PERFORMANCE CHARACTERISTICS OF COLO STORAGE UNDER FROSTING CONDITIONS

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MY WIFE:

MARIA EMMA KUCHA
FOR HER LOVE AND PATIENCE

A B S T R A C T

In the Refrigerating Industry, the frosting of refrigerating coils and its effect on heat transfer is a problem of obvious importance. Amount of frost on the evaporator is dependent on the ambient temperature and air humidity. The performance of any cold storage unit depends upon maintaining storage space and evaporator temperatures within a range of most favourable conditions.

This research project therefore is aimed at investigating the peculiarities of frosting, especially the variations of the heat transfer coefficient as a function of the duration of use during frosting with the influence of air conditions as close as possible to those in practice and to present some recommendations for defrosting on the basis of the results obtained.

The theory of frost formation is outlined and its complexity identified.

Design procedure of a cold storage is briefly given. This is to give readers an idea of how the various components of the system, eg, the evaporator, the condenser, etc were selected; the various factors that contributed to the successful working of the unit.

Measurements on a house hold refrigerator are present tedj the results obtained are analysed and on the basis of these results and the experience obtained during the performed tests, some conclusions and recommendations are made.

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NOMENCLATURE

Symbol	Description	Units
A	Heat Transfer area	m^2
C	Humid Heat of Moist air [c = 1.005t + 1.88H]	$kJ/kg.^oC$
d	Diameter of tubing	mm
F	Usage factor	$kW/m^3.^oC$
h_e	Effective Heat Transfer Coefficient defined by $h_e = \frac{q/A}{t_a - t_s}$	$kW/m^2.^oC$
f	friction factor	dimensionless
h_G	film coefficient of sensible heat transfer for air	$kW/m^2.^oC$
h_i	Coefficient for Enthalphy transfer defined as $h_i = \frac{(dq/dA)_{x_f = 0}}{(i_a - i_s)}$	$\frac{kg/s.m^2}{(H_a - H_f)}$
H_a	Humidity of Bulk of air	$kg_v/kg_{air,dry}$
H_f	Humidity of Saturated air at temperature of the frost-air interface	$kg_v/kg_{air,dry}$
i	enthalphy of air given by $i = ct + rH$	kJ/kg.
k_G	Mass Transfer Coefficient	$kg/s.m^2.(H_a - H_f)$
K_a	thermal conductivity of air	$kW/(^oC/_m)(m^2)$
K_f	thermal conductivity of frost	$kW/(^oC/m)(m^2)$
dq/dA	Local heat flux at any Instant	$kJ/s.m^2$

Symbol	Description	Units
v	Specific volume	m^3/kg
L_e	Total length of evaporator	m
L_c	Total length of condenser	m
m_v	mass of water vapour	kg
ma	mass of dry air	kg
G	fluid mass velocity	$kg/s.m^2$
G_f	Growth rate of frost	kg/hr
r	Latent heat of sublimation	kJ/kg
t_a	bulk temperature of air	^oC
t_f	Temperature at Frost-air Interface	^oC
t_s	Temperature of coil surface	^oC
x_f	Frost thickness	mm
t	time	min
t_o	ambient temperature	^oC
t'_o	temperature of refrigerated space	^oC
TD_e	Evaporator Temperature Difference	^oC
tS	Condenser temperature split	^oC
t_{cr}	'critical' time	mim
t_r	'reduce' time	min
W_a	air rate	kg/hr
ρ_a	air density	kg/m^3
ρ_f	frost density	kh/m^3
ω	Specific Humidity	$kg_v/kg_{air,dry}$

Symbol	Description	Units
Φ	relative air Humidity	%
Φ_{cr}	'critical' relative air Humidity	%
Φ_r	'reduced' relative air Humidity	%
$\bar{v}$	fluid velocity	m/s
ΔV	pressure drop	bar

Subscripts

a	air
e	evaporator
f	frost
c	condenser
0	origin
i	enthalpy
v	vapour
s	coil surface
1,2	inlet and outlet of evaporator
t	total
u	usage
cc	cooling capacity
cr	'critical'
r	'reduced'
ac	acceleration
fr	firctional

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CHAPTER 1

INTRODUCTION

Refrigeration is concerned with heat transfer from a lower temperature (heat source) to a higher temperature (heat sink). More specifically, it deals with the process of reducing and maintaining the temperature of a space (a chamber or room) below the temperature of the surroundings.

1.1 OBJECT

A cold storage is a space maintained at a temperature below the surrounding in order to keep a product at the desired temperature [21]. However, the overall thermodynamic performance of the system is determined by many factors. This research aims at ascertaining the specific influences of frost formation on the evaporators of cold storage units. It aims at recommending procedures for reducing those influences that are adverse.

From basic heat transfer knowledge frost on evaporators will have the following effects:-

(i) It will cause an increase in thermal resistance and hence a drop in the heat transfer rate (cooling capacity).

(ii) It will bring about a reduction in temperature potential causing a reduction in effective heat load.

(iii) because of reduced cooling capacity there is the tendency for the amount of superheat to drop and this may result in wet vapour entering the compressor.

Frequent defrosting to remove the above adverse effects is therefore necessary. The defrosting system however, is dependent on the cooling capacity and air condition in the storage space. This research aims at recommending defrosting cycles with varying initial air conditions.

1.2 MOTIVATION

One motivation that carried me to study frosting and humidity in refrigerating systems stems from my experience with cold rooms for vaccine storage, at the National Veterinary Research Institute, Vom, Nigeria. Here vaccines are stored in these cold rooms at about $-20^{o}C$. It is observed that during the wet season or to be more specific, during the period just before the rains arrive, when the relative air humidity is very high, frost begins to occur on the vaccine bottles in addition to the formation on the evaporator unit and this is undersirable. Furthermore, during this period, the assigned temperature ($-20^{o}C$) is hardly met.

The second motivation, still from my experience at N.V.R.I, Vom, came from the observation that there seems to be seasonal variations of refrigeration loads which

affect the running time of the cold rooms. These cold rooms control both humidity and temperature and the seasonal variations in refrigerating loads are observed to affect this function.

The observations have moticated me to find the causes of these happenings and if possible design or recommend their remedy.

1.3 LAY-OUT OF THE RESEARCH

With the goal proposed by the motivations, the next task is to set out the possible experimental procedure to achieve this goal.

A working cold storage is chosen and measurements are required of its performance under frosting conditions at varying initial relative air humidities. The tests are performed under conditions as closely as possible to those obtained with the cold rooms at N.V.R.I., Vom. Steam is injected into the storage space at varying amounts to increase the space relative air humidities.

These measurements together with their graphical representations are presented.

In the discussion of the results, attempt has been made to correlate the experimental findings with the practical observations. On the basis of these results, a conclusion is drawn.

The theory of frost formation is analysed, indicating the governing basic equations and relations (Chapter 3)

A design procedure of a cold storage is also briefly given (Chapter 4) Appendices A, B, C and D deal with the fundamentals of Refrigerating processes, Refrigeration system components, Air Humidity and Defrosting methods respectively.

CHAPTER 2

LITERATURE SURVEY

It is worthwhile mentioning here some of the important efforts and investigations on the complex process of frost formation. P.Sale, C. Oesseux and I. Simonato [26] together investigated the effects of frost on the heat transfer coefficient of extended surface air coolers, tested in a cold chamber where real operating conditions were simulated . Investigations were also made on the testing of air coolers under frosting conditions, carried out on the stand of the aerodynamic channel type where the air conditions could be varied within broader limits [15, 14, 17, 25]. However, testing by this method was found to have the serious disadvantage that the conditions differ to a greater or lesser degree from those of real conditions.

K.O. Beatty, E.B. Finch and E.H. Schoenborn [8] investigated the effects of frosting on the transfer of heat from humid air to a metal surface under frosting conditions. Measurements were made on a four-foot long surface and a four-in long unit, showing some of the influence of air velocity, air conditions and cold surface temperature.

Frost thickness was measured by probing to the frost surface as seen through the transparent tubing but value depended on the care and judgement of the operator, owing to the fuzzy nature of the frost. In conclusion, they suggested the use of an enthalpy driving force concept to predict heat transfer rates under frosting conditions.

Another investigation was made by T.S. Gatchilov [12] who studied the characteristics of extended surface air coolers during operation under frosting conditions. For different working conditions, they determined the influence of the relative inlet air humidity, the evaporator temperature difference and the working time duration on the air flow rate, the frost layer thickness, the frost quantity formed and the heat transfer coefficient.

Other investigators of importance included W.H. Stoeker [27] who also studied frost formation on refrigerating coils. S. Kamei, T. Mizushina and T. Koto [20] undertook research on frost formation in low temperature dehumidifiers.

The above mentioned investigations no doubt provide considerable contributions to the problems and many of the features of the complex phenomenon of frost formation.

Unfortunately the results obtained are not universally applicable because of the presence of many difficulties during testing, for instance, measurement of the frost layer thickness. This means that these results apply only to individual constructions and operating conditions considered. Therefore any efforts at further clarifying the influence of frost formation on refrigerating coils may be of considerable practical value.

CHAPTER 3

FORMATION OF FROST

3.1 HEAT TRANSFER:

The laws of heat transmission are of controlling importance in the design and operation of many diverse forms of steam generators, furnaces, preheaters, exchangers coolers, evaporators and condensers in numerous industries like Refrigeration and air-conditioning. In some cases, the primary objective is to obtain the maximum heat transfer rate per unit surface compartible with economic factors. In other cases the objective is the salvaging of heat as in exchangers, recuperators and regenerators, In still other cases the purpose is to minimise the heat flux by the use of insulation.

Frequently in a given problem like in Regrigeration and Air-conditioning all the three objectives are important.

3.2 MECHANISM OF FROST FORMATION

When a mixture of air and water vapour (humid air) is exposed to a surface below the freezing point of water, some frost forms. The compound mechanism of heat transfer and dehumidification is made more complex. A layer of frost forms on the cooling surface and a film of a mixture of air and water vapour forms next to the frost layer.

The concentration of vapour in this air film is lower than in the main body. The concentration difference causes diffusion (mass transfer) of vapour from the main body through the air film forming frost on the frost-air interface. In this way air is simultaneously cooled and dehumidified by contact with a surface which is undergoing a continuous change in temperature and in nature because of the build-up of frost or ice. That is, air is cooled by convection heat transfer and dehumidified by diffusion of water vapour across the air film at the frost-air interface due to difference in partial pressure [24].

ANALYSIS:

Total heat transferred is given by

$$q/A = h_e (t_a - t_s) \text{---------- (3-1)}$$

and is made up of (1) sensible heat and

(2) latent heat lost by the vapour.

The rate of frost formation is governed by the laws of diffusion of vapour through the air-film, whereas, sensible heat transmission is governed by the laws of heat transfer by conduction and convection.

The total heat flux is given by

$$\frac{dq}{dA} = h_G(t_a - t_f) + k_G r(H_a - H_f) \text{----------(3-2)}$$

Lewis [24] proposed an approximate relationship that

$$\frac{h_G}{k_G} = C = - \quad - \quad - \quad - \quad - \quad - \quad - \quad - \quad - \qquad (3\text{-}3)$$

= Humid heat of moist air.

If this holds then equation (3-2) may be transformed into enthalpy driving force equation. The enthalpy of humid air is given as $i = ct + rH$ --- ------- (3-4)

Hence for the bulk air, the enthalpy is

$$i_a = ct_a + rH_a$$

and for humid air at the frost-air interface, the enthalpy is

$$i_f = ct_f + rH_f.$$

Then $i_a - i_f = c(t_a - t_f) + r(H_a - H_f)$ Multiplying both sides by k_G,

$$k_G(i_a - i_f) = ck_G(t_a - t_f) + rk_G(H_a - H_f)$$

The right hand side of this equation is same as that of equation (3-2) after replacing h_G with ck_G. Therefore

$$\frac{dq}{dA} = k_G(i_a - i_f) \quad - \quad - \quad - \quad - \quad - \quad - \qquad (3.5)$$

Define h_i, coefficient for enthalpy transfer as

$$h_i = \frac{dq/dA \text{ at } x_f = 0}{(i_a - i_s)} \qquad -- \quad -- \quad -- \quad -- \qquad (3\text{-}6)$$

That is, at the beginning (when frost thickness $x_f = 0$) it may be assumed that the frost-air interface temperature is substantially the metal surface temperature t_s and

enthalpy at this temperature is

$$i_s = ct_s + rH_s.$$

h_i is numerically equal to k_G [8] therefore

$$\frac{dq}{dA} = h_i(i_a - i_f) \text{ ------- ------} \qquad (3\text{-}7)$$

the enthalpy driving force equation. Unless t_f is known, equation (3-7) cannot be integrated. In general t_f varies with time because of the continuous build-up of frost. Assuming that all of the heat delivered to the frost-air interface must pass through the entire frost layer, then

$$\frac{dq}{dA} = k_f \frac{(t_f - t_s)}{x_f} = h_i(i_a - i_f) \quad \ldots\ldots\ldots \quad (3\text{-}8)$$

$$\text{i.e.} \quad \frac{k_f}{x_f}(t_f - t_s) = h_i(i_a - i_f)$$

But $i_f = ct_f + rH_f$ Substituting for i_f in above equation we have

$$\frac{k_f}{x_f}(t_f - t_s) = h_i i_a - h_i(ct_f + rH_f)$$

$$\frac{k_f}{x_f} t_f + ch_i t_f = h_i i_a + \frac{k_f}{x_f} t_s - rh_i H_f$$

$$\therefore \quad t_f = \frac{h_i i_a + \frac{k_f}{x_f} t_s - rh_i H_f}{\frac{k_f}{x_f} + chi} \text{ ----------} \qquad (3\text{-}9)$$

Assuming that air at the frost-air interface is saturated and if k_f, the thermal conductivity of frost is known equation (3-9) may be solved at any value of frost thickness x_f.

At any time for which the value of x_f/k_f is known, the performance of an apparatus may be predicted by step-wise integration over the area. The change in performance with x_f/k_f may also be predicted.

However, for certain design problems, one may be concerned with predicting the manner in which the TOTAL HEAT TRANSFER RATE will vary with TIME, rather than how it will vary with the frost thickness itself. It is then necessary to predict the relationship between frost thickness and time in order that t_f may be found from equation (3-9) and the value obtained be used in solving equation (3-7).

Consider a differential length over which the interface temperature and air properties, may be considered constant. In this case the value of x_f is given by [8]

$$x_f = k_G \int_o^t \frac{H_a - H_f}{\rho_f} dt \quad \text{----------} \qquad (3\text{-}10)$$

Since the frost density ρ_f continuously varies with the changing conditions as deposition progresses, and since H_f is also a function of the frost thickness, the integration of this equation becomes virtually impossible barring the use of many unwarranted assumptions. The average frost thickness $\bar{x}_f$ over the entire surface A, may also be expressed as

$$\bar{x}_f = k_G \int_o^t \int_o^A \frac{1}{\rho_f} (H_a - H_f) dA dt \text{ ----------} \quad (3\text{-}11)$$

This equation is subject to the same integration difficulties as equation (3-10) and is further complicated by the fact that ρ_f is probably a function of position along the length of the tubing.

Although it is apparent that the general solution cannot be obtained from these equations, the flux may readily be determined for the two unique conditions representing the limits of operation. The first of these is the initial condition at which the frost thickness $x_f = 0$ and t_f may be taken as $t_f = t_s$, the surface temperature of the tube. Where the dew point of the air is above 0^oC, the second condition obtains when t_s reaches 0^oC. In each of these conditions i_f is known, is constant and may be used to integrate equation (3-7) .

When $t_f = 0^oC$, frost thickness is given by

$$\left(\frac{x_f}{k_f}\right)_o = \frac{0 - t_s}{(i_a - 28.6)} \quad - - - - - - - \quad (3\text{-}12)$$

x_f is dependent on k_f [16]. When frost surface is at 0^oC, it is expected that the water vapour will condense as a liquid and if the frost is porous, this liquid will soak into the frost layer and gradually freeze producing a continuous increase in k_f approaching the value for solid ice. Until the maximum value of k_f is reached, the frost thickness may continue to increase even though the surface temperature is 0^oC. When this maximum is reached dehumidification will still take place with the liquid water continuously draining off.

But where the air dew point is below 0^oC deposition of frost will stop when the frost surface reaches the dew point. Since heat transfer at this point is by sensible cooling alone, equation (3-1) reduces to

$$\frac{dq}{dA} = h_G(t_a - t_f) \quad -- \quad ------ \quad -- \quad (3\text{-}13)$$

It may be useful to predict the value of thermal resistance, $1/h_e$ as a function of frost thickness, x_f. In order to do this use may be made of the following parametric equations:

Recalling that

$$\frac{dq}{dA} = h_e(t_a - t_s) \quad \text{- - - - - - - - - - - - (3-1)}$$

and

$$\frac{dq}{dA} = h_i(i_a - i_f) \quad \text{- - - - - - - - - - - - (3-7)}$$

$\therefore \quad h_e(t_a - t_s) = h_i(i_a - i_f)$

or $$\left(\frac{1}{h_e}\right) = \left(\frac{1}{h_i}\right)\left[\frac{t_a - t_s}{i_a - i_f}\right] \quad \text{----------} \quad (3\text{-}14)$$

also

$$\frac{dq}{dA} = k_f \frac{(t_f - t_s)}{x_f} \quad \text{- - - - - - -} \quad (3\text{-}8)$$

combining (3-8) and (3-7), we have

$$\frac{k_f}{x_f}(t_f - t_s) = h_i(i_a - i_f)$$

or $$\left(\frac{x_f}{k_f}\right) = \left(\frac{1}{h_i}\right)\left(\frac{t_f - t_s}{i_a - i_f}\right) \quad \text{-------- --} \quad (3\text{-}15)$$

From the condition of the problem, i_a and t_s are known. The enthalpy transfer coefficient, h_i may be calculated by using equation (3-3), noting that h_i is numerically equal to k_G. Since i_f is taken as enthalpy of saturated air at t_f, the parametric equations (3-14) and (3-15) may now be solved for $\frac{1}{h_e}$ and $\frac{x_f}{k_f}$ respectively for a series of arbitrary values of t_f. Assuming k_f is a constant and is known, x_f may be determined and $\frac{1}{h_e}$ plotted against x_f.

3.3. DEFROSTING

The evaporator (cooling unit) is the coldest spot inside the storage space; if it is operating below the freezing point of water ($0^{\circ}C$), unless prevented by some special means, it will collect frost. But where this frost is formed, provision must be made to get rid of it. Defrosting is therefore a process of removing frost from the evaporator surface. The common defrosting methods are discussed in Appendix D.

CHAPTER 4

DESIGN OF A COLD STORAGE

4.1 INTRODUCTION:

Designing a cold storage, generally aims at sizing and balancing the various components from the performance and economic point of view. For this research project, a cold storage is not designed from the scratch owing to the difficulties of building an enclosure to meet the requirements mentioned in appendix B.6 (for non availability of materials and equipment). In view of this, a house-hold refrigerator cabinet is adapted for a storage enclosure where the requirements are assumed to have been considered. This cabinet is however modified to accommodate the experimental arrangement.

It is then required to size the other components in order to produce a system that is suitable for the tests in anticipation. The components to be sized and balanced are

(1) the evaporator (2) the condenser and (3) the refrigerant charge.

There is need too to select the suitable

(1) compressor (2) expansion valve and (3) refrigerant lines.

The proper assemblying of the above components constitute a working cold storage unit.

4.2 DESIGN CALCULATIONS

4.2.1 DATA

Temperature of Surroundings $t_o = 27^oC$

Evaporator Temperature Difference $TD_e = 11.1^oC$

Temperature of Refrigerated Space $t'_o = 5^oC$

Evaporating temperature $t_e = -5^oC$

Condensing Temperature $t_c = 40^oC$

Dry bulb temperature of entering air $tdb_1 = 25^oC$

Volume of Cabinet $V = 0{:}28m^3$

Usage factor (from table E.1 appendix E) $F = 41\ ^W/_{m^3}.^oC$

Thermal conductivity for Copper (table E.3 appendix E) $K = 142\ ^W/_{m}{}^2{}^oC$

Rated Relative Humidity $\phi_o = 75\%$

Assumptions:

1. Usage Load only
2. Dry Expansion Evaporator (Bare tube)
3. Average or Normal Usage.

4.2.2 EVAPORATOR SIZING

Neglecting cooling loads due to

(i) Leakages through coldroom walls (wall gain Load)

(ii) Other sources (supplementary Load) (refer to Appendix E)

total cooling load is therefore

$$Q_t = Q_u \qquad \text{----------} \qquad (4\text{-}1)$$

Usage Load is given by equation E-2 (Appendix E)

$$Q_u = F\, v(t - t_o) \text{- - - - - -} \quad (E\text{-}2)$$

For reach-in cabinets at average or normal usage

$F = 41\ W/m^{3o}C$ (from data)

$$\therefore\ Q_u = (41\ W/m^{3o}C)(0.28m^3)(27\text{-}5\ ^oC)$$

$$= (41)(0.28)(22)W$$

$$= 252.6W$$

Heat Transfer through the walls of the evaporator is by conduction and is given by

$$Q_e = A_e K(TD_e) \text{ ---- ----- } (B\text{-}2)\text{(from appendix B).}$$

It is assumed that cooling load is due to usage only

$$\therefore\ Q_e = Q_t = Q_u = 252.6W$$

From equation (B-2)

$$A_e = \frac{Q_e}{K(TD_e)} = \frac{252.6W}{142\ W/m^2\ ^oC)(11.1^oC)}$$

$$= \frac{252.6}{(142)(11.1)}m^2$$

$$= 0.16m^2$$

From table E.2 (appendix E) it takes 33.477 linear meter of 9.53mm diameter copper tubing to make $1m^2$

∴ this evaporator requires

$$(\frac{0.16m^2}{1m^2})\ (33.477)$$

or $\underline{5.4m}$ of 9.53mm diameter copper tubing. The diameter 9.53mm is chosen because it is easily obtained.

4.2.3. CONDENSER SIZING

From practical experience it has been found that due to some losses in the system, heat transfer in the condenser is 1.25 times that in the evaporator [21]

[Heat in condenser = heat in evaporator + heat of compression]

Hence in this system, heat in condenser

$$Q_c = 1.25Q_e = 1.25Q_u \text{ - - - - - - - - } (4\text{-}3)$$

$$= (1.25)(252.6)W$$

$$= \underline{315.8W}$$

Here too heat transfer through the condenser walls is by conduction and is given by

$$Q_c = A_c K_c\ (ts) \text{ - - - - - - - - } (4\text{-}4)$$

again using copper tubing

$$A_c = \frac{Q_c}{K_c(t_s)} = \frac{315.8W}{142.W/m^{2o}C)(40\text{-}25)}$$

$$= \frac{315.8}{(142\ )(15)}m^2$$

$$= \underline{0.148m^2}$$

From table E.2 it takes 50.16 linear, meter to make $1m^2$ of 6.35 mm diameter copper tubing.

∴ this condenser requires

$$\left(\frac{0.148m^2}{1m^2}\right)(50.16)$$

or 7.42m of 6.35mm copper tubing.

This diameter is easily available in the market.

4.2.4 REFRIGERANT CHARGE

(From Evaporator Analysis)

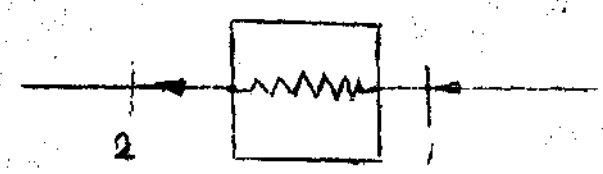


Fig. 4.1 Evaporator Unit.

$t_1 = 40^oC$

$t_2 = -5^oC$

From tables of thermodynamic and transport properties of fluids [22]

at $t_2 = -5^oC$ $\quad h_2 = h_g = 185.4kJ/k_g$

at $t_1 = 40^oC$ $\quad h_1 = h_f = 74.6kJ/k_g$

Refrigerating effect is given by

$$Q_{ref} = h_2 - h_1 \text{ ----------} \qquad (4-5)$$

$= (185.4 - 74.6)kJ/kg$

$= 110.8kJ/k_g$

i.e $1k_g$ of refrigerant gives 110.8kJ of refrigeration.

$\therefore$ Cooling capacity of Q_{cc} = 252.6 J/sec requires the circulation of $\dot{m}$ refrigerant given by $m = \frac{Q_{cc}}{Q_{ref}}$ ----(4-6)

$$\therefore \quad m = \frac{252.6 \times 10^{-3} \text{kJ/s}}{110.8 \text{kJ/kg}}$$

$$= 2.28 \text{ g/s}$$

4.2.5. COMPRESSOR SELECTION

All work on the refrigerant in the compressor is converted into heat. It has been shown too that the difference in enthalpies of vapour entering and leaving the compressor is a measure of the power necessary to compress it (appendix B.2)

$t_2 = -5^oC$

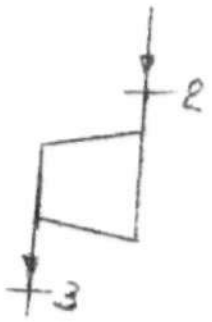


Fig. 4.2 Compressor Unit.

Entering the Compressor:-

at $t_2 = -5^oC$, $h_2 = h_g = 185.4 \text{kJ/kg}$

$s_2 = s_g = 0.6991 \text{ kJ/kg}^oK$.

from table of thermodynamic properties [22].

Leaving Compressor: assuming process is isentropic.

$$s_3 = s_2 = 0.6991 \text{ kJ/kg } ^{o}K.$$

From tables again, absolute condensing pressure at condensing temperature of $40^{o}C$ is Pabs = 9.607 bar. For $s_3 = 0.6991$ kJ/kg^{o}K, and Pabs = 9.607 bar, from super heat tables of Freon 12

$$h_3 = 204.80 \text{ kJ/kg.}$$

Work of compression is given by

$$Q_w = h_3 - h_1 \quad \text{(8-6)}$$

$$Q_w = (204.6 - 185.4)\text{kJ/kg}$$

$$= 19.4 \text{ kJ/kg}$$

Theoretical power in put

$$P_{th} = \dot{m}Q_w$$

$$= (0.0228\times10^{-1}\text{kg/s})(19.4 \text{ kJ/kg})$$

$$= 442 \text{ W}$$

i.e a compressor of 442 W capacity is required for the circulation of 2.28g/sec of refrigerant.

4.2.6 REFRIGERANT AND REFRIGERANT CONTROL

For this system, Freon 12 is chosen as the refrigerant. An externally equalized thermostatic expansion valve of 'Danfoss' make (TE2 type) is selected. This has the principal advantage that control of refrigerant flow into the evaporator is done by both the temperature of evaporator

outlet and the low-side pressure; it gives completely automatic evaporator flow control, responding readily to variation in load, and balancing the entire coil without allowing and flóodbacks.

4.2.7 Refrigerant Lines

In choosing or sizing refrigerant lines a compromise must be reached between low pressure drops and high velocities.

Pressure Drop

Background:

Whenever a fluid flows through a tube or in general contacts another substance while moving, it looses pressure. This pressure drop depends on

(i) specific flow pattern

(ii) specific void fraction

(iii) coil geometry and orientation

(iv) fluid quality

and it is the result of thrae basic phenomena

(a) acceleration B (b) gravity and (c) friction Gravitational pressure drop is usually negligible in Refrigerating systems. Single-phase frictional pressure drop calculations are reasonably well understood and amply documented to permit accurate evaluation for any Newtonian fluid.

Frictional pressure drop

$$\Delta p_{fr} = 4f \frac{L}{D} \rho \frac{\bar{v}^2}{2g} \quad - - - - - - - - - - - - - - - \quad (4\text{-}6)$$

acceleration pressure drop

$$\Delta p_{ac} = \frac{1}{g} \nu G^2 \quad - - - - - - - - - - - - - - - - \quad (4\text{-}7)$$

These expressions are valid for either liquid or vapour (single-phase) flow but not two-phase flow. However, in evaluating two-phase pressure drop, a simple analysis for freon refrigerant was developed by J.R.S. Thom [13] who stated that

$$\Delta P_{TPT} = r_2[\Delta P_{ac}] + r_3[\Delta P_{fr}] \text{----------} (4\text{-}8)$$

where Δp_{ac} and Δp_{fr} are the single-phase values of acceleration and friction pressure drop respectively with correction factors r_2 and r_3.

Calculations:

For this system only frictional pressure drops are considered in choosing the refrigerant lines (suction, Discharge and Liquid Lines).

Suction Line:-

at $-5^{o}C$ and 2.6 bar, refrigerant vapour density

$\rho_g = 15.38 \text{ kg/m}^3$

and dynamic viscosity $\mu = 316.72 \times 10^{-6} \text{kg/m.s}$.

Refrigerant mass flow rate

$$\dot{m} = \rho_g A \bar{V} = 0.228\times10^{-2}\ kg/s$$

$$\text{or } \bar{V} = \frac{\dot{m}}{\rho_g A} = \frac{\dot{m}}{\rho_g (\pi/4)(d)^2} \qquad \text{-------- (4-9)}$$

$$= \frac{(4)(0.228\times10^{-2})}{(15.38)(\pi)(9.53\times10^{-3})^2}$$

$$= \underline{2.1 m/s}$$

$$\text{Reynolds Number } R_e = \frac{\rho d \bar{v}}{\mu} \qquad \text{-------- (4 - 10)}$$

$\therefore$ For the refrigerant vapour flow in suction line

$$R_e = \frac{(9.53\times10^{-3})(15.38)(2.1)}{316.72\times10^{-6}}$$

$$= \underline{\underline{970}}$$

Flow is laminar.

For laminar flow in smooth pipe the friction factor, f is given by

$$f = \frac{16}{R_e} \qquad \text{-------- (4-11)}$$

$$= \frac{16}{970} = 0.0165$$

from equation (4-6)

$$\therefore = \frac{\Delta p_{fr}}{L} = \frac{2\rho}{d} \cdot \frac{\bar{v}^2}{g} f \qquad \text{-------- (4-6)}$$

$$= \frac{(2)(15.38)(2.1)^2(0.0165)}{(9.53\times10^{-3})(9.81)}$$

$$= \underline{\underline{0.024\ bar/m}}$$

Similar calculations for the discharge and liquid lines of 6,35 mm diameter tubing give

Discharge Line : $\frac{\Delta p_{fr}}{L}$ = 0.03 bar|m

Liquid Line : $\frac{\Delta p_{fr}}{L}$ = 0.0064 bar|m.

The total pressure drops for the lines are as tabulated below.

	Line Size a(mm)	Velocity of flow $\bar{V}$(m/s)	Pressure Drop Δp_L bar/m	Length of tubing L(m)	Total Pressure Drop Δp (bar)
Suction Line	9,53	2.1	0,024	0,183	0,0044
Discharge Line	6,35	1.3	0,03	0,152	0,0046
Liquid Line	6,35	0.3	0,006	0,162	0,00097

The higher the velocity of flow, the greater the pressure drop, this change forces the refrigerant to flow in the lines. In choosing the lines of any refrigerating system, the lines should be of sufficient size that pressure drops do not interfere with the efficiency of operation. In practice, charts are provided in selecting the lines by either the pressure drop method or the velocity method.

4.4 THE COMPLETE UNIT

The evaporator (bare tube and dry expansion) and the condenser (air cooled) are formed into 16 coils (fig. 4.3) and 17 coils (fig. 4.4) respectively. The various components are assembled together, carefully joining the lines by brazing- suction line (from evaporator to compressor), discharge line (from compressor to condenser) and liquid line (from condenser to expansion valve. An additional component, the drier, is installed to remove any moisture content from the liquid refrigerant or this may freeze out at the expansion valve forming an ice plug which will block refrigerant flow.

The system is charged with 2.8g/s freon 12 refrigerant and 149W compressor. Figure 4.5 is a sketch of the complete assembly unit illustrating components arrangement and refrigerant circuit.

CHAPTER 5

EXPERIMENTS, RESULTS AND DISCUSSION

5.1 EXPERIMENTAL ARRANGEMENTS

A household refrigerator cabinet is adapted for a storage enclosure. Evaporator and Condenser (dry expansion bare coil type and air-cool type respectively) are designed and constructed. A hermetic compressor of 149.0W capacity is selected to match. Freon 12 refrigerant and an externally equalized thermostatic expansion valve are also sized for the system.

Two pressure gauges are mounted at inlet and outlet of the evaporator. Three copper-constantan thermocouples are soldered, one to the evaporator, one to the inlet to the expansion valve and the third to the outlet of the evaporator.

A sling psychrometer (wet - and dry - bulb thermometers) is mounted to the inside of the cabinet space as shown in Fig. 5.1. The cabinet space is connected to a boiler for supply of steam.

A cold-junction of $-10^{o}C$ - $-20^{o}C$ is obtained with a 1:3 salt/crushed ice mixture.

5.2. EXPERIMENTAL PROCEDURE

The general procedure here is to make measurements of the refrigerating effect (Q_{ref}) [a direct measure of the rate of heat transfer between the humid (moist) air in the refrigerated space and the evaporator coils]. The cooling Load on the system could be regarded as that due to air infiltration by door openings - the condition of surrounding air.

Scope of Experiment

The test was first run with air at the prevailing atmospheric condition. Initial condition of the air was recorded by the sling psychrometer (the wet - and dry-bulb thermometer readings - t_{db} and t_{wb} respectively). Moisture content and Relative Humidity were deduced. The equipment was allowed to run and at intervals of 15 minutes, readings of t_{db}, t_{wb} and t_1, t_2, t_s corresponding to the thermocouple positions were recorded. This continued until t_{db} was fairly constant ($t_{db} \approx t_a$).

For the next runs, steam at 7.0 bar was injected into the cabinet space and allowed to diffuse. A new air condition was obtained when t_{db} was fairly steady. The equipment was again switched on and the above set of readings taken. This was repeated for further injecting steam.

After each test run, the frost formed was observed and apparatus allowed to defrost completely by the off-cycle method.

5.3 CALCULATIONS AND RESULTS

5.3.1 Total evaporator length used

$L_e = 5.433m$

Diameter of tubing used

$d = 9.53mm$

∴ Total surface area of evaporator coil

$$A_e = \pi dL_e$$
$$= (\pi)(9.53\times10^{-3}m)(5.433m)$$
$$= \underline{0.163m^2}$$

Effective Coefficient of Heat Transfer (h_e)

Effective Coefficient of transfer of heat from the humid air in the cabinet to the evaporator surface is defined as

$$q/A = Q_{cc}/A_c = h_e(t_a - t_s) \quad - - - - - - - - \quad (5\text{-}1)$$

$$\text{or } h_e = \frac{Q_{cc}/A_e}{t_a - t_s} \quad ---------- \quad (5\text{-}2)$$

This coefficient includes

(i) the effect of the frost deposit (thermal resistance of the frost layer).

(ii) the effect of sensible and latent heat transfer rates.

The cooling capacity, Q_{cc}, is determined by heat balance on the basis of the refrigerant.

5.3.2 REFRIGERATING EFFECT (Q_{ref}) AND COOLING CAPACITY (Q_{cc})

This is cooling capacity per unit mass of refrigerant.

From Fig. 4.1.

$$Q_{ref} = (h_2 - h_1) \ \frac{kJ}{kg} \qquad \text{----------} \ (5\text{-}3)$$

and $$Q_{cc} = m.(h_2 - h_1) \ \frac{kJ}{sec} \qquad \text{--------} (5\text{-}4)$$

From the charging of the system

$$m = 2.8g/s$$

$$\therefore \quad Q_{cc} = 2.8(h_2 - h_1) \qquad \text{---------} (5\text{-}5)$$

5.3.3 MOISTURE CONTENT (ω)

Volume of dry air

$V_a = 0.28\ m^3$ = Volume of cabinet, mass of dry air

$m_a = \rho_a V_a$

at 1atm,

$\rho_a = 1.177\ kg/m^3$

$$\therefore \quad m_a = [1.177\ kg/m^3][0.28m^3]$$

$$= 0.33kg.$$

But $$\omega = \frac{m_v}{m_a} \qquad \text{----------} (c\text{-}1)$$

$$m_v = \omega m_a$$

$$= 0.33\omega\ [kg] \qquad \text{--------} (5\text{-}6)$$

Equations (5-2), (5-5) and (5-6) are used in computing the parameters indicated and results tabulated as shown in tables of results.

TABLES OF CALCULATIONS AND RESULTS

RUN 1 - Table 5.1a

Initial Air condition:-

Dry Bulb Temp T_{ab} = 26°C

Wet Bulb Temp T_{wb} = 16.2°C

Relative Humidity ϕ = 35%

t (min)	T_1 (°C)	h_1 (kJ/kg)	T_2 (°C)	h_2 (kJ/kg)	h_2-h_1 (kJ/kg)	T_a (°C)	T_s (°C)	T_a-T_s (°C)
0	36	70.558	3	188.748	118.190	25	-3	28.
15	35	69.550	3	188.748	119.198	22	-4.5	26.5
30	34.7	69.590	2	188.382	118.832	19	-4	23
45	35.3	69.350	-3.7	185.810	116.260	17	-5	22
60	36	70.558	-2.2	186.670	116.110	16	-5	21
75	36	70.558	-4.1	185.810	115.252	15.5	-6	21.5
90	38	72.574	-3.0	186.240	113.666	15.3	-6	21.3
105	38	72.574	-5.0	185.380	112.806	15.0	-6	21
120	39	73.582	-6	184.942	111.360	14.5	-6.5	21
135	40.	74.590	-7	184.904	110.310	14.5	-6	20.5
150	41	75.534	-8.1	184.066	108.532	14.5	-6.5	21

RUN II - Table 5.1b

Initial Air Condition:-

Dry Bulb Temp T_{db} = 28.2°C

Wet Bulb Temp T_{wb} = 22.3°C

Relative Humidity Φ = 55%

t (min)	T_1 (°C)	h_1 (kJ/kg)	T_2 (°C)	h_2 (kJ/kg)	h_2-h_1 (kJ/kg)	T_a (°C)	T_s (°C)	T_a-T_s (°C)	Q_c (kJ/s $\times10^{-1}$	Q_c/A kJ/s.m^2 $\times10^{-1}$	$\frac{Q_c/A}{T_a-T_s}$ (kJ/s.m $\times10^{-1}$
0	40.8	75.409	20	195.78	120.37	27	-2	29	3.37	20.677	0.713
15	39	73.582	14	193.272	119.69	23	-2	25	3.35	20.552	0.822
30	38	72.574	11	192.148	119.574	18	-4	22	3.35	20.540	0.934
45	36	70.558	6	190.096	119.518	15	-5	20	3.35	20.540	1.08
60	37	71.566	3	188.808	117.242	14	-5	19	3.28	20.14	1.06
75	39	73.582	7	190.492	116.910	13	-6	19	3.27	20.06	1.056
90	38	72.574	-7	184.904	112.330	12.8	-7	19.8	3.24	19.88	1.004
105	38	72.574	-5	185.380	112.806	12.5	-7	19.5	3.18	19.50	1.00
120	39	73.582	-5	185.380	111.798	12.3	-7-7	20	3.13	19.20	0.966
135	39.5	74.590	-5	185.380	110.790	12.2	-7.5	19.2	3.10	19.02	0.95
150	40	74.590	-8	184.066	109.476	12.2	-8.0	19.5	3.10	19.02	0.941

RUN III - Table 5.1c

Initial Air Condition:-

Dry Bulb Temp T_{db} = 30.5°C

Wet Bulb Temp T_{wb} = 25 °C

Relative Humidity Φ = 65%

t (min)	T_1 (°C)	h_1 (kJ/kg)	T_2 (°C)	h_2 (kJ/kg)	h_2-h_1 kJ/kg	T_a (°C)	T_s (°C)	T_a-T_s (°C)	Q_c kJ/s $\times10^{-1}$	Q_c/A kJ/s.m^2 $\times10^{-1}$	$\frac{Q_c/A}{T_a-T_s} = h_e$ kJ/s.m^2°C $\times10^{-1}$
0	40	74.590	22.4	196.732	122.142	29	-2	31.0	3.42	20.98	0.70
15	38	72.574	15.5	194.00	121.428	22	-3	25.0	3.40	20.86	0.83
30	36.5	71.062	11.8	192.47	121.428	17	-4	21.0	3.40	20.86	0.99
45	35	69.55	5.2	189.545	120.000	15.5	-3.8	19.3	3.36	20.61	1.07
60	34	68.588	1	187.956	119.398	15	-3.0	18.0	2.34	20.49	1.14
75	32	66.574	-2	186.67	120.096	14	-4	18.0	3.36	20.61	1.15
90	35	69.55	-2.7	187.401	117.851	12.5	-6	18.5	3.30	20.245	1.10
105	35	69.55	-5	185.38	115.830	12.1	-7	19.1	3.24	20.00	1.04
120	36	70.558	-5	185.38	115.822	12	-7	19.0	3.22	19.75	1.04
135	38	72.574	-7	184.904	112.332	11.8	-7.6	19.4	3.14	19.26	1.00
150	39.5	74.086	-7	184.904	110.814	11.8	-7.6	19.4	3.14	19.02	0.98

RUN IV - Table 5.1d

Initial Air Condition:-

Dry Bulb Temp T_{db} = 34.4°C

Wet Bulb Temp. T_{wb} = 30.0°C

Relative Humidity Φ = 75%

t (min)	T_1 (°C)	h_1 (kJ/kg)	T_2 (°C)	h_2 (kJ/kg)	h_2-h_1 kJ/kg	T_a (°C)	T_s (°C)	T_a-T_s (°C)	Q_c kJ/s $\times 10^1$	Q_c/A kJ/s.m^2 $\times 10^{-1}$	$\frac{Q_c/A}{T_a-T_s} = h_e$ kJ/s.m^2°C $\times 10^{-1}$
0	39	73.582	26	198.11	124.6	32	-2	34	3.40	22.81	0.670
15	36	70.558	19.5	195.58	125.00	23	-2	25	3.50	21.47	0.86
30	36.5	71.062	19	195.38	124.32	17.5	-3.5	21	3.48	21.35	1.02
45	35	69.550	9	191.32	121.77	14	-4	18	3.41	20.92	1.16
60	35	69.550	6.5	190.28	120.73	13	-4	17	3.38	20.74	1.22
75	34	68.558	5.0	189.66	121.10	12.5	-4.2	16.7	3.40	20.86	1.25
90	34	68.558	3.0	188.808	120.25	12.2	-5	17.2	3.37	20.67	1.20
105	34.5	69.062	2.0	189.382	120.32	12.1	-5.5	17.6	3.37	20.67	1.17
120	35	69.550	-0.5	187.315	117.77	11.6	-6	17.6	3.30	20.25	1.15
135	35	69.550	-2	186.67	117.12	11.4	-7	18.4	3.28	20.12	1.09
150	36	70.558	-3.7	185.94	115.38	11.0	-7	18	3.26	20.0	1.11

B). CHANGES IN AIR CONDITION-VALUES OF CHANGES IN MOISTURE CONTENTS

RUN 1 - Table 5.2a

t (min)	$T_{ab} = T_a$ (°C)	T_{wb}	Φ (%)	ω (kg_v/kg_{dryair}) $\times 10^{-3}$	$\times 10^{-3}$ mv (kg)	$\times 10^{-3}$ Δm_v (kg)	Total reduction in m_v (kg)$\times 10^{-3}$
						0.0	0.0
0	24	15.2	35	3.97	1.046	0.386	0.386
30	19	10.6	32	2.00	0.660	0.195	
60	16	7.8	27	1.41	0.465	0.062	0.581
90	15.3	7.0	25	1.22	0.403	0.06	0.643
120	14.5	5.8	22	1.04	0.343	0.043	0.703
150	14.5	5.6	20	0.91	0.300		0.746

RUN II - Table 5.2b

0	27	22	55	5.53	1.826	0.0 0.78	0.0
30	18	12.8	51	3.17	1.046	0.358	0.78
60	14	8.3	46	2.09	0.688	0.089	1.138
90	12.8	7.2	42	1.81	0.599	0.045	1.227
120	12.3	6.0	39	1.68	0.554	0.030	1.272
150	12.2	5.8	33	1.59	0.524		1.302

RUN III - Table 5.2c

t (min)	$T_{ab}=T_a$	T_{wb}	Φ (%)	ω ($kg_v/kgdryair$) $\times10^3$	$\times10^{-3}$ m_v (kg)	$\times10^{-3}$ Δm_v (kg)	Total reduction in m_v (kg)$\times10^{-3}$
0	30	29	65	7.98	2.63	0.0 1.43	0.00
30	17.0	14.4	63	3.63	1.20	0.257	1.43
60	15.0	9.4	60	2.86	0.943	0.195	1.687
90	12.5	8.0	56	2.27	0.748	0.104	1.882
120	12.0	7.0	50	1.95	0.644	0.045	1.986
150	11.8	6.0	45	1.81	0.599		2.031

RUN IV - Table 5.2d

0	33	30	75	10.88	3.59	0 2.24	0.0
30	17.5	14.4	72	4.08	1.35	0.55	2.24
60	13	10.6	65	2.72	0.90	0.15	2.79
90	12.2	8.3	60	2.27	0.75	0.11	2.94
120	11.6	7.2	55	1.95	0.64	0.06	3.05
150	11.0	6.7	51	1.77	0.58		3.11

Table 5.3 - Values of Constants a & b for some typical operating conditions.

	ϕ (%)			
	35	55	65	75
a %	36	54.5	67	74
b % min	-0.125	-0.125	-0.134	-0.148

Table 5.4 - Values of t_{cr} and ϕ_{cr} for some typical operating conditions.

	INITIAL RELATIVE HUMIDITY ϕ (%)			
	35	55	65	75
t_{cr}(min)	54	60	66	75
ϕ_{cr}(%)	28	47	58	63

Figs. 5.5a - 5.5d - Values of Φ_r and t_r for some typical operating conditions.

RUN 1 - Fig. 5.5a

$t_r = t/t_{cr}$	$\Phi_r = \Phi/\Phi_{cr}$
0.00	1.25
0.56	1.14
1.11	0.96
1.67	0.89
2.22	0.79
2.78	0.71

RUN II - Fig 5.5 b

$t_r = t/t_{cr}$	$\Phi_r = \Phi/\Phi_{cr}$
0.00	1.18
0.50	1.08
1.0	0.98
1.50	0.89
2.00	0.83
2.50	0.70

RUN III - Fig. 5.5c

0.00	1.16
0.45	1.09
0.91	1.03
1.36	0.96
1.82	0.86
2.27	0.78

RUN IV - Fig. 5.5d

0.00	1.19
0.40	1.14
0.80	1.03
1.20	0.95
1.60	0.87
2.00	0.81

5.4 DISCUSSION OF RESULTS

5.4.1 OBSERVATIONS & TABLES

It is observed that the distribution of frost across the evaporator coil is uneven. The frost surface is also not sharply defined because of the fuzzy nature of the frost deposit. These have made it practically difficult to measure the frost layer thickness. It is observed however, that the frost layer is greater at higher air humidity.

The heat transfer characteristic during frost formation (or 'on' time duration) is judged by the values obtained for the effective coefficient, h_e, given by equation (5-2) and the cooling capacity, Q_{cc}, given by equation (5-5). These values, for each test run, are presented in tables 5.1a - 5.1d.

In tables 5.1a - 5.1d is also shown the drop in air temperature (t_a) in the refrigerated chamber. The drop can be seen to be faster initially, corresponding to the increasing heat exchange, and then to a fairly constant values for a particular operating condition.

In a design problem, the mass of frost precipitated could be computed from the estimated change in moisture content where the surface temperature is below the freezing point of water.

But when the moisture phases out partly as frost and partly as liquid water, the problem becomes more complicated [9]. The quantity of moisture transformed is influenced by the relative air humidity and air temperature of the refrigerated chamber. This is shown in tables 5.2a - 5.2d for the different test runs.

There is also a change in relative air humidity in the chamber during the 'on' time duration of the system.

5.4.2 GRAPHS

Figure 5.2: This shows the variation of the cooling capacity, Q_{cc}, with time for four values of initial relative air humidity. In general, the higher relative air humidity has higher cooling capacity. This is probably due to the availability of more water or gas molecules acting as carriers for the heat which is naturally convected. The cooling capacity also decreases with time. Each curve of $Q_{cc} = f(t)$ tends to have a small slope initially and as the storage temperature approaches that of the coil, the cooling capacity tends to decrease very slowly to a values corresponding to 'dry' operating conditions. The trend is probably due to the formation of frost on coils which increases the thermal resistance and simultaneously decreases the temperature potential.

Figure 5.3:

There exists a considerable variation of heat exchange in comparison with 'dry' operating conditions. This could be attributed to the precipitation of frost on the evaporator coil. The variation may be judged by the character of the change in the heat transfer coefficient, h_e. Figure 5.3 is a plot of the variation of h_e with time. The exhibition of a maximum point for the curve $h_e = f(t)$ is very significant, corresponding to the most intensive heat exchange. Up to the maximum point, the cooling is probably done to supply only the necessary sensible heat for the air and moisture to attain the freezing temperature. Beyond this point, cooling is done only to offset the infiltration and to provide latent heat of fusion. The coefficient tends to a constant valve which is typical for the low relative air humidity. This may correspond to the trend of change in moisture content to a constant valve. The influence of relative air humidity on h_e can be seen from the same figure 5.3. At higher initial relative air humidity, the maximum point of the curve $h_e = f(t)$ is shifted to the right, the steepness of the curve is also increased and at the same time h_e reaches its value in conditions of dry heat exchange more rapidly.

This may probably be due to higher frost growth rate on the coils at higher relative humidity, causing greater thermal insulation.

Figure 5.4:

The variation of the quantity of moisture transformed (either in frost or liquid form) is of particular importance for the selection or design of the defrosting system. Figure 5.4 represents the relation $\Delta m_v = f(t)$, showing that the quantity of moisture precipitated or condensed increases initially and then tends to a constant value. It depicts the rate of 'freeze drying' obtained relative to an initial relative humidity. The qualitative trend of this graph may be given as

$$G_f = Ae^{(c.t)}$$

where $c = c(Q_{cc}, h_e)$

$A = A(t)$, defined as gradient from the origin, to curve $G = f(t)$ at time of consideration.

If the change in moisture content Δm_v, is considered as a measure of quantity of frost precipitated, then Fig. 5.4 may be interpreted as representing the frost growth rate $G_f = f(t)$.

Figure 5.5:

There is also a variation in coil surface temperature, t_s, with time. During operation, t_s is seen to decrease. The variation is linear as is shown in figure 5.5 and may correspond to increasing frost build-up. As the frost build-up progresses, the decreass in t_s means a colder refrigerant temperature which results in a lower back pressure and may affect the compressor.

Figure 5.6.

The relative air humidity in the chamber follows a linear variation with time as is depicted in figure 5.5. The relation $\phi = f(t)$ may be presented in the form $\phi = a + bt$ where a and b are constants evaluated for the operating conditions under consideration as is given in table 5.3.

Derived Graphs and Functions:

It may be useful to combine the variation of relative air humidity, ϕ, in the chamber and the variation of the coefficient h_e, both functions of time (figure 5.5 and 5.3 respectively). The combination or correlation may be useful in view of the fact that most cold storages are meant to control simultaneously air humidity and temperature. It is seen in figure 5.3 that at a higher relative air humidity the maximum of the coefficient h_e is higher leading

to a higher drying capability of the cold storage. Hence a correlation between Φ, h_a and time t may help in arriving at when to lower this maximum and also control air humidity and temperature in the chamber.

The time at which the maximum h_e is attained may be defined as the 'critical' time t_{cr}, this is found from figure 5.3 (values at four different initial relative humidities are shown in table 5.4). These values are used to read-off the corresponding Φ, designated Φ_{cr}, from Fig. 5.5. Using Φ_{cr} and t_{cr} as the reference values, Φ and t are 'reduced' and defined as

$$\Phi_r = \Phi/\Phi_{cr}$$

$$t_r = t/t_{cr}$$

Table 5.5 contains these 'reduced' values.

Figure 5.7:-

This is a plot representing the function $\Phi_r = f(t_r)$ which can be expressed as $\Phi_r = \alpha + \beta t_r$, α and β being constants evaluated to be $\alpha = 1.20$ and $\beta = -0.2$ giving

$$\Phi_r = 1.20 - 0.3t_r$$

Thus by scaling down Φ and t, a single function in the reduced form is obtained which describes the condition of air in the chamber for its operation.

Figure 5.8:-

For the expression

$$\phi_r = 1.20 - 0.2t_r$$

to be useful, a relation of the critical values ϕ_{cr} and t_{cr} is necessary. This relation is given by figure 5.8. The curve obtained is of the form

$$\phi_{cr} = m(t_{cr} - 40) \text{ with } m = 2.$$

$$\text{ie } \phi_{cr} = 2(t_{cr} - 40)$$

Therefore to describe air condition in the chamber at any given time during operation, these four equations are necessary

$$\phi_r = 1.20 - 0.2t_r$$

$$\phi_{cr} = 2(t_{cr} - 40)$$

$$\phi_r = {}^{\phi}/_{\phi_{cr}}$$

and $$t_r = {}^{t}/_{t_{cr}}$$

Example:

Consider for this system an initial air condition with relative humidity of 75%. It is required to find the condition after a time of 30 minutes of operation.

Solution

Here, at $t = 0, \rightarrow t_r = 0$,

$\phi = 75\%$

$\therefore$ from $\phi_r = 1.20 - 0.2t_r$

$\phi_r = 1.20$

and from $\phi_r = \phi/\phi_{cr}$,

$\phi_{cr} = \phi/1.2$

$= 75/1.2$

$= 62.5\%$

from $\phi_{cr} = 2(t_{cr} - 40)$

$62.5 = 2(t_{cr} - 40)$

or $t_{cr} = \frac{62.5}{2} + 40$

$= 71.25$min

or $t = 30$min.,

$t_r = t/t_{cr} = \frac{30}{71.25}$

$= 0.421$

Substituting in

$\phi_r = 1.2 - 0.2t_r$ we have

$\phi_r = 1.2 - (0.2)(0.421)$

$= 1.116$

ie $\phi/\phi_{cr} = 1.116$

$\therefore \phi = (1.116)(62.5) = \underline{69.75\%}$

That is, after 30 min of operation this system provides in the chamber air condition with relative humidity of 69.75%. This compares closely with the value of 72% obtained experimentally (table 5.2d).

5.5 CONCLUSIONS & RECOMMENDATIONS

The results present a picture of the influence of initial relative air humidity in the chamber (prevailing ambient air condition) on the change in the moisture content (a measure of the mass of frost precipitated under condition when the surface temperature of refrigerating coil is below the freezing point of water) and influence of this on the

effective heat transfer coefficient. On the basis of these tests and the experience obtained during their performance, the following conclusions and recommendations may be made for this system and similar ones.

(i) The defrosting cycle for the cold storage at the investigated operating conditions may be determined in accordance with

(1) the change in effective heat transfer coefficient

(2) the change in moisture content(or quantity of frost precipitated)

and (3) considerations of the design.

In a design where air humidity and temperature control is involued simultaneously, it could be inferred that more frequent defrosting is required at the initial stage of operation, in conditions of higher air humidity, in order to lower the point of maximum h_e.

(ii) The result on quantity of frost precipitated may be used as initial data for defrosting system calculations of similar cold storages operating under similar conditions.

(iii) The results propose two relationships

$$\Phi_r = 1.2 - 0.2t_r$$

$$\Phi_{cr} = 2(t_{cr} - 40)$$

where $\Phi_r = \Phi/\Phi_{cr}$, $t_r = t/t_{cr}$

that may be used to describe air condition in the chamber at any given time of operation for this system and similar ones. The 'critical' time may be used as the optimum time of operation before defrosting.

Recommendations for Further Work

It has been stated earlier (and from the foregone tests too) that frost formation is a big hazard in the Refrigerating Industry. Further work on this subject could be done with the aim of solving this problem. In view of this I make the following recommendations:

(i) Further efforts for preventing frost formation rather than employing defrosting methods (such methods are generally messy) will be useful. The use of foreign gas or gases, like N_2 , CO_2, etc to delay frosting may be investigated.

(ii) The validity of the derived equations in this report,

$$\phi_r = 1.2 - 0.2t_r$$

$$\phi_{cr} = 2(t_{cr} - 40)$$

where $\phi_r = \phi/\phi_{cr}$, $t_r = t/t_{cr}$,

which describe air condition in the refrigerated chamber under test at any time during its operation,

could be tested

(iii) Further attempts could be made in integrating the equations

$$x_f = k_G \int_0^t \frac{H_a - H_f}{\rho_f} dt \qquad \text{------(3-10)}$$

and $$\bar{x}_f = k_G \int_0^t \int_0^{\Delta} \frac{1}{\rho_f}(H_a - H_f) d\Delta dt \qquad \text{-------(3-11)}$$

Since the designer is more concerned in predicting the variation of Total Heat Transfer Rate with Time. The two expressions give the relationship between frost thickness and time.

OR (iv) It would be useful to further investigate on the proper means of measuring the frost layer thickness, x_f as the equipment is operating under frosting conditions. This establishes the frost thickness-time relationship and therefore the designer can easily predict the variation of total heat transfer rate with time.

APPENDIX A

REFRIGERATION PROCESSES

Refrigeration is one important aspect of thermal environmental control. The refrigerating effect may be produced by several means. Some of the common ones are:-

A.1 VAPOUR COMPRESSION SYSTEMS

This method is used in the majority, of refrigerati systems with compression accomplished mechanically. Cooling is accomplished by the evaporation of a liquid refrigerant (in the evaporator) under reduced pressure and temperature. Mechanical compression (in the compressor, elevates the temperature and pressure of the vapour. The high pressure vapour condenses (in condenser) by heat rejection. The relatively high pressure liquid is expanded (in the expansion value) and fed to the evaporator, thus completing a thermo-dynamic cycle [28]. Fig. A.1

A.2 VAPOUR ABSORPTION SYSTEMS

This provides the means for raising the pressure of the refrigerant without appreciably altering its volume. In this system, the evaporator, the condenser and expansion valve are quite similar to those used in the mechanical compression system. Fig. A.2

The compressor has been replaced by a heater and generator [28]. The vapour leaving the evaporator is absorbed by a weak liquid solution in the absorber by removal of heat, producing a solution stronger in the refrigerant. The pressure of this solution is raised to the generator pressure by the pump. By addition of heat in the generator, refrigerant vapour is driven out of the solution and to the condenser.

A.3 USE OF PHASE CHANGES

Flash cooling or partial vaporization of the liquid to be cooled, and ice cooling were the methods employed in early days. These are still in use today. Flash cooling is used in cooling and dehumidifying atmospheric air in comfort air-conditioning-and water ice is important commercially. Solid CO_2 or dry ice sublines at approximately -80^{o}C at atmospheric pressure and may be used in producing low temperatures.

A.4 THERMOELECTRIC COOLING

This is based on the principle that when current flows through a thermocouple (usually made of semi conductors) heat is relased at one junction while heat is absorbed at the other junction. This effect could be used for refrigeration purposes. Fig. A.3

A.5. STEAM-JET-EJECTION SYSTEM

In this system flashing of some liquid to vapour is accomplished by creating a 'vacuum' and the remaining liquid is cooled to the saturation temperature corresponding to the chamber pressure. Fig. A.4

A.6 MAGNETIC COOLING

Lower temperature may be achieved by adiabatic demagnetization of certain paramagnetic salts. This requires that work be done by the salt molecules at the expense of their internal energy, resulting in the lowering of salt temperature. The salt molecules may be considered as tiny magnets. [28] Fig. A.5

APPENDIX B

SYSTEM COMPONENTS

A compression refrigerating system is made up essentially of, and its performance depends on the proper functioning of the following components:

(1) Evaporator (2) Compressor, (3) Condenser (4) Expansion valve and (5) Refrigerant.

These components must be balanced for the system to operate properly at design conditions plus every other condition which may be expected during operation [4].

B.1 THE EVAPORATOR

Evaporating process is the purpose of the refrigerating system. Absorption of heat to effect evaporation is what constitutes COOLING.

An evaporator requires [10].

(1) Total surface proportional to the required load. This is obtained by making the evaporator in form of tube coils.

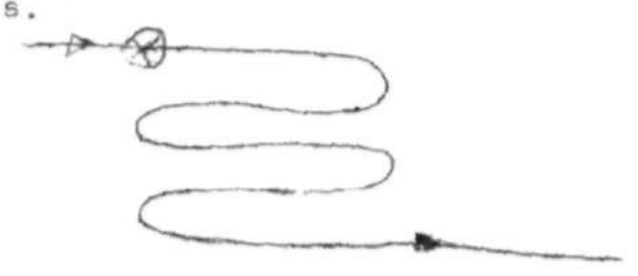


Fig. B.1 General form of evaporator.

(2) Chamber to hold the liquid refrigerant and to allow vapour separation.

(3) Circulation without excessive pressure drop. This is essential to avoid the clinging of bubbles on the inside walls of evaporator.

Two types of evaporators are found:

(a) flooded, when the inside surface is full of liquid refrigerant and.

(b) Dry Expansion with $\frac{1}{4}$ or $\frac{1}{3}$ full of liquid refrigerant.

Heat Transfer in Evaporator (: Evaporator Analysis).

Heat reaches the evaporator walls by conduction, convection and radiation and reaches the refrigerant by conduction through the walls. The capacity of evaporator is determined by the factors governing the rate of heat flow by conduction given by

$$Q = A\ U\ D \qquad \text{- - - - - - - -(B-1)}$$

or $$Q_t = AK(TD) \qquad \text{- - - - - - - -(B-2)}$$

Knowing heat transfer rate (Q_t), conductivity (K) of material of surface and the evaporator temperature difference (TD) the surface area of the evaporator can be calculated and therefore the size of the tubing.

The selection of an evaporator for a system is generally based on the temperature difference, TD.

COOLING LOAD

Associated with the evaporator is the cooling load. It is the load on the refrigerating equipment and is the summation of the heat which usually evolves from several different sources. Some of the common sources of heat are:

(1) Heat given off by the air from door openings.

(2) Heat that leaks into the refrigerated space from outside by conduction through the insulated walls and floor (leakage load)

(3) Heat that enters the refrigerated space through the ceiling top

(4) Heat given off by the warm product as its temperature is lowered to the desired level. (Product Load)

(5) Heat given off due to respiration by living products (Respircetory load)

(6) Heat given off by the ventilating air stream (Infiltratory load) and

(7) Heat given off by people occupying the refrigerated space and any other heat-producing equipment located inside the space, e.g. electric motors, lgiths etc [10]

B.2 THE COMPRESSOR

A compressor generally admits fluid from low pressure region, compresses it and delivers it to a high pressure region. In compression refrigeration, the compressor is the heart of the system. It supplies the forces necessary to keep the system operating [1].

Vapour compressors used in refrigeration are of three types.

(1) Reciprocating (2) Rotary and (3) Centrifugal

The reciprocating compressor is by far the most frequently used. For smaller units, the hermetic compressor is used. Compressor Analysis [23].

Considering steady flow reversible process in the compressor, the energy equation is

$$Q = \dot{m} \int_2^3 Tds \qquad \text{.---------- (B-3)}$$

But $Tds = dh - vdp$

$$\therefore \quad Q = \dot{m} \int_2^3 (dh - vdp) \qquad \text{- - - - - - - -(B-4)}$$

assuming adibatic process in compressor and applying 1st Law of Thermodynamics

$$Q + W = 0 \rightarrow W = 0$$

and (B-4) becomes

$$\dot{m} \int_2^3 (dh - vdp) = 0 \qquad \text{- - - - - - - - - - - - -(B-5)}$$

But $vdp = dw$

∴ Integrating (B-5) gives

$$h_3 - h_2 = W \qquad \text{- - - - - - - - (B-6)}$$

That is to say, the difference in enthalpies of vapour entering and leaving compressor is a measure of the work required to compress it.

B.3 THE CONDENSER

This is also a heat transfer component. It removes both the heat absorbed in evaporator, and the heat equivalent of the energy input to achieve compression. The refrigerant is convected back into liquid phase. The condenser has similar requirements to those of the evaporator [10]

(1) heat transfer surface

(2) sufficient volume for vapour delivered and

(3) good separation of condensed liquid from uncondensed vapour.

Three forms of condensers classified on the basis of the cooling medium

(a) air-cooled, employing air as a condensing medium

(b) water-cooled, utilizing water as condensing medium

(c) evaporative, using both water and air; the cooling of refrigerant initially results from the evaporation of the water; the function of the air is to increase rate of evaporation by carrying away.

Condenser Analysis:

Heat is transmitted from the refrigerant to the cooling medium by conduction. Heat flow rate is given by

$$Q = KA(t_s) \qquad \text{---------(8-7)}$$

$$\text{or} \quad A = \frac{Q}{K(t_s)}$$

Knowing the total surface area for a particular system, the condenser can be sized.

8.4 REFRIGERANT CONTROL DEVICE

The control of refrigerant flow is an essential feature for any refrigerating system. The functions of a control device are two-fold [21]

(1) it regulates the flow of liwuid refrigerant according to the cooling load.

(2) it maintains presaure differential between high and low pressure sides of the system.

at the control device, four basic things take place:

(i) pressure drops instantly from high to low.

(ii) temperature drops from condensing to evaporating valves.

(iii) there is no change in enthalpy

(iv) part of liquid refrigerant instantly changes to vapour (flash gas) to effect the change in temperature.

Types of Controls

Six basic types are available:

(1) Lowside float (2) High side float

(3) Hand Expansion valve (4) Capillary tube

(5) Automatic Expansion Valve and (6) Thermostatic Expansion Valve.

8.5 REFRIGERANTS

Generally speaking, a refrigerant is any body which acts as cooling agent by absorbing heat from another body [10]. In the vapour compression cycle, the refrigerant is the working fluid which alternately vapourizes and condenses as it absorbs and gives off heat respectively.

A refrigerant must possess the following qualities:-

(1) chemically inert

(2) non-explosive

(3) non-flamable

(4) non-toxic and non-irritating

(5) possess critical temperature well above the condensing temperature.

B.6 COLD STORAGE ENCLOSURE

This is any enclosure where temperature is maintained lower than the surroundings [21]. Such an enclosure must

(1) be well insulated

(2) provide a vapour barrier

(3) have doors to bring products in and out

(4) be protected against excess heat

A cold storage can be classified under WALK-IN or REACH-IN. [10]

Walk-in Fixtures

These are primarily storage fixtures found in retail stores, markets, hotels, institutions, etc. where products are removed from or placed in the space by walking into the space: e.g the cold rooms for vaccine storage at the National Veterinary Research Institute, Vom. Here, vaccine bottles can be loaded or removed by walking into the cold room.

Reach-In Fixtures

These are widely used, they are typically found in grocery stores, meat markets, bakeries, drug stores, etc where desired products are merely picked or placed by extending the arm, eg removal of meat from a deep freezer.

APPENDIX C

H U M I D I T Y

This is a term used to describe the presence of moisture (low pressure steam) in the air. The amount of moisture that air can hold depends on the temperature of the air [2].

Specific Humidity (ω)

Ratio of mass of water vapour (m_v) to the mass of dry air, (m_a) in a given volume is the absolute humidity

$$\omega = m_v/m_a \qquad \text{- - - - - - (C-1)}$$

$$\omega = \frac{m_v/V}{m_a/V} = v_a/v_v \qquad \text{- - - - -}$$

Treating the water vapour as a perfect gas

$$p_v V = M_v R_v T$$

Similarly for dry air,

$$p_a V = M_a R_a T$$

$$\therefore \quad p_v/p_a = \frac{m_v R_v}{M_a R_a} = \omega \frac{R_v}{R_a}$$

$$\text{or} \quad \omega = \frac{p_v R_a}{p_a R_v}$$

$$= \frac{p_v R_a}{R_v (p - p_v)} \qquad (p_a = p - p_v).$$

$R_v = 0.4619 kJ/kg\ ^oK$. $R_a = 0.2871 kJ/kg\ ^oK$

$$\therefore \quad \omega = 0.622 \frac{p_v}{p - p_v} \qquad \text{- - - - - (C-2)}$$

Relative Humidity (ϕ)

This is the ratio of actual partial pressure of vapour (p_v) to the partial pressure of the vapour when the air is saturated at the same temperature (p_g). It is the amount of moisture in a given sample of air compared to the amount of moisture the air would hold if totally saturated at the temperature of the sample.

$$\phi = p_v/p_g = m_v/(m_v)_{sat}. \qquad \text{----------(C-3)}$$

The relative humidity is a measure of how air will affect other things.

Dew Point (t_{dp})

The temperature to which unsaturated air must be cooled at constant pressure to become saturated.

Sling Psychrometer

This is used to measure relative humidity of the air and is made-up of

(1) a dry-bulb thermometer - measures the actual temperature of the air (t_{db})

(2) a wet-bulb thermometer - this thermometer has its mercury bulb placed in a moist wick and evaporation from the wick lowers the thermometer reading (t_{wb}). The difference between t_{db} and t_{wb} is a measure of relative humidity.

Humidity in Cold Storages

In a cold storage, it is aimed at maintaining some amount of moisture (humidity) in the storage space. Evaporation of moisture from stored products (for instance food stuffs) takes place if the space contains very dry air; the rate of evaporation depends on the air relative humidity in the space.

This moisture content, however, is affected by the evaporator coil temperature. The desired temperature difference between the evaporator and storage space air humidity is therefore something of a compromise. A large temperature difference requires a lower evaporator. But this requires a lower back pressure resulting in very dry air and thus low humidity. A smaller temperature difference on the other hand requires a larger evaporator coil and less moisture is taken out of the air as it passes over the coil producing a higher space humidity [21].

APPENDIX D

DEFROSTING METHODS

Defrosting is the means of removing frost from the evaporator surface. Frost acts as insulation and reduces the evaporator capacity. Defrosting of the evaporator is accomplished in a number of different ways and because frosting is an important problem in the Refrigeration Industry, it is worth looking at these different ways. Defrosting methods, can be classified as 'natural defrosting' or 'supplementary-heat defrosting' according to the source of heat used to melt-off the frost [10].

D.1 NATURAL OR 'OFF-CYCLE' DEFROSTING

These methods utilize the heat of the air in the cold storage chamber, to melt the frost after the system has been shut down. They are not, however, applicable when the storage chamber temperature is equal or less than the freezing temperature of the water.

(a) Manual Defrost Method

This involves manual stopping of compressor or refrigerant flow to the evaporator to be defrosted.

brine. Concentrators are also available to evaporate some of the water to maintain the brine concentration.

(c) Hot Gas Method

This entails the use of a by-pass to get hot refrigerant from the compressor to flow through the evaporator. Modifications are however made to present large slug of liquid refrigerant getting into the compressor.

(d) Hot Gas Thermobank Method

In this method, a water 'bank' stores some portion of the heat discarded in the condenser during the normal operation. This is used in vaporizing condensed refrigerant in the suction line during frost back.

(e) Vapomatic Defrost Method

A specially designed accumulator in the suction line traps some liquid from evaporator during defrosting. This liquid is then metered to the compressor along with the suction vapour.

(f) Electric Defrost

Electric resistance heaters are attached to or built into the evaporator. The electric defrost cycle can be started and stopped mannually or a defrost timer may be used to make defrosting completely automatic.

APPENDIX E

T A B L E S

(Repraduced from Ref. 21)

TABLE E.1 - COOLER USAGE FACTORS (For Coolers below 45.3m^3)

USAGE	USAGE FACTOR, F($W/m^3.^oC$)	
	REACH-IN CABINETS	WALK-IN CABINETS
LIGHT	33,5	11.2
MEDIUM (NORMAL)	41.0	14.9
HEAVY	56.0	22.4

TABLE E.2 - SIZES AND WEIGHTS OF COPPER TUBING

OUTSIDE DIAMETER, D_o(mm)	WALL THICKNESS, x(mm)	INSIDE DIAMETER, D_i(mm)	WEIGHT, W($kg/_m$)	LENGTH PER M^2 EXTERNAL SURFACE (m)
7.15	0.76	3.24	0.078	67.54
6.35	0.76	4.83	0.12	50.16
8.16	0.81	6.31	0.16	40.00
9.53	0.81	7.90	0.20	33.48
12.70	0.81	10.95	0.27	25.05
16.33	0.89	14.10	0.37	20.03
19.59	0.89	17.27	0.45	16.72

TABLE E.3 - HEAT TRANSFER IN REFRIGERATION EQUIPMENT
CONDUCTIVITY, K($^{KW}/_{m^2.{}^oC}$)

	COOLING AIR				
		AIR VELOCITY (km/h)			
	STILL AIR	1.01	1.52	2.03	2.54
DIRECT EXPANSION COIL	0.142	2.840	3.29	3.749	4.370
BRINE COIL	0.1704	4.023	5.68	6.248	6.816
PLATE OR HOLD-OVER TANK	0.128				

NOTES TO TABLES E.1 AND E.3

The Usage Factor, F (TABLE E.1)

In most refrigerating applications, simplification is made on cooling load calculations by dividing the total cooling load into

(1) the wall gain load

(2) the air change load

(3) the product load and

(4) the supplementary load.

Further simplification is made for small refrigerating equipment and 45.30m^3 capacity, used for general-purpose storage.

Here the product load is frequently unknown or varies fom day to day so that it is not possible to compute the product load with any real accurary [10]. In such cases, use is made of the USAGE FACTORS, F (or load factors) which have been determined by experience. Using this method of calculation, the entire cooling load is divided into two parts:

(a) the wall gain or leakage load, given by

$$Q_w = UA \Delta T \qquad \text{--------(E-1)}$$

and (b) the usage or service load given by

$$Q_u = FV \Delta T \qquad \text{----------(E-2)}$$

In determining F, allowance is made for normal usage (where the quantity of product handled is normal) and heavy usage (where room temperatures are likely to be high and large quantities of warm products are involved). The factor also varies with the type of fixture, either walk-in or reach-in as can be seen in Table E.1.

The K Factor (Table E.3)

The heat leakages through some surfaces is better given for the average thickness such surfaces are made. Thus the factor K replaces $^{k}/_{x}$ in the heat leakage formula

$$Q = \frac{Ak(t_o - t_o')}{x} \qquad \text{-------(E-1)}$$

$$= A(\frac{k}{x})(t_o - t_o')$$

$$= AK(t_o - t_o') \qquad \text{------(E-2)}$$

or $K \equiv \frac{k}{x}$, in units of $(W/m^2.{}^oC)$

Thickness x is therefore eliminated.

GLOSSARY OF SOME TECHNICAL TERMS

SPECIFIC HUMIDITY: amount of moisture in the air
$\{kg_{water\ vapour}\ ^{kg}_{dryair}\}$

ABSORBENT: Substance with the ability to absorb or take up another substance.

ABSORPTION REFRIGERATION: Refrigeration by using the cooling effect formed when a refrigerant is absorbed by chemical substance

ADIABATIC COMPRESSION: Compressing refrigerant gas without removing or adding heat.

AIR CONDITIONING: Control of the temperature, humidity, air movement and cleaning of air in a confined space.

AIR.COOLED CONDENSER: Heat of compression is transferred from condensing coils to surrounding air.

AIR COOLER: Mechanism designed to lower temperature of air passing through it.

AIR INFILTRATION: Leakage of air into rooms through cracks, windows, doors and other openings.

AMBIENT TEMPERATURE: Temperature of fluid (usually air) which surrounds object on all sides.

AUTOMATIC DEFROST: System of removing ice and frost from evaporators automatically

BACIL PRESSURE: Pressure in low side of refrigerating system, also called suction or low-side pressure

BRAZING: Method of joining metals with non-ferrous filler

BRINE: Water saturated with salt

CASCADE SYSTEMS: arrangement in which two or more regrigerating systems are used in series produces ultra-low temperatures.

CENTRIFUGAL COMPRESSOR: Pump which compressor gaseous refrigerants by centrifugee force

CHARGE: amount of refrigerant placed in a refrigerating unit.

COEFFICIENT OF CONDUCTIVITY: measure of the relative rate at which different materials conduct heat

COEFFICIENT OF PERFORMANCE (COP): Ratio of work performed as compared to the energy used.

COLD JUNCTION: That part of a thermoelectric system which absorbs heat as the system operates.

COMPRESSION: Denotes increase of pressure on a fluid by using mechanical energy

COMPRESSOR: Pump of a refrigerating mechanism which draws a low pressure on cooling side of refrigerant cycle and squeezes or compresses the gas into the high-pressure or condensing side of the cycle.

CONDENSATION: Liquid or droplets which form when a gas or vapour is cooled below its dew point

CONDENSER: The part of refrigeration mechanism which receives hot, high-pressure refrigerant gas from compressor and cools gaseons refrigerant until it returns to its returns to its liquid state

CYCLE: Series of events or operations which have tendency to repeat in the same order.

DEFROST CYCLE: Refrigerating cycle in which evaporator frost and ice accumulation is melted-off.

DEFROSTING: Process of removing frost accumulation from evaporators

DEHUMIDIFIER: Device used to remove moisture from air.

DEW POINT: Temperature at which vapour begins to condense.

DRIER: Device used to remove moisture from a refrigeration system

EVAPORATION: Changing of a liquid to a gas. Heat is absorbed.

EVAPORATOR: Part a refrigerating mechanism in which the refrigerant vapourizes and absorbs heat.

EXPANSION VALVE: Device in refrigerating system which reduces the pressure from the high side to the low side and is operated by pressure

FROST BACK: Condition in which liquid refrigerant flows from evaporator into suction line indicated by sweating or frosting of the suction line.

HEAT EXCHANGER: Device used to transfer heat from a warm or hot surface to a cold or cooler surface.

HEAT LOAD: Amount of heat removed (in Watts).

HEAT SINK: Relatively cold surface capable of absorbing heat.

HERMETIC COMPRESSOR: Compressor which has the driving motor sealed inside the housing or done

HOT JUNCTION: That part of thermoelectric circuit which releases heat.

HUMIDIFIERS: Device used to add to and control humidity.

HUMIDITY: Moisture, Dampress of air.

NONCONDENSABLE GAS: Gas which does not change into a liquid at operating temperatured and pressures.

OFF CYCLE: That part of a refrigeration cycle when the system is not operating.

PRESSURE DROP: Pressure difference at two ends of a circuit or part of a circuit.

PSYCHROMETER: Instrument for measuring the relative humidity of atmospheric air.

REFRIGERANT: Substance used in refrigerating mechanism which absorba heat in evaporator and releases heat in condenser

REFRIGERANT CONTROL: Device which meters flow of refrigerant between two areas of a refrigerating system

RELATIVE HUMIDITY: ratio of amount of water vapour present in air to greatest amount possible at same temperatura.

SATURATION: Condition existing when substance contains all of another substance it can hold for that temperature and pressure

SENSIBLE HEAT: Heat which causes a change in temperature of a substance.

SUBLIMATION: Condition where a substance changes from a solid to a gas without becoming a liquid.

SWAETING: Condensation of moisture from air on cold surface

THERMOCOUPLE: Device which generates electricity, using principle that if two unlike metals are welded together and junction is heated, a voltage will develop across the open ends.

THERMOSTATIC EXPANSION VALVE: Control valve operated by temperature and pressure with in evaporator controlling flow of refrigerant.

TUBING: Fluid-carrying pipe which has a thin wall.

VAPOUR BARRIER: Thin plastic or metal foil sheet used to prevent water vapour from penetrating insulating material.

CRITICAL RELATIVE AIR HUMIDITY: Relative Humidity at maximum heat exchange.

CRITICAL TIME: Time taken to attain maximum heat exchange

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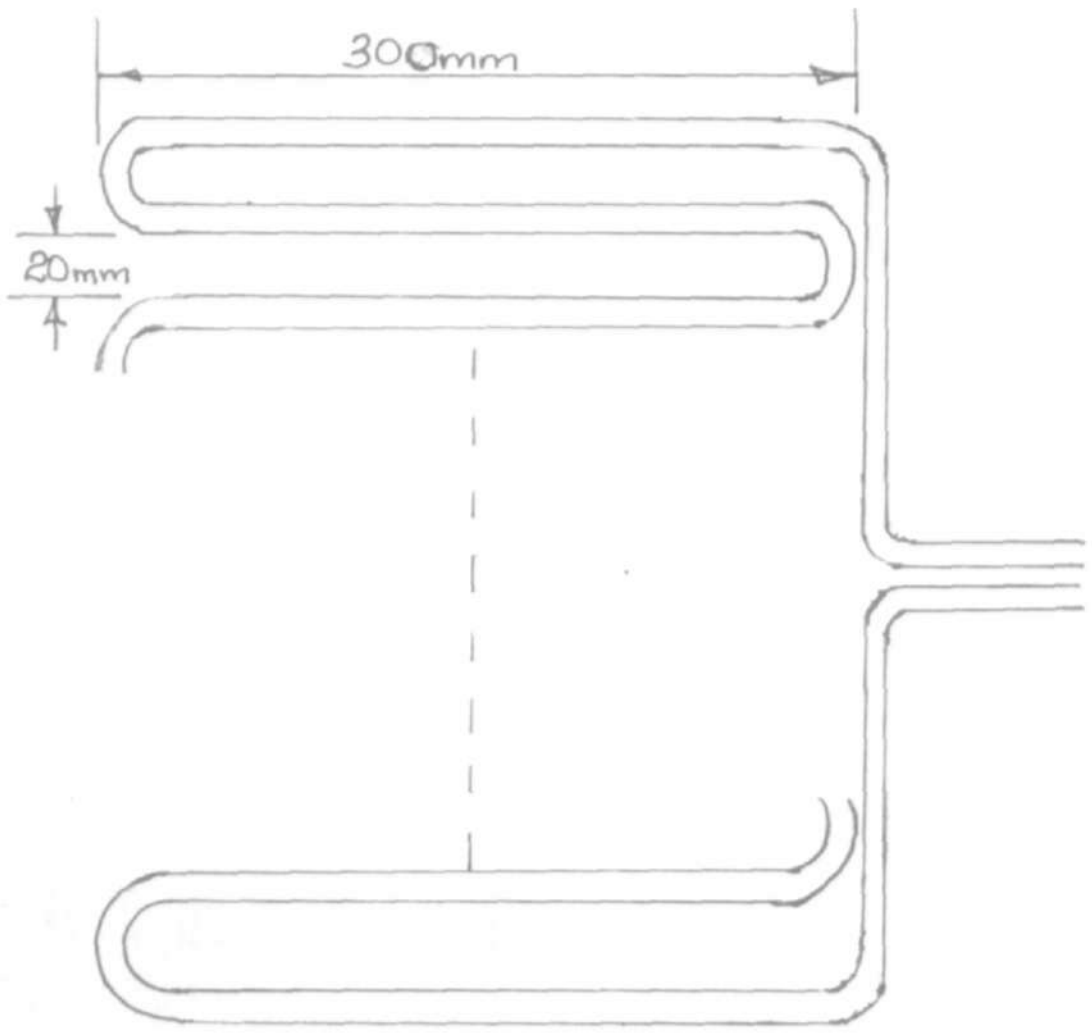
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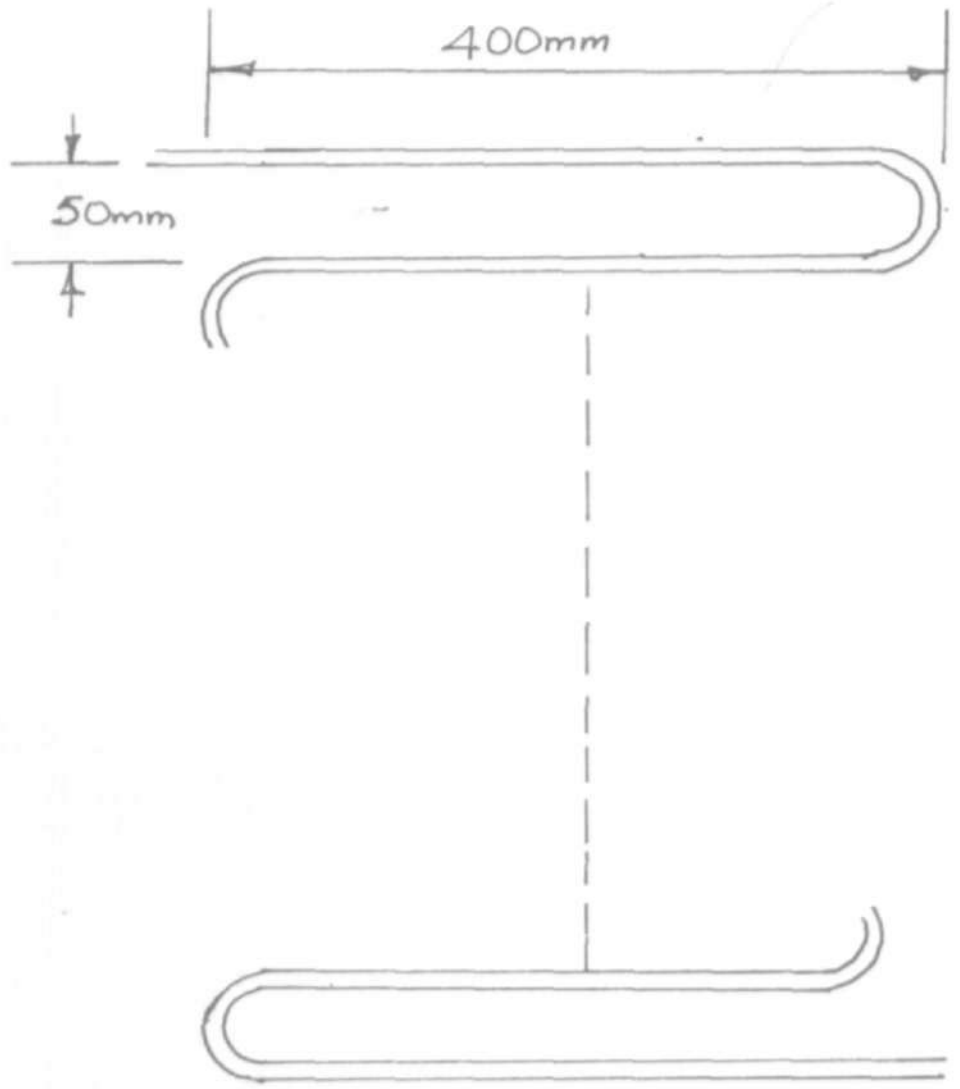
Scale:- 1:3



Total Length of tubing
L = 5.43m
No of Coils N = 16
O.D. of tubing d = 9.53mm

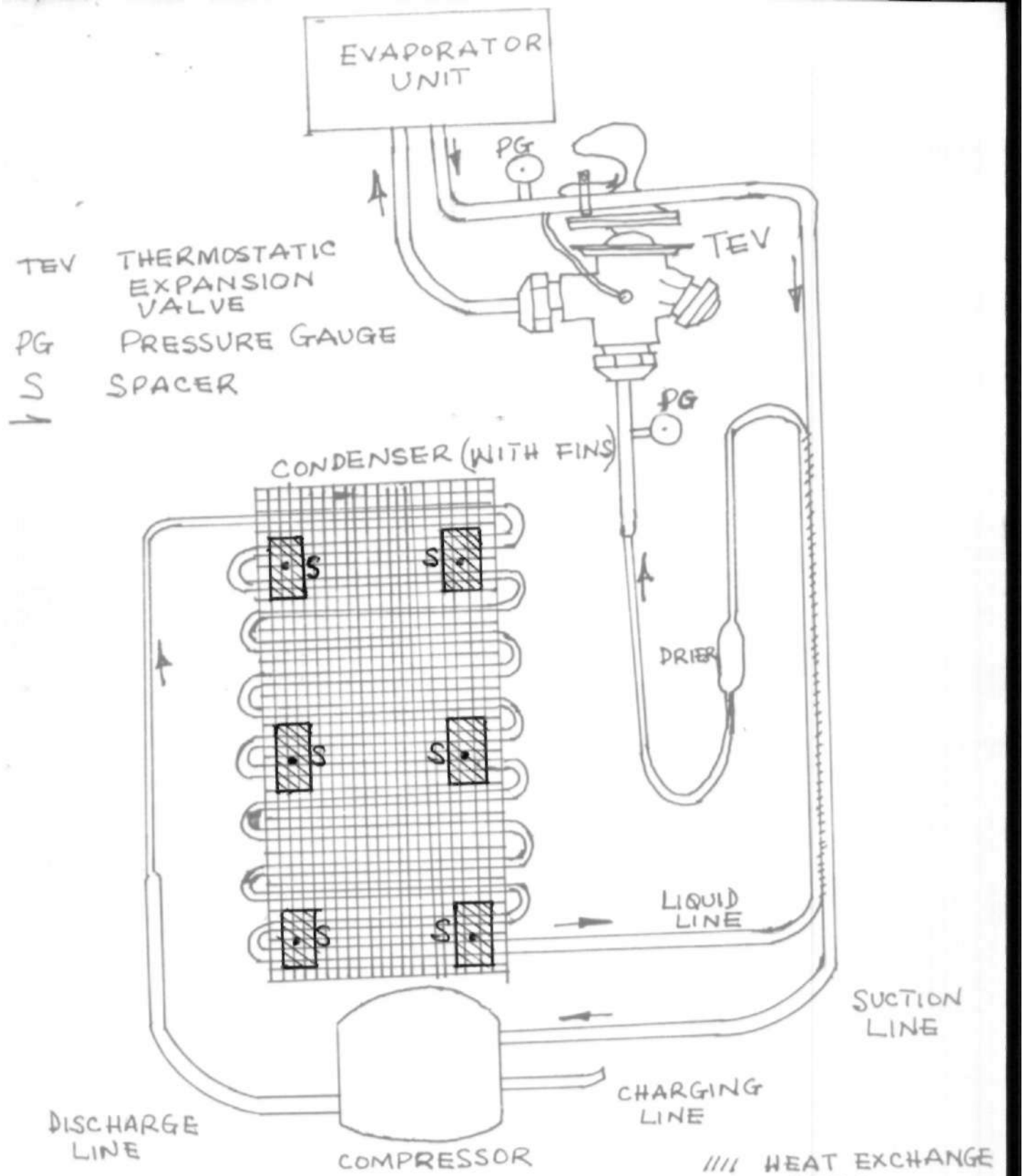
FIGURE 4.3 — EVAPORATOR SIZE & SHAPE

Scale:- 1:4



Total length of tubing L = 7.46m
No of Coils N = 17
OD of tubing d = 6.35mm

FIGURE 4.4. CONDENSER SIZE & SHAPE WITHOUT FINS



URE 45 — SKETCH OF ASSEMBLY UNIT SHOWING REFRIGERANT CIRCUIT AND COMPONENT ARRANGEMENT.

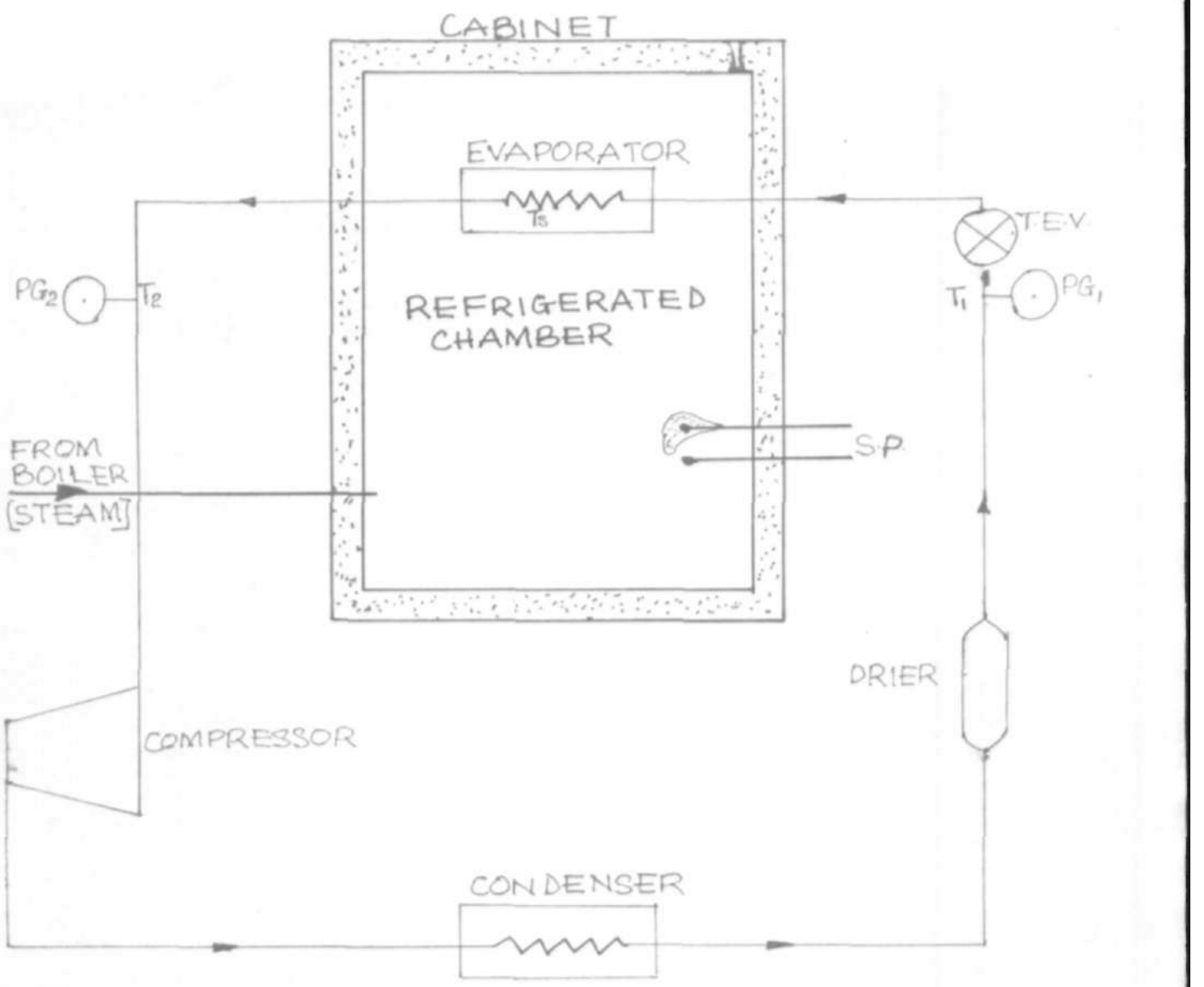


FIGURE 5·1 SCHEMATIC [FLOW] SHEET SHOWING GENERAL ARRANGEMENT OF THE EQUIPMENT.

KEY

T.E.V = THERMOSTATIC EXPANSION VALVE
T_1, T_2, T_5 = THERMOCOUPLES
S.P. = SLING PSYCHROMETER
P.G. = PRESSURE GAUGE

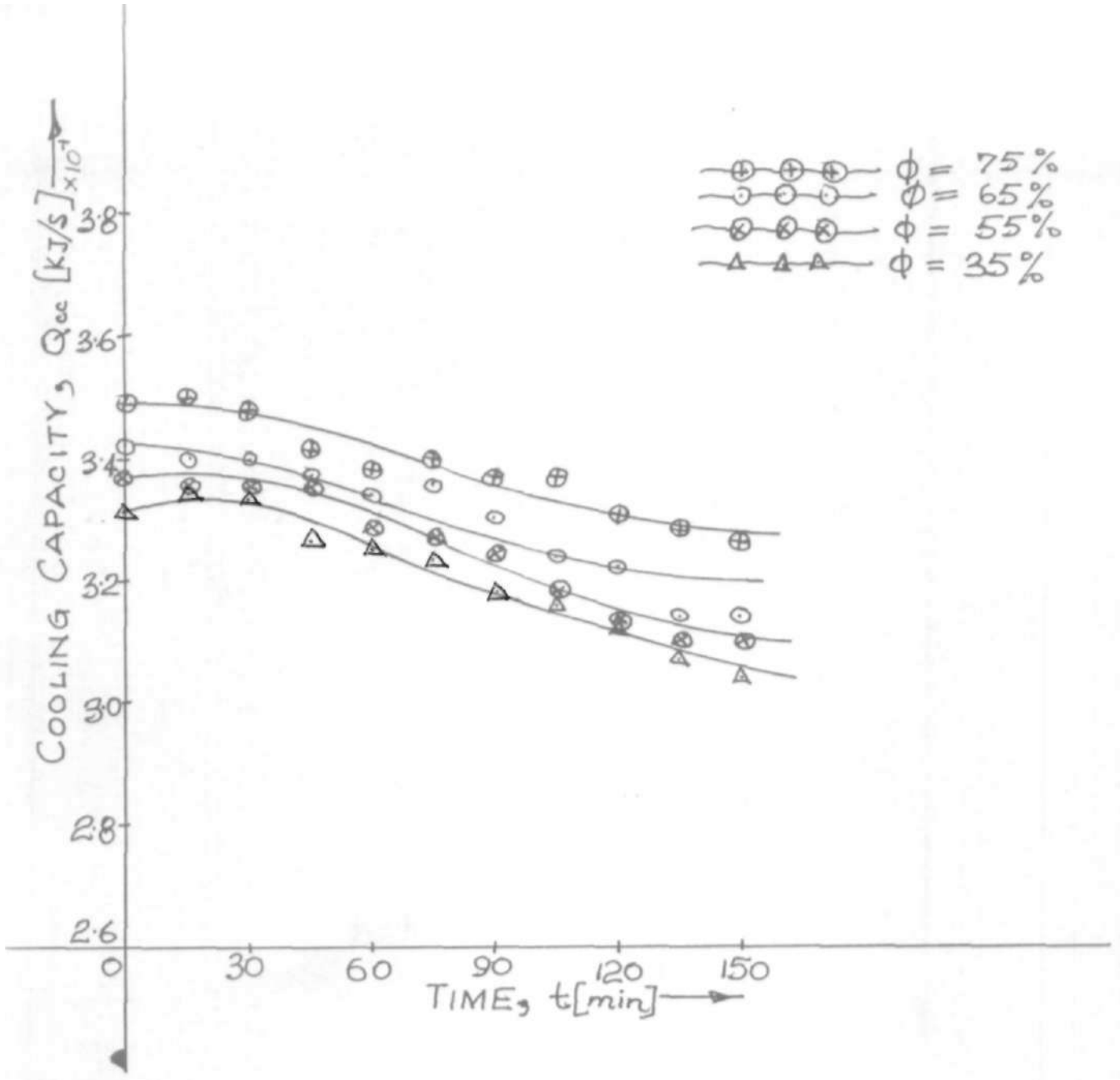


FIGURE 5·2—VARIATION OF COOLING CAPACITY WITH TIME—INFLUENCE OF INITIAL RELATIVE AIR HUMIDITY

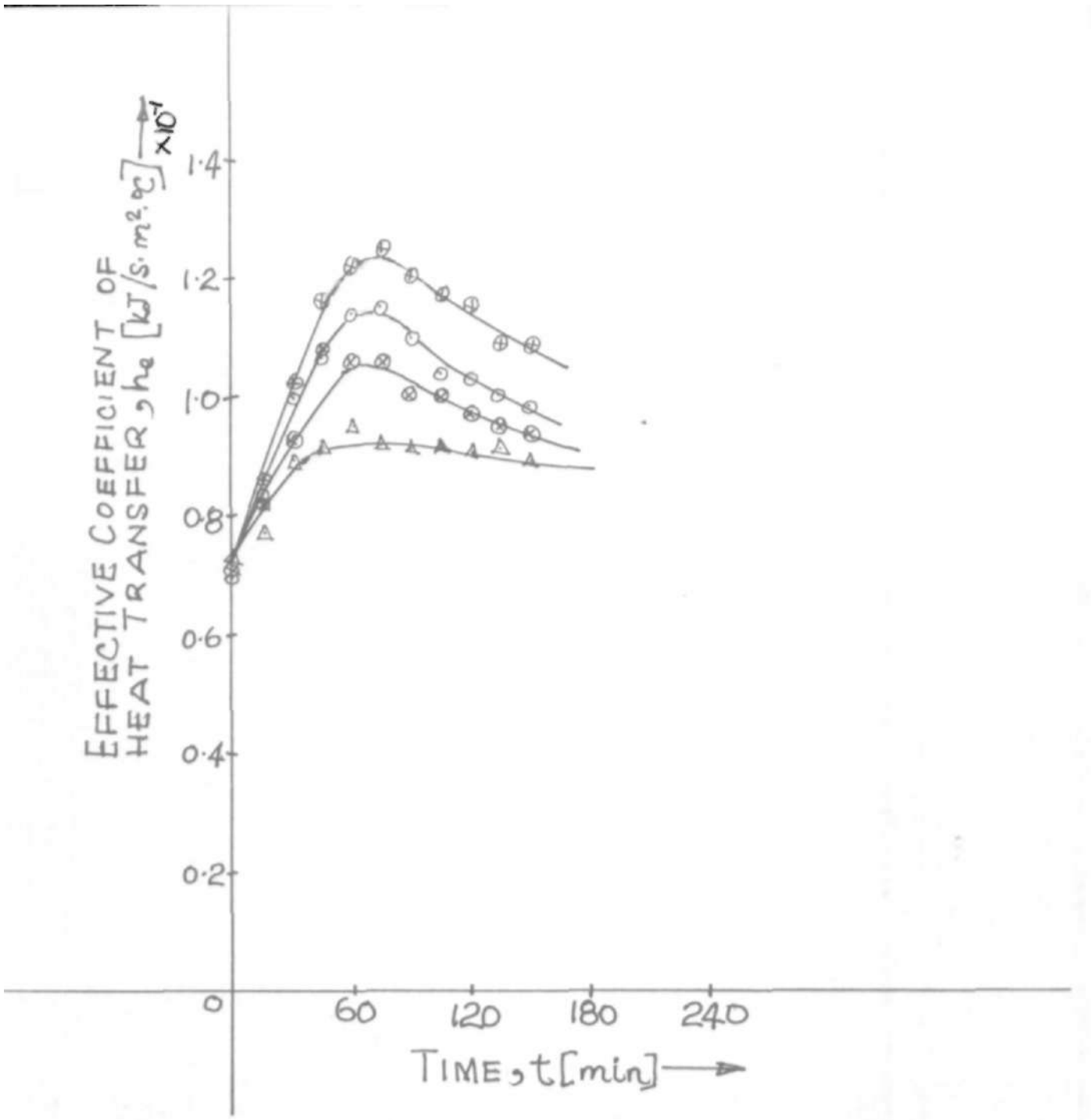


FIGURE 5·3 — VARIATION OF EFFECTIVE COEFFICIENT OF HEAT TRANSFER — INFLUENCE OF INITIAL RELATIVE AIR HUMIDITY

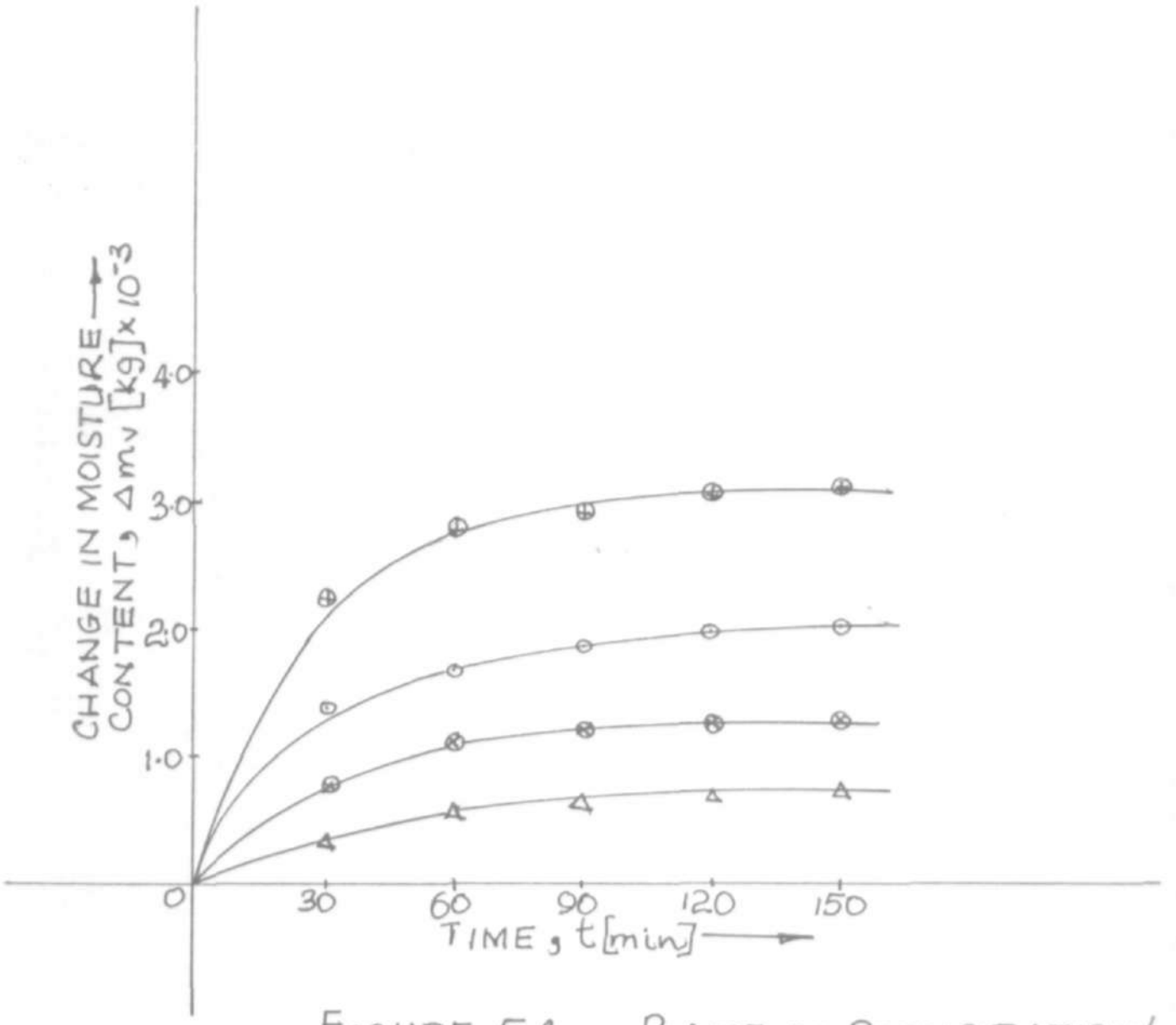


FIGURE 5.4—RATE OF PRECIPITATION/CONDENSATION—INFLUENCE OF INITIAL RELATIVE AIR HUMIDITY

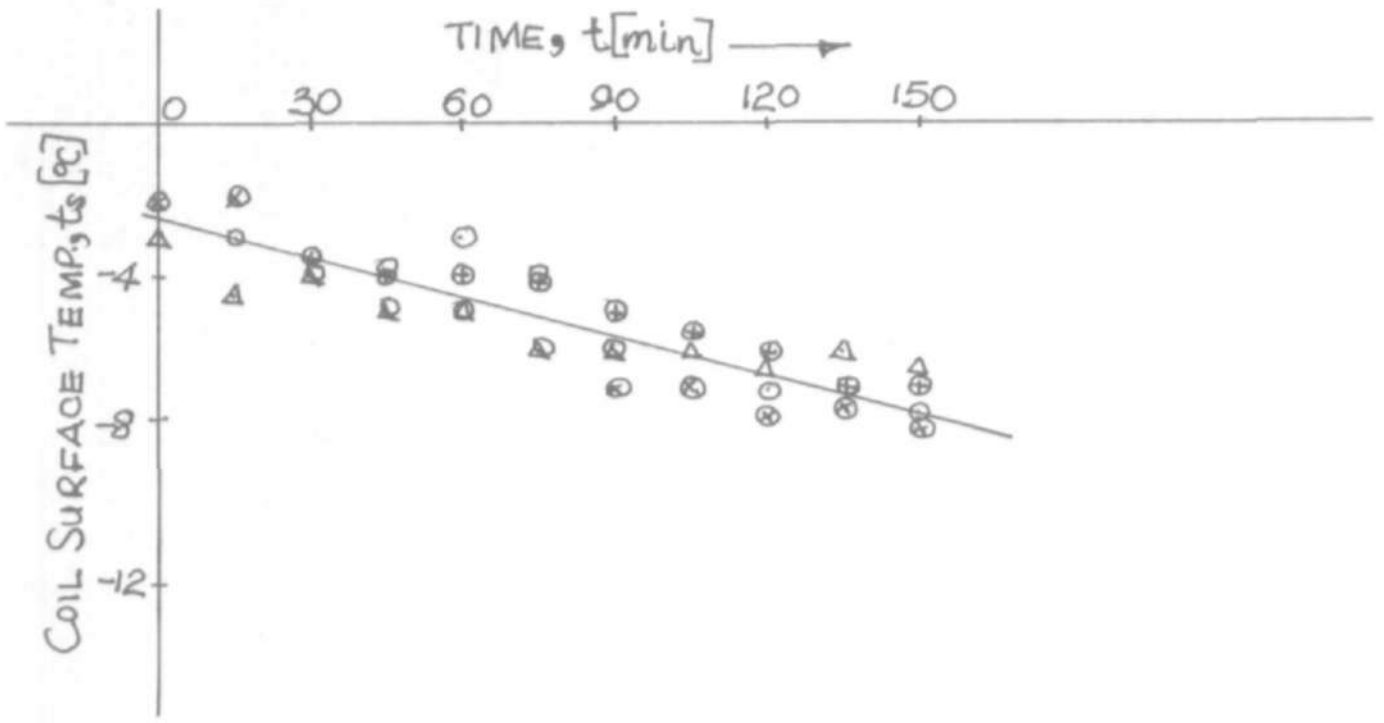


FIGURE 5.5— VARIATION OF COIL SURFACE TEMPERATURE WITH TIME.

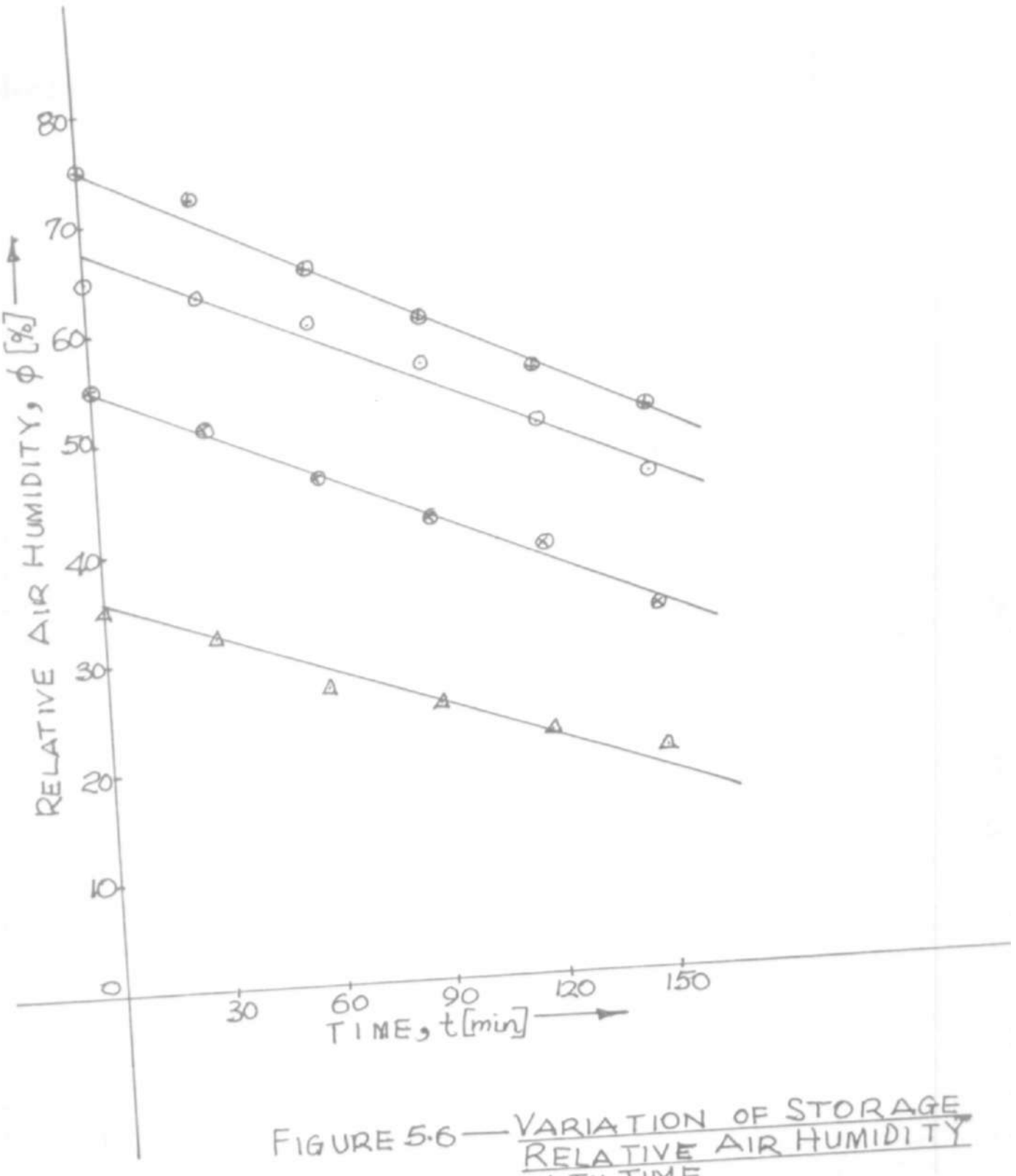


FIGURE 5.6 — VARIATION OF STORAGE RELATIVE AIR HUMIDITY WITH TIME

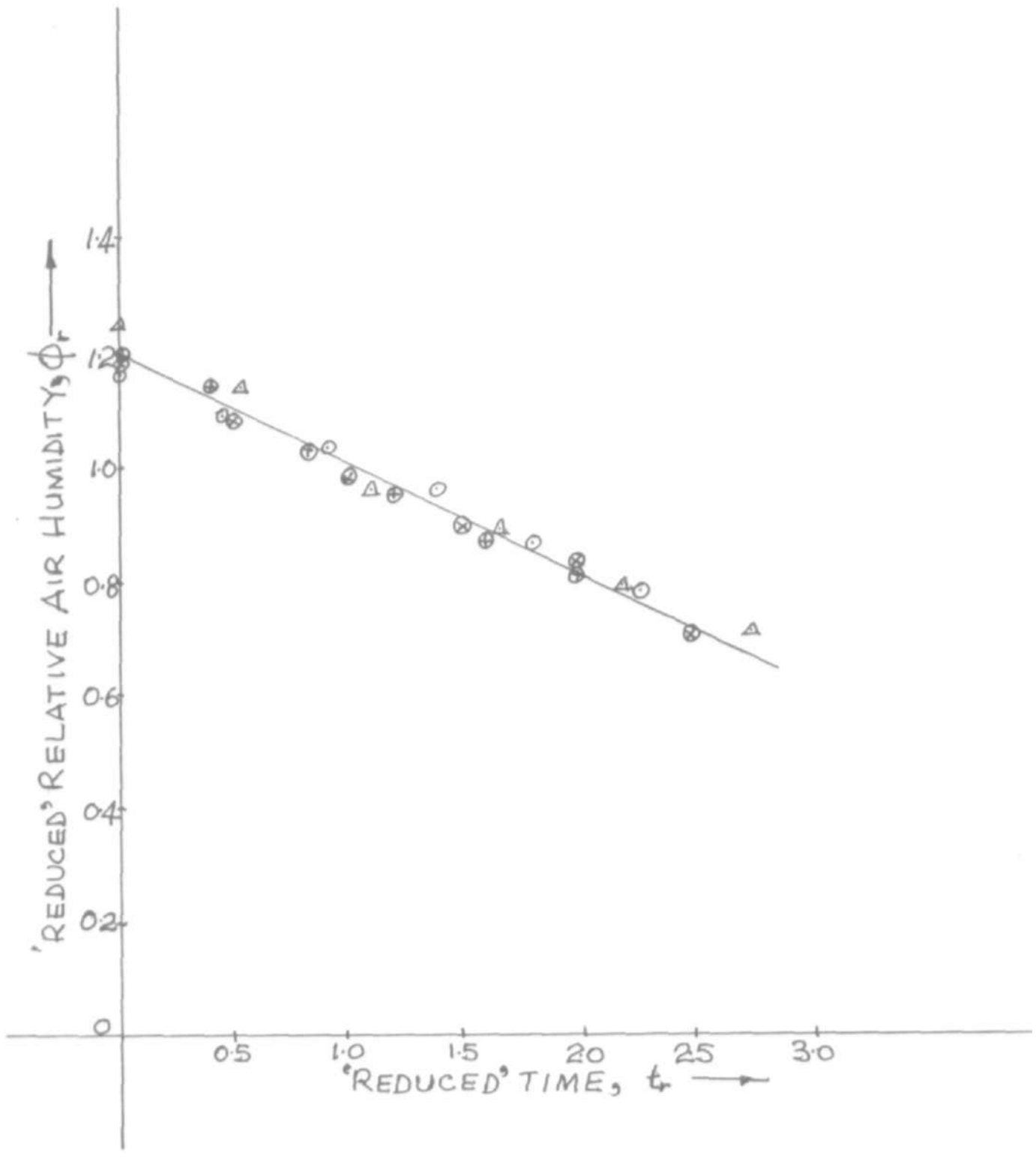


FIGURE 5·7 — GRAPH OF 'REDUCED' RELATIVE AIR HUMIDITY Vs 'REDUCED' TIME.

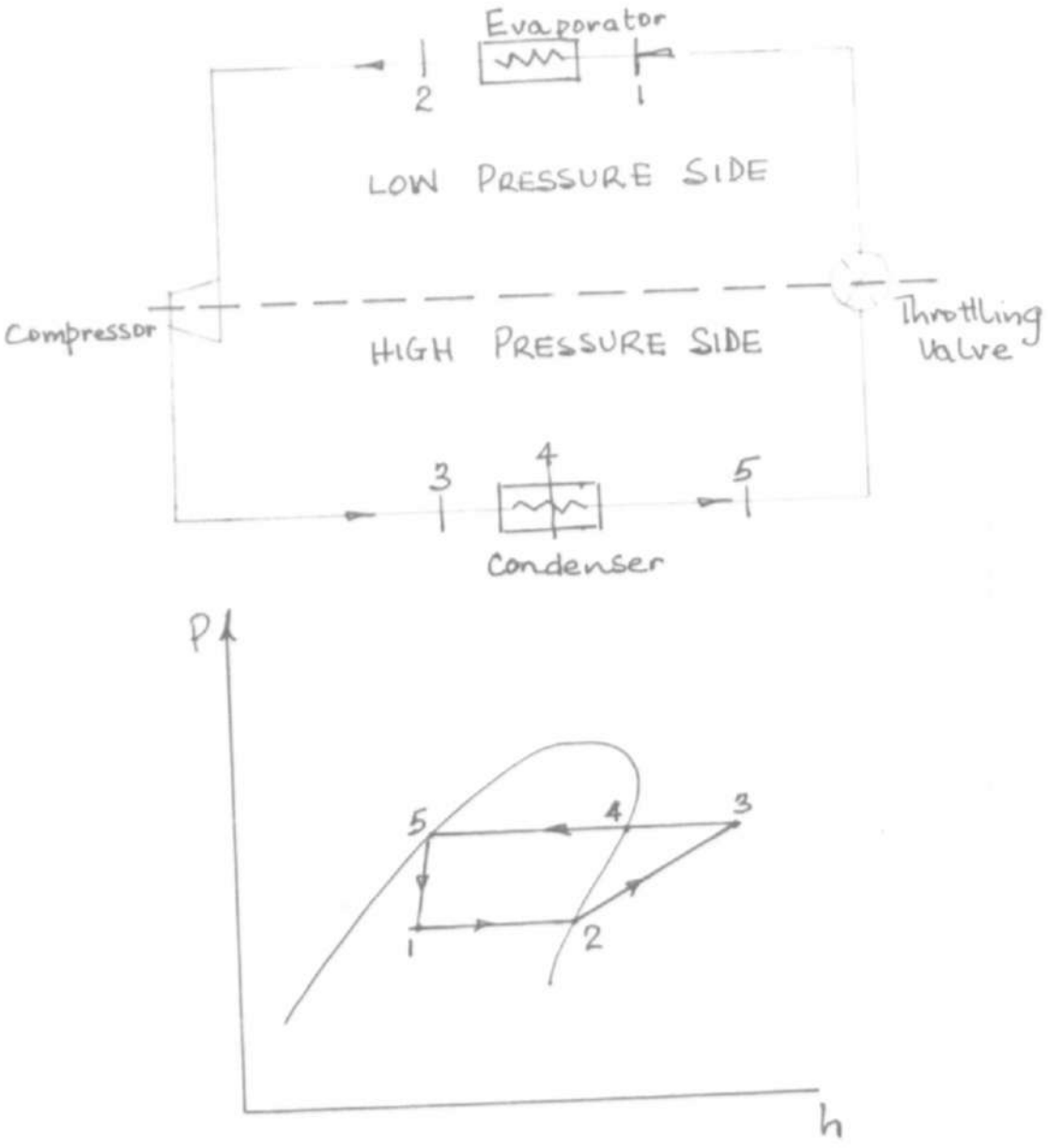


FIGURE A·1—VAPOUR COMPRESSION CYCLE

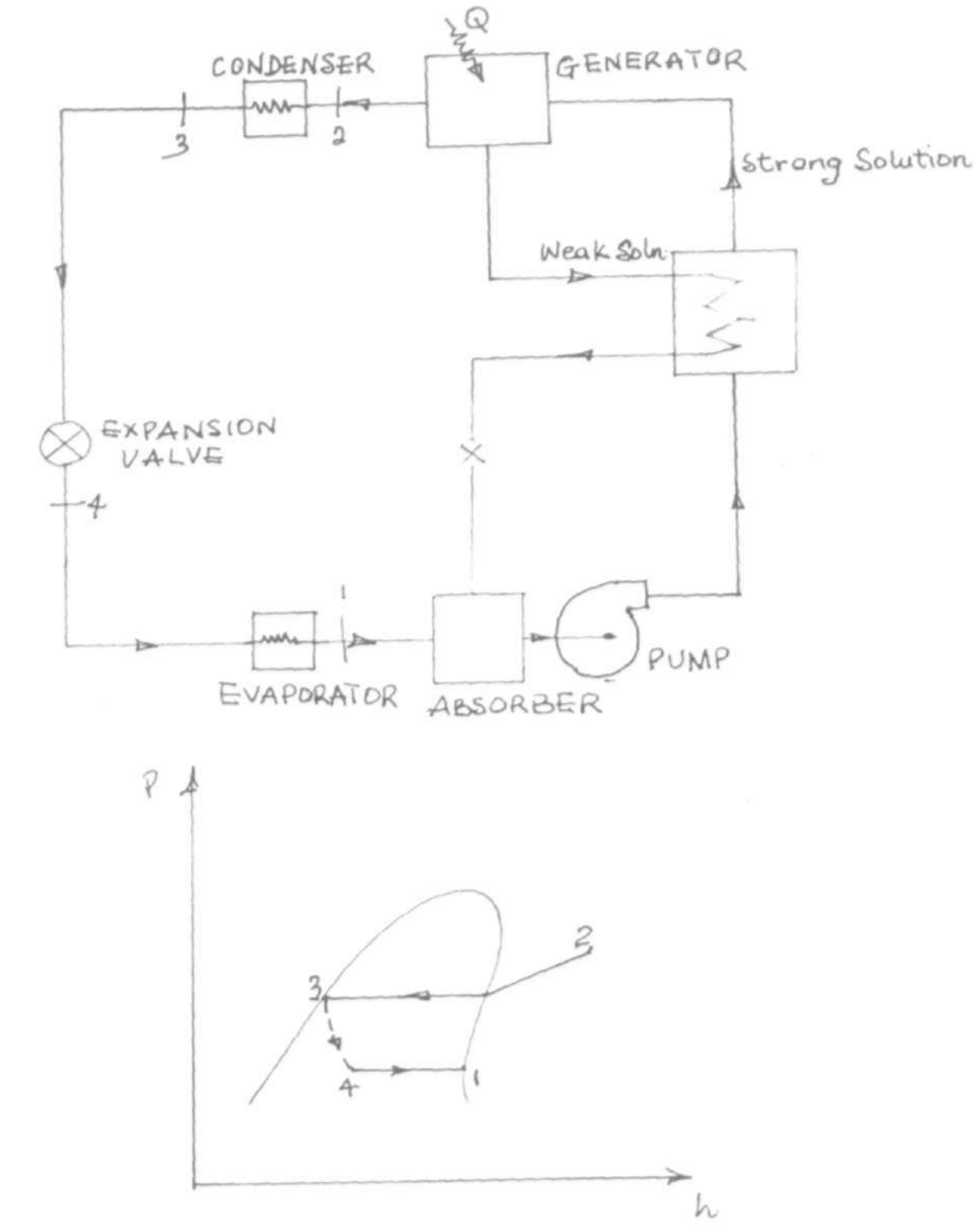


FIGURE A·2—SIMPLE ABSORPTION SYSTEM

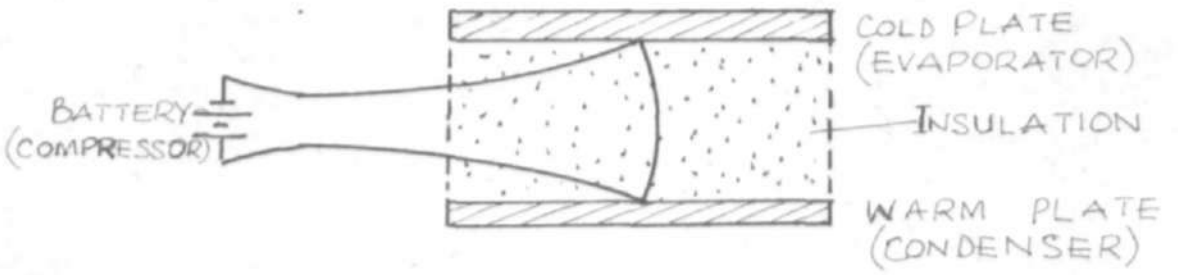


FIGURE A·3—SCHEMATIC THERMOELECTRIC COOLING SYSTEM

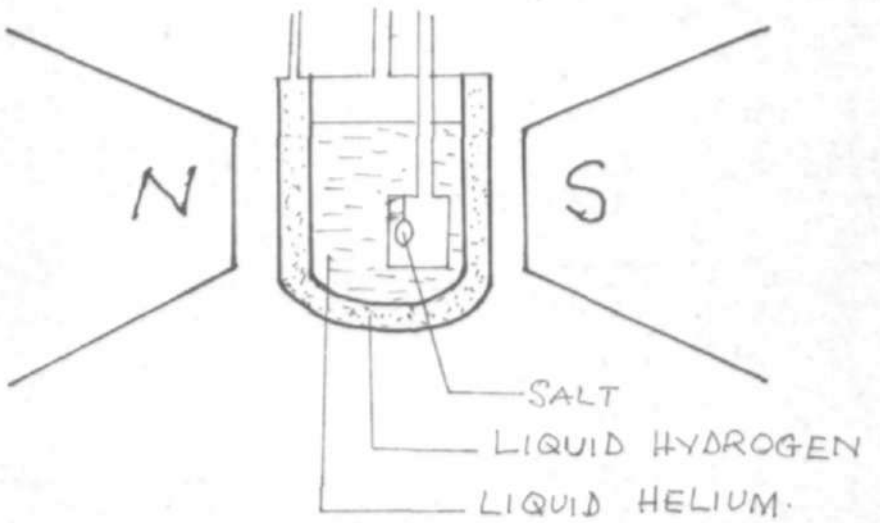


FIGURE A·4—SCHEMATIC ADIABATIC DEMAGNETIZATION APPARATUS

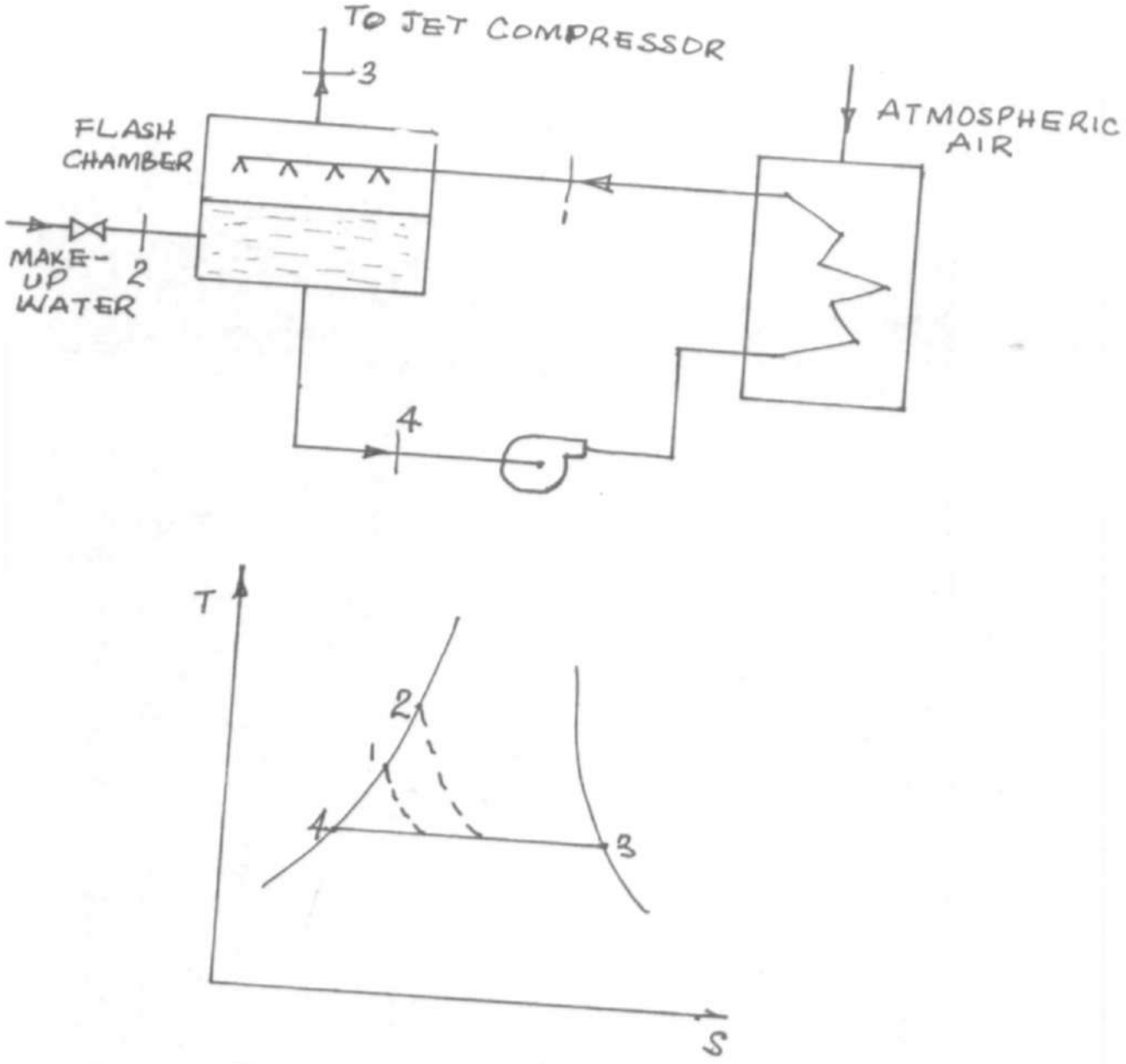


FIGURE A.5—SCHEMATIC FLASH COOLING OR STEAM-JET-EJECTION SYSTEM.