BULB TEMPERATURE

It may be noted that a slight error in the test data such as leaving air wet bulb temperature may cause a considerable error in determining the wetted surface temperature t_w and the bypass factor S.

The method of analysing the test data is as follows :

1. Plot the inlet and outlet conditions of air on the psychrometric chart, extend the line joining these points to intersect the saturation line. The temperature at the intersection point will represent the wetted surface temperature t_w.

2. Determine the values h_1 , h_2 and h_w either from Tables or chart, Calculate S , the bypass factor by the following equation

$$S = \frac{h_w - h_2}{h_w - h_1} \qquad \ldots(2.33)$$

3. Determine the air flow rate G from the equation:

$$(h_2 - h_1)\, G = q \qquad \ldots(2.34)$$

4. Determine the various values of S corresponding to different values of heat input. Plot a graph between heat input and bypass factor S. Draw a smooth curve passing through the plotted values of S , as shown in the Fig.(3.4) . The adjusted value S' may be obtained from the smooth curve for any particular heat input . Take different values of S' from the curve.

5. Substitute Z' in the equation

$$h'_w = h_1 + \frac{q}{G(1 - Z')}$$

h'_w can be calculated . Determine corresponding values of t'_w , the adjusted wetted surface temperature.

6. Determine value of adjusted U' by substituting t'_s in the equation

$$U' = \frac{q}{A_o(t_r - t_w)}$$

7. Plot U' versus load as shown in Fig.(3.4) If the datas are determined accurately, the graph should follow a smooth curve.

3.4.2. Rating Curves:

These two curves define the condenser performance for any combination of entering air wet bulb temperature and condensing refrigerant temperature. The method of preparing rating curves is as follows :

1. Assume any desired heat input and from the graph find out the corresponding values of U and Z.

2. Determine t_w from the equation

$$t_w = t_r - q/UA_o$$

find out the corresponding value of the enthalpy h_w .

3. Determine the entering wet bulb temperature t_1 from the enthalpy of h_1 in the equation.

$$h_1 = h_w - q/G(1 - z)$$

4. By selecting different values of condensing temperature corresponding entering air wet bulb temperatures are determined for a fixed assumed value of heat input. At different values of heat input, different sets of values for condensing and entering air wet bulb temperature can be found out. The values are plotted as shown in Fig.(3.5).

3.4.3. Effect of Air Velocity Variation:

Z, the bypass factor should increase with the increase of G , the air flow rate. But the test results show that Z remained practically constant when G was increased from 2600 to 3100 cfm. This may be expla-ined by the increased wetted surface area due to the greater turbulence created as a result of increasedvelocity. At increased velocity of air the water droplets in the tube bundle will form additional wetted surface in contact with air stream. The wetted surface is considerably greater area then the actual area of condenser tubes.

3.4.4. Effect of Loading on Bypass Factor Z and Coefficient U.

The graph in the Fig (3.4) shows that the bypass factor Z is decreased with the increase in loading. On the otherhand equation $Z = e^{-(fs_1 A_0 /G)}$ .....(2.36) shows that the bypass factor Z is independent of load if physical relations remainedconstant. Air quantity slightly varies with load at constant fan speed. Moreover according to tests Z remained practically unchanged with the increased air flow rates.

An increase in loading will result an increase in air and water temperature along with an increase in moisture content of the leaving air. This will cause a change in the air thermal conductivity and viscosity and will tend to increase the heat transfer coefficient from water surface to air. This will reduce the bypass factor. At increased water temperatures viscosity and surface tension of water is lowered resulting in the formation of more number of droplets thus giving rise to additional wetted surface. This further reduces the bypass factor. These factors are small but they combine in the same direction. U is only slightly affected by the increase in loading . This is evident from Fig.(3.4).

3.4.5. Variation of Water Temperature within the Tube Bundle.

The water surface temperature rises as it enters the tube bundle to a maximum near the centre and then decreases towards the bottom of the bundle. Since cooling of the water takes place

FIG 3·6

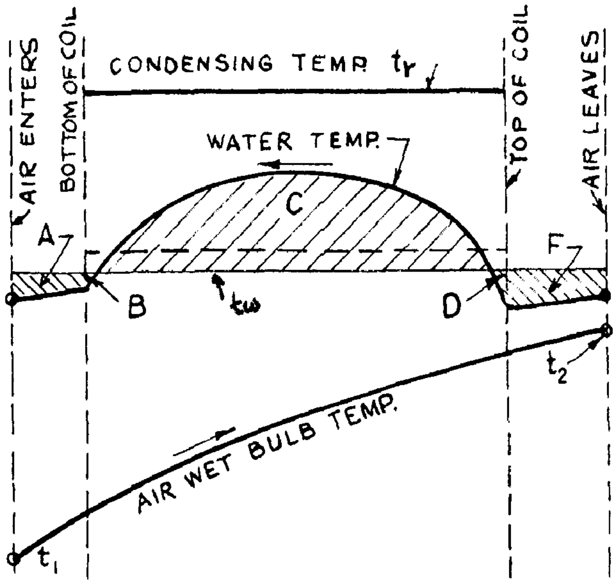
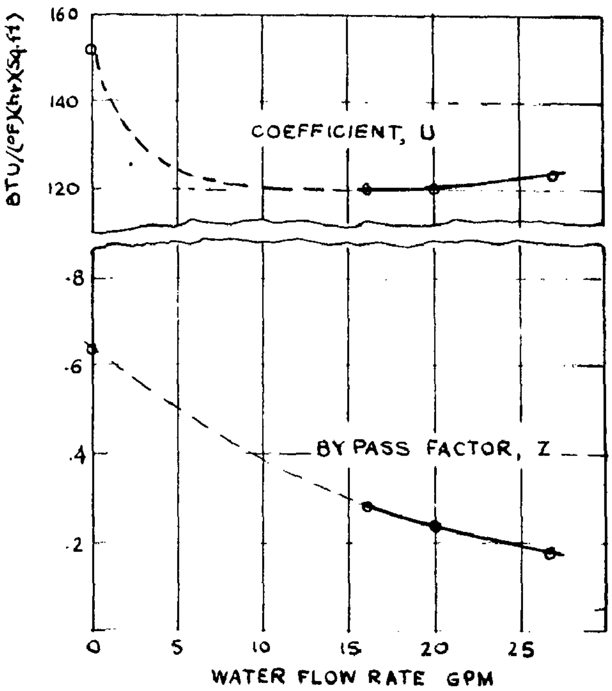

CONDENSING TEMP. t_r
AIR ENTERS
BOTTOM OF COIL
TOP OF COIL
AIR LEAVES
WATER TEMP.
C
A
F
B
t_w
D
t_2
AIR WET BULB TEMP.
t_1

FIG 3·7


BTU/(°F)(hr)(Sq.ft)
160
140
120
COEFFICIENT, U
·8
·6
·4
·2
BY PASS FACTOR, Z
0
5
10
15
20
25
WATER FLOW RATE GPM

as it falls from the tube bundle and additional cooling occurs in the sprays above the tube bundle. Further more a considerable amount of wetted surface exists in the eliminator at the discharge of the condenser which will also cool the spray water. In the Figure (3.6), the solid line t_w is the theoretical wetted surface temperature defined by the equation

$$h_1 = h_w - q/G(1 - Z)$$

Its position is such that

$$C = A + B + D + F$$

where they refer the respective areas in the Fig. (3.6)

It follows therefore, that

$$C > B + D$$

And the dotted line representing t_w defined in equation

$$U = q/A_0(t_r - t_w)$$

must lie above the solid line with the result that U and F_r should be some what larger than indicated by the equation ~~mentioned~~ above. Water temperatures were measured during several test runs by inserting the thermometer to the various depths in the tube bundle. A small perforated cup was provided to catch a part of the water flow without contacting the tube surfaces. At a heat input of 2,56,000 Btu/hr Wile noted a rise of 10 ^oF in the centre of the bundle with the inlet and outlet water temperatures. This shows that the wetted surface temperature should be higher than the temperature of the recirculated water. This difference is of the order of 2 $^{\circ}F$ for maximum loading.

3.4.6. Effect of Varying Water Flow Rates:

The effect of varying water flow rate is shown in graphs of Fig. (3.7). The dotted portion do not indicate actual readings but are showing the tendency when the dry performance is approached. By increasing the water flow rate, bypass factor Z is decreased and the coefficient U is increased. Both combine to improve the performance of the condenser.

This is explained by the increased turbulence and the greater wetted surface created, thus reducing the bypass factor Z. Since f_r is practically constant U depends on f_w with the increased water rates, turbulence in the water film increases f_w and with it the coefficient U.

3.4.7. Effect of Refrigerant Superheat on Heat Transfer and Scale Formation:

It is generally believed that superheat of refrigerant in the condenser causes rapid accumulation of scale. This is said to be due to the higher temperature attained by the coil wall. However, according to Wile superheat has no effect on scale formation. He carried out several tests & found that the spperheat of the refrigerant had no affect on scale formation.

CHAPTER IV

THE DESIGN OF EVAPORATIVE CONDENSER.

The graphical method for determining the spray water temperature is found very useful in the design work.

The minimum values of spray water temperature, surface area and the air quantity that can theoretically be used in an evaporative condenser are also discussed. Selection of air quantity and coil area are dealt.

Finally, there is an illustrative design of the evaporative condenser.

SYMBOLS

A_o	Outside surface area of the coil in sq.ft.
A_i	Inside surface area of the coil in sq.ft.
F_r	Coefficient of heat transfer of the refrigerant film Btu/ deg. F/ hr/ sq.ft. of internal surface.
l	Thickness of pipe in inch.
f_m	Coefficient of heat transfer through the metal wall of the pipe Btu/ deg.F/ hr/sq.ft/ inch of thickness.
f_w	Coefficient of heat transfer of the water film from the metal wall to the wetted surface Btu/hr/deg.F/sq.ft.
B	Ratio of the outside to inside surface of the coil.
q	Total heat surrendered by the condensing refrigerant that is , the total heat capacity of the condenser in Btu/hr .
f_c	Coefficient of heat transfer between air and water (Air film coefficient) Btu/hr/sq.ft./deg.F.
Z_2	A quantity that depends on M, defined by equation 4.1.
M	A quantity depended by equation 4.1.
G	Weight of air circulated through the condenser lbs/hr.
N	A quantity defined by equation 4.8
t_w	Temperature of spray water in deg. F.
t_r	Temperature of the condensing refrigerant in deg.f
t_1	Wet bulb temperature of the entering air.
t_{si}	Temperature of inside Surface of metal tube deg. F.
t_{so}	Temperature of outside surface of metal tube ~~of~~ deg. F.
S_A	Scale of abcissa, inch per deg. F.
S_o	Scale of ordinate, in per Btu.
c_p	Specific heat of Air Btu/lb.
U	Overall coefficient of heat transfer from refrigerant vapour to wetted surface Btu/hr/sq.ft./deg.F.

h_1 , h_2 Enthalpy of air entering condenser and leaving condenser Btu/lb. respectively.

h_w Enthalpy of air saturated at a ~~wet bulb~~ temperature equal to the temperature of the spray water Btu/lb.

θ A quantity defined by equation 4.15.

D Diameter of the coil.(outside) in inch.

H Head in feet of water.

CHAPTER IV

THE DESIGN OF EVAPORATIVE CONDENSERS.

4.1. The heat capacity of a condenser is the amount of heat that can be transferred from the refrigerant to the air. This must always be greater than the tonnage rating of the compressor to which it is to be connected. This is because the total heat rejection by the condenser includes both the heat of compression and the net refrigeration effect. Although a compressor can remove 12000 Btu/hr/ton it rejects to the condenser this amount plus the heat of compression.

In order to obtain the total heat rejection to the condenser, the heat of compression and the refrigerating effect should first be computed.

A " freon " compressor operating between a condensing temperature of 102 ^{o}F. and suction temperature of 40 ^{o}F will reject theoretically, slightly more than 228 Btu/min. to the condenser per ton of refrigeration capacity or 13680 Btu/hr

This will be best understood by representing the process on the P - H chart, shown in fig 4.1

The refrigerating effect = 83 - 31 = 52 Btu/ lb.

Heat rejected to conden-ser = 90 - 31 = 59 Btu/lb.

FIG. 4.1

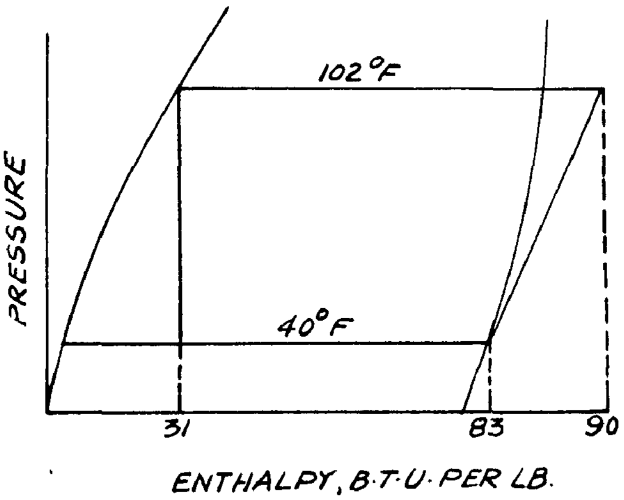
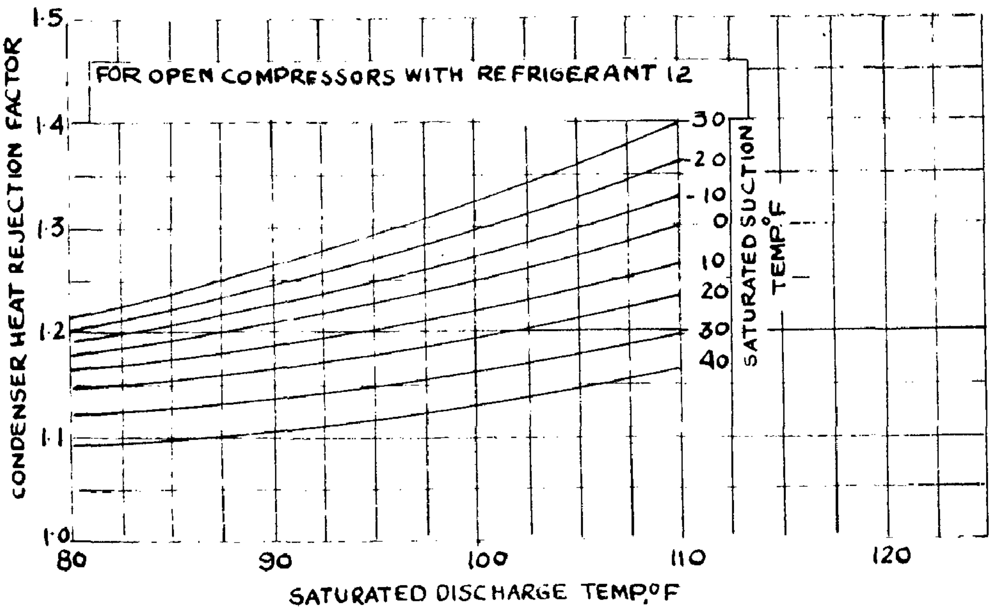

102°F
PRESSURE
40°F
31
83
90
ENTHALPY, B.T.U. PER LB.

FIG. 4.2


FOR OPEN COMPRESSORS WITH REFRIGERANT 12
CONDENSER HEAT REJECTION FACTOR
1.5
1.4
1.3
1.2
1.1
1.0
-30
-20
-10
0
10
20
30
40
SATURATED SUCTION TEMP. °F
80
90
100
110
120
SATURATED DISCHARGE TEMP. °F

Therefore,

$$\text{Heat rejected / Ton} = \frac{59 \times 200}{52}$$

$$= 228 \text{ Btu/min.}$$

Based on the above calculations a number of curves can be plotted on a graph between condenser heat rejection factor as the ordinate and saturated discharge temperature as abscissa for the different constant saturated suction temperatures.

The heat rejection factor is defined as the ratio of the condenser heat rejection divided by the refrigeration effect. The Fig.4.2 showes the curves obtained for open compressors using freon12 as refrigerant.

$$\text{Condenser heat rejection factor} = \frac{\text{Condenser heat rejection}}{\text{Refrigerating effect.}}$$

These curves are very useful in the design and save a good amount of time involved in computing the total heat rejection to the condenser.

4.2. COMPUTATION OF TOTAL HEAT TRANSFERRED:

There is no need for saperately evaluating the transfer of sensible heat and the moisture between air and water. The total heat transferred ~~by~~ to air in evaporative condenser can be easily computed by the following equation.

$$q = G\,(h_w - h_1)\,Z_2 \qquad \ldots\ldots(4.1)$$

Where $Z_2 = (1 - e^{-M})$ and $M = \dfrac{f c A_o}{c_p\, G}$

This is the same equation as that equation 2.13 .

After calculating M for a given set of conditions, the factor Z_2 can be found . The total quantity of heat transferred from the water wetted surface of the coil to air can be computed by means of equation 4.1. Before aquation 4.1 could be used, the spray water temperature must be known in order to find h_w .

Generally the condensing temperature and the initial wet bulb temperature is known and the spray water temperature is to be computed.

4.3. DETERMINATION OF SPRAY WATER TEMPERATURE.

Since the temperature of the condensing refrigerant and the temperature of the spray water may be taken constant throughout the condenser, the temperature of the metal wall may also be considered constant throughout. The temperature of the metal surface must be at a point between the temperature of the condensing refrigerant and the temperature of the spray water.

The heat transferred from the condensing refrigerant to the metal wall is given by :

$$q = f_r (t_r - t_{si}) \frac{A_o}{B} \qquad \ldots.(4.2)$$

And $$q = f_w (t_{so} - t_w) A_o \qquad \ldots.(4.3)$$

In as much as the metal wall of the tube is thin and its conductivity high, the resistance of the metal wall to the flow of heat is negligible. Hence, there will not be any appreciable error in considering the temperature at the outside and inside surfaces of the tube being equal

i.e. $t_{si} = t_{so}$

Eliminating t_{so} and t_{si} in the equation (4.2) and (4.3)

$$\frac{q}{A_o f_w} + t_w = - \frac{qB}{A_o f_r} + t_r$$

Therefore

$$\frac{q}{A_o}\left(\frac{1}{f_w} + \frac{B}{f_r}\right) = (t_r - t_w) \qquad \ldots.(4.4)$$

Let $$\frac{1}{U} = \frac{1}{f_w} + \frac{1}{f_r} \qquad \ldots.(4.5)$$

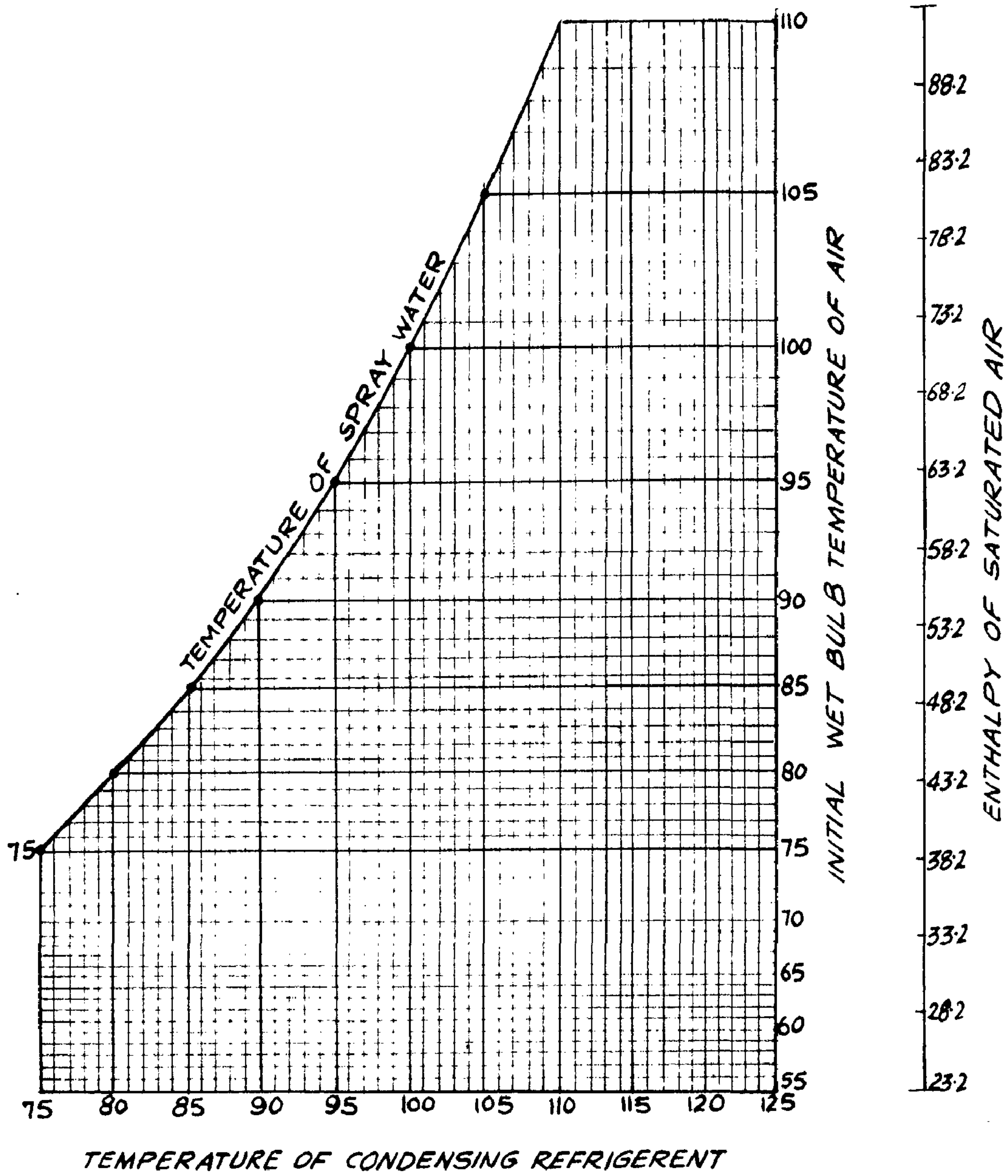
Substituting (4.5) into equation (4.4)

$$q = UA_o (t_r - t_w) \qquad \ldots(4.6)$$

FIG. 4.3

CHART FOR DETERMING TEMPERATURE OF SPRAY WATER IN EVAPORATIVE CONDENSER

[THIS CHART CAN BE USED FOR ANY REFRIGERENT]



Dividing equation (4.1) by equation(4.6)

$$1 = \frac{G(h_w - h_1)\,Z_2}{(t_r - t_w)\,UA_o} \qquad \ldots.(4.7)$$

$$\text{Let} \quad N = \frac{GZ_2}{A_oU} \qquad \ldots..(4.8)$$

$$\text{Therefore,} \quad N = \frac{t_r - t_w}{h_w - h_1} \qquad \ldots.(4.9)$$

For constant value of N, this represents the equation of a straight line on a chart whose abcissa is water temperature and whose ordinate is the enthalpy of air, Figure (4.3) i-s prepared in this way. The enthalpy values are not shown in Figure (4.3) instead, values of wet bulb temperature corresponding to enthalpy of saturated air were used for simplicity in solving problems. A set of values for air dry bulb temperatures are selected and corresponding values of enthalpies of saturated air are determined from the psycrometric chart and are tabulated as in the Table 4.1. The curve in the graph was obtained by plotting values of Enthalpy of saturated air against dry bulb temperature.

TABLE (4.1)

D.B. Temperature	Enthalpy, of saturated air.	D.B. Temperature	Enthalpy of saturated air
55	23.2	90	55.2
60	26.45	95	63.2
65	30.5	100	71.4
70	34.1	105	81.2
75	38.6	110	92.2
80	43.65	115	105.0
85	49.4	120	119.7
		125	126.0

Refering to the skeleton Chart in Fig. 4.4.

$$\frac{CB}{AB} = \tan \theta \qquad .....(4.10)$$

Length of the line CB is given by the relation

$$CB = S_A (t_r - t_w) \qquad .....(4.11)$$

Similarly length of the line AB is

$$AB = S_o (h_w - h_1) \qquad ...(4.12)$$

Dividing equation (4.11 by (4.12)

$$\frac{CB}{AB} = \frac{S_A (t_r - t_w)}{S_o (h_w - h_1)} \qquad ....(4.13)$$

FIG. 4.4

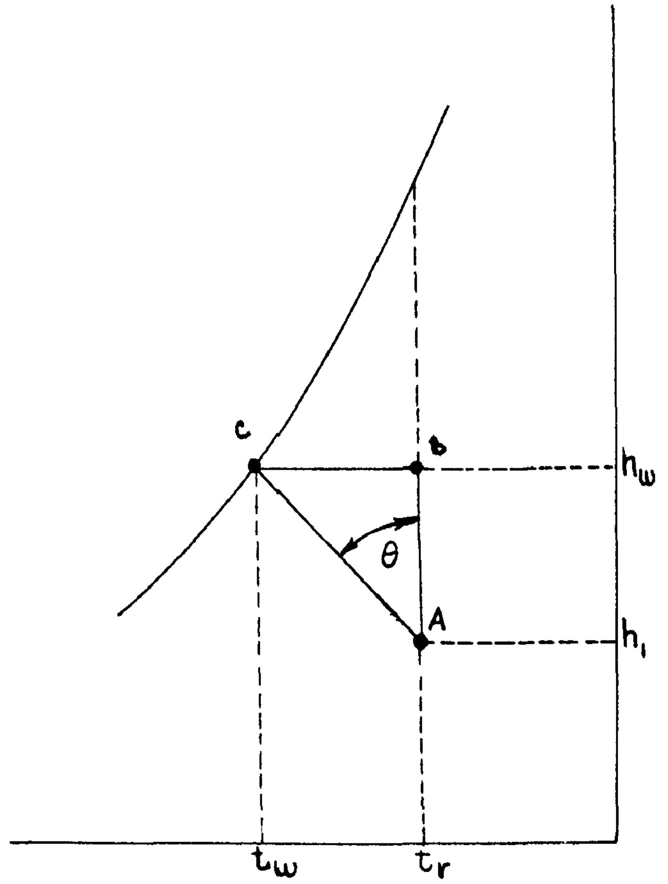
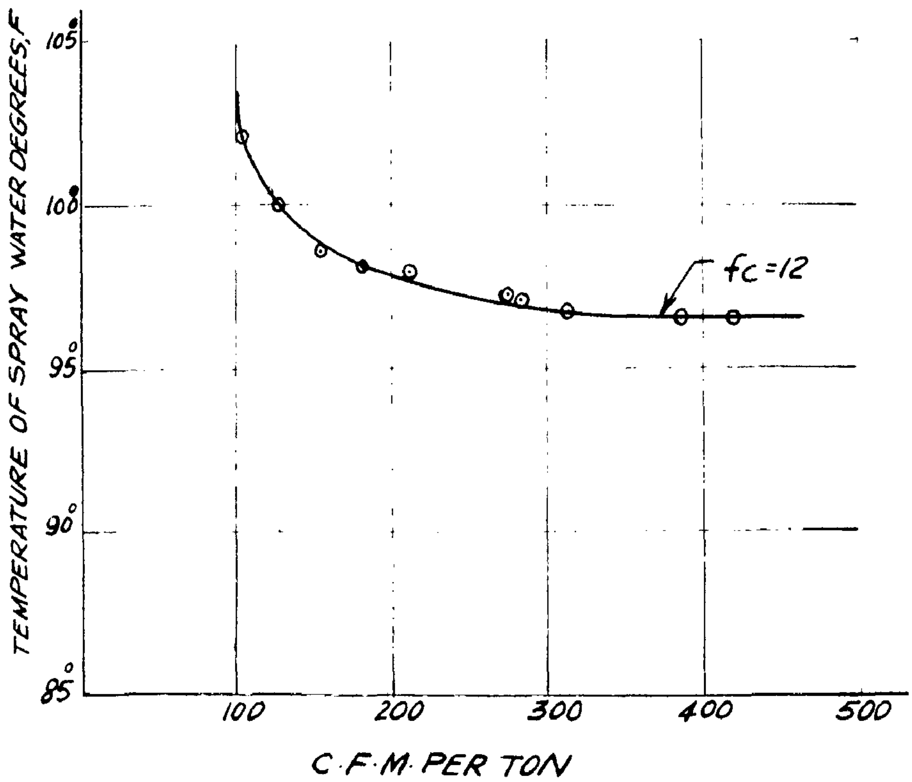


FIG. 4.5

CONDENSING TEMPERATURE = 103 °F
INITIAL WET BULB TEMPERATURE = 80°F
U = 133 B.T.U./HR./SQ.FT./°F



$$\frac{S_o}{S_A} \tan\theta = \frac{t_r - t_w}{h_w - h_1} \qquad \ldots.(4.14)$$

Chart in Fig. 4.3 was drawn so that $\frac{S_o}{S_A} = 1$

$$\tan\theta = \frac{t_r - t_w}{h_w - h_1} \qquad \ldots.(4.15)$$

Equating equations (4.14) and (4.15)

$$N = \operatorname{Tan}\theta \qquad \ldots.(4.16)$$

The chart in figures is prepared on the basis of the method suggested by Goodman[8].

The chart shown in Fig. (4.3) is very useful in design and reduces lot of labour, otherwise, would have involved, in calculations for determining ~~spraying~~ water temperature by solving the equation (4.9) by tiral and error method. With the help of this chart the spray water temperature for any given operating conditions may be readily obtained. This chart is useful for any refrigerant. The method of using this chart is illustrated by the Fig. 4.4. After computing N for any particular condenser, the corresponding angle θ can be calculated. The point A in the Figure is located by the intersection of the vertical line representing the temperature of the condensing refrigerant and the horizontal line representing the initial wet bulb temperature of the air. Through A, draw a straight line at an angle θ with the vertical line. This line intersects the curve at B, at which point the temperature of spray water is read.

4.6. SELECTION OF AIR QUANTITY AND COIL AREA.

When a condenser is designed for a given load, condensing refrigerant temperature and initial wet bulb temperature any desired combination of air quantity and coil area may be selected. As a matter of fact there is a range in which coil area and the air quantity can be chosen. A reduction in the air quantity below a certain limit will result in a tremendous increase in the surface area and any advantage of the decrease in the air quantity is offset by the greatly increased surface area of the coil. Similarly a reduction in the surface area below a certain limit results in an increase in the air quantity that is out of all proportions to the decrease in the surface area. Again any advantage obtained by the decrease in the surface area is totally offset by the greatly increased air quantity.

4.7. THE MINIMUM SURFACE TEMPERATURE AND THE THEORETICAL MAXIMUM CAPACITY OF CONDENSERS:

It shall be noted that the spray water temperature gradually decreases as the air quantity is increased. This has been shown in the graph of the Fig. 4.5, where the spray water temperature is plotted against the air quantity per ton.

At low air quantities the fall in the spray water temperature is rapid while at large air quantities the drop

in the spray water temperature is negligible. The maximum
maximum capacity of a condenser is obtained at the minimum
possible spray water temperature. The minimum spray water temperature will be obtained at the theoretically infinite air quantity.

Although the capacity of a condenser with infinite quantity of air has no practical significance but it is of value in indicating the limitation in ~~quantity~~ capacity for an increase in air volume.

In order to find the maximum possible capacity of a condenser as the air quantity is made infinitely large, the lowest possible spray water temperature must first be determined.

$$N = \frac{G\ Z_2}{AU} \qquad \ldots(4.8)$$

Where $Z_2 = 1 - e^{-M}$ and $M = \cancel{f\ A}$ $f_c\ A/C_pG$. ..

The above equation cannot be used to find the angle θ when $\frac{G}{G} = \infty$, because at this condition it becomes indeterminate.

Substituting the value of Z_2 and M in the equation (4.8) it reduces to:

$$N = \frac{G\left(1 - e^{-f_c A/C_p G}\right)}{AU_0} \qquad \ldots(4.17)$$

$$N = \frac{\left(1 - e^{-f_c A/c_p G}\right)}{AU/G} \qquad \text{....(4.18)}$$

The limiting value of N, when $G = \alpha$ may be found by differentiating the numerator and denominator seperately with respect to G,

$$N = \frac{\left(\frac{f_c A_o}{c_p}\right)\left(e^{-f_c A/c_p G}\right) d(1/G)}{UA_o \cdot d(1/G)} \qquad \text{....(4.19)}$$

The factor $d(1/G)$ is eliminated by cancellation as it appears in both numerator and demoninator.

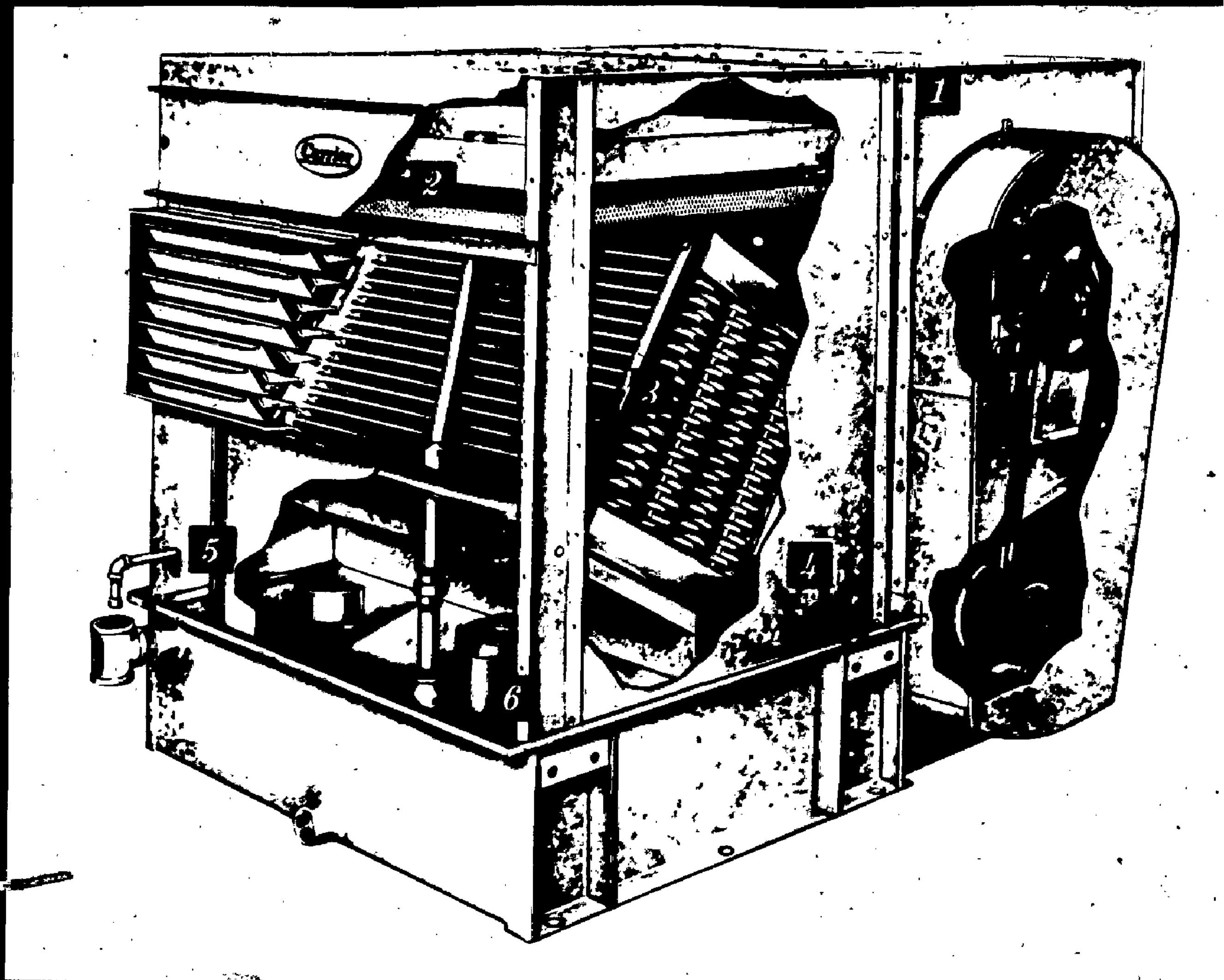
When G = infinity and ~~N =~~ $N = N_1$, the above equation reduces to,

$$N_1 = \frac{f_c}{c_p U}$$

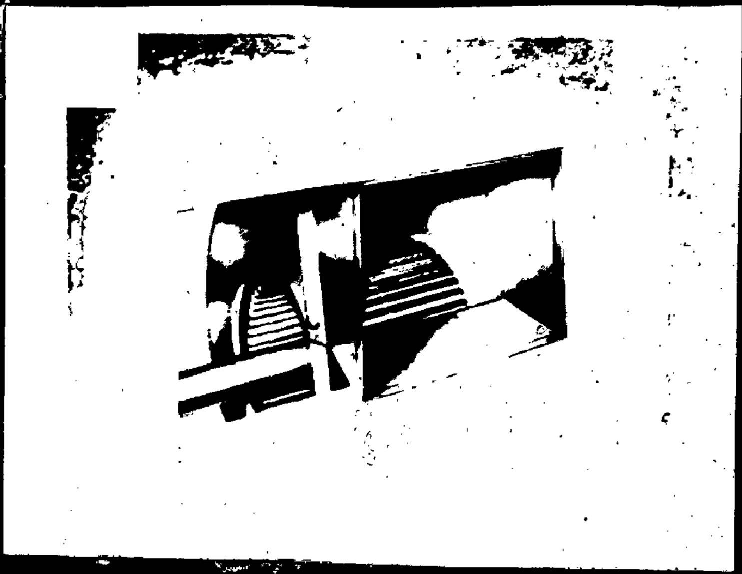
Put $c_p = .24$ for air.

$$N_1 = \frac{f_c}{0.24\ U} \qquad \text{....(4.20)}$$

The above equation can be successfully used for calculating the minimum possible spray water temperature. After computing the minimum spray water temperature the maximum capacity of the condenser per sq.ft. of the surface may be

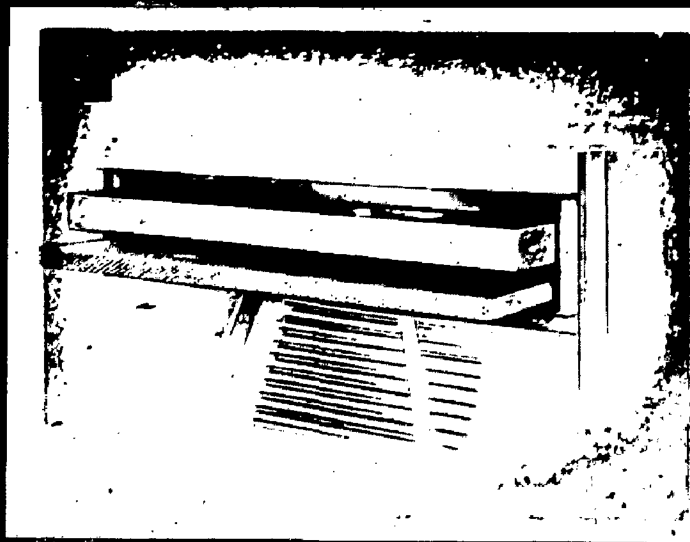


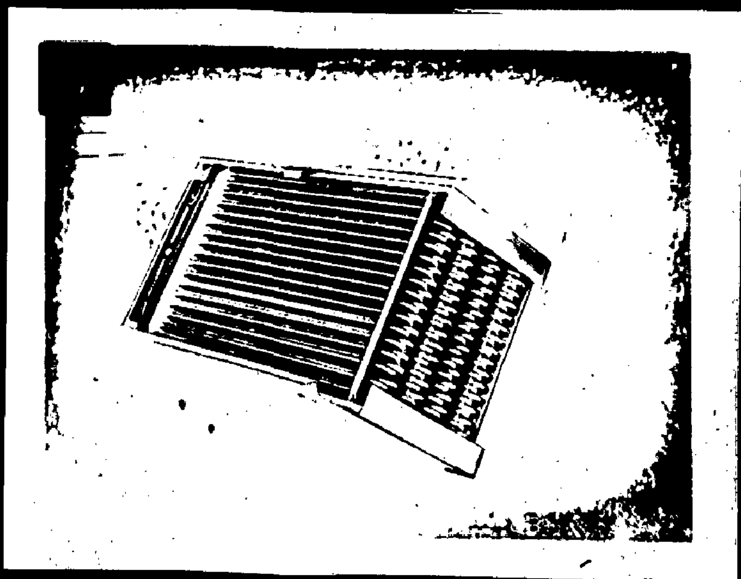
EVAPORATIVE CONDENSERS



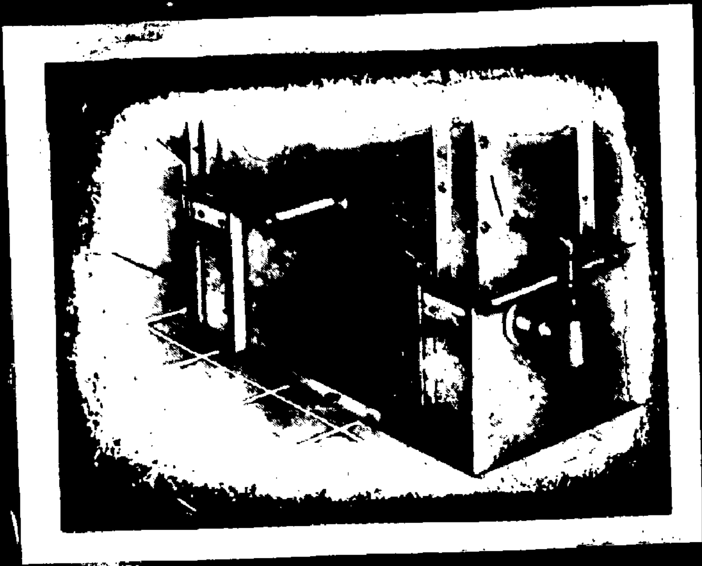
Powerful, low-speed fans draw liberal quantities of air through the wet coils to provide top condensing performance.

The two drawer like water distributing pans meter and break the water into tiny droplets which drench the entire condensing coil



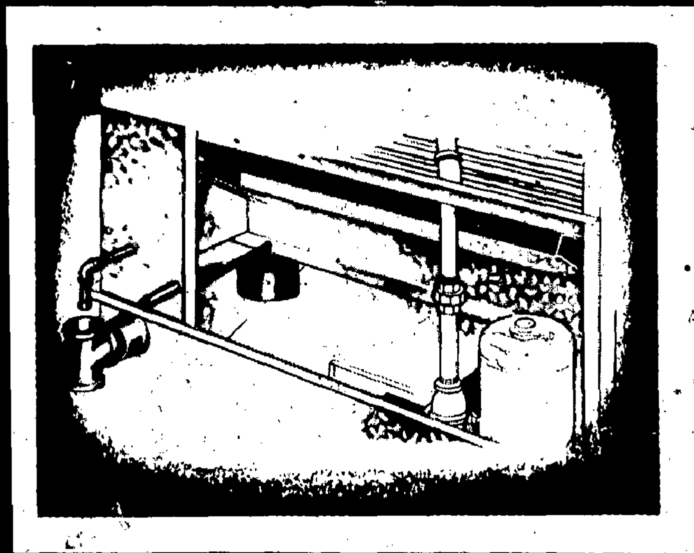


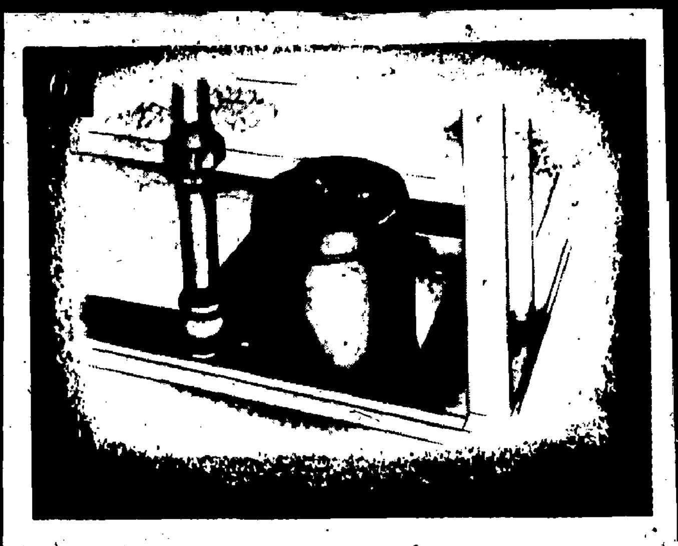
Condensing coils are serpentine formed from single lengths of prime surface steel tubing headered into one or more circuits.



The Unit casing is constructe
throughout of heavy gauge
galvanized sheet steel rigidl
attached to a sturdy steel
frame

In hard water areas, chemical treatment and bleedoff of circulated water will keep the coil free of air restricting mineral encrustation thus maintaining its full rated capacity





A small centrifugaltype pump circulates water from the base pan to the water distributing pans above the condensing coil.

determined by using the equation (4.6)

$$\frac{q}{A_o} = U (t_r - t_w)$$

Conversely, the minimum possible surface per ton which can theoretically be used can easily be computed. The actual amount of the surface that should be used, is of course, greater, but knowledge of theoretical minimum surface is of value and interest.

It is evident ~~from Fig. 4.3~~ that the minimum quantity of the air is required when the surface area is infinite. When surface area is infinite M will be infinite. Correspondingly Z_2 will be equal to 1 for $Z_2 = 1$ the value of $N = 0$, It is clear from Fig. 4.3. that at this condition $t_r = t_w$. In other words for this condition, the temperature of the spray water would be equal to the temperature of the condensing refrigerant.

A knowledge of these theoretical minimum values of cfm and area is of importance when designing a condenser for a given set of conditions. The actual air quantity and area must be greater than these minimum values.

4.8. ILLUSTRATIVE DESIGN OF AN EVAPORATIVE CONDENSER:

Design an evaporative condenser suitable for a compressor having a rating of 75 Tons. The saturated discharge temp.

is $105^{o}F$, the saturated suction temperature is $80^{o}F$. Refrigerant F-12 has to be used.

4.8.1. Solution:

Overall coefficient of heat transfer:

U, the overall heat transfer coefficient from refrigerant to the water film is given by the equation:

$$\frac{1}{U} = \frac{B}{f_r} + \frac{1}{f_m} + \frac{1}{f_w} \qquad \ldots(4.5)$$

For computing U, the coefficients f_r , f_m and f_w should be determined first.

F_r , the refrigerant film coefficient have been determined by Hull[11] and James[12]. It varies between 180 to 300. A value for f_r = 300 Btu/hr/sq.ft/ deg. F. may be selected as find out by James[12] experimentally for evaporative condensers.

Condensing coil of 3/8" nominal size steel tubing may be selected, as recommended by "Carriers Internationals"[4]. The coils are designed for 300 psi working pressure.

The conductivity for steel is approximately 360 Btu/hr/ sq.ft/ Deg. F / inch. used by Thomson[22] and James.[12]

McAdams[15] gives an emperical equation for film conductance in evaporative condensers.

$$f_w = 408 \ (w')^{\frac{1}{3}}$$

Where w' is the water rate in gallons per sq.ft. of the projected area.

A water flow rate of 0.7 gallons per sq.ft. of projected area recommended by Marlo Coil Company (14) in their bulletin, for evaporative condensers

$$f_w = 408 \ (0.7)^{\frac{1}{3}} \qquad = 364 \text{ Btu/hr/sq.ft/}^{\circ}\text{F}$$

A 3/8" nominal size standard pipe will have a wall thickness equal to 0.09 inch found out from Refrigerating data book.

The outside diameter will be 0.675 inch and inside diameter 0.493 inch.

B, the ratio of the outside area to the inside

$$\text{area} = \frac{0.675}{0.493} = 1.37$$

Substituting the values of the film coefficients in the equation 4.5 we get,

$$\frac{1}{U} = \frac{1.37}{300} + \frac{0.091}{360} + \frac{1}{364}$$

$$= 0.007562$$

Therefore, U = 133 Btu/hr/sq.ft. / Deg. F.

The convective coefficient of heat transfer f_c has been computed by McAdams[15] . An average value of 12 Btu/hr./sq.ft./ Deg.F may be selected for the practical use.

Select a value of G/A = 65 . recommended by "carriers Internation[4] " .

$$M = \frac{f_c \quad A_o}{0.24\ G} \quad \text{as used in equation (4.1)}$$

$$= \frac{12 \quad x\ 1}{0.24 \ x\ 65} = 0.77$$

Z_2 is given by the relation

$$Z_2 = 1 - e^{-M} \quad \text{as defined in equation (4.1)}$$

$$= 1 - \frac{1}{e^{0.77}} = 0.538$$

4.8.2. <u>Heat Rejected to the Condenser:</u>

Heat rejected to the condenser = Heat rejection factor x Refrigerating effect.

Using the chart in Fig. 4.2., the heat rejection factor for 105 °F saturated discharge temperature and 40°F saturated suction temperature is 1.14 . Therefore, total heat rejected to the condenser = 1.14 x 75 = 85.5 Tons of refrigeration.

4.8.3. Condensing Temperature:

Assume a loss equivalent of $2^{\circ}F$ in line between compressor and evaporative condenser. The condensing temperature will be

$$105\ ^{\circ}F - 2\ ^{\circ}F = 103\ ^{\circ}F\ .$$

4.8.4 Spray Water Temperature :

Knowing the values of G/A, Z and U, the factor N can be computed by the relation.

$$N = \frac{G\ \cancel{N}R\ Z_2}{AU} \qquad \ldots.(4.8)$$

$$= \frac{65 \times 0.538}{133} = 0.263$$

Since $N = \tan\theta$ $\qquad\ldots.(4.16)$

$$\theta = 14^{\circ}\ 25'$$

Once the value of θ is known, the Fig. 4.3. can be used to find out the spray water temperature. Locate the point of intersection of the initial wet bulb temperature $80^{\circ}F$ and the condensing refrigerant temperature $103\ ^{\circ}F$. Draw a straight ~~straight~~ line at an angle θ with the vertical to cut the curve, the spray water temperature is read at the point of intersection of this straight line with the curve.

The spray water temperature determined for this case is $97^{o}F$.

4.8.5. Coil:

(a) Total Surface Area.

After computing the spray water temperature the equation (4.6) may be used to determine the total surface area of the coils required.

$$q = U.A_o (t_r - t_w) \qquad ...(4.6)$$

substituting the values for U, t_r and t_w

$$12000 \times 85.5 = 133 \; A_o (103 - 97)$$

$$A = \frac{12000 \times 85.5}{133 \times 6}$$

$$= 1285 \text{ sq.ft.}$$

Total surface area = 1285 sq.ft.

(b) Length of the coil:

$$\pi D L = \text{surface area.}$$

$$L = \frac{1285 \times 12}{\pi \times 0.675}$$

$$= 7260 \text{ ft.}$$

Length of the coil = 7260 ft.

(c) Total coil face area considering the coils to be 8 row deep. Assuming a gap of 1/8 inch between the coils.

$$\frac{\text{Face Area} \times \text{no. of rows}}{(\text{Outside dia.} + 1/8")} = \text{Length of piping.}$$

$$\text{Face area} = \frac{7260 \times (0.675 + 0.125)}{8 \times 12}$$

$$= 60 \text{ sq.ft.}$$

(d) Casing dimensions.

Assume width = 10 ft.

$$\text{Height} = \frac{\text{Face Area}}{\text{width}} = \frac{60}{10}$$

Height = 6 ft.

(e) The gap is 1/8 inch between the coils.

Total no. of parallel circuits

$$= \frac{\text{Height}}{(D + 1/8")}$$

$$= \frac{6 \times 12}{(0.675 + 0.125)}$$

$$= 90$$

Total no. of parallel circuits = 90

4.8.6. Fan:

Low speed, centrifugal type fans are recommended . Fan assembly should be light weight and statically and dynamically balanced for suitable performance.

(a) Air Quantity :

$G/A = 65$ as assumed.

$G = 65 \times 1285 = 83{,}500$ lbs/hr.

Taking the density of air = 0.075 lbs/cu.ft.

$$\text{Air quantity required in cu.ft.per min.} = \frac{83500}{0.075 \times 60}$$

$$= 20900$$

Air quantity required = 20900 cfm.

(b) H.P. Fan Motor:

Assume pressure drop through 8 row coil = 1 inch of water.

and pressure drop in distributing pans = 0.3 inch of water.

The total pressure drop taking into account the losses in the eliminators and other unaccounted losses = 1.7 inches of water.

$$\text{H.P. fan motor} = \frac{0.000157 \times \text{cfm} \times \text{Head in inches of water}}{\text{efficiency of motor.}}$$

Assuming the efficiency of fan motor 0.8

$$\text{H.P. required} = \frac{0.000157 \times 20900 \times 1.70}{0.8}$$

$$= 7 \text{ H.P.}$$

Select a motor of 10 H.P. for the fan.

4.8.7. Water Circulating Pump:

Vertical shaft water circulating pump centrifugal type, requiring no suction piping are recommended.

(a) Water Quantity circulated:

Total water recirculated in Gpm = 0.7 x projected area

$$= \frac{0.7 \times 7260 \times 0.675}{12.}$$

Total water flow in gallons per min. = 286.

(b) H.P. of Pump Motor:

Assume a static head of 15 feet and a head of 2' for other unaccounted losses such as in pipe friction ~~nozzles~~.

Total head against which pump has to work

= 17 feet of water.

H.P. of the ~~fan~~ motor for the pump will be given by the relation :

$$H.P. = \frac{WH}{33000 \times \text{efficiency of the motor}}$$

Assume efficiency of pump motor = 0.95.

$$\text{Therefore, H.P of pump motor} = \frac{286 \times 10 \times 17}{33000 \times 0.95}$$

$$= 1.6 \text{ H.P.}$$

Select a motor of 2 H.P. for the water pump.

4.8.3. Make up water required.

$$\text{Water Evaporated in lbs.} = \frac{\text{Total heat rejected to condenser}}{\text{Latent heat of moisture.}}$$

$$= \frac{85.5 \times 12000}{1000}$$

$$= 1030 \text{ lbs./hr.}$$

Water evaporated 1.72 gallons per minute. Keeping in allowance for bleed off, the make up water required will be 2.5gallons/ min approximately.

CHAPTER V

FOULING IN EVAPORATIVE CONDENSER AND ITS CONTROL

Scale, Corrosion and Algae are the principal problems which may develope from water use in an evaporative condenser. The various design elements which affect fouling in the condenser are given. Bleed off, effective water treatment and Chemical cleaning of the condensers have been discussed extensively.

CHAPTER V

FOULING IN EVAPORATIVE CONDENSER AND ITS CONTROL .

Unfortunately water we obtain for the use is generally a very complex chemical liquid. Because water is an excellent solvent, we find all kinds of mineral and gases dissolved in it, which comlicates its use in heat exchanger equipment. These impurities in water are solely responsible for the fouling in condensers using water as a cooling media.

A good design of an evaporative condenser can reduce as much as 50% of the normal fouling. Although no condenser will remain clean, without periodic servicing, a good design, effective water treatment, and timely servicing can keep it efficiently clean.

In a well designed condenser with adequate water velocity and flow rate the tendency to foul is much reduced, hence the cost of water treatment is kept down much.

Fouling is considered to be the deposition of foreign material on the condenser heat transfer surface which reduces the heat transfer efficiency and also gives rise to friction to water flow.

Keeping in view of this fouling effect on the heat transfer surface, manufacturers generally keep an allowance in the condenser capacity by providing of additional heat transfer surface which is termed as " fouling factor".

By perfect water treatment and cleaning the fouling factor can be reduced to a minimum. A condenser having no additional heat transfer surface allowance and has tendency to foul, will require careful water treatment.

5.1. TYPES OF FOULING:

Scalying, corrosion and organic or bio-fouling all come under the heading of fouling.

5.1.1. Scaling is a deposition of inorganic salts on the heat transfer surfaces precipitated from water, most commonly calcium carbonate. In case of evaporative condenser, scaling is very prominent since water evaporates leaving a thin film of solids deposited on the condensing surface. This film becomes thicker as the deposition is continued following the evaporation.

5.1.2. Corrosion is a direct attack of water on metals of the system. Rust is the example of this on the ferrous metals.

5.1.3. Organic or bio-fouling is the deposition of organic matters, algae or other micro organisms. Bio-fouling has usually the tendency of increasing scaling and corrosion.

Fouling is practically negligible in the condensers using once-through or waste water systems, since the dissolved solids are carried over with the water itself.

But in the case of those employing recirculated water systems such as in case of evaporative condensers fouling is a real problem. This is evident from the fact that the concentration of the dissolved solids goes on increasing because of the continuous evaporation of the pure water. It should be noted that the evaporation is of the distilled water and it does not carry any dissolved solids with it. This results in gradual increase of the concentration of solids.

5.2. DESIGN AFFECTS FOULING :

Following are some of the elements of design which have a direct effect on scaling or fouling condensers :-

a. The rate of water circulation should be high as compared to the rate of evaporation, this will have a cleaning or washing effect on the condenser coils. If the rate of water circulation is low, the equilibrium temperature established will be high. Scaling is directly proportional to the equilibrium temperature. This shows that the scaling tendency will be reduced by keeping a high rate of circulation.

b. Even distribution of water over the entire condensing surface will provide a uniform washing action . In some condensers where the water is not evenly distributed, some of the coils get ample water while the other starve for it or remain mainly dry. Without exception these have a high fouling factor.

c. Condenser tube bundle will influence both water and air circulation and distribution. Mostly the evaporative condensers are designed with counterflow of air and water. The spacing of the tubes should be such that the air and water flow can take place effectively.

In some cases finned coils in evaporative condenser is used to increase the heat transfer area. The balance of the heat transfer rates on both sides of the condenser tube should be maintained. That is the rate of the heat transfer from the gas to ~~be~~ the tube and from the tube to the water. If the fin spacing is too narrow or the tubes are packed closely then air and water flow cannot be efficient.

Although there is no hard and fast rule for the fin and tube spacing, according to the results of tests and experience, Ralph M. Westcott has arrived to an arbitraily emperical formula for finned surfaces: "Fin spacing should be at least equal to the height of the fins and the spacing between the finned coils should be at least equal to the height of the fins".

d. It is a belief of most of the investigators that scaling rate is increased with the superheat of the refrigerant.

D.D.Wile[23] carried out certain experiments and noted that the superheat has no effect on the rate of scaling.

i.e. scaling is independent of superheat.

Anyway, the control on this effect of superheat if it is , is achieved by providing a dry tube for desuperheating, before the refrigerant enters wet coils below the sprays. However, in attempt to remove this superheat care should be taken not to place the dry coil (superheat coil) just above the sprays or below the eliminators. Because the water entrainment ~~entertainment~~ with the air will evaporate immediately leaving solids baked on the superheat coil. Furthermore, the cleaning of these coils is difficult and no water treatment will prevent scaling as all the moisture is evaporated to dryness.

e. If the water sprays are too high above the coils, there will be a greater tendency for the water to be carried upto the eliminators and to cause corrosion there in the fan section.

f. In some of the cases the subcooling is achieved by submerging the receiver in the pan of the condenser. The advantage being the saving in floor area. But it is offset by the greater disadvantage of the increased tendency of corrosion due to valves and connections always under water.

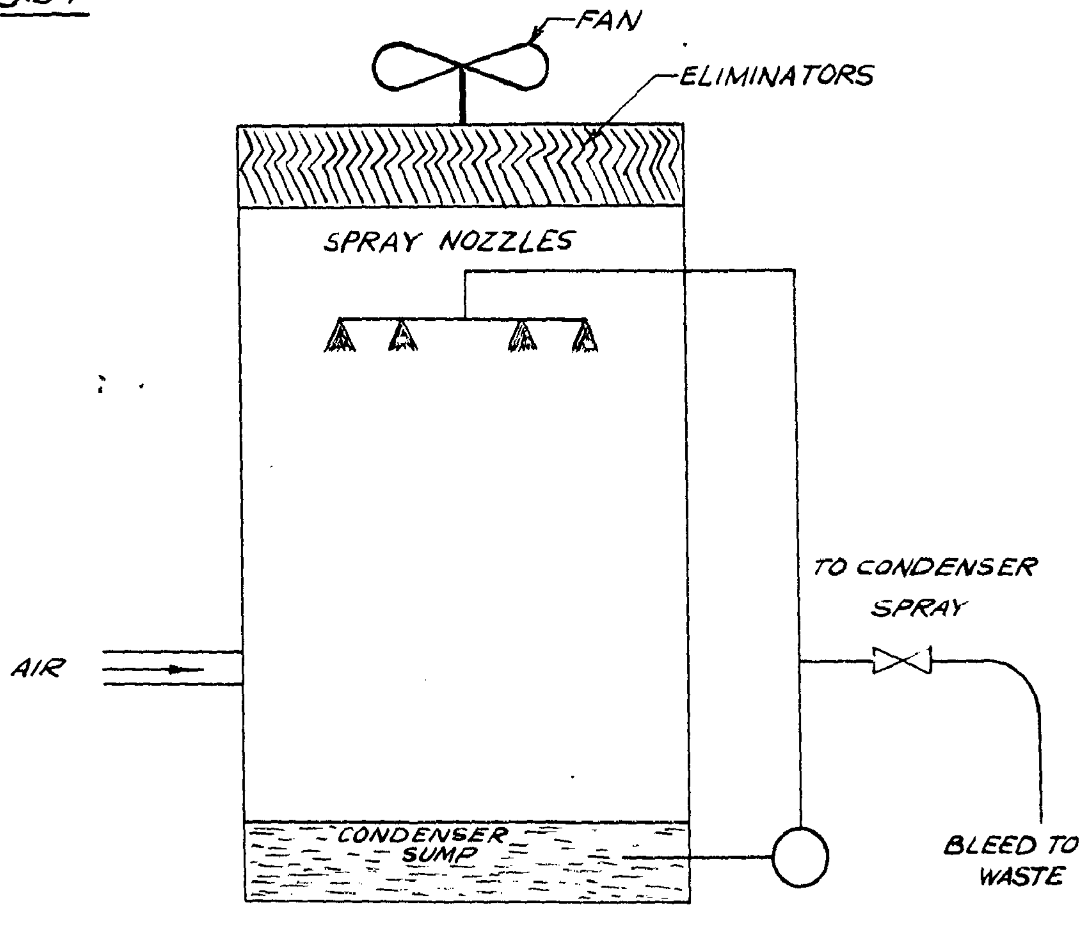
g. Corrosion normally occurs in water pan, casing, eliminators, fan section, duct work and coil section. Dissimilar metals may contribute to electrolytic corrosion in water circulation system. Suction screen of the pump intake of brass connected to steel water lines or to the steel pan of the condenser. Proper insulation between dissimilar metals will reduce this sort of corrosion. Elimination of corrosion in fan section is obtained by efficient eliminators.

h. Lastly, accessibility of elements of design is necessary to keep evaporative condenser clean. The evaporative condenser should be cleaned periodically, so all the essential parts like to become fouled should be accessible . In some cases it is necessary to cycle either the air or water circulation during the change of load. Whenever this practice is necessary the fans or the air circulation should be cycled rather than the circulation of water. Otherwise due to repeated wetting and drying of the condenser, the minerals will be deposited on the dry cycle, which will not be completely washed out during wetting cycle.

5.3. <u>WATER QUALITIES:</u>

The water analysis is based on the ions and not on the combined natural salt form. Thus calcium and Sulphate are determined as individual constituents and it would be very rare that the quantities of each would be present in exact amount that will require each fully to form calcium sulphate. The ions responsible for fouling are such as of Magnesium, Sodium, Potassium, Aluminium, Iron, Sulphates and Chlorides, etc. Since water contains

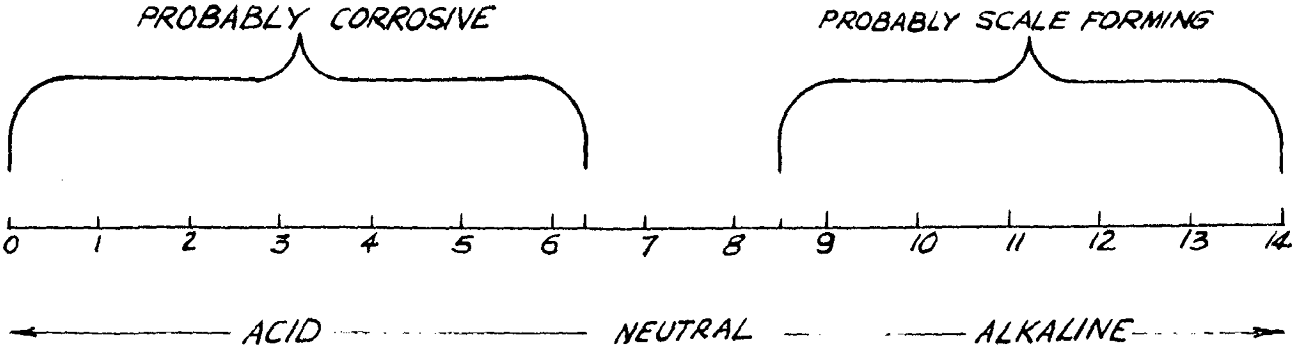
FIG. 5.1



EVAPORATIVE CONDENSER SHOWING THE POSITION OF BLEEDING POINT

FIG. 5.2

PH· CONTROL



different quantity of many impurities, no thumb rule can be formed regarding its fouling action. However, the untreated water which contains 100 ppm of soap hardness or about 6 gr. per gallon will cause scaling in condenser if bleed is not provided.

5.4. CONDENSER BLEEDING:

Bleeding is the first step in prevention of fouling of condensers. This is the method of draining out a portion of water, and thus effectively reducing the concentration of dissolved solids. The amount of bleed depends upon the ~~a~~ evaporation rate and the quality of water used. However, this ranges from half the quantity to three to four times of the evaporated water in worse cases. The quantity of bleed increases as the quality of water is poorer. By correct bleeding and make up water supplied, a constant concentration of solids can be maintained in the system.

Bleed should be taken from a point higher than the water level as shown in Figure (5.1), so that the water is being circulated even though the system is not operating.

5.5. WATER TREATMENT:

Although scale formation can be reduced by bleeding but it cannot be completely eliminated without water treatment.

There are two broad classifications of water treatment viz. (a) Pretreatment or External Treatment and

(b) Internal treatment.

(a) Prewater treatment or External Treatment of the make up water is provided by methods of zeolite softening, filtering or chemical treatment. This is to remove scale forming salts from the make up water. Zeolite softening removes ions of Calcium and Magnisium, thus reducing scaling tendency. Some internal treatment still remains necessary for the control of corrosion and bio-fouling and the resideual hardness which could not be removed by the external treatment. External treatment is costly and requires additional space and maintenance.

(b) Internal treatment of water is done by introducing some chemical directly in the circulating water system. There can be two types of internal water treatment.

Firstly, the chemical is added so as to precipitate the scale forming salts, which can be removed as sludge from the basin or by bleeding.

Secondly, to feed a chemical at low concentrations which will tie up or isolate the scale forming salts and inhabit their deposition.

5.6. SCALE CONTROL WITH CHEMICAL TREATMENT

The raw water or recirculated water can be tested for chloride content and hardness, so as to get an idea of the quantity of bleed.

Chloride ion is relatively soluble in water and generally does not precipitate and may be used as an indicator for hardness. Hardness can be determined by the soap solution. Care should be taken during the soap test that the end point of calcium should not be mistaken as the final end point, thus excluding the hardness of magnesium. In some cases marked lather is seen of Calcium but this is broken by adding more soap solution.

The method already stated above of precipitation of scale forming salts with the addition of chemicals is not so favourable. Because the sludge is formed in the basin ~~tk~~ and many times gives rise to difficulty of clogging of the spray nozzles. This requires therefore careful bleeding. Also the sludge is carried over the coils and eliminators with air where it deposits. Periodic manual cleaning is essential when using this method of treatment.

The second method is accepted widely in which scale forming constituents are isolated with the help of chemicals added. This method is economical also since a high concentration of scale forming salts is kept without permitting deposition to form scale.

But there is a limit to concentrations that can be kept in the solution due to the limit of their respective solubilities. Thereafter bleeding is the only alternative. There are so many chemicals which can be used for wat-er treatment. The most extensively used is Glassy Phosphate or Metaphosphate. This is added to the make up water preferably by a feeder and is dissolved slowly. Glassy Phosphate dissolves at a fairly constant rate of about 25 percent per month. A calculated amount of the chemical is fed for a fixed quantity of make up water. This is then again recharged after a month with one quarter of dissolved, during the previous month.

The most important ion in water is Hydrogen ion. Upon the concentration of this Hydrogen ion, depends its acidic or alkaline properties. The Hydrogen is a powerful catalyst, and the whole cause of reaction may be modified by Hydrogen ion concentration in the medium in which the reaction takes place. The Hydrogen ion concentration is expressed in a special way.

The pH of a solution ~~of~~ is defined as the logarithim to the base 10 of the reciprocal of Hydrogen ion concentration in gram. mols/liter.

$$pH = - \log_{10} (H^+) = \log_{10} \frac{1}{(H^+)}$$

Formation of scale depends upon pH of the water. Generally as the pH exceeds 8.3, scale occurs. With the lower pH of water in the acidic media the tendency will be corrosive.

Scale or corrosion may very rarely occur simultaneously. This is easily understood when it is recognised that there exists an equilibrium of concentration of impurities in water, that is a balance point, where corrosion occurs on one side of balance point and scale on the other side.
This is evident from fig 5.2

Professor Langlier of University of California developed equations based on the probable action of water. This is known as Langlier's equation of saturation. The index gives fairly accurate results of what to expect from a given water.

The index is calculated by knowing the concentration of Calcium, Alkalinity, total solids, pH and temperature involved. A value known as pH of saturation on pH_s then corrosion is expected or if actual pH is greater than the pH_s, scale formation can be predicted.

$$\text{i.e. } pH - pH_s = \text{Index.}$$

This is very reasonable since scale generally occurs at higher pH values and corrosion at the lower pH values.

5.7. CORROSION:

There are two forms of corrosion, electrolytic corrosion and Oxygen corrosion.

Electrolytic corrosion is already been discussed under the heading of "Design effects fouling". Electric currents are set up between two dissimilar metals in the water system. This results corrosion. This is not so common as it is believed to be.

Oxygen corrosion is much more common. Oxygen is the principal cause of corrosion in water. It does not mean that all water with dissolved oxygen will have a corrosive action. But on the otherhand if all the oxygen is removed from the water, the corrosion tendency is materially reduced or eliminated.

Carbon Dioxide dissolved in water also contributes to corrosion. The langlier's index again proved valid because if the index is positive, the effect of Oxygen will be minimised. The scale forming tendency of water forms a coating on the metal surface and saves it from the corrosive act of Oxygen. If the saturation index is negative then no film is formed on the surface and Oxygen readily attacks metal.

The control of corrosion generally means pH control. As shown in the diagram of pH scale, no corrosive action is expected if the pH value is kept between 6 to 9.

Fig. 5.2 Algae growth on evaporative condenser coil.

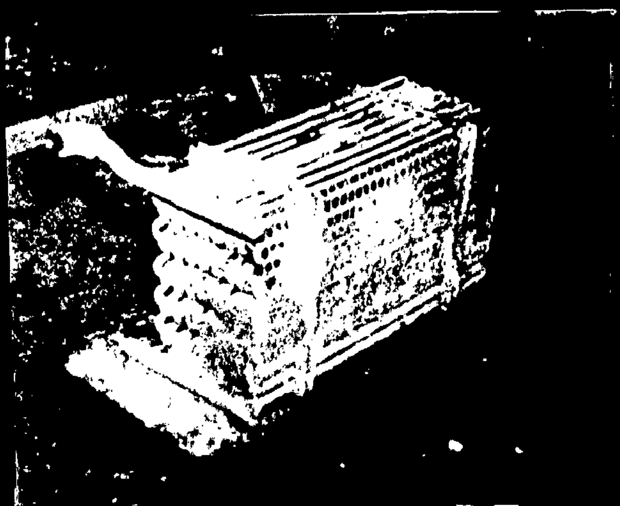


Fig. 5.3. Heavy deposits of scale on condenser coils

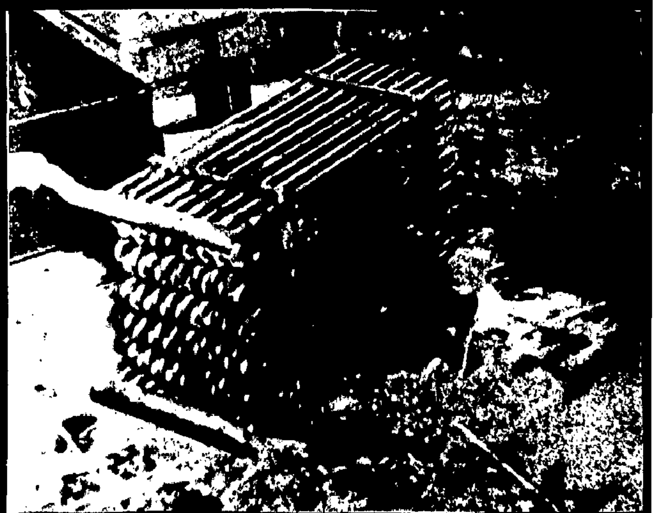


Fig. 5.4. Scale formation on the condenser coils.

The pH value can be adjusted by proper introduction of suitable chemicals like alkalies and acids. If the pH is low, alkalies are added or if high, acid is introduced.

The chemical control of corrosion is done by the chemicals like glassy phosphate, chromates or nitrites. In case of glassy phosphate, the concentration required for corrosion protection are fairly high.

Chromates are poisonous and should be handled care-fully. The glassy Phosphate has the inherent advantage of being non-poisionous and is therefore widely used.

Metaphosphate alone is capable of scale control as well as corrosion control to some extent. A portion of the Metaphosphate is absorbed on the metal and thus provides a protective coating but not a scale as it is only molecularly thick.

5.8. ALGAE AND SLIME CONTROL:

Algae or biofouling is generally found in dark places. Previously chlorine was extensively used for algae control. Later on Bromine is also successfully used for the prevention of biofouling. Both chlorine and Bromine have toxic effects on algaes and slime, yet their method of treatment differs somewhat.

Batch or intermittent feed has proved very satisfactorily for algae control. There is simple test to determine the quantity of free chlorine and Bormine in water. When either of these algaecides is present, the addition of orthodolene gives a yellow colour which increases in intensity with the quantity of algaecides.

Other than Bromine and Chlorine, Pentachlorphenate, Copper sulphate, Pottasium Permagnate, and some ammonium coumpounds can be used for above purpose. Although their quantities will differ with the quality of the water and their own strength. For instance, Bromine and Chlorine give effective results at 1 to 5 ppm on an intermittent feed basis. Sodium Pentachlorophenate is generally used of the order of 60 to 100 ppm-

It is obvious that the uncontrolled feed of any of these algaecides will result worse problem than by the algaes itself. Since Bromine and Chlorine are strong oxidising agents will promote corrosion if used in excess. Copper sulphate is a good electrolyte and may produce another form of electro-chemical corrosion. Pentachlorphenate has no such disadvantage and can be used freely but its cost cannot be overlooked.

It is always better that the use of Chemicals for water treatment should be under proper care and supervision.

5.9. CHEMICAL CLEANING OF CONDENSERS:

The plants in which periodic cleaning and water treatment is ignored, they soon become inoperative and cleaning becomes essential. Photos 5.2, 5.3, and 5.4. show the results of such cases.

Manual cleaning can also be done in some cases, but where the plant cannot be stopped for cleaning, chemical cleaning is the only alternative.

Generally muriatic or commerical hydraulic acid is used to dissolve scale or corrosion. The acid should be used with some inhibition so that the metal is not attached by the acid. Formaldehyde is an excellent inhibitor but it has an irritating odour and a tendency to vaporize. About 2% by weight of the acid is sufficient. Care should be taken that the dissolved scales and corrosion may not enter the pump which may cause erosion in the coils by the high velocity sprays.

The inhibitor forms a coating on the surface of the metal so that it is saved from the attach of the acid. After the chemical cleaning of the condenser, new fresh water should be refilled and corrosion controlling chemicals are added to retard corrosion of the prime metal. Serious loss may occur if the acid is not properly inhibited, too high concentrated acid is used or is left for too long a period.

There is an interesting example of trouble developing from overdoing a treating job. A zeolite softener was installed with an evaporative condenser where the water was high in concentration of chlorides and sulphates. When completely softened by removing calcium and Magnesium it became very corrosive. The softener eliminated the scale but it almost eliminated the evaporative condenser too.

Another is a classic example of evaportive condenser with a trouble of growth of algae. This was installed in powder factory where the odour of the volatile oils was quite noticeable.

In few weeks the growth of algae was intolerable. Manual cleaning proved hopeless as the stuff could not be washed out. Numerous chemicals were tried without much of success. It was finally determined that the growth of algae was the result of oils in air combined with moisture. The equipment was then sealed in a room and the fresh supply of air was taken from outside and this resulted the complete control of algae.

Evaporative condensers are being used as exhaust fans for many conditioned spaces or the air exhaust for food storage spaces, where atmosphere contains organisms for slime growth.

In the words of Westcott " To make a piece of equipment serve too many purposes. It might backfire".

CHAPTER VI

COST ANALYSIS

AS COMPARED WITH OTHER REFRIGERANT CONDENSING METHODS

Too often it is seen, judgement has been based either on initial cost alone or on operating cost alone without regard to overall investment. Ultimate objective in the economic analysis is to find the total annual owning and operating cost.

A comparative study of evaporative condensers cooling tower and condenser combination and "once through" water cooled condensers shows that the evaporative condenser enjoys the economical advantage over the other two. There are cases where a particular condensing method is to be adopted. The factors governing the selection of a specific condensing method have also been included.

CHAPTER VI

COST ANALYSIS

AS COMPARED WITH OTHER REFRIGERANT CONDENSING METHODS

6.1. The rapidly expanding demand for air conditioning puts a great strain on municipal supplies as a source of cooling water and regulations have become necessary to restrict its use and disposal.

For this reason several types of refrigerant condensing methods and water saving devices were developed. Apart from the traditional air cooled and "once through" water cooled condensers , in recent years cooling towers and evaporative condensers have gained popularity and have greatly reduced the water requirements and also resulted power saving to some extent.

For reasons only of comparision and simplicity the condensing refrigerant F-12 for air conditioning cycles will be considered in tonnage rating of 10 to 100. In this category are three methods in common use :

i. Evaporative condenser.

ii. Cooling tower (induced draft) and condenser combination.

iii. Water cooled condensers ('Once through' using city water.)

Each of the three methods have certain advantages over the other two, and the selection requires good engineering and econimics analysis for application and locality involved.

The natural draft cooling tower and condenser combination proves to be very uneconomical due to its greatly increased size and does not permit its use beyond a certain capacity of about 50 Tons. Consequently, the Natural draft cooling tower and condenser combination is not included in further discussions.

6.2. <u>BASIC CONSIDERATIONS:</u>

The factors governing the selection of particular condensing method for a specific application are :

a. Design considerations.
b. Water availability.
c. Economics.
d. Space availability.
e. Type of application and size of system.

(a) <u>Design Considerations:</u> The design factor such as dry and wet bulb temperatures of outside air, water supply temperature, pressure and hardness will have a direct bearing upon the selection of a particular type of condensing equipment. In a combination where the wet bulb temperature of air is low and the temperature of water available is high, the use of an evaporative condenser or a cooling tower combination is recommended . On the other hand a low available water temperature combined with a high wet bulb temperature of air would indicate the use of "Once through" water cooled condenser.

(b) Water Supply: Before making a choice between the condensing methods it is of vital importance to judge the adequacy of water supply.

Many cities have acute water shortages and it is very expensive affair to think of a water cooled condeser using city water. In such cases evaporative condenser or cooling tower and condenser combination are the best alternatives and prove to be very economical.

In the localities where the water is available in abundance from the surrounding lakes or rivers and the fresh water is available at moderate temperatures, the 'once through' water cooled condensers should be preferred.

It may be noted here that the make up water requirements of an evaporative condenser are about 10% of the water consumed by a city water cooled condenser.

(c) Economics: In order to study the economics of the various condensing methods it is necessary to find out the total owning and operating cost which is composed of the installation cost and the operating cost. A judgement should not be based on either the installation cost alone or the operating cost alone. The best way would be to treat the large sized ai-r conditioning installation as long term investments and consider all the factors which may affect the result.

These factors may include the following:

i. Installed cost of equipment including auxiliaries and piping.

ii. Annual operating cost.

iii. Period of amortization and interest rate.

iv. Overhead expenses (Taxes, insurance and rent)

i. <u>Installed Cost of Equipment:</u> The initial cost consists of:

(1) Cost of the condensing apparatus including condenser (water or evaporative), cooling tower, auxiliaries such as pumps , water treatment devices, refrigerant receivers, water valves, motors, stores. This cost should also include freight and trucking charges from factory to job site.

(2) Cost of refrigerant and water piping required to connect the condenser and cooling tower into the rest of the cooling system. Valves, fittings and hangers should be included in this cost.

(3) Cost of labour and supervision to install the equipment including rigging, handling, piping and controls.

ii. Annual Operating Cost: The annual cost for operation include:

(1) Water cost based on total water requirements, taking into account for evaporation, bleed off and windage losses.

(2) Power cost for operation of fan and pump motor (Compressor not included).

(3) Maintenance cost including replacement and labour.

iii. Period of Amortization and Interest Rate: Especially when selecting the evaporative condenser or a cooling tower where initial investment is relatively high, the amortization period is of importance in determining the annual owning and operating cost.

The useful life of this type of equipment may vary from 10 to 20 years, depending upon the type of application, the hours of operation and the degree of maintenance. An average life of 15 years may be taken with good maintenance programme.

A suitable interest rate of 4 to 6% may be taken.

iv. Overhead Expenses: Allowance should be made for the increase in the taxes , insurance and a proper share of it should be added to the air conditioning equipment.

FIG. 6.1

DESIGN WET BULB 76°F, CITY WATER 87°F

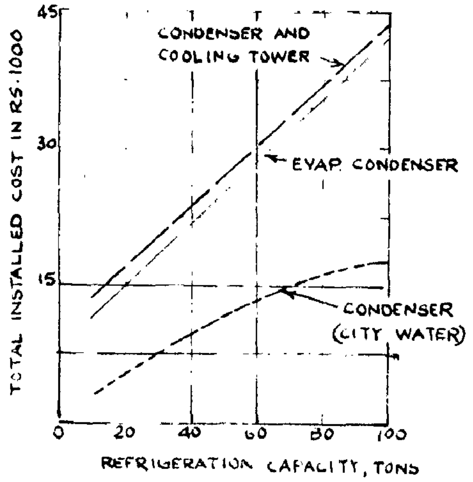
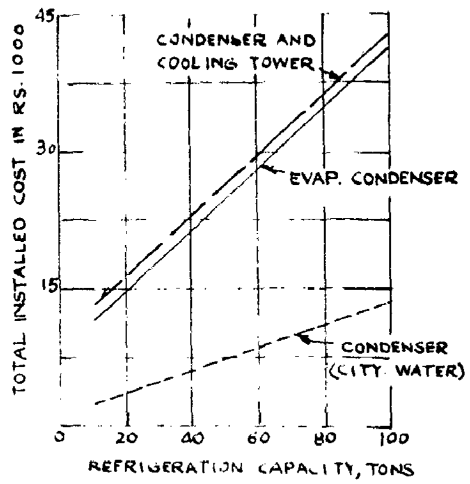


FIG. 6.2

DESIGN WET BULB 75°F, CITY WATER 73°F



COMPARISION OF TOTAL INSTALLED COSTS FOR VARIOUS TYPES OF CONDENSING EQUIPMENT USING FREON-12 REFRIGERANT

FIG. 6.3

DESIGN WET BULB 76°F, CITY WATER 80°F

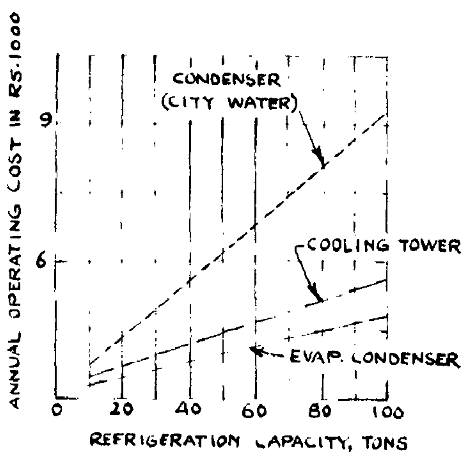
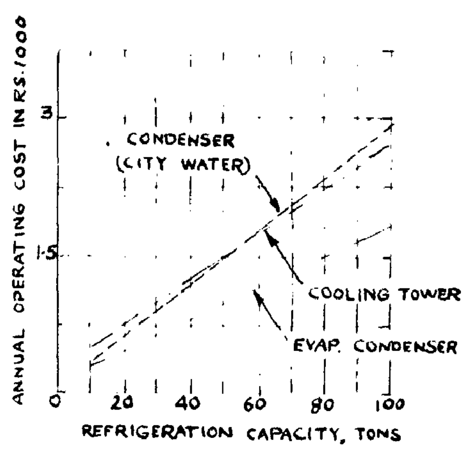


FIG. 6.4

DESIGN WET BULB 75°F, CITY WATER 64°F



COMPARISION OF ANNUAL OPEARATING COSTS FOR VARIOUS TYPES OF CONDENSING EQUIPMENT USING FREON-12 REFRIGERANT

[ALL SYSTEM USE CONDENSING TEMP. 105°F AND SUCTION 40°F]

If the equipment is located in rentable space , an addition should be made to allow for loss of the space occupied. In case of evaporative condensers and cooling towers installed this space can usually be neglected as it is unlikely to be useful.

(c.1) Installed Cost:

Installed cost or the first cost is affected to some extent by the loading and design factors such as the wet bulb temperature of air, entering water, and condensing temperature. When compared with the evaporative condenser, the installed cost of the "Once through" city water condenser varies between 25 to 50%. depending on the factors mentioned above. The cooling tower combination for the capacities between 10 to 100 tons will have installed cost , a little greater than that for the evaporative condensers. This is illustrated by the graphs shown in Fig. 6.1 and 6.2 , where total installation cost is plotted against the refrigeration capacity in Tons for each type of condensing methods.

(c.2.) Operating Cost:

Figures 6.3 and 6.4 show the results of the cost of operation including water, electricity, maintenance and water treatment for three types of condensing equipment.

It is evident from Fig. 6.3 that the operating cost of "Once through " water condenser is very much unfavourable as compared to the evaporative condenser or a cooling tower. Fig. 6.3 is drawn on the basis of relatively high water cost and temperature and Fig. 6.4 is based on relatively low water cost and temperature. It will be noticed from Fig. 6.4 that the operating cost of the conden ser using city water has materially reduced as the available water temperature is lowered.

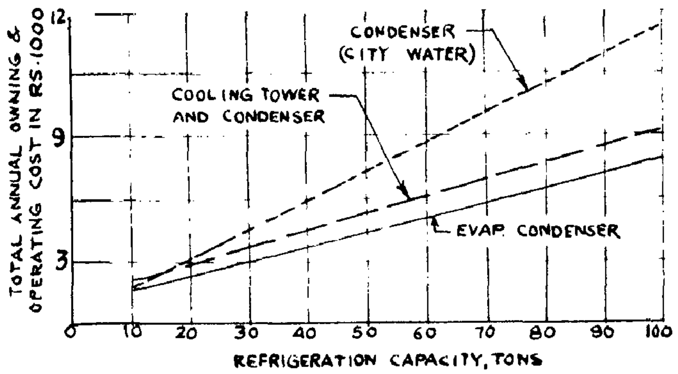
The advantage that the evaporative condenser has over the cooling tower and condenser combination is due to the additional power required to circulate the water between tower and condenser.

In addition to the water and power costs, the maintenance cost should also be included in the estimate for operating cost. Maintenance cost will include the cost for chemicals used for water treatment, painting, lubrication, cleaning, labour and parts replacement. It is evident that the maintenance cost of cooling tower and evaporative condenser will be more as compared with "Once through" condenser using city water.

(c.3.) Total Annual Owning and Operating Cost :

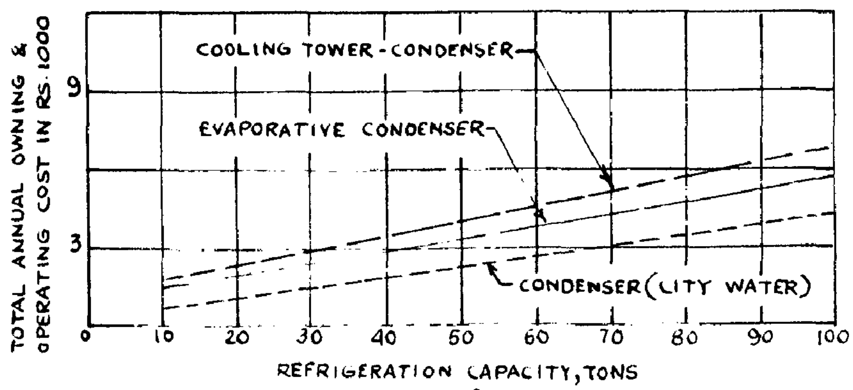
The annual cost to own the equipment is determined by dividing the total installed cost by the amortization period (15 years say) and adding to the annual interest rate of(5% say) T-he operating cost is determined as previously discribed.

FIG. 6.5



DESIGN WET BULB TEMP. 76°F, CITY WATER TEMP. 80°F

FIG. 6.6



DESIGN WET BULB TEMP. 75°F, CITY WATER TEMP. 64°F

COMPARISION OF TOTAL ANNUAL OWNING AND OPERATING COSTS FOR VARIOUS TYPES OF CONDENSING EQUIPMENT USING FREON-12 REFRIGERANT

[ALL SYSTEMS USE CONDENSING TEMP. 105°F, SUCTION TEMP. 40°F]

Figures 6.5 and 6.6 show the resultsof two sets of data~~s~~ used for comparision purposes. Fig. 6.5. is for the localities where the water cost and the available water temperature are both high. While Fig. 6.6. is for comparatively lower water cost and low water temperature.

It is apparant from Fig. 6.5 that the evaporative condenser~~s~~ is an economic necessity in localities where city water cost and temperature are relatively high, regardless of the availability.

In the other case where water cost and temperature are both low (Fig. 6.6) the water cooled condenser using city water is recommended. Figures 6.1 to 6.6 are based on the datas collected by Groseclose C. E.[7].

However, it may be emphasized here that such instances occur only in urban cities or underdeveloped localities, and are relatively few.

(d) <u>Space Availability:</u>

The evaporative condenser claims the advantage that it can be located outdoors or on the roofs of the buildings, consequently saving the valuable indoor space. In addition to this evaporative condenser handling outdoor air produces an additional saving by operating without water as an air cooled condenser during comparatively cold weather. The modern arrangement of evaporative condenser is compact,requiring little space.

(e) Type of Application and Size of System:

The application may influence the type of condensing equipment selected. For example in a big restaurant where the number of operating hours per day is usually very high, the city water condenser would be unfavourable. Conversely , in a Church where the operating hours per week are small the low water requirements would favour the city water condenser with its low first cost. Another example in which an industrial process application requires high humidity make up air, the evaporative condenser could be most effectively utilized.

The size of a proposed air conditioning system may be a factor in the selection of a condensing method. In the case of an evaporative condenser where first cost is dominating, its installed cost per ton of refrigeration in a small system (10 tons) is about three times the installed cost per ton of a large (100 tons) evaporative condenser. Contrast tothis with city water condenser, where operative cost is the primary factor, the water consumption is directly proportional to the tonnage, and the cost per ton remains more nearly constant. For this reason the evaporative condensers are most suitable for large capacity systems.

6.3 . SUMMARY:

It is evident that, with the boom in air conditioning already well started and reliably predicted to reach major proportions, this industry must solve water conservation problem otherwise, the normal growth of air conditioning could very easily be stunted. We cannot expect city water supply systems to be expanded at the rate which would be required if unbridled use of water cooled condensers is allowed. It is difficult, in first place, to find the new sources of water in some localities and it is certainly impracticable to expect city and State Governments to move fast enough to expand facilities as an accommodation to one industry. Water saving devices and schemes must be developed and promoted by air conditioning industry so that they are inexpansive and capable of reducing quantity of water cooled condenser requirement.

B I B L I O G R A P H Y

1. Alton E. " A recent evaporative condenser design" Ref.Engg. Sept. 1940.

2. A.S.H.V.E. Guide.

3. Avery Lester T. " Evaporative Condenser Guilty too" Jr. Heating Piping and Air Conditioning Dec; 1959.

4. Carrier Corporation " Bulletin on Evaporative Condensers".

5. Deverall C.N. " Evaporative Condensers" Ref. Engg. Jr. May,1942.

6. Ereckson E.V. " Evaporative Condensers" Ref. Engg. Jr.Nov. 1936.

7. Groseclose C.E. " Cost Comparision of Air Conditioning Refrigerant Condensing Units" Ref. Engg. Jr. June, 1954.

8. Goodman W. "Evaporative Condenser" Heating Piping and Air-Conditioning Jr. March, 1938. and April 1938.

9. Homes R.E. "Water Conservation Devices" Ref. Engg. Jr. Aug,1953.

10. Herman Welter "Effect of Water Distribution on Evaporative Condenser Efficiency " Ref. Engg. Jr. April, 1951.

11. Hull H.B. " Household refrigerators" Collins Co, 4 th edition.

12. James "A study of Heat transfer in Unit Refrigerant Condensers which use evaporative Cooling" Ref. Engg. Jr. March,37.

13. Jordon " Ref. & Air conditioning."

14. Marlo Coil Company " Bulletin on Evaporative Condenser.

15. McAdams " Heat Transmission" McGraw Hill Book Co.

16. Montrel Joshep " Does your water Saver really save water" Ref. Engg. Jr . Sept. 1954.

17. Ralp M. West Cott (i) "Control of Condenser fouling by water treatment" Ref. Engg. May, 1949.

    (ii) " Water quality Control for air conditioning and Refrigeration" Heating Piping and Air conditioning Jr. Sept. 1951.

    (iii) Evaporative Condenser Scale Control " Ref. Engg. Jr. July , 1947.

18. Reichman Irving " Use of water in Refrigeration and Air Conditioning" Ref. Engg. Jr. Feb. 1944.

19. Roltmayer S. " Notes on Cooling Tower and E vaporative Condenser" Ref. Engg. Jr. 1946.

20. Rose J. Martins " Evaporative Condenser, their use in water shortage area" Ref. Engg. Jr. May 1948 .

21. Sherwood T.K. " Economic balance in the design and operation of the ammonia condensers" Ref. Engg. Jr. Feb, 1927.

22. Thomson E.J. " Heat transfer in an Evaporative Condenser" Ref. Engg. Jr. May , 1946.

23. Wile D.D. " Evaporative Condenser performance factors" Ref. Engg. Jr. Jan, 1950.