DEFROSTING IN REFRIGERATION SYSTEM

BY

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C E R T I F I C A T E

Certified that the thesis entitled " DEFROSTING IN REFRIGERATION SYSTEMS " , which is being submitted by Shri HARI KRISHNA VARMA in partial fulfilment for the award of degree of MASTER OF ENGINEERING in Applied Thermodynamics - Refrigeration and Air-Conditioning, of University of Roorkee, is a record of student's own work carried out by him under my supervision and guidance. The results embodied in this thesis have 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 about one year and seven months from 1st April, 1963 to 5th November, 1964 for preparing thesis for Master of Engineering Degree at this University.

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A C K N O W L E D G E M E N T S

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A B S T R A C T

The present work was taken up to expedite the performance of a refrigeration system under different defrosting conditions. A finned evaporator coil was selected as the object of study and its behaviour was investigated. A theoretical background to the problem is discussed and the general behaviour verified. A review of existing literature has also been made.

CONTENTS

CHAPTER 1

FROST, ITS EFFECT & INFLUENCING FACTORS

Frost, its effect on refrigeration plant operation, and factors influencing its formation are reviewed.

1.1 INTRODUCTION:

In refrigeration systems many parts of the equipment exist at temperatures below freezing point of water ($32^{o}F$). The evaporator coil, expansion valve, suction line and some other parts of the system are exposed to sub-freezing temperatures.

It is a well known fact that the atmospheric air contains certain amount of humidity. When the humid air comes in contact with a solid surface, nothing will happen to the composition of air so long the solid surface is at a temperature above the dew point of air although heat transfer will take place if a temperature difference exists. If the solid surface has a temperature below the dew point of air then a portion of humidity content in the humid air will condense on the solid surface. In such a case a compound phenomenon of heat and mass transfer occurs. Yet interesting is the case when the solid surface is at subfreezing temperatures. In such case, the moisture in air will deposit on the surface in the form of minute crystals of ice. This deposit of ice is called FROST. This phenomenon has its peculiarity in that the compound mechanism of heat and mass transfer is much more complex because under such conditions air is cooled and dehumidified by a surface, of which the nature is continuously changing because of the build-up of the frost, and the nature of the build-up of the frost, too, is changing.

1.2 TYPES OF FROST:

Frost produced on evaporator coils during refrigeration will vary and will be found of very different density and structure depending upon the conditions of its formation and age. When the surface temperature is slightly below $32^{o}F$, frost formed is hard and clear and does not contain any air bubbles. At lower temperatures, below 15°F nearly, the frost formed has a porous and weak structure. In general colder the surface, the finer are the ice crystals. It has a white appearance. Looking through a magnifying glass it can be observed that this frost forms an intricate network with a multitude of minute air spaces between them. The later type of frost is called RIME FROST while the hard tenacious frost is called CRYSTALLINE FROST. The two can be easily distinguished as the rime frost has white appearance and fluffy structure while the other type has a flint like structure and is hard.

Because of the porous nature of the rime frost, more vapour would diffuse into it and more moisture will travel in the frost (as in insulation) and will freeze gradually in the air spaces. This will affect the thermal and physical properties of the frost as will also change the density of the frost. Given enough time the density may increase even ten times (1).

While at temperatures below 0°F, the frost is snow like and can easily be scraped even with a hard broom or light mechanical scraper. At such low temperatures, it will not stick to floor or surfaces. The crystalline frost on the other hand is hard to melt and remove.

1.3 FACTORS AFFECTING FORMATION OF FROST:

A number of variable factors affect the quantity and nature of frost formed on surfaces at sub-freezing temperature. Of these we have already indicated in previous article, how the temperature and duration of operation affect.

1.31 FUNCTION OF EVAPORATION:

The temperatures to be maintained in the refrigerators depend upon the usage and the type of service it is intended to be put to. In case of display cases, the temperature of the evaporator should be lower than that required otherwise for the same usage. This should be so because in the later case a smaller temperature differential between the refrigerant and the product area exists.

Different perishables are to be maintained at different temperatures. Ice-cream display cases must operate with a fixture temperature between $-20^{\circ}F$ to $-10^{\circ}F$ and the refrigerant under such conditions would be required to evaporate at about $-40^{\circ}F$; forzen food display cases are to be maintained at somewhat higher temperatures, nearly $-10^{\circ}F$ to $0^{\circ}F$ and the refrigerant would evaporate at about $-30^{\circ}F$. Fresh meat will require fixture temperature of 28° to $32^{\circ}F$ and evaporator temperature of approximately $8^{\circ}F$; those values for fresh vegetables and dairy products are: fixture temperature 36° to $42^{\circ}F$ and an evaporator temperature of $12^{\circ}F$ nearly. (2)

1.32 <u>CONDITION OF COIL</u>:

The state of coil surface also affects the rate of formation of frost. The rate of frost deposit on a clean tube is much greater than the rate when it is with frost (3). Messrs [illegible].E. Sabbitt, W.E. Fontaine and J.P. Doston have published some interesting data regarding its formation on metal surfaces (4). Presence of dust particles or other foreign material promotes the formation of frost. Many a coatings are known that would condense supercooled water at temperatures below $32^{\circ}F$, which could be removed without frosting. It is possible to treat and design cold surfaces, with a thin coating of Polystyrene or a silicone oil or rosin, that will condense supercooled water upto -40° F. Minimum values of -38° F have been obtained under favourable conditions of the clean surfaces and dust-free atmosphere. Presence of dust particles reduces this limit considerably and below -17° F it would be extremely difficult to prevent sizeable quantity of frost formation over the coils; and under useable conditions this imposes lowest limit to temperature with coated surface.

Adhesion of the frost to metal surface also decreases by coating the metal with proper materials. In general any non-wettable substance was considered of promise. The inhibition to frost formation tended to increase with the decreasing tendency of the surface to the adhesion of ice. It can again be said that presence of dust affected inhibition and that its presence cannot be eliminated when the circulation of air was dependent upon the operation of the fan. Hydrophillic compounds, antifogging fluids and a number of antirust compounds tended to promote the initiation of frost. On surfaces quoted with such materials, layers of ice are formed much more rapidly than on clean surfaces. It is also interesting to note that many bugs prevent water to freeze at temperatures below 32° F.

1.33 <u>HUMIDITY</u>:

The temperature and humidity of the surroundings also affect the formation of frost, particularly in the open type display cases. Thus if the display case is placed in an air conditioned space with a different temperature and humidity, the quantity of frost accumulated will be different. The humidity of the warm air leaking from outside will also affect the rate of its deposition. It will also depend upon ..

the moisture picked up from the products or its containers. However, this factor produces considerably small accumulation of frost.

1.34 MISCELLANEOUS FACTORS:

The amount of frost accumulated will also depend upon a number of other factors:

i. Excessive refrigeration load: hot foods or steaming foods or liquids placed in the cabinet, poor quality of insulation or that in which water has been soaked, or system with smaller capacity for actual load at desired temperatures will put excessive load on the refrigerator thus exposing the evaporator for larger frost formation.

ii. The size of the refrigerator and evaporator coil design: larger evaporators will collect more fost. In finned coils the spacing of the fins is also important, closer fins deposit greater amount of frost.

iii. The quantity of air leakage from outside: at doors and walls the protection against leakage would effect this factor.

iv. Doors: opening the doors will communicate the cold space with the ambient and will allow some fresh air to be picked up by the refrigerator. Doors opened often and for longer time will allow more frost to be accumulated. The tendency to collect more frost will also be enhanced if the doors are loose on hinges or latches or if they have broken or flattened gaskets.

v. Cycling Control incorrectly set: if set for lower temperatures or pressures than optimum, frosting would be promoted.

vi. Restricted air circulation: in systems where air is circulated, an inadequate circulation may be caused by;

(a) crowding the food containers which may impede the flow of air from the evaporator, through the entire cooled space and back to the evaporator;

(b) baffles or flues either omitted or in-correctly placed;

or (c) evaporator is too wide, too high, too long or incorrectly placed to allow free circulation of air.

vii. Too low temperatures: evaporator being too small has to be operated at too low temperatures.

viii. More refrigerant is being admitted to the evaporator than can be evaporated with the load on the system and pressure in the evaporator. This will cause frosting on the suction line even.

ix. In some cases the liquid line may also develop frost. This may happen when the receiver valve is partly closed or the liquid line strainer gets clogged at receiver or condenser (5).

x. Whole of the snow condensed does not stick to the evaporator. A variable which may be interesting to note is the proportion of snow condensed that actually adheres to the tube wall.

xi. The possibility of supersaturation can be regarded as unlikely. The cooling surface will become coated with ice crystals and further crystals will be held in suspension in the air stream. This will furnish sufficient nuclei for further condensation.

xii. Increasing the air velocity will increase the amount ..

of adhesion. It will increase the amount of vapour admitted to it and shall also affect the pressure drop for the flow (6).

1.4 INFLUENCE OF FROST ON PERFORMANCE OF REFRIGERATION SYSTEMS.

1.41 INSULATING EFFECT:

The frost accumulated acts as an insulator and reduces the heat transfer capacity of the coil. A coat of frost on the evaporator coil wastes power and causes needless wear. For example consider a cold storage room piped with 1,000 sq. ft. of 2 in. direct expansion coils. Assume that it is figured on heat transfer coefficient from still air to evaporating surface of 2 Btu/ (sq. ft.) (hr.) ($^{\circ}$F). The temperature in the room is 30°F and that of the refrigerant is 5°F. The refrigerating capacity under these conditions would be 4.16 tons. If the coil is covered with 2 in. of fine dry frost the coefficient of heat transfer may be as low as 0.9 Btu/ (sq. ft.) (hr.) ($^{\circ}$F). So the capacity of the plant would be decreased about 50 to 60 percent (7).

Frost on the coil does not possess same insulating proportios all the time. There are multitude of minute air spaces between the network of ice crystals which render the insulating effect. (As moisture travels through these air spaces the insulating effect of the deposit varies. Pure ice is nearly 20% as effective an insulation as the cork but the frost containing air spaces has got greater insulating effect. One inch of frost can attain the insulating value of ½ in. of cork (7), while in extreme cases the coil surface thus covered with a multitude of air cells between ice crystals is as efficiently insulated against heat transfer as if it were insulated with cork or asbestos (8).

Some good work has been reported by Mr. W. F. Stoecker (9). He performed his experiments with finned coils wherein he has studied the effect of frost accumulated on the overall heat transfer coefficient Uo. Two graphs plotted between Uo based on L.M.T.D. between the air and the refrigerant, and the air side surface area of the frost free coil, and the quantity of frost accumulated on the coils are reproduced here. These graphs are for different fin spacings a close finned coil with nine fins per inch and the other a wide finned coil with four fins per inch; refer to figures 1.41 and 1.42.

The graphs reveal that the nature on the two cases of variation is qualitatively the same although actual figures vary. With first few pounds of frost deposition, the value of Uo increases and reaches a peak which depends upon the rate of air flow through the evaporator coil and the fin-spacing. From the peak the value of Uo gradually drops as more and more of frost collects. A possible explanation for such a behaviour of the values of Uo has been given as follows: A slight layer of frost provides a rough extended surface area for the coil. Another reason given was that as the flow rate of the air was kept the same, the air velocity increased because of the reduced area for air flow, area having been reduced because frost accumulated occupied certain volume. This increased velocity increases the air side coefficient. The eventual drop in the value of Uo can be assigned to the increasing insulating effect of frost as it collects in larger quantities. For further information table 1.41 with its discussion may be consulted in Art. 1.43.

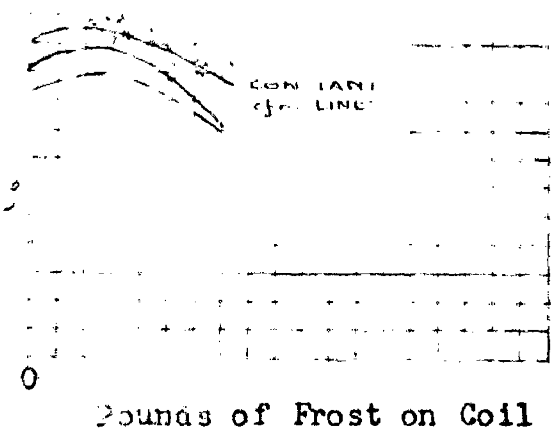

Fig. 1.41 EFFECT OF FROST ACCUMULATION ON U_o (CLOSE FINNED COIL)

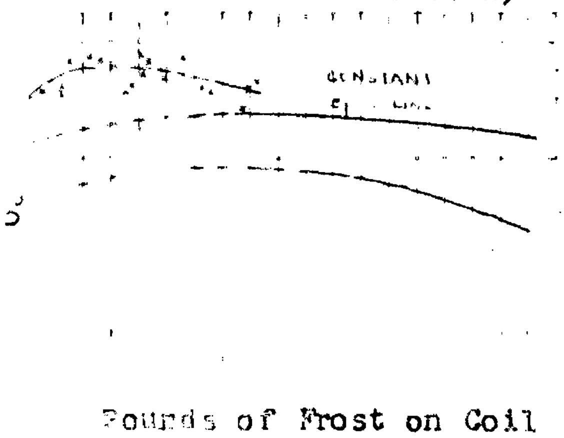

Fig. 1.42 EFFECT OF FROST ACCUMULATION ON U_o (WIDE FINNED COIL)

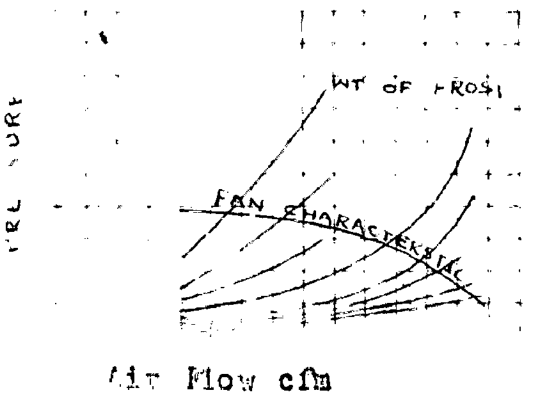

Fig. 1.46 FAN CHARACTERSTIC AND PERFORMANCE CURVE DURING FROSTED CONDITION (WIDE FINNED COIL)

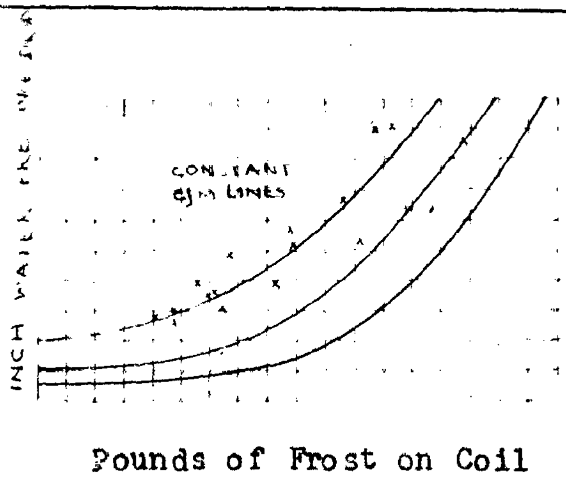

Fig. 1.43 EFFECT OF FROST ON PRESSURE DROP OF AIR (CLOSE FINNED COIL)

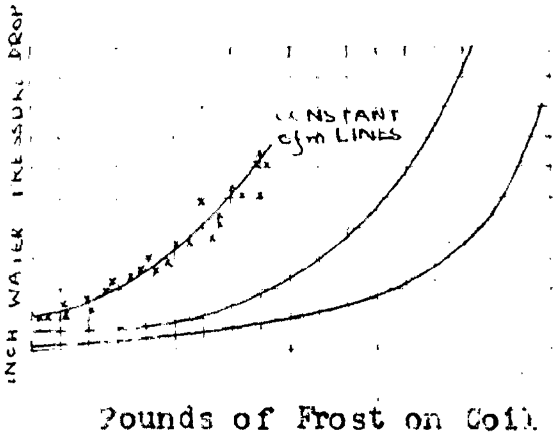

Fig. 1.44 EFFECT OF FROST ON PRESSURE DROP OF AIR (WIDE FINNED COIL)

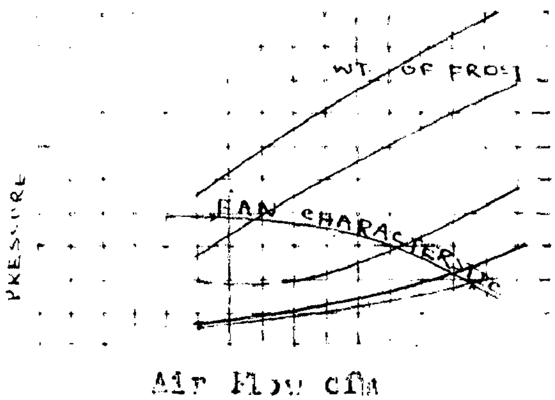

Fig. 1.45 FAN CHARACTERSTIC AND PERFORMANCE CURVE DURING FROSTED CONDITION (CLOSE FINNED COIL)

curve between air pressure drop in inches of water and air flow in cfm shown in fig. 1.45 is sumerimposed on fig. 1.43. This yields fig. 1.45 and represents the air flow in cfm for various values of frost accumulated. This figure reveals the truth of the above statement. Similarly fig. 1.46 is reproduced for wide-finned evaporator.

The effect of frost in diminishing air flow is important because this would require a larger L.M.T.D. to produce same refrigerating effect, necessitating thereby a lower refrigerant temperature. We have already discussed that more power per ton of refrigeration would be consumed when the evaporator temperature is lower.

1.43 PRESSURE DROP vs. INSULATING EFFECT - FINNED COIL:

The effect of frost deposit is more important than on the heat transfer coefficient. This fact can be illustrated by the following example: A simple calculation would show that the effect of decrease in Uo is not so pronounced quoting from Stocker (9), if the coil is providing one ton of refrigeration with an air flow of 1,450 cfm.; air entering at 32°F and leaving at 24.6°F and a refrigerant temperature of 16°F; the L.M.T.D. would be 11.9°F. A decrease of 12% Uo (assuming air flow to be maintained constant) will warrant for an increase of 12% in L.M.T.D. i.e. to 13.3°F or will require the refrigerant temperature to change from 16° to 14.6°F which is quite small and would probably not be of much concern in refrigeration plant.

To show that the effect of blockage in air flow is more important, he further cites another example: consider a plant of one ton refrigeration capacity, entering and leaving air temperatures being 32°F and 24.8°F respectively, refrigerant temperature

Further with the help of calculations it has been shown that for an actual case when the system be run to meet a constant load the value of overall heat transfer coefficient increases first with frost accumulation but later drops rapidly with additional frost accumulating.

If Uo decreases, for the same refrigerating effect, L.M.T.D. must proportionally increase to compensate this effect and that an excellent guide to the efficient operation of a refrigeration system is the refrigerant temperature that is required to produce the desired refrigerating effect. Lower refrigerant temperatures would need more power input per ton of refrigeration. Thus a decrease in the value of Uo tends to impair the performance of the coil.

1.42 PRESSURE DROP OF AIR-FINNED COIL:

The frost accumulated keeps the air from flowing into the cooling unit and reduces the free area for the passage of air. This factor is, however, not important in case of non-finned coils and where no air flow may be involved. Stoecker in continuation of his experiment also studied this effect. (9) Figures 1.43 and 1.44 are reproductions of his results plotted in graphical form between air pressure drop, inches of water, against the pounds of frost deposited on the above mentioned two coils. The common effect on either coil is that first few pounds of frost on the coils do not affect the pressure drop appreciably but after a certain stage has been reached, the further accumulation of frost increases the pressure drop considerably. This will produce a pronounced effect on the quantity of air flowing through the system. A fan characterstic which is a curve between air pressure drop in inches of water and air flow in cfm shown in fig. 1.45 is sumerimposed on fig. 1.43. This yields fig. 1.45 and represents the air flow in cfm for various values of frost accumulated. This figure reveals the truth of the above statement. Similarly fig. 1.46 is reproduced for wide-finned evaporator.

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To show that the effect of blockage in air flow is more important, he further cites another example: consider a plant of one ton refrigeration capacity, entering and leaving air temperatures being 32^{o}F and 24.2^{o}F respectively, refrigerant temperature

15.9°F, therefore L.M.T.D. is 12.2°F, Uo value is 2.66 Btu/hr. ft^{2}. $^{\circ}$F. With 6½ lb. of frost collected air flow and Uo values are read from graphs 1.45 and 1.41, as 540 cfm. and 2.10 Btu/hr. ft^{2}. $^{\circ}$F. Because of reduced air flow, the outlet temperature would decrease as temperature drop of air through the coils must increase to give 1 ton of refrigeration. This would involve the leaving air temperature as 12.2°F, and with the estimated value of Uo, to produce one ton of refrigeration, L.M.T.D. should be 15.41°. Calculating it will be found that the refrigerant temperature under such conditions would be 6.7°F. Thus the required refrigerant temperature will vary from 15.9°F to 6.7°F and would be of concern to the operator since it would mean an increased power consumption of about 23 percent.

The table 1.41 is extracted from the aforementioned article by Mr. W. F. Stoecker and shows the effect of frost accumulated on the performance of the coil. Under different frost conditions the value of discharge is read from chart 1.45 and the value of Uo is read from chart 1.41, the area of the coil being 370.2 sq. ft. Entering air temperature was adjusted at 32°F and so leaving air temperature per ton of refrigeration could be found. Knowing the UoxA, leaving and entering air temperatures, would make it possible to calculate the refrigerant temperature to produce the required refrigerating effect.

TABLE 1.41 (9)

REFRIGERANT TEMPERATURE FOR CLOSE FINNED COIL

Lb. of frost	cfm.	Air flow lb. per min.	UoxA	Leaving air temperature.	Refrigerant temperature.
0	1,335	102.3	1,007	26.91	22.00
1	1,345	101.0	1,036	26.85	22.17
2	1,310	98.2	1,043	26.70	22.20
4	1,140	85.5	1,036	25.90	21.75
6	680	51.0	825	21.80	18.32

A study of the figures reproduced in table 1.41 will reveal that the effect of reduced air flow is more important than the decrease in the value of Uo. First few pounds of frost deposited does not appreciably reduce air flow and heat transfer coefficient but with time, frost would build up to a stage that the pressure drop curve (Fig. 1.43 and 1.44) become steep. In this operating range the pressure drop affects the refrigerant temperature more than the reduction in the value of Uo.

4.5 FROST MUST BE REMOVED PERIODICALLY. (ENERGY, TEMPERATURE AND SPACE CONSIDERATIONS)

With the compressor controlled by temperature of refrigerated space, the percent operating time and the energy requirements will be expected to increase during running. This increase, however, will depend upon the type of system for freezer box it was found to be small. With the compressor controlled by temperature of refrigerant the temperature of the refrigerated space will go up owing to a blanket of frost deposit around the tubes (10).

In addition the accumulation may occur to such a depth that inconvenience will result and a large reduction in storage space may accrue. This is usually a problem of freezer boxes.

The accumulation of frost, therefore, impairs the performance of a refrigeration system. For intelligent design and operation of a plant its performance must be studied. Additional coil surface will be required to meet the retarded operating conditions and the designer must know and provide for it. In actual operation the frost would keep on accumulating gradually and in order that the coil performance and operating conditions are not adversely affected it must be removed periodically. The user shall know that, for best economy and/or convenient operation, at what stage he should defrost the plant. It has been shown for the finned coils the air flow is the best indication of the optimum time to defrost and the above discussion shows that this corresponds to the time when the pressure drop curve becomes steep. An air flow meter or static pressure indicator in the air duct would appear to be the best monitor to show when the air flow has dropped to a point when coil performance is suffering and defrosting is needed. In case of freezer boxes the space blockage may be the main criterion.

CHAPTER 2

DEFROST METHODS

Various defrost methods, their merits and demerits are discussed.

2.1 INTRODUCTION:

Frost accumulation is detrimental to the performance of the unit. As more and more frost accumulates the heat transfer capacity of the coil decreases. Whether a commercial refrigerator or a cold storage plant, the problem of deposition of frost and its removal is quite involved. Refrigerator can not be interrupted very often or for too long, on the other hand the growing layers of frost on the evaporator coil or pipe so inhibit the transfer of heat that the compressor cannot meet the load and hold down the temperature even if it is to run continuously.

Any defrosting operation comprises of two problems one is how to defrost and the other concerns the control of operation. The first point will be discussed here in details while the second point will be discussed in the proceeding chapter. There are a number of possible variations in the methods adopted for removing the frost from the coils. They have been used by different manufacturers and users, and have found varying adaptability. All such problems concerned with the removal of frost, however, have simple solutions, at least in principle, but tend to become expensive and may involve complicated mechanisms in actual applications.

2.2 MECHANICAL METHODS:

The one and quite crude method consists of removing the frost by scraping it off. This method is barbarous and may result in various leaks in the refrigerant piping, that may be caused by chosing sharp instruments indiscriminately.

Another method requiring force works only in case the -

frost is fluffy. The evaporator compartment is pressurised gradually until a pressure of about 50 lb. per sq.in. is built. This pressure is then suddenly released to the atmospheric pressure. The entrapped air within the structure of frost escapes, the frost bursts and is carried away by the outrushing stream of air.

It has a disadvantage in that it requires a pressure proof evaporator, so that the air does not leak into the refrigerant line. This process is used in liquid air plants where the refrigerant also is air and any leakage if it occurs inwards, will not deter the refrigerant.

2.3 SPRAY SYSTEMS:

2.31 De-icing fluids can be used to remove the deposited frost. These fluids are allowed to flow over the frosted surfaces where they loosen and melt the frost, pick it and remove if off.

Figure 2.31 shows a portable type of brine coil defroster. In this the brine is heated by means of electricity. The brine coil to be defrosted is shut off from the circulation and is connected with hose pipe to the warm brine mains. The temperature of the brine is not above 150°F. Hot brine will then circulate through the brine coil and the frost will loosen, melt and drift away.

A variation of the above device can be in the mode of heating the brine. The warm brine coil may be thermally attached to the condenser or to its cooling water discharge and connected to a pair of defrosting mains with a pump. Heat is thus carried from the condenser to the defrosting mains to melt the frost and it also affects at least a saving of cooling water.

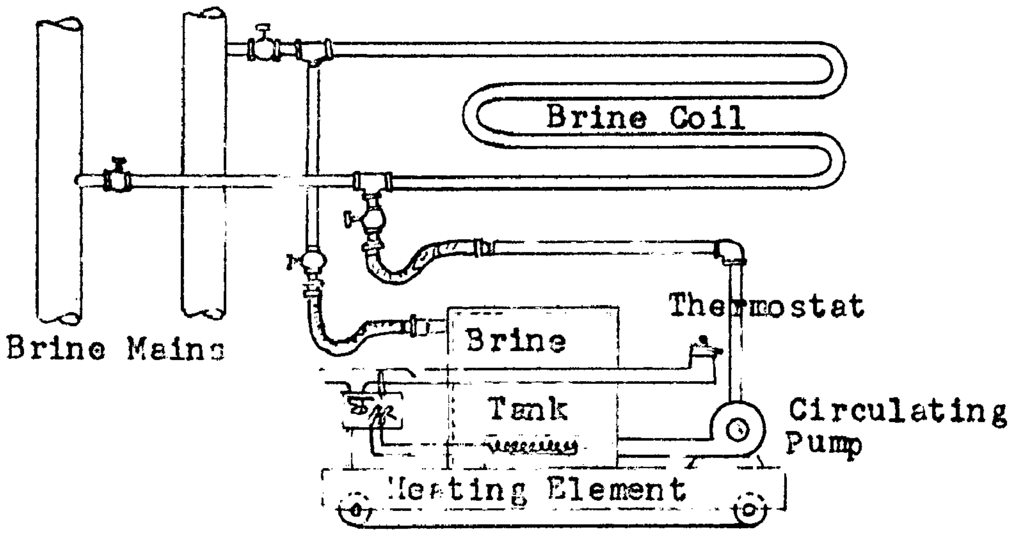


Fig. 2.31 PORTABLE BRINE COIL DEFROSTER

In some applications to meet emergent cases, servicemen pour water over thickly frosted coils whereby the frost is removed. The defrost methods using water can be employed in permanent installations but for the freezing of water in the piping carrying water during the normal operation. Improvements have been suggested and in some units water is used as the defrost medium. (Art. 2.4)

2.32 UNIT COOLERS:

These are brine spray refrigeration units which are a technical variation of the above method. The evaporator is placed in the brine collecting tank and the brine is sprayed over the evaporator, rejects heat to the evaporating refrigerant thereby getting itself cooled. This cold brine performs an additional task of heat carrier and cools the air. The pressure of the spray is very low and it requires a small horse power brine circulating pump. This is called brine spray unit of refrigeration or the indirect open spray method.

These units are generally floor typed, fig. 2.32 shows one such unit. It consists of a steel frame, and located in this frame are evaporator coils, brine spray header with sparay nozzles and eliminators. Other accessories required are a brine circulating pump, salt bins and a sump, and a fan to maintain the circulation of air. In this case no frost will ever be accumulated on the evaporator coil so long as the concentration of brine is maintained at a proper level depending upon the temperature of the refrigerant in the evaporator coil. Any moisture that precipitates out from the air will dissolve in the downward spray of brine and no frost will be formed. The

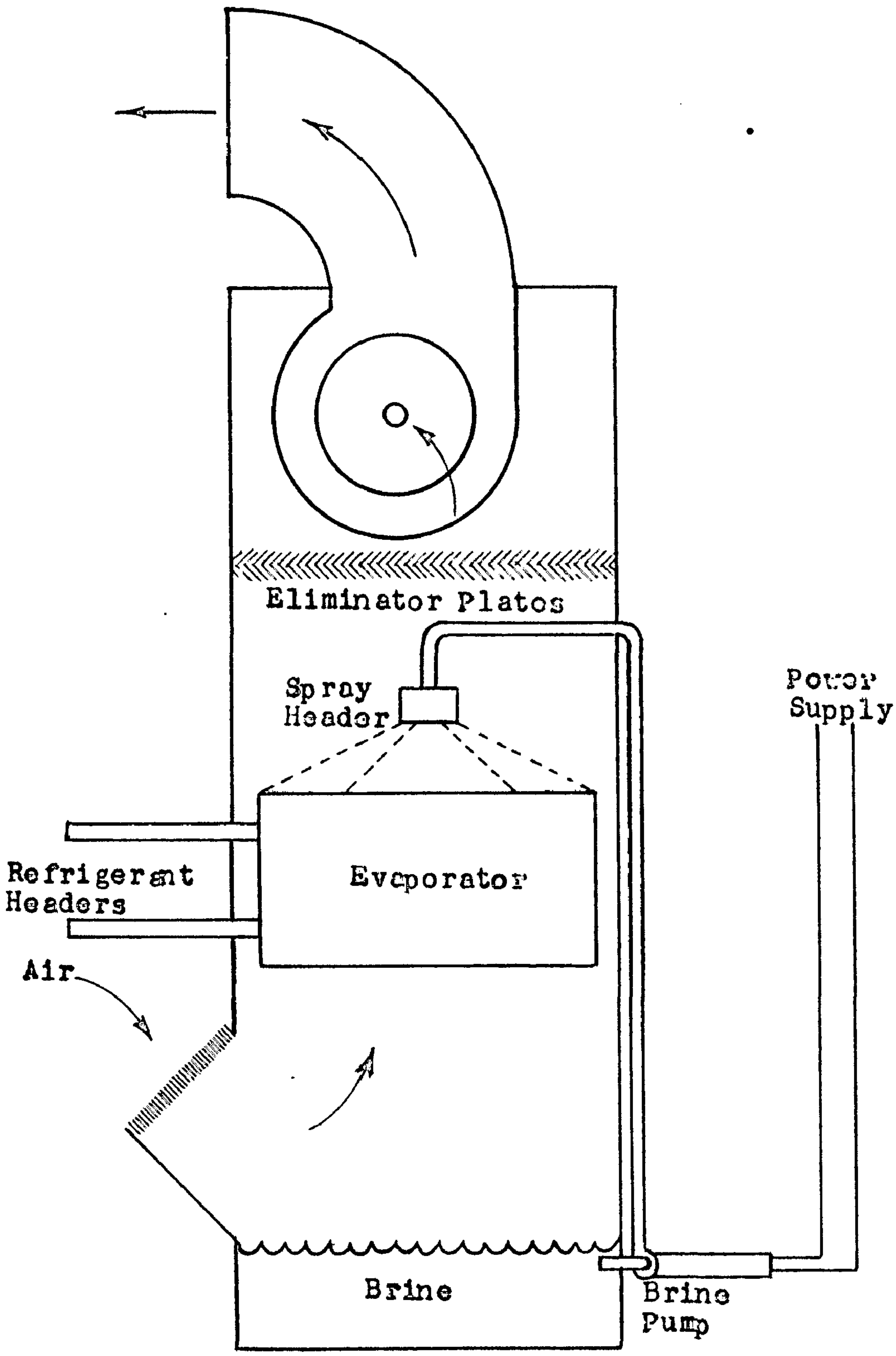


Fig. 2.32 BRINE SPRAY UNIT

brine, however, will be diluted and in order to maintain the strength, it must be concentrated by adding more of salt.

The advantage of the spray units is that the evaporator coil is always free of frost so long the proper concentration of the antifreeze solution is maintained and thus allows maximum performance of the evaporator coil. It has certain disadvantages too, in the form of corrosion possibilities, replenishing the brine strength and maintenance difficulties etc.

2.4 WATER DEFROST SYSTEM:

It is a system in which an external source of heat viz. water is used to melt the frost from the metal surface. It consists of a water spray chamber above the coils and the water is sprayed by the nozzles that are arranged on a header. Fig. 2.41(a) shows such a system. Water for spraying is usually obtained from the supply line and the rate of flow is to be maintained at about 40 gallons of water per ton refrigeration capacity of the coil, for 10 T.D. (14). The time taken will be about 2 to 5 minutes for daily defrosting assuming an adequate quantity of water not too low in temperature. This shows that a large amount of water will flow during this period and therefore proper precautions need be taken to drain the water completely lest it freezes in the spray chamber after the defrost period. The spray-header is pitched so that the water drains from it at the end of the cycle; vents are provided in the water line from the valve to the distribution header to allow complete draining of the remaining water after defrosting cycle, where otherwise it will freeze.

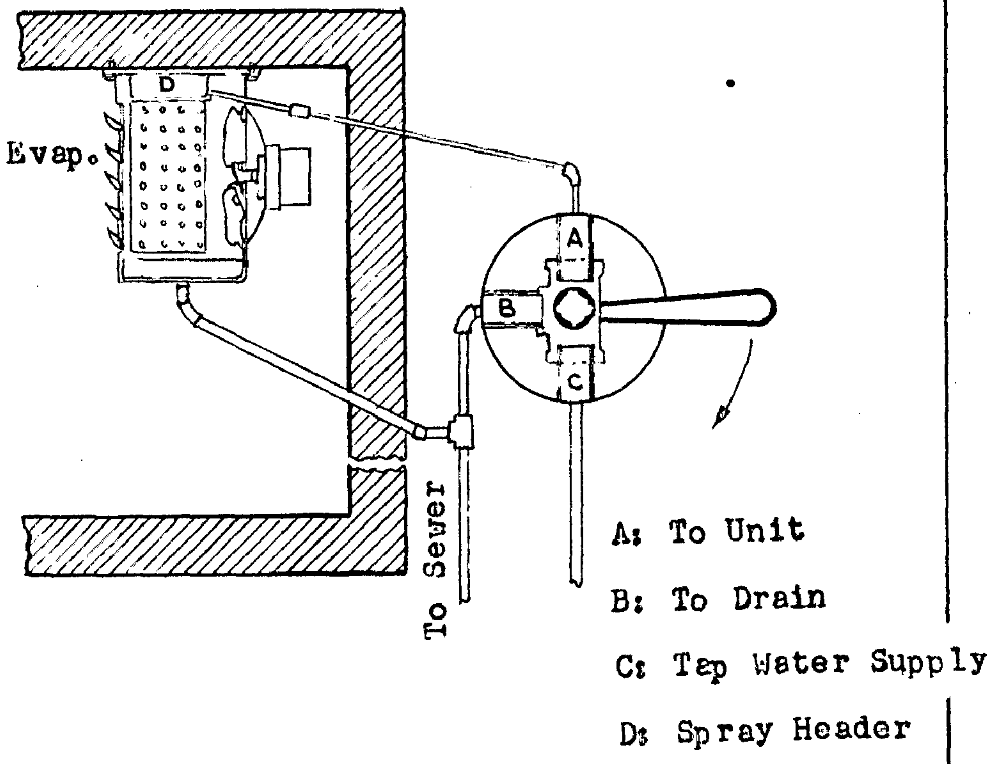
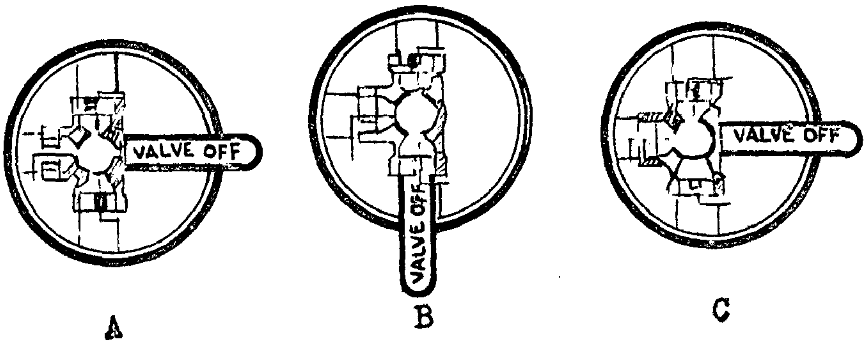


Fig. 2.41(a) WATER DEFROST SYSTEM



A: Normal Operating Position

B: Defrost Position

C: Valve Returned to Normal Position

Fig. 2.41(b) THREE-WAY VALVE CONTROLLING WATER FLOW

A special three-way valve is used to control the flow of water, to drain the water from the water-line at the end of the cycle, and also to prevent building up of water in the supply line in case the valve should leak. An illustration on the working of this valve is shown in fig. 2.41 (b). When the valve is in the normal position the water supply is cut off and the lines from the spray chamber and the drain pan are connected to the drain. This ensures that any water that may have remained during previous cycle will drain off. During defrost period the valve would be turned to open the water supply to the spray chamber and the water will spray over the coils thus removing the frost from over them. The water from the drain pan below the coils will flow out through the sewer. Again, when at the end of cycle the valve would be returned to normal operating conditions, the water will be quickly drained off from the water lines and the spray header.

The three way valve that has been just described may be dispensed with and can be substituted by separate supply and drain valves. The arrangement shown in fig. 2.42 does not use it and has been often used on large installations to feed several coils at one time. In some cases the drain valve has been substituted by a restrictor tube shown in figure 2.43 so that it can drain the supply line reasonably fast after defrosting cycle, while at the same time it shall not waste water excessively during defrost. A ¼ inch OD copper tubing of suitable length has worked satisfactorily in most of the installations. Automatic control can be applied to the water defrost system with the help of a timer, fig. 2.44.

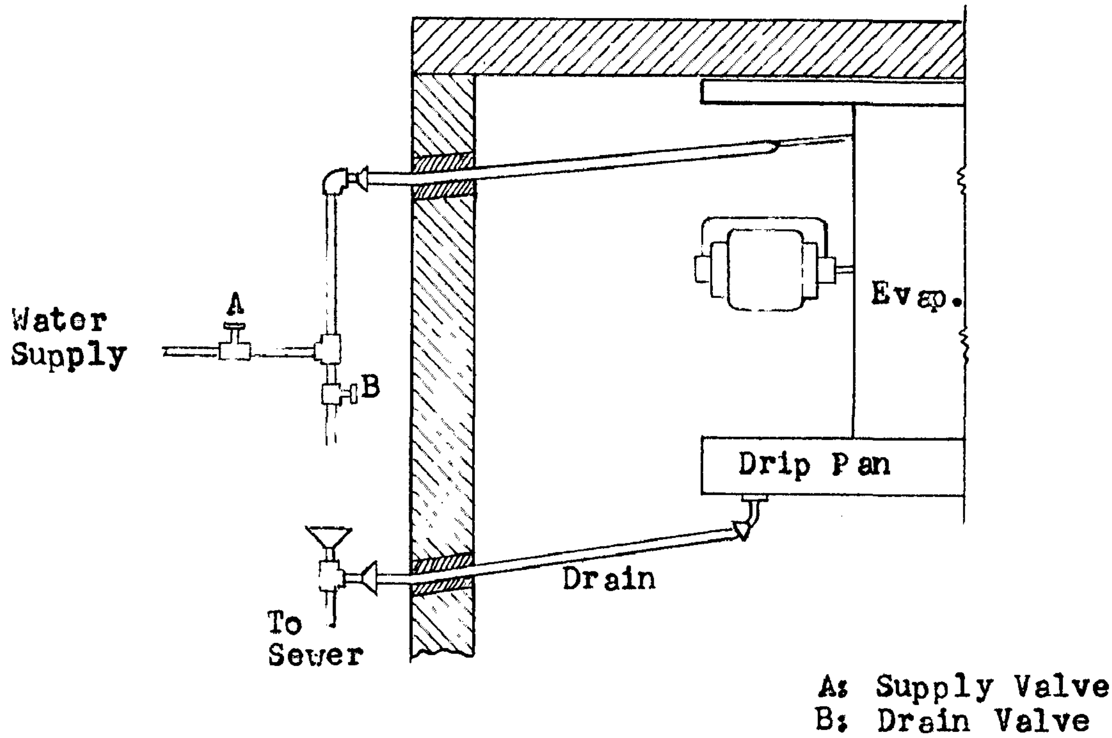


Fig. 2.42 WATER DEFROST WITH SEPARATE SUPPLY AND DRAIN VALVES

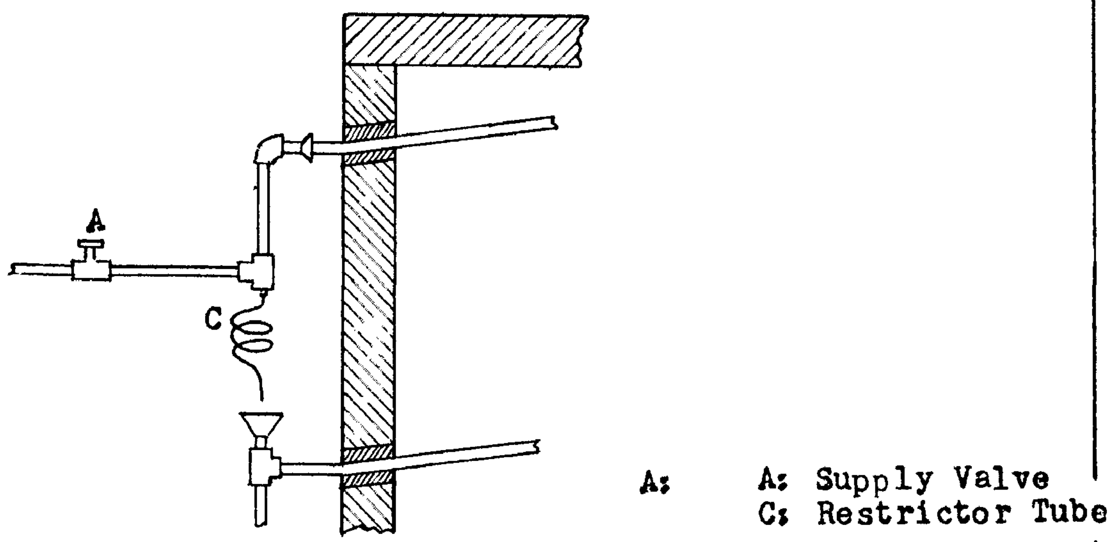


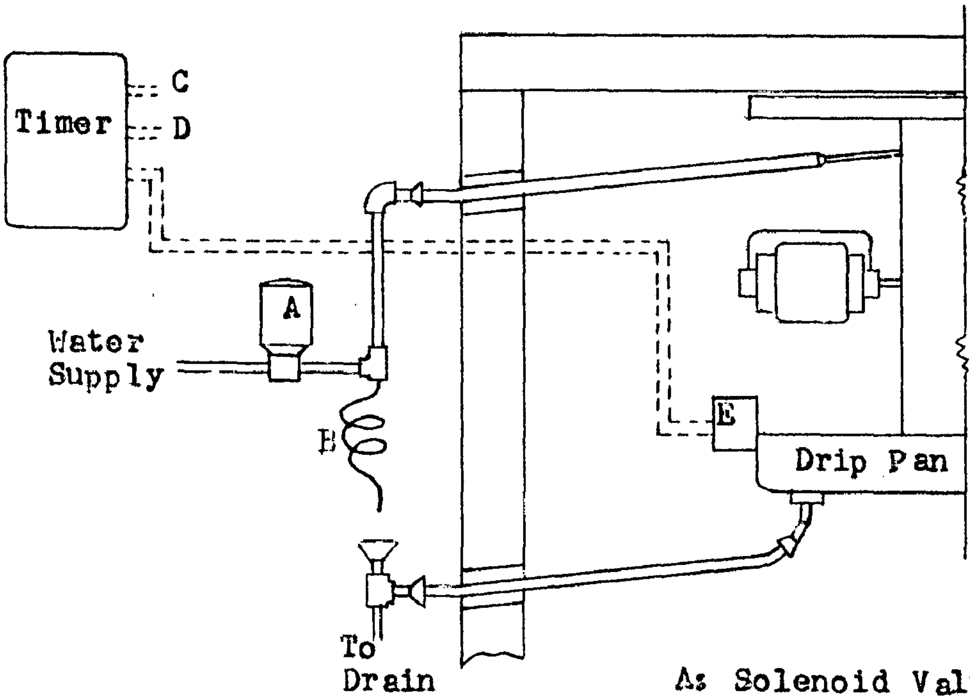
Fig. 2.43 WATER DEFROST USING SUPPLY VALVE & CAPILLARY DRAIN

The advantages of water defrost systems are its quick operation, and applicability for all evaporator temperatures. The disadvantages are the limitation of its usefulness only where water is available at temperature above 45^{o} F, difficulties with automatization, and considerable plumbing requirements.

2.5 NATURAL DEFROST (OFF PERIOD DEFROST):

The earliest and most elementary method is the manual shutdown where the operator stops the compressor and allows the frost on the coils to melt. It is the simplest but time consuming as no heat is added other than from the ambient. After the removal of frost the compressor is switched on.

It takes a long time, perhaps a few hours for the convection currents to melt the frost on the evaporator, and the temperature of refrigerated products would rise and its quality will suffer. It will also cause the articles placed in the freezer to melt. It is of common knowledge that products like ice cream etc. cannot be kept under such conditions and it is advocated that frozen food stuff shall not be thawed until they are ready to be eaten. For these applications and alike the off period defrost is not suitable. This method can only be applied where medium operating temperatures are involved (above 28^{o} F). In small refrigerators e.g. in house hold, in butcher boxes, interruptions being permissible, sufficient time can be allowed to remove the frost by picking up heat partially across the cabinet and partly from within the box itself. Thus in a way, the heat consumed in frost deposition will be reutilised by cooling the box with it during the defrosting. It is



A: Solenoid Valve

B: Restrictor Tube

C: To Fan Motors

D: To Compressor Motor or Refrigerant Solenoid Valve

E: Safety Float Switch

Fig. 2.44 AUTOMATIC DEFROST CONTROL: WATER DEFROST

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This method of manual shutdown is also used in large plants attended by competent operators, in general it is not satisfactory on present day commercial equipment commonly found in food and beverage dispensing equipment.

Automatically regulated shutdown methods are preferable in many applications. Man is not as dependable as an automatic control. Moreover the cost of performing the job manually or of failing to perform a function at a desirable time can be more than that of automatisation. There are four ways to control this method ; (i) Pressure control; (ii) Temperature control; (iii) Time control and (iv) Time initiate and Pressure terminate controls.

2.61 DEFROSTING BY ADJUSTING SUCTION PRESSURE:

This method is used in dairy and vegetable refrigerators where the air is circulated through the evaporator by fans, refer fig. 2.51. The compressor is switched off with the help of a pressure control cut out when the suction pressure falls below the cut-off point, and will be restored to normal operation when the pressure rises above the cut-in point. A cyclic operation will thus accrue. In the figure the low pressure cut out is shown in position and is located in the compressor circuit.

Fig. 2.52 shows the cyclic variation in suction pressure with respect to the time (2). The portion D represents the refrigeration period, while A,B, and C represent off periods following the switching off of the compressor. A is the period when the ice, the metal of the evaporator and the metal of

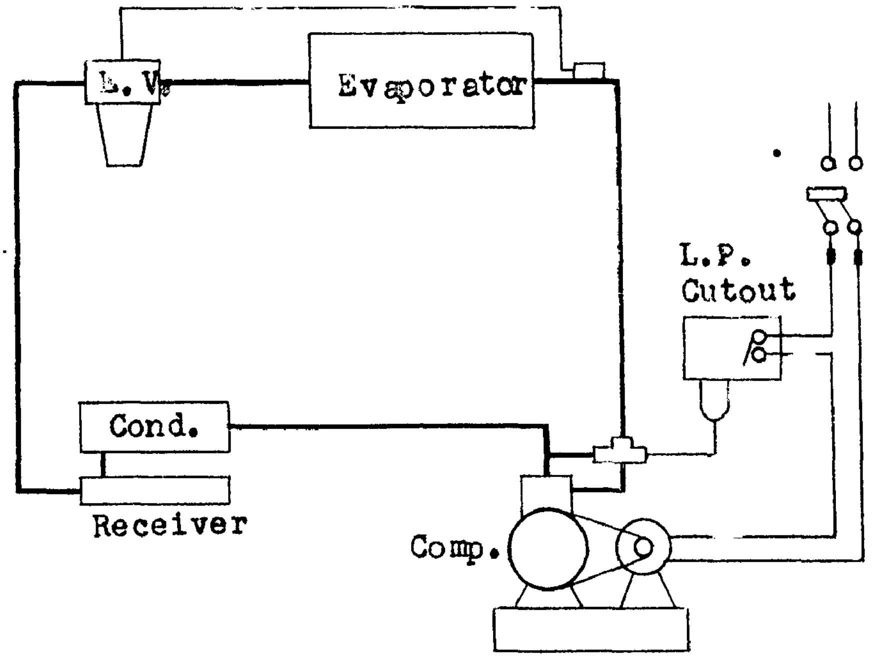


Fig. 2.51 PRESSURE DEFROST

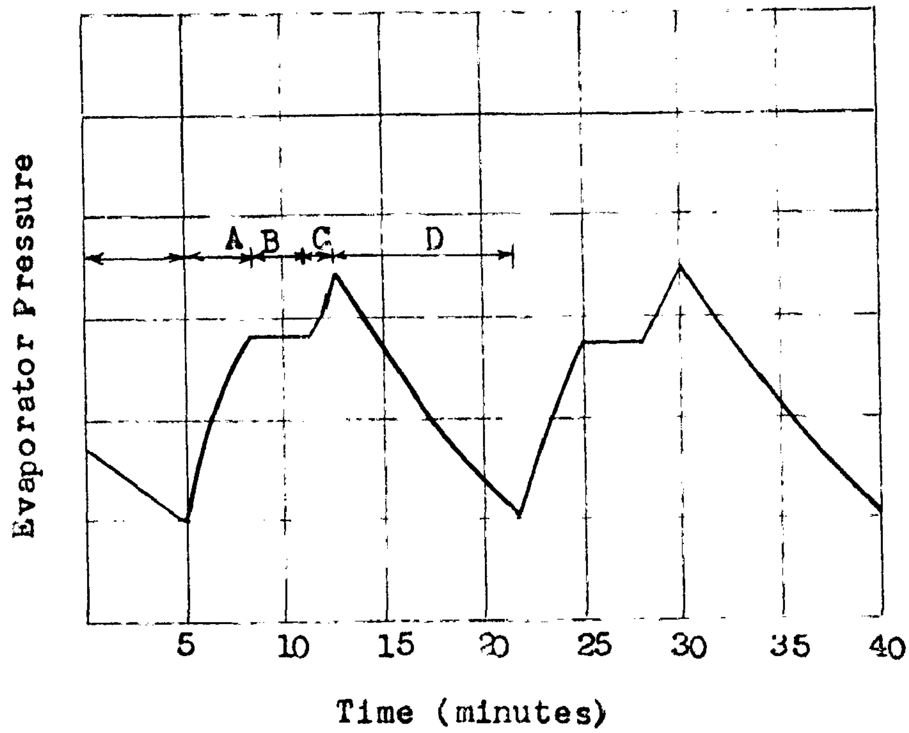


Fig. 2.52 VARIATION OF SUCTION PRESSURE WITH TIME FOR OFF-CYCLE SYSTEMS

walls surrounding evaporator are rising to 32° F; B is the duration of the actual melting of ice at a constant temperature; and C represents the time during which the refrigerant is rising above the temperature of 32° F. Indidentally C is the period of maximum importance as too long a period will increase the temperature undesirably while at the same time should be long enough to allow defrost water to drain clear from the fixture. With this defrost method large amounts of frost will never accumulate on the coil as an off period occurs whenever pressure falls corresponding to large frost accumulations.

In this case even if the heat load be such that no off can be reached, the evaporator pressure will eventually drop to the pressure control cut-out point should the frost be accumulated. This will allow the off period defrost cycle. A drawback of this method is, when it is used where remote condensing units are involved and an ambient at the condensing unit exists at a temperature lower than the evaporator temperature. It will cause the suction pressure to linger at a value below the pressure control cut in point. This will keep the compressor off for a longer time and the temperature of the fixture will rise. A similar condition will arise if the refrigerant lines from the fixture to remote condensing units pass through trenches or conduits with other cold refrigerant suction lines.

It should be noted that where temperature nearly 35° F or below are desired it is usually impractical to defrost every cycle by the low side pressure method.

2.52 <u>TEMPERATURE CONTROL:</u>

A thermostat is used to control the fixture temperature.

Fig. 2.53 represents a line diagram of such a system. As compared to the pressure defrost, a temperature control is substituted in place of low side pressure control. The cut in point is so adjusted that it corresponds to a defrosted evaporator. In some cases this method is preferable to the control by means of low side pressure switch such as in cases where conditions exist so as to cause the suction pressure to linger below the pressure control cut-in point.

A variation for normal temperature applications incorporates the use of special control involving control through a two temperature switch as in fig. 2.54. One temperature bulb is clamped with the evaporator and the other is mounted to sense the air temperature. The later bulb stops the compressor when the desired air temperature has been attained although it cannot close the switch to restore the normal refrigerating operation. This is done by the bulb on the evaporator coil, which is so adjusted as to switch on the compressor at a temperature of the evaporator corresponding to defrosted condition. The bulb on the evaporator surface cannot open the switch. It has the advantage that the defrost period comes and the compressor switches off when the circulating air has reached a particular low temperature, and the system is switched on to normal refrigeration operation as soon as evaporator has been completely defrosted.

This method again provides the defrosting action every cycle and has the limitation of becoming impractical where temperatures of nearly $35^{\circ}F$ or below are desired.

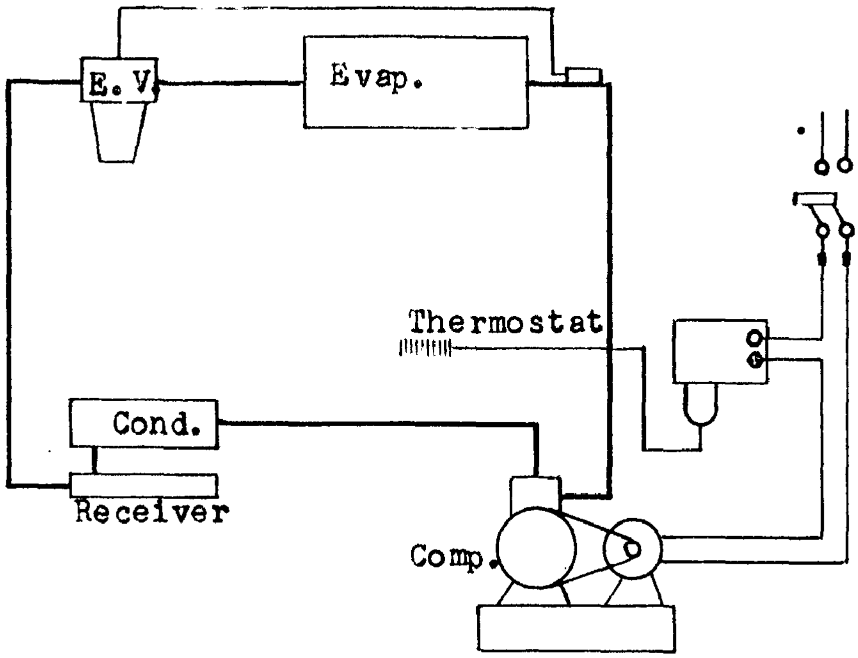


Fig. 2.53 TEMPERATURE DEFROST

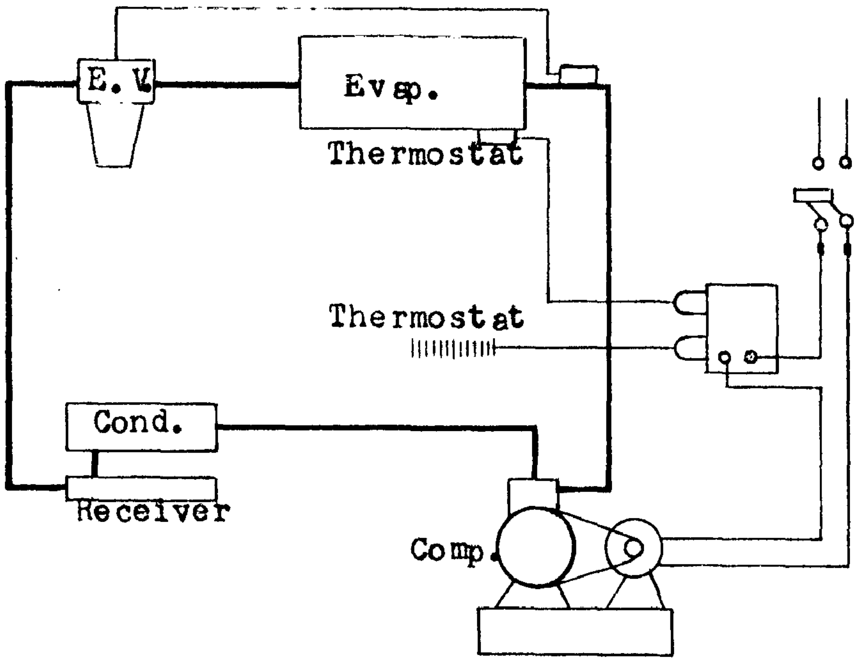


Fig. 2.54 TWO TEMPERATURE DEFROST

2.53 TIME CONTROL:

Time clocks are utilised to initiate and terminate the defrosting period. The timer clocks are wired so that the circuit to the condensing unit breaks for a predetermined time after fixed intervals. Fig. 2.55 carries a timer switch connected in the compressor circuit which will switch off the compressor after a fixed time period. After a predetermined time of off period to which the timer switch has been set it will connect the circuit of the compressor and start it. The frequency and duration of defrosting operations are selected and adjusted based upon the operational experience or under instructions from the manufacturers.

If the shut down period is not long enough to allow complete melting of the frost and its dripping off then ice and frost will complicate the operation. On the other hand if this period is adjusted to maximum load conditions, it will be too long under light load conditions and the temperature of the fixture will increase thus exposing the products stored to the danger of spoiling. The setting of timer presents complications in that the shut-down period varies with numerous factors and one setting will not be well suited under different conditions. Generally the setting of timer is done seasonally. As a rough guide the defrosting period in commercial refrigerators may be something from 45 to 90 minutes for forced air evaporators as used in dairy, vegetable and meat fixtures. A period of three hours or more is required for gravity air flow refrigerators. (2).

2.54 TIME INITIATE AND PRESSURE TERMINATE METHOD:

It is a definite variation of the previous method. A

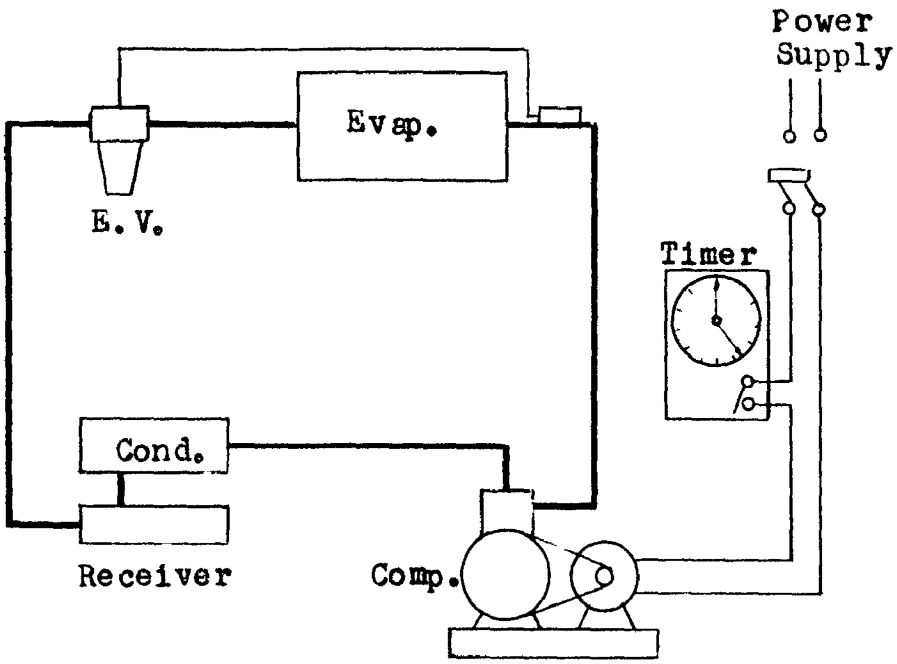


Fig. 2.55 TIME SHUT DOWN DEFROST

timer switch starts the defrost by opening the compressor circuit but does not close it. This period is terminated by a pressure cut-in control, refer fig. 2.56. Most of these systems have a safety time limit to operate and close the circuit should the pre-set value of pressure cut-in point not be reached. This safety time limit is generally adjustable. The frequency of defrost can be regulated from four to six times a day.

A graph between evaporator pressure and time of operation in fig. 2.57 is a typical representative of meat display unit utilising two to four defrostings per day (2). Vegetable and dairy display cases will have similar evaporator pressure characterstics and require approximately the same frequency of defrosts per day. It is clear from the pressure-time graph that as the frost is removed the low side pressure increases gradually, and as soon as the last bit of frost falls from the coil surface this pressure rises rapidly. When the pressure rises to the pressure cut-in point the system is turned on to normal refrigerating operation. The period of defrost is varied automatically depending upon the evaporator conditions.

In this case the uncertainty of adjusting the cut-in point to suit varying operating conditions, as involved in simple time defrost, is avoided. Defrosting is done at regular intervals while the normal operation is restored by a definite factor related to defrost viz. the evaporator pressure that corresponds to defrosted evaporator. Therefore, although the system is shut for long enough to defrost but no longer than what is necessary. Consequentially in this type the seasonal adjustments are eliminated and this system affords an obvious advantage over control by

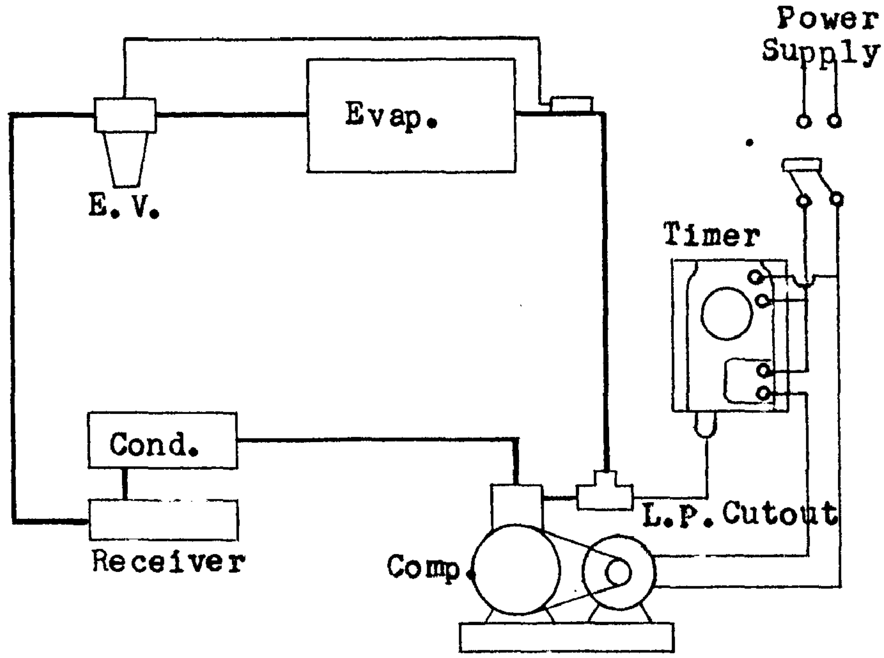


Fig. 2.56: TIME - PRESSURE DEFROST

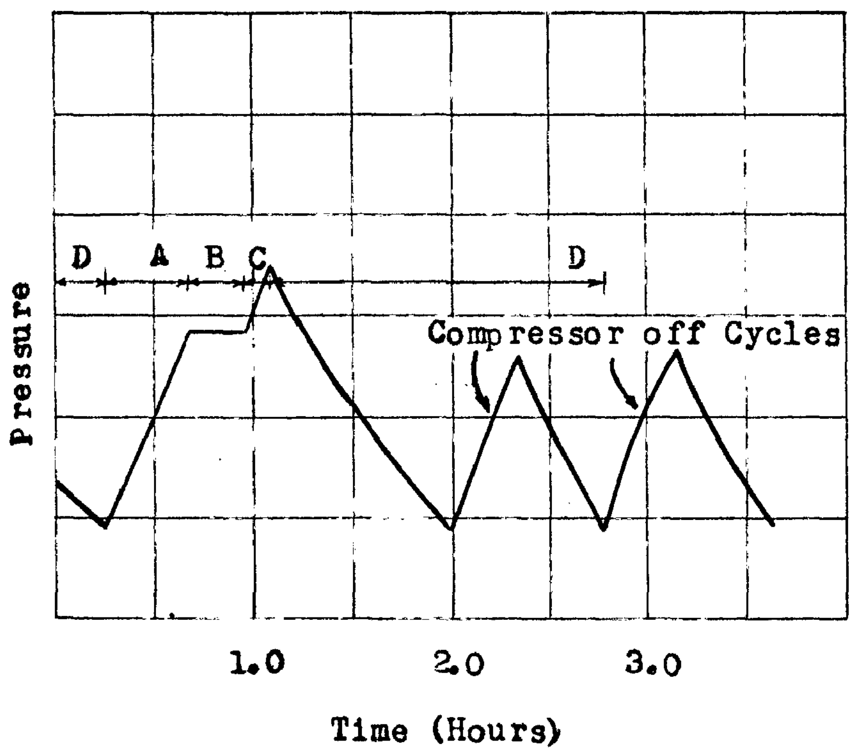


Fig. 2.57 VARIATION OF SUCTION PRESSURE WITH TIME FOR TIME INITIATE AND PRESSURE TERMINATE SYSTEM

plain timers.

2.6 WARM AIR DEFROST:

In this method warm air is used as the heat carrier for defrosting. To defrost, the evaporator is first isolated from the low temperature space by means of baffles. The high and low pressure sides are disconnected from the unit. Warm air is then passed over the coils till the frost has been removed. Heat from an electric source or from outside the refrigerated space i.e. the ambient, is utilised to supply a blast of warm air. After the frost has been removed, the system is restored to normal refrigerating operation. A few operators prefer to defrost the coils without disconnecting it from the low side. In this case the time required will be more as compared to that required after isolating the evaporator.

When the temperature of warm air is 70^{o}F it takes about half an hour for defrosting and with lower temperature of air, say 45^{o}F, longer time to the tune of one hour may be required. Obviously the defrosting time will depend upon ambient temperature and its relative humidity.

This method is in fact an attachment to refrigerating systems and is, therefore, preferred when the same are to be transported overseas. Two sets of doors are used with the evaporator, one communicating with the warm space and other with the cold space. By opening the set to warmer space and closing that to the cold space defrosting is accomplished, and during refrigerating operation the doors to cold space are open while the set to warm space is closed. This system lends itself to particular applications. Instead of adapting the refrigeration job to a standard unit, air defrosting units are usually designed to suit specific applications and fit in the space available for the equipment. The air handling equipment and the refrigerator are varied in design with the requirements of particular application.

It is one of the safest methods which does not disturb the conditions in cooled space. The defrosting cost is low, and there is flexibility in location of defrost units. The disadvantages include higher installation cost, special engineering and design cost for each unit, and inapplicability in low temperature ambients.

2.7 HOT GAS DEFROST:

Paradoxically enough it is an old but current method of defrosting the coil. For the last many years this system has been used on a number of commercial units and large cold storage plants. This is one of the most economic methods although the installation cost may be slightly higher than other systems. In this method all or a part of the hot gas from the compressor is used for defrosting the evaporator. Figure 2.71 shows the basic arrangement.

When the defrosting of the evaporator coil is desired the bypass valve is operated to disconnect the condenser etc. from the discharge line, and the bypass line is connected to discharge. The hot gases from compressor are led through the bypass line and enter the evaporator. The refrigerant vapours condense in the evaporator and reject heat to the evaporator and through it to the frost deposited upon it, which thereby loosens, melts and drifts away from the evaporator coil. Caution should be exerted that the valve is opened slowly and not too wide.

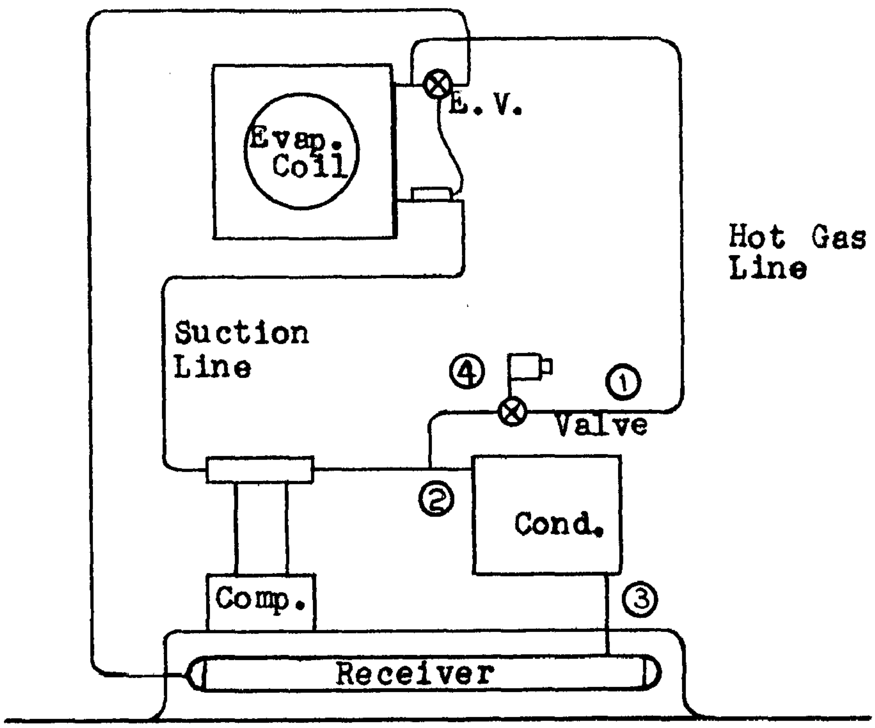


Fig. 2.71 BASIC HOT GAS DEFROST SYSTEM

High pressure refrigerant should be metered to pass through the hot gas line at a rate fast enough to prevent any appreciable pressure drop at the restriction i.e. bypass valve or solenoid valve in the bypass line but at the same time slow enough so that the refrigerant condensed in the evaporator may be re-evaporated completely before returning to the compressor. It is a very important aspect of the defrosting by hot gases.

In this type of defrosting the heat is supplied internally to the coil and some of the frost adjacent to the evaporator coils may melt and loosen the layer of frost above it, a part of which may fall off without melting. Adequate drain heaters shall, however, be provided. The basic cycle had many drawbacks:

(a) The hot refrigerant vapours that condense in the evaporator may not re-evaporate and liquid thus produced may either run back or be drawn back by the compressor cylinders. This may cause serious damage to them by liquid pumping.

(b) When defrosting, the coil is not absorbing heat and the re-evaporation may be too slow and in-adequate. Some refrigerant may remain in it after condensing. This may continue till eventually not enough quantity of refrigerant returns to the compressor. In such a case sufficient hot gas will not be available for complete defrosting and the system will run out of heat.

(c) It also depends upon high ambient temperatures and high condenser pressures. If the condensing pressures in the evaporator of the system during defrost (which are fairly lower than normal condensing pressures) correspond to a temperature near 32°F, little or no heat transfer from the refrigerant vapour to the frost on the evaporator will occur. The condensed liquid from the evaporator is to be re-evaporated by ambient at the same pressure and enough heat transfer from ambient, in suction line, for re-evaporation will occur only if the ambient temperature is high enough. Therefore this system is not dependable when the heat requirements for defrosting are large, particularly in winter.

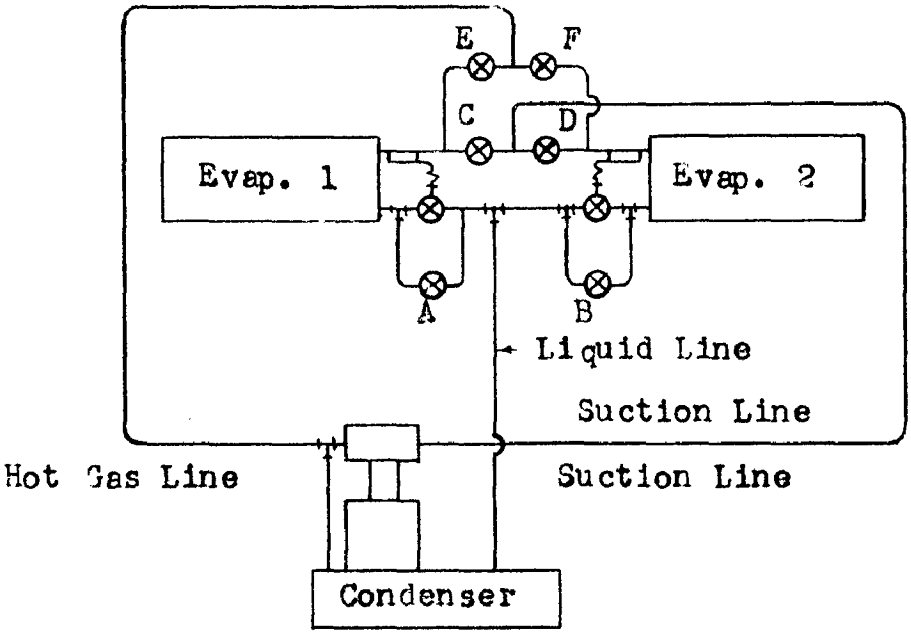
2.71 REGENERATIVE SYSTEM:

With the help of the foregoing discussion it can be observed that in a practical hot gas defrosting system there are two basic requirements that must be fulfilled.

(a) Liquid refrigerant must be prevented from slugging the compressor , and

(b) Sufficient quantity of heat must be available to be supplied to the system to obtain a rapid, complete and unfailing defrost.

As already pointed out the basic system does not work well, nevertheless it will lend itself well with multi-evaporator system. When more than one evaporator is connected to the same compressor, one evaporator may be defrosted with the heat available from the other evaporator. Fig. 2.72 gives it and shows the valve operation sequence too. This system also has certain limitation in that during defrosting if there are two evaporators of approximately the same size, the time taken is too large. This is because of the fact that it takes much more heat to remove frost than an equal size evaporator can pick up under normal refrigerating conditions. However, where there are more than two evaporators this method is normally reliable and rapid enough to



Defrosting Evap. 1

Valves open A and D then E

Valves closed B, C and F

Normal Operation

Valves open C and D

Valves closed A and B

E and F

Fig. 2.72 REGENERATIVE HOT GAS SYSTEM

remove the frost in reasonable time. The same holds when two evaporators of largely different sizes are used on a plant. The smaller evaporator shall be defrosted first immediately followed by the larger evaporator.

Fig. 2.72 also reveals that in regenerative method a large number of valves is required. This is no handicap when the operation is manual but is an outright bottleneck in the way of automatization of this system with the help of standard valves and instruments on the market.

2.72 AIR RE-EVAPORATOR:

The ambient air is used as the heat source to re-evaporate the refrigerant which has been liquefied in the evaporator during defrost. In fig. 2.73 is included a long suction line through the ambient, which may be an adequate heat source provided that the temperature of air is fairly above 32°F. But the rate of condensation is so rapid that a part of the liquid may not be re-evaporated and may pass on to the compressor. To overcome this difficulty the flow rate of the refrigerant is restricted with a hold back valve placed before the re-evaporator. The valve is so set that it remains wide open during normal refrigerating operation but as the suction pressure increases during defrost it starts modulating, thus checking inrush of liquid refrigerant. The modulating point of this hold-back valve is set at a pressure higher than normal suction pressure and shall be at the same time lower than that corresponding to the temperature of the air. At the end of defrost there will be considerable liquid refrigerant behind the hold-back valve. To ensure that whole of it is evaporate before the restoration of normal operation certain delay is allowe

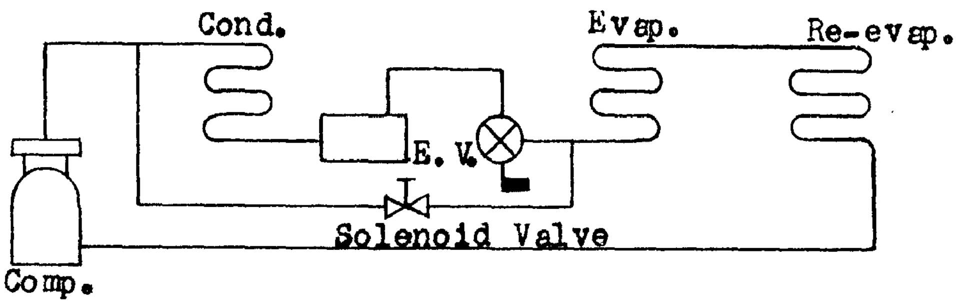


Fig. 2.73 HOT GAS DEFROST WITH AIR RE-EVAPORATOR

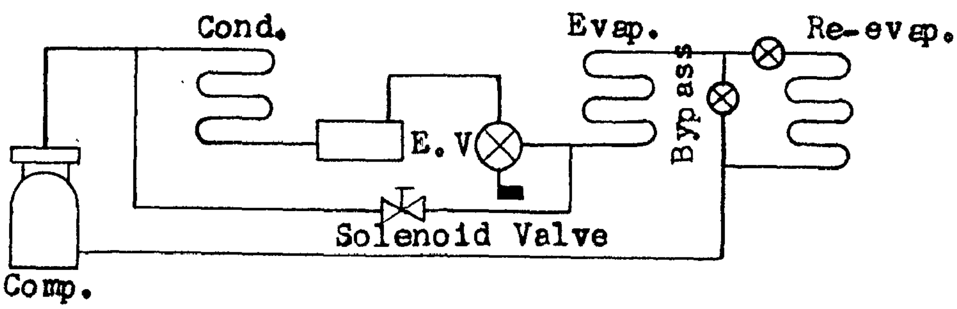


Fig. 2.74 HOT GAS DEFROST WITH AIR RE-EVAPORATOR AND BYPASS

between the hot gas line valve closing and the starting of the evaporator fans.

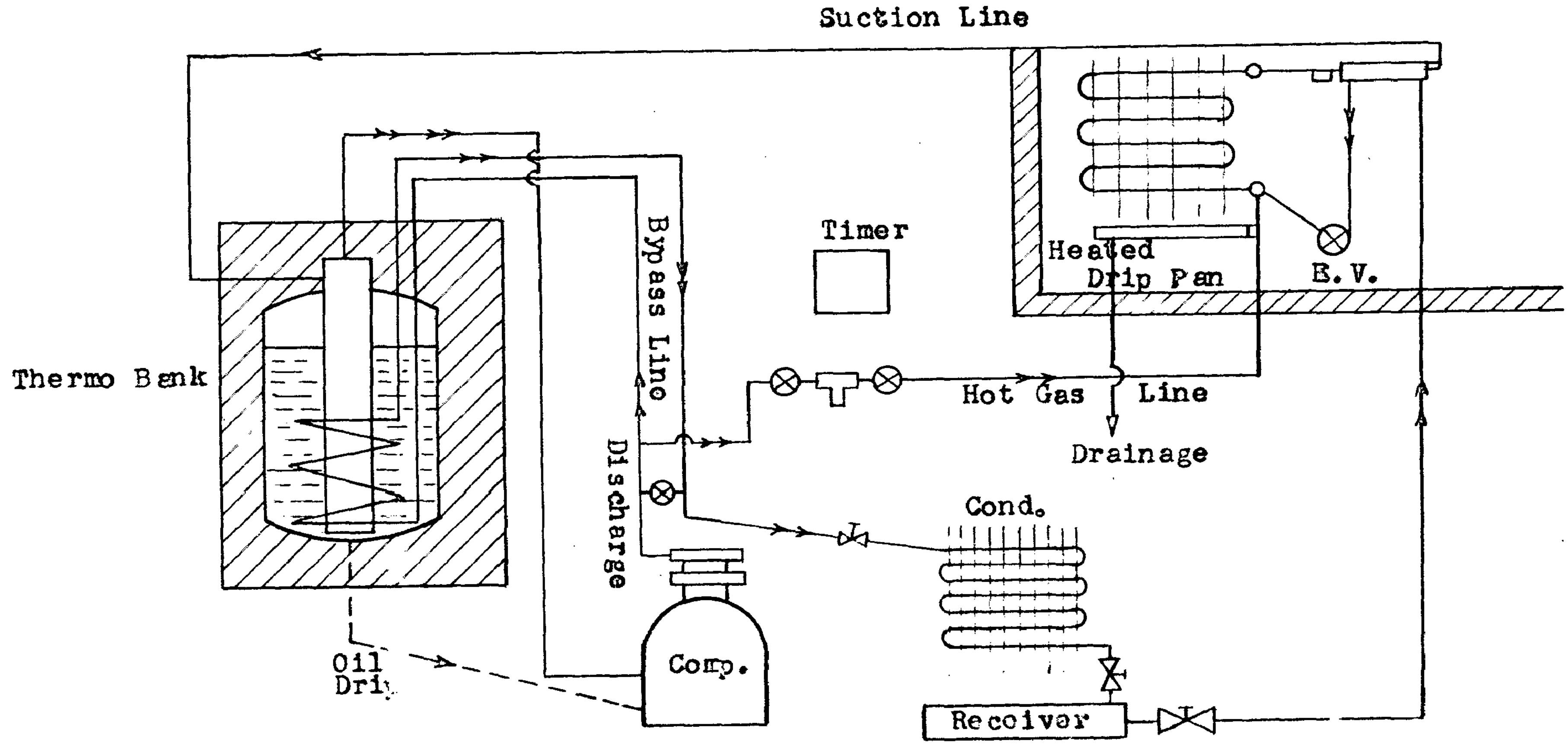
The drawbacks of this type of re-evaporator include its dependence on high ambient temperatures. Hunting may be caused by two modulating valves viz. the expansion valve and the hold back valve located in series on the same line. It will cause intermittent working of the refrigeration system and reduce its capacity. The modulating valve and re-evaporator in the suction line cause a pressure loss to the extent of $2\frac{1}{2}$ psi. and will reduce the capacity of compressor. However, with proper design this loss can be effectively reduced to values between $\frac{1}{4}$ to $\frac{1}{2}$ psi.

A variation of the above system intended to overcome above mentioned difficulties may incorporate a bypass line across the re-evaporator, refer fig. 2.74. During normal operation the bypass is open thus cutting off the re-evaporator and the hold-back valve. The pressure drop in them would be eliminated. During defrost the bypass is closed. Since the re-evaporator line is in operation during defrost only the modulating valve can be set for any pressure. Air temperatures of even $0^{\circ}F$ at the re-evaporator have produced satisfactory results.

2.73 <u>THERMO BANK SYSTEM</u>:

The thermo bank is essentially a heat exchanger and a heat reserve or heat bank at the same time. It is installed on the compressor discharge line and before the condenser. It contains a heat storing means (usually an anti-freeze solution or sometimes water even). During normal operation the warm discharge gases from the compressor heat up the heat reserving means. The heat is withdrawn from it during the defrost period. A bypass to the heating coil of the discharge line is also installed so that when the temperature in the heat bank rises the coil may be cut off from discharge line.

Fig. 2.75 shows the heat bank in details. There is an outer tank that contains the heat storing medium and in which is partially immersed an inner tank. The two are insulated to reduce any loss of heat and the outer vessel is sealed hermetically thus avoiding the necessity of replenishing the medium. During nromal refrigerating operation the compressor discharge gases enter the outer tank of heat bank through coil 1, transfer some heat to the reserve, and leave through coil 2. All suction gases pass through the inner tank entering through coil 3 and leaving by coil 4. Almost no pressure drop occurs in this coil although the temperature may rise by about 4° to 6°F. (14) During defrost period the hot gas bypass line is opened, hot gases enter the evaporator, condense there and the liquid passes through the suction line till it reaches the thermobank. It enters through coil 3 and drops down in the inner tank, picks up heat, evaporates and returns to compressor. There is a small hole to drain thcoil from inner tank to oil sump. The size of this hole should be small enough to ensure the escape of oil and at the same time it should allow no liquid refrigerant to drain. In some cases it may happen that the thermobank is not capable of supplying sufficient heat to remove the frost completly and the liquid refrigerant may slug the compressor. To overcome this difficulty, a hand operated throttling valve is installed which will be normally wide open but in case the liquid slugs the compressor it will throttle them. This, however, will



Refrigeration Cycle ouble Arrows
Defrost Cycle ingle Arrows

Fig. 2.75 THERMOBANK DEFROST SYSTEM

increase the defrosting time and will not be automatically operating.

When water is used as the heat storing means it may be allowed to freeze in the heat bank. This will allow the refrigerant to pick up not only the sensible heat of the reserve but also the latent heat of fusion. A hold-back valve will be placed before the heat bank and that will maintain a low suction pressure in the re-evaporator (such that the water in the heat bank may actually freeze). This will also hold the suction pressure below an excessive amount.

This method offers automatic defrost with the exception of hand operated solenoid valve, establishes quick defrosting, utilises the waste heat of compression through the thermo-bank, and the equipment can be installed outside the refrigerated space. The disadvantages are higher initial cost and greater care in selection and application.

2.74 GENERAL REMARKS ON HOT GAS DEFROSTING.

This method of defrosting has been used in a number of small commercial units without making any special provisions for re-evaporation of the hot gases condensed. Danger of refrigerant slugging the compressor, however, exists. As a matter of fact only a part of the refrigerant within a system is circulated during defrosting. The only part involved is that which is trapped in the evaporator and suction line at the time the by-pass valve opens. The refrigerant is more or less pressure sealed in the receiver and liquid lines. In a dry expansion system the amount of refrigerant partaking in defrost is particularly very small. This reason helps to reduce the possibility of liquid slugging the compressor .

To eliminate the liquid passing on to compressor from other causes, following points should be considered:

1. The system should be designed preferably for dry expansion so that during defrost the compressor discharge gases entering the evaporator have a minimum amount of unevaporated liquid refrigerant to push.

2. The bypass line should be installed in such a way that it drains under gravity into the evaporator and does not accumulate liquid refrigerant during normal operation, that will have to be pushed by hot gases during defrost.

3. Traps on the suction line within the refrigerated space should be avoided as liquid may condense in them which will have to be re-evaporated before it reaches the compressor.

2.75 REVERSE CYCLE:

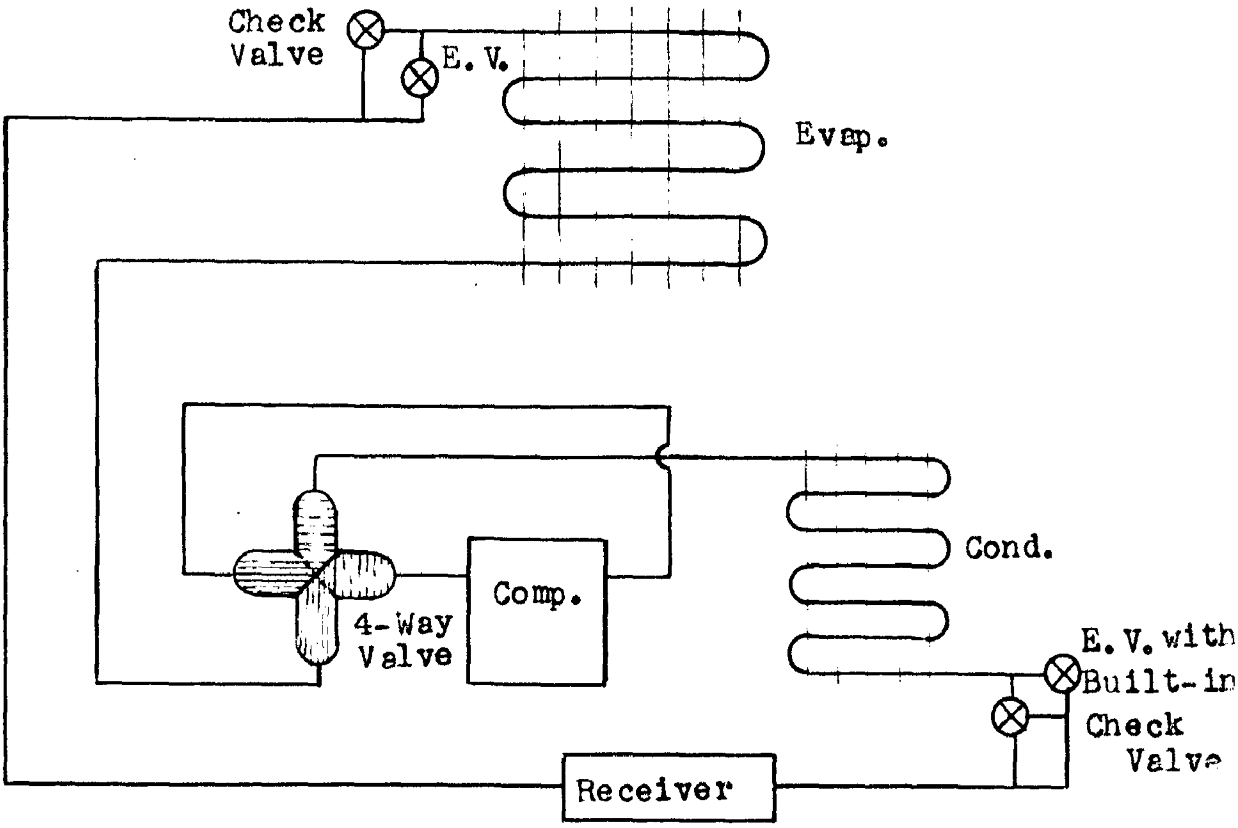
In this method, the flow of refrigerant is reversed for defrosting. During this period evaporator acts as a condenser and the frost is removed; the condenser acts as an evaporator. Although it utilises hot gases from compressor discharge to defrost, it can be classed as a system different from that.

During normal operation the refrigerant flows from the compressor through a four-way valve to the condenser and through the check-valve built in the automatic expansion valve. The check-valve allows free flow of the refrigerant in this direction. The liquid refrigerant goes on to the receiver. At the evaporator the check valve is closed and the liquid flows through the expansion valve thus setting up a lower pressure in the evaporator where it furnishes cooling. Evaporated refrigerant passes through the four way valve back to the compressor.

As frost builds up on the evaporator coils the suction pressure decreases. A reverse-acting low-pressure control is used to energise one of the solenoids in the four-way valve and will reverse the flow of refrigerant. The refrigerant vapours flow from compressor to the evaporator, which will now act as a condenser, and the vapours will condense, pass through the check valve that is open and by-pass the expansion valve. The liquid refrigerant will collect in the receiver and moves on to condenser through the automatic expansion valve (with built incheck valve keeping closed). The refrigerant evaporates quickly in the condenser, which acts as an evaporator. As the defrosting continues the pressure of the refrigerant in the evaporator rises and it actuates the reverse-acting high-pressure control that energises the other solenoid in the four-way valve and reverses the direction of flow to restore the system to normal operation.

In diagram 2.76 the two cycles are clear. The four way solenoid valve will connect as shown by dotted line during normal operation and will be in position represented by solid line. Although the check valve after condenser is shown as a separate unit in actual operation, automatic expansion valve with a ball check built into it are used.

This method provides an automatic arrangement. The evaporator is defrosted quite frequently and not much frost is allowed to deposit on the coil. It will therefore operate under favorable conditions. Reversing the cycle establishes defrosting



4-Way Valve Position

Dotted Line Normal Operation

Solid Line Defrost

Fig. 2.76 REVERSE CYCLE DEFROST

quite rapidly and the condenser heat is utilised efficaciously. In this period the load on the compressor is low.

2.8 CHEMICAL METHODS:

A deliquescent salt is placed in perforated throughs which are placed above the evaporator coil. As the air passes through the salt, it forms brine by giving up part of its humidity. This brine drips over the coils washing off the frost. In this method the refilling of the troughs is combursome and is a disadvantage. Moreover, certain quantity of the salt is also lost.

2.81 Another method involves the treatment of circulating air by certain dehumidifying agent, the vapour pressure in the air would be reduced such that its dew point would be depressed below the surface temperature of the dehumidifying agent. The operation of such a system can be described with the help of an air washer. If some dehumidifying agent like Lithium Chloride be added to the solution its vapour pressure can be lowered for the same temperature of the solution. In a Lithium Chloride water solution - 44 percent concentration, the vapour pressure at 92°F corresponds to a pressure of water at 35°F. The air leaving such a washer will have a dry bulb temperature of about 100°F and a dew point of 40°F.

It is obvious that this method will lend advantageously to applications where relatively high dry bulb temperature and a relatively low dew point is required. We know that frost problems are involved because air possessing a moderate dew point, that is above the temperature of evaporator coils, passes over them at sub-freezing temperatures. A system utilising the above principle may produce air at a dew point well below its dry bulb temperature which may itself be below 32°F. If the dew point is depressed to such an extent that it is less than the temperature of the evaporator coil, no frost will ever form and it will contribute towards the solution of the frost problem.

Figure 2.81 shows Kathabar system of U.S.A. utilising the above principle. It uses a Kathene solution for dehumidification treatment. Kathene is a trade name applied to certain Lithium Chloride solution with some other additions. The air that is to be cooled and dehumidified to a low dry bulb temperature and still lower dew point, is passed through over the cooling coils. The Kathene solution is sprayed over those coils and flows downwards under gravity to the sump. Either direct expansion in the tubes or chilled brine therein can be used to cool the air stream. The Kathene solution during dehumification of the air stream absorbs certain moisture from it. In order to maintain a particular strength of the solution, it is removed to the regenerator. However, the concentrated solution leaving the regenerator will be hot and will result in a certain load on the cooling coil. Therefore, only a small amount is delivered to the regenerator heater, which it leaves in a very concentrated state such that it may be sufficient to maintain the specific gravity of the main body of the solution. The resulting load from this source is very small. It has been claimed (23) that the equipment can be designed such that the refrigeration load from the regenerator is 150 Btu/lb. of water removed from the air. This compares well with refrigeration requirement of 144 Btu/lb. of ice formed.

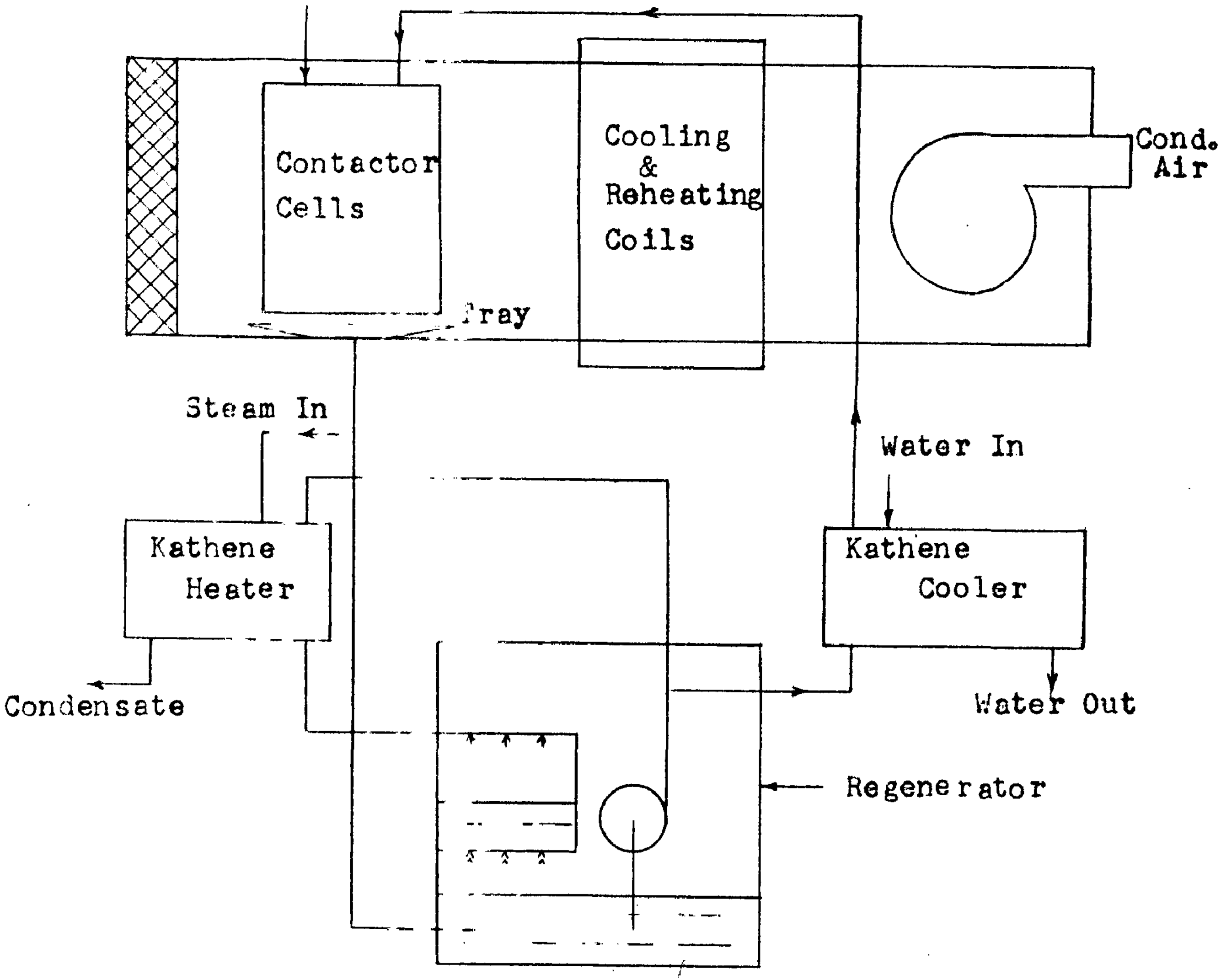


Fig. 2.81 CHEMICAL NO-FROST SYSTEM

Kathene solutions are reported to be capable of being used at very low temperatures. It has been claimed that (23) that temperatures as low as -110°F have been obtained without the freezing of the solution and air has been successfully cooled to -50° dbt dry bulb temperature and -65°F dew point.

The advantages of the above system are that the coil remains completely covered by the Kathene solution because of its viscosity and the frosting does not occur even at very low temperatures, therefore the operation is continuous regardless of the moisture load. Also air can be delivered at very low temperatures.

2.9 ELECTRIC DEFROST SYSTEMS:

The heat for defrost is obtained from electric source. In most cases the heat is applied externally although it can be applied internally too. Such systems, therefore, take a longer time for defrosting than the hot gas methods, usually 1½ times or more than that of hot gas methods.

The heating element used during defrost may be installed directly in contact with the evaporator, depending upon heat transfer by conduction; or may be located between evaporator fans and the evaporator, depending upon convective heat transfer, or a combination of the two for defrost. In any case a temperature limiting device should be used on or near the evaporator to prevent excessive temperature rise.

2.91 In figure 2.91 the heating element is built into the evaporator. The heating cable is applied in correct lengths in direct contact with the evaporator surface, the bottom of the

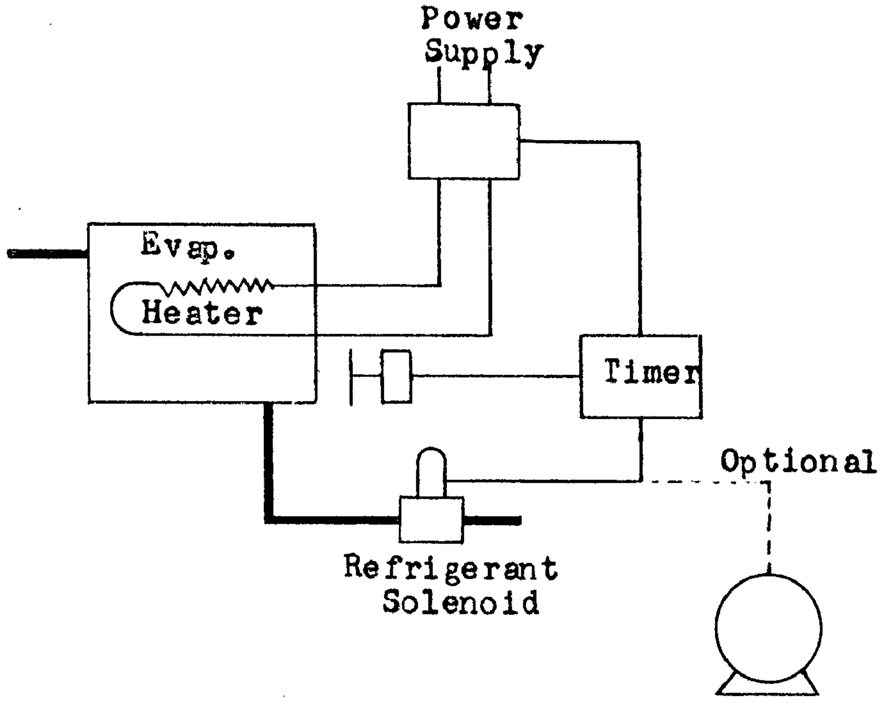


Fig. 2.91 AUTOMATIC ELECTRIC DEFROST

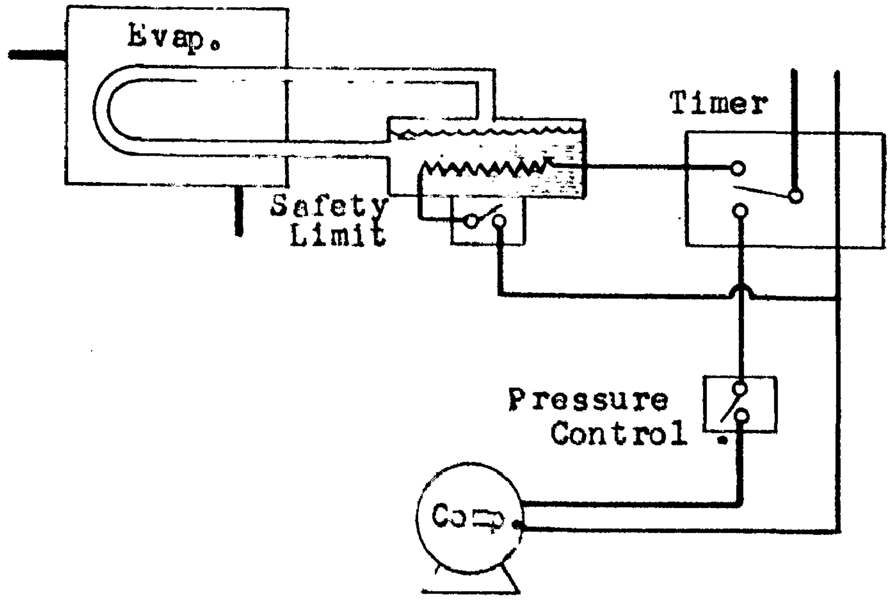


Fig. 2.93 INDIRECT ELECTRIC DEFROST

drain pan, and the drain-line from the drain pan leading to the outside of the cold room. This system is usually automatically operated by a timer switch that, when the defrost period comes, will close a solenoid valve in the refrigerant feed line, stop motor and fans, and will energise the heaters. To prevent overheating of the evaporator coils, a thermostat is usually installed within the frame-work of the unit, and that is set to cut out the heaters as soon as the temperature of the unit rises above 40^{o} F. Frequently the required heating load during defrost may exceed the current carrying capacity of a standard timer equipment and so relays may be necessary.

Figure 2.92 shows the variation of temperature with time. There are three distinct period: first the temperature increases corresponding to the period when the frost temperature is increasing to a value of 32^{o} F, and then the temperature remains constant till the frost melts, this is the second period; then again the temperature increases and this period corresponds to the dripping away of water that has been formed because of melting of the frost in the proceding period.

In case when the heaters are inserted into the evaporator coil, manufacturing and field-servicing problems may be involved. Therefore, in long evaporators, it is usually best to utilise the flowing stream of air as the heat transfer medium and the heating elements used to supply heat for defrost may be located between evaporator fans and the evaporator coil. Tests have demonstrated that since the flowing air provides a good heat exchanger vehicle, the period of defrost is about the same as that for other direct contact methods. (2).

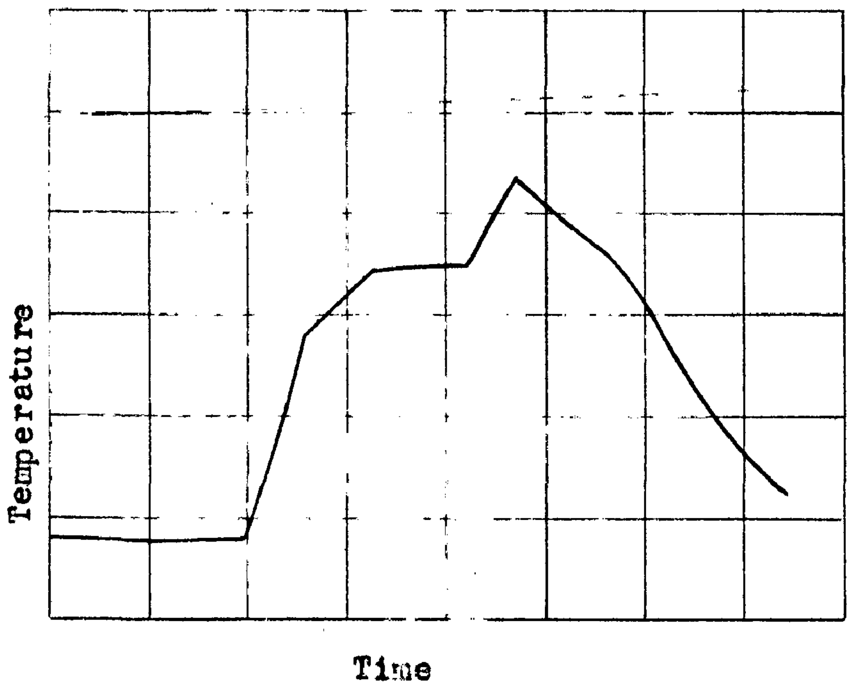


Fig. 2.92 VARIATION OF TEMPERATURE WITH TIME FOR ELECTRIC DEFROST

2.92 INDIRECT ELECTRIC DEFROST:

As shown in figure 2.93, a separate system containing the refrigerant is built into the evaporator. There is a container of refrigerant that can be heated by electrical elements during the defrost period. The hot refrigerant gas passes through the tubes in the evaporator thereby providing necessary heat to the frost. Circulation of the gas is maintained by thermosiphonic action. During defrost safety limit switches are employed and the operation is automatically controlled by timer. These controls stop the condensing unit and the blowers of the evaporator.

2.93 ELECTRIC LAMPS:

Electric lamps focussed on the coil are also used to defrost. Heating is by radiation. Infra-red lamps have been applied successfully only to belts in continuous food-freezers. The woven wire-belts used in continuous food freezing become iced in a short time. In one installation a belt 6 feet wide, travelling at a speed of approximately 1½ fpm. was completely freed of all frost by twelve 500 watts infra-red lamps placed at a distance of 3 feet from the belt and outside the freezer space. The only fault of this system lies in the high breakage of the lamps caused by clean-up gang which took over at the end of each day's operation. (15).

CHAPTER 3

ANALYSIS AND RECOMMENDATIONS

Performance from energy considerations is analysed. Different defrost methods are analysed and recommendations for handling frost problem are made.

ABBREVIATIONS

A = Area

E = Electric energy consumed

h = Enthalpy

h_s = Enthalpy of air at compressor suction

h_d = Enthalpy of air at compressor discharge

Q = Heat transfer

Q_w = As defined in Art. 3.54

Q_r = Total refrigeration load

q_f, q_a, q_s, q_r = As defined in Art. 3.2

t = Temperature

T = Time

T_o = Time of refrigeration cycle (hours)

T_d = Time of defrost (hour)

U = Overall heat transfer coefficient

v = Specific volume

W = Specific humidity of air

w = Weight rate of flow

W_c = Work of compressor per pound of refrigerant

Subscripts

re = Re-evaporator

c = Compressor

3.1 INTRODUCTION:

From the point of view of the operating economy and energy consumption the performance of an evaporator is ideal when the evaporator remains free of frost. But practical applications collect frost, the performance deteriorates, and so need defrosting. If it may be desired to keep the evaporator frost free a large number of idle periods of defrost will have to be utilised. It will require larger equipment to handle the load and may also cause spoilage of the products.

Apart from the considerations of product spoiling, it is to be noted that while selecting any particular defrost frequency, following points shall be considered:

(a) For a given refrigeration plant, what is the maximum refrigeration rate that can be obtained with different defrost frequencies?

(b) What is the minimum average energy consumption of compressor motor that will handle the refrigeration load?

It is quite important to note that each system has an optimum interval and duration of defrost. It may be just as uneconomic to defrost too often as too seldom. The variables for optimum performance should, therefore, be carefully selected.

In this chapter the performance of a refrigeration system is analysed in relation to frost deposit, and different defrost methods adopted. In reference to the later energy equations are developed for different systems and a basis for comparison of defrosting cost is expedited.

3.2 HEAT REQUIRED DURING DEFROST:

For any defrost method the quantity of heat to be supplied will be a composite term Equation 3.21 is the expression for the total quantity of heat supplied q_d.

$$q_d = q_f + q_a + q_s + q_r \qquad (3.21)$$

where q_d = total quantity of heat required during defrost.

q_f = quantity of heat necessary to melt the frost and remove it.

q_s = quantity of heat that contributes towards the rise of temperature of the coil surface and case walls to a value of above about 34° F.

q_a = quantity of heat lost to ambient air during defrost.

q_r = quantity of heat lost to refrigerant that may, in some cases, be entrapped within the evaporator coil during defrost.

Some of these terms may be inoperative in certain cases. The amount of heat required to melt and remove the frost, q_f, depends upon the temperature and quantity of the frost deposited. It is composed of heat required for sensible heating of frost to its melting point, melting it and heating a bit further for easier removal, and can be roughly taken at 200 Btu per pound of frost removed under ordinary conditions. The exact weight of the frost deposited can be measured in the laboratory. However, experience may tell us the approximate amounts deposited for particular applications.

The magnitude of heat supplied to the evaporator coils and shell i.e. q_s will be determined by the weight of the wall, its specific heat and the temperature range through which it is heated.

Heat lost to the ambient air depends upon a number of factors and may vary largely with different installations. Whereas it may be negligible in case of a few systems, it may be the largest factor in a few others. The influencing factors are: the defrost period, the temperature difference between the air and the products, the type of separation between these two, and the extent of circulation of air if any. Since their overall influence cannot be simply determined, this term is hard to be computed. However, it can be determined indirectly if necessary, by measuring q_d and finding out the terms q_f , q_s , and q_r . Then from equation 3.21 the value of q_a can be determined.

In addition to these quantities of heat, in some cases where the refrigerant remains entrapped in the evaporator coil, certain amount may also be required to heat up this portion of the liquid. This term will be absent in case of hot gas and reverse cycle defrost (Art. 2.7).

Considering the expression for the amount of heat required for defrost i.e. q_d in equation (3.21), it can be noted that whole of the heat required is not for heating up and melting the frost. On the other hand, this requirement amounts to only a part of the total. The warming up of the evaporator coil and walls may have quite a large share, and in some cases viz. in electric defrost methods air may take a large portion of heat. Under similar operating conditions, the temperature of the coil not changing much, the magnitude of q_s will not vary considerably with different quantities of frost to be removed. The same has been found to be true with the factor q_a. Therefore, with different amounts of frost accumulated the change in the heat required during defrost will be relatively small and the defrost time will remain more or less unaffected.

Equation 3.21 also shows that smaller amounts of heat will be required for medium or relatively high temperature evaporators, as they will defrost more rapidly. The amount of heat transferred to air, wherever it is significant, is proportional to the defrost period. A quicker defrost will result in smaller quantity of heat wasted in warming the air.

3.3 REFRIGERATION CYCLE:

The refrigerant pressure enthalpy diagram is shown in figure 3.31. The condenser pressure is P_d and the evaporator pressure is P_s. The cycle of operation will be sdce. The heat is picked up in the evaporator at constant pressure P_s during the process es.

The Refrigerating Effect per pound of refrigerant $= h_s - h_c$. The amount of work required by the compressor will depend upon the operating pressure range, the refrigerant and the type of compression. If the compression were adiabatic, the work required per pound of refrigerant passing through the compressor $= h_d - h_s$. For a general non-adiabatic process

$$\frac{W_c}{J} = (h_d - h_s) + Q_c$$

The liquid leaving the condenser is assumed to be saturated. When the compressor does not operate the pressure throughout the system will equalise at some intermediate pressure value P_3.

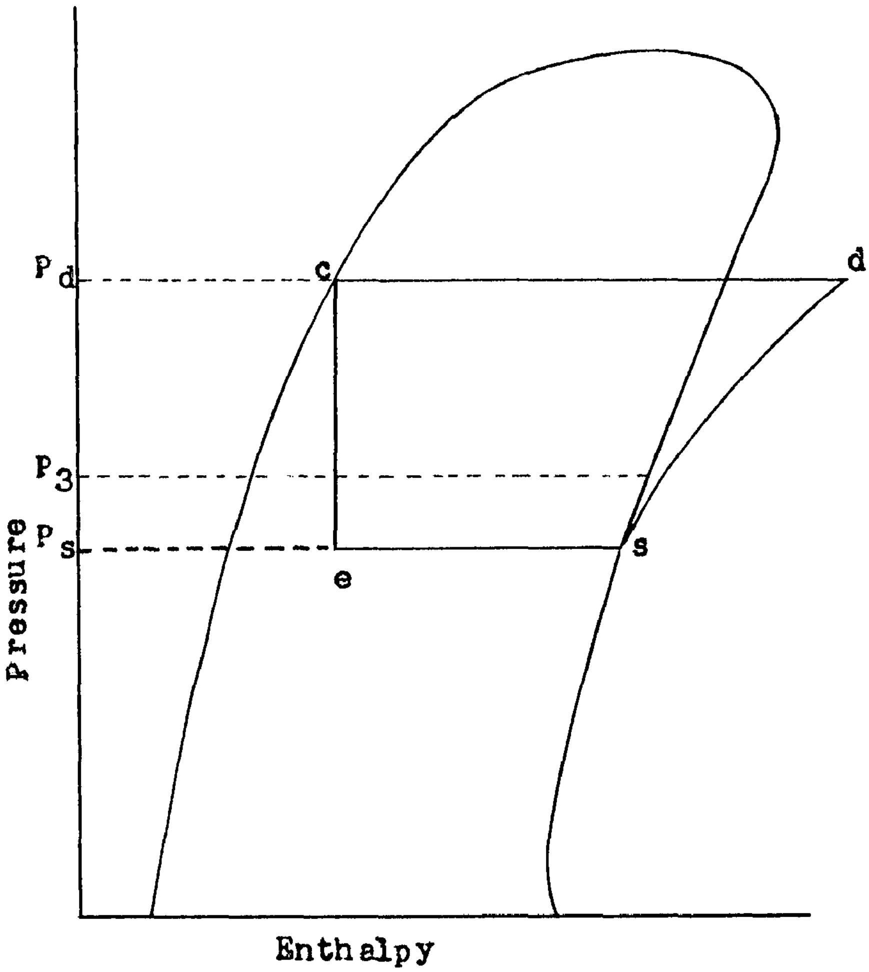


Fig. 3.31 REFRIGERATION CYCLE

3.4 ANALYSIS.

3.41 AVERAGE RATE OF REFRIGERATION:

Let us consider an arbitrary cycle of operation and defrost, for which

Average rate of cooling per hour for the cycle $= \dfrac{Q_r}{T_o + T_d}$

which can be rewritten as $$= \frac{Q_r / T_o}{1 + T_d / T_o} \qquad (3.41)$$

In this expression it can be noted that the numerator is the average rate of cooling by the refrigerator when only normal operating time is considered as affecting.

If T_o is small the amount of frost on the coil will also be small and it will not inhibit the rate of heat transfer so that the heat will be removed by the evaporator at more or less uniform rate. Immediately after defrost, some additional load is imposed up on the plant which is also to be handled by the evaporator and the useful cooling effect is reduced. This factor, however, will be important only in the starting hours of the plant and in the beginning the refrigerating effect will be small. Once this load has been offset its influence on term Q_r / T_o subsequently will be absent. In the initial stages of operation Q_r / T_o will therefore, increase slightly with increasing running time. However, when the plant has run for some time the heat will be removed at more or less a uniform rate. In the denominator occurs the term T_d / T_o ; the variation of T_d has already been discussed, it does not change much with the quantity of frost accumulated (Art. 3.2), particularly when the time of operation T_o is low, the amount of frost on the coil is small and the value of q_f is small (Equation 3.21). Therefore in the lower range of T_o when frost accumulation is less, with increasing values of T_o the denominator in the expression for average rate of cooling during the cycle decreases while the numerator tends to increase. For a given plant the average rate of cooling shall consequently be expected to increase.

As T_o increases further, the numerator becomes constant and the variation of the denominator may also be small due to small magnitude of T_d / T_o . If the stage when Q_r / T_o is constant is reached when the variation in the denominator is small, the magnitude of average rate of refrigeration will be more or less constant. However, a general trend that the average refrigerating effect shall tend to become constant, is indicated.

Now when the term T_o is large, T_d / T_o in the denominator becomes an insignificant quantity (for 25 minutes defrost once a day T_d is about 1.77 percent of T_o) and this factor will not change materially with the change of T_o. Nonetheless, with increasing time of operation the frost accumulation on the coil would have increased which will require lower refrigerant temperatures in the evaporator to meet the load (9). It will result in a lowered coefficient of performance and Q_r / E will decrease. For a system in which the compressor continuously runs and where the energy is supplied at more or less same rate by electric motor, the refrigeration rate Q_r / T_o will be reduced. Due to this factor the average rate of cooling during the cycle will decrease.

It leads to a conclusion that for a continuously running machine while defrosting too often T_o will reduce the average cooling rate between two defrosts, the same would be true if the frequency of defrost is very small. Therefore, it indicates the possibility that there exists an optimum defrosting frequency at which the average refrigerating rate is maximum.

Where the refrigerating effect is produced at the same rate by having a controlled operation, during the period when the compressor can cope with the load, the average refrigeration rate can be maintained by the plant. But it is expected to be accompanied by an increase in percent running time of the compressor, operating cost will go up. The defrost may not be important, in such a case, from the point of view of maximum rate of refrigerating effect but will be important due to an increase in cost of operation.

3.42 <u>MINIMUM POWER REQUIRED PER TON:</u>

Next let us consider a refrigeration system operating under conditions such that it is automatically controlled for the temperature of the refrigerated space by a thermostat.

$$\text{Average daily cooling load} = \left(\frac{Q_r}{T_o + T_d} \right) \times 24$$

Note that $\frac{24}{T_o + T_d}$ is the frequency of defrosting per day.

$$\text{Average electrical energy consumed per hour} = \frac{E}{T_o + T_d}$$

$$\text{Tonnage capacity of the plant} = \frac{Q_r}{(T_o + T_d) \times 12{,}000}$$

$$\text{Average energy consumed KWH/(ton) (hr.)} = \frac{E \times 12{,}000}{Q_r} \qquad (3.42)$$

and considering the operating time T_o only

Average power of the motor $= \frac{E}{T_o}$ KW

Average power of the motor KW/ton

$$= \frac{E}{Q_r} \times \left(\frac{T_o + T_d}{T_o}\right) \times 12{,}000 \qquad (3.43)$$

Analysing expression (3.43), it can be noted that it is a product of two variable terms E/Q_r and $(1 + T_d / T_o)$. With increasing values of T_o, obviously the term within parenthesis decrease while E/Q_r , as discussed earlier, decreases in the beginning while at a later stage has a tendency to increase. Whichever way these two factors vary, and their relative magnitudes govern the direction of the variation of average power required per ton.

When T_o is small, in magnitude of the same order as T_d , the value of the bracketed term decreases with increasing time of operation. E/Q_r is also decreasing during first hours and so the average power required per ton of refrigeration decreases as the time of operation increases. It is a desirable feature. As the value of T_o increases further, its effect on the expression $(1 + T_d / T_o)$ becomes more and more insignificant. However, E/Q_r passes through a period of constancy and will start increasing and its effect may be more dominant. Thus if T_o is large, average power per ton increases. This is undesirable. Summing up we find that increasing the time of normal operation i.e. reducing the defrost frequency while first reduces the average power required per ton, at a later stage increases the same. This shows a tendency that the average power requirements per ton will be minimum under certain optimum operating conditions i.e. under a proper combination of T_o and T_d.

Although it can be predicted that there exists an optimum defrost frequency on the basis of the above criterion yet the exact nature of performance near the optimum operating conditions is not known. What the magnitude of deviation from optimum will be for a variation of T_o from optimum appears to be dependent upon the operating conditions of the plant and its design, and shall be studied as a case influenced by these factors for individual plants.

The significance of term average power of the motor need not be over emphasized and it shall be noted that a motor of just the capacity found from equation 3.43, may not be enough to handle the load under more stringent conditions of frosting at a later stage even if continuously operating, and a larger motor may be required to meet the demand.

3.5 ENERGY EQUATIONS FOR VARIOUS DEFROST METHODS:

We will next consider and develop the energy equations that will apply during the defrost period for different methods. Spray systems and chemical defrost methods will not be covered as they do not involve independent defrost periods.

3.51 TIME DEFROST (OFF PERIOD):

The compressor is stopped and the heat from surroundings is allowed to move into the refrigerator and melt the frost. The total amount of heat required during this period will contribute towards increasing the temperature of the frost to 32°F, to melt it, to increase the temperature of the evaporator shell and the refrigerant contained therein. Different types of heats that will be involved during defrost (Art. 3.2, equation 3.21) can be determined individually in the following way.

Sensible heat required to raise the frost temperature to 32°F

$$q_1 = w_f \times c_f \times (32 - t_f)$$

Latent heat required to melt the frost $q_2 = w_f \times h_{fs}$

heat required to heat the water $q_3 = w_f \times (t_d - 32)$

Therefore $$q_f = q_1 + q_2 + q_3 \qquad (3.51)$$

Heat required to raise the temperature of the evaporator shell

$$= q_s = w_e (t_d - t_e)\, c_e \qquad (3.52)$$

In order to find out the heat required to raise the temperature of refrigerant in the evaporator when defrost comes, it is to be noted that the refrigerant undergoes a constant volume non-flow process. When the compressor is stopped first the pressure equalises throughout the system at P_1 and then the heating occurs at constant volume and the pressure will increase to some value P_2. This process is represented on pressure-enthalpy chart by process line 1-2 in fig. 3.51. It is represented superimposed on the P-h chart for conventional refrigeration cycle sdce of fig. 3.31.

Heat picked up by the refrigerant = qr =

$$w_r (h_2 - h_1) - (w_r \times v/J)(P_2 - P_1) \qquad (3.53)$$

Total quantity of heat required during defrost =

$$q_f + q_r + q_s + q_s$$

Heat transferred through the insulation = Q_d =

$$A\, U \times (t_{out} - t_{in}) \qquad (3.54)$$

Since this heat will be supplied to above four items

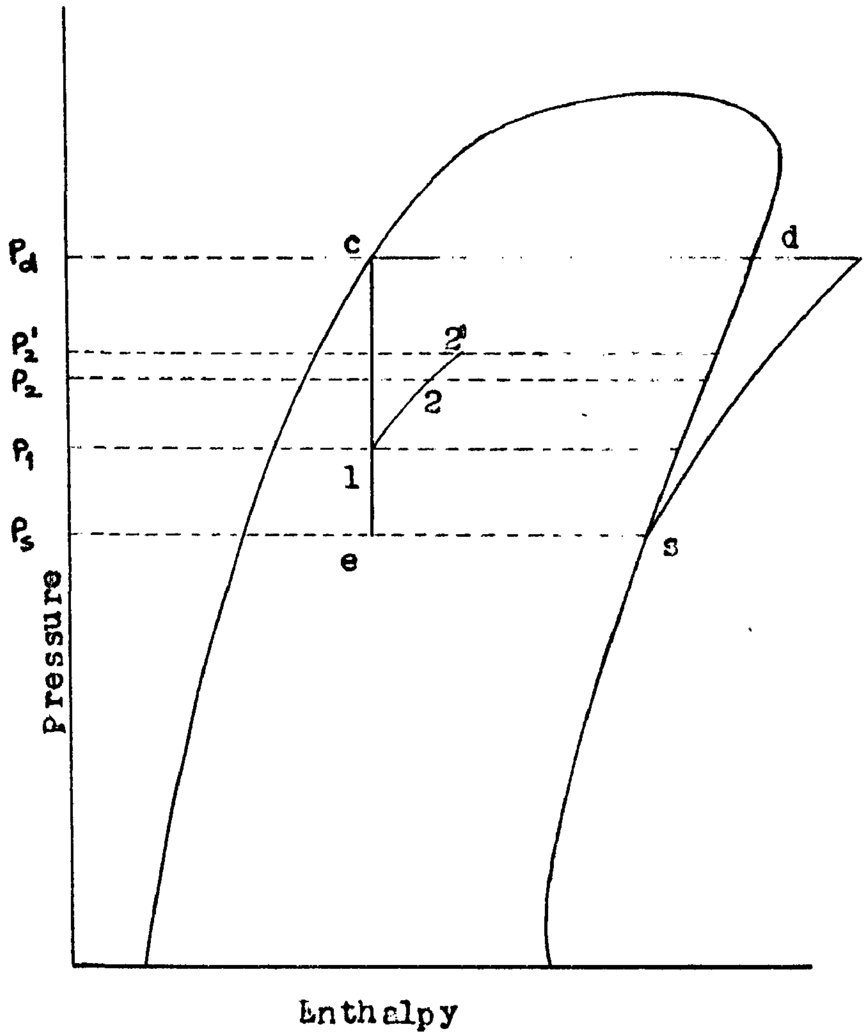


Fig. 3.51: P-h CHART FOR OFF CYCLE DEFROST

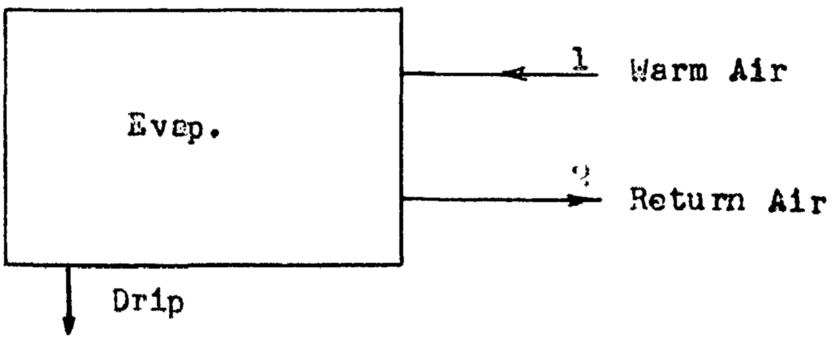


Fig. 3.52: BLOCK DIAGRAM FOR WARM AIR DEFROST

$$q_d = q_f + q_r + q_s + q_a \qquad (3.21)$$

In case of domestic refrigerators q_a may be small enough to be neglected. The equation (3.54) clearly indicates that the operation will have to be interrupted for smaller defrost periods as t_{out} exceeds t_{in}. If the overall conductance is more this period will again be less but this will entail greater losses during refrigeration period and is undesirable.

3.52 WATER DEFROST:

Water is sprayed over the evaporator surface. The frost is melted and carried away with water, the evaporator surface will be heated up and so would be the refrigerant contained therein. The final temperature will depend upon the rate of water circulation and its temperature, and the duration of defrost.

Heat required to melt the frost can be found by the equation 3.51 in which t_d is to be considered from actual conditions. The quantity q_s can be determined by equation (3.52) in which the final temperature of the evaporator shell will have to be considered. Heat supplied to the refrigerant will again be during a constant volume process from state 1 to another state 2' compared to point 2 on fig. 3.51. The pressure at 2' will be determined by the temperature of water used for defrost. This having been estimated, heat required towards this item can be found from equation 3.53. Since the defrost is very rapid and also not much of heat transfer is expected to the air in-side the refrigerated room, q_a can be safely assumed negligible. Whole of this heat required for defrost is being transferred from the circulating water and if it is q_{in} then for a circulation of water for T minutes

$$q_{in} = gpm\ 8.33\ (t_{entering} - t_{leaving}) \times T\ (min.) \qquad (3.55)$$

The energy equation, from equation (3.21), will be

$$q_{in} = q_f + q_r + q_s$$

In this case it is to be observed that the heat carrier water also acts to remove the frost, to carry it away by melting it and then entraining it alongwith itself. Since the temperature of water is to be well above 32^{o} F, about 45^{o} F or above (Art. 2.4), the temperatures of the evaporator shell and of refrigerant will be higher in this as compared to the time defrost.

The expression (3.55) clearly brings out the fact that greater is the rate of water circulation, the shorter will be the period for defrost and a higher temperature of circulating water will produce a similar effect.

3.63 WARM AIR DEFROST:

Warm air is blown over the evaporator coils. The frost shall be heated and melted by picking up the heat from the air. The air that is blown over the evaporator is generally the outside air which will be usually existing at a dew point higher than the temperature of the frost. As it comes in contact with frost, in addition to the sensible cooling of air humidity in air will precipitate and latent heat will be evolved. This latent heat will be supplied to frost to heat and melt it. Therefore, in order to account for the heat transferred from the air, not only the sensible heat but also the latent heat associated with the precipitation of humidity shall be considered. The mass of water vapours in the leaving air will not be the same as in the entering air. The water vapour precipitated will leave with the drip from the evaporator.

For this case the energy equation can be written by considering block diagram 3.52. At section 1 the warm air is entering the evaporator and after warming the frost it is leaving at section 2.

$$(cfm/v) \times (h_1 - h_2) \times T(min.) = q_f + q_s + q_r + q_w$$

where q_w = heat in precipitated water. The first three terms have been already derived and their values can be determined from equations 3.51, 3.52 and 3.53. However for q_r, in considering the process 1-2 on fig. 3.51 the final pressure in the evaporator will be approximately corresponding to the dew point of the warm air. These pressures are well within the working limits of the evaporator coil. Only the term heat in precipitated water is new and this will be given by

$$q_w = (W_1 - W_2)\ (cfm/v) \times h\ T(min.)$$

All other factors remaining the same, the time period will decrease with increasing rates of air circulation or when the value of h is high. Ofcourse this depends upon the dry bulb temperature and humidity of air and will be smaller as the temperature or humidity will increase. The term q will be small and can be neglected without any effect.

3.54 HOT GAS DEFROST:

A bypass line is added between compressor and evaporator short circuiting the condenser and expansion valve. When the bypass valve is opened the gases from the compressor, which are undoubtedly hot enter the evaporator. The head pressure will drop down within a minute or so and within a few minutes the head and suction pressures will stabilize at some intermediate values. The values of stabilized pressures will depend upon the amount of resistance in the bypass line valves. It is for this restriction that a pressure difference exists. The magnitudes of these pressures will determine the rate of defrost.

The pressure-enthalpy diagram for this method compared to the ordinary refrigeration cycle is given in fig. 3.53. The cycle adcc is the normal cycle, while 1234 is that during defrost. Process 34 represents heat losses in bypass line. The head pressure P_3 is less than P_d while P_2 is greater than P_a. The head and suction pressures during defrost i.e. P_3 and P_2 respectively should be such that the saturation temperature corresponding to the head pressure should be greater than $32^{o}F$ and temperature corresponding to suction pressure P_2 must be less than the temperature of the heat source. The cycle 1234 assumes that the refrigerant entering the evaporator during defrost is completely condensed. In some cases only partial condensation will occur and the cycle of operation will be something like 1' 234.

Heat picked up by the refrigerant during re-evaporation

$$Q_{RO} = w_c (h_2 - h_1)$$

Compressor work = $w_c \times W_c / J$

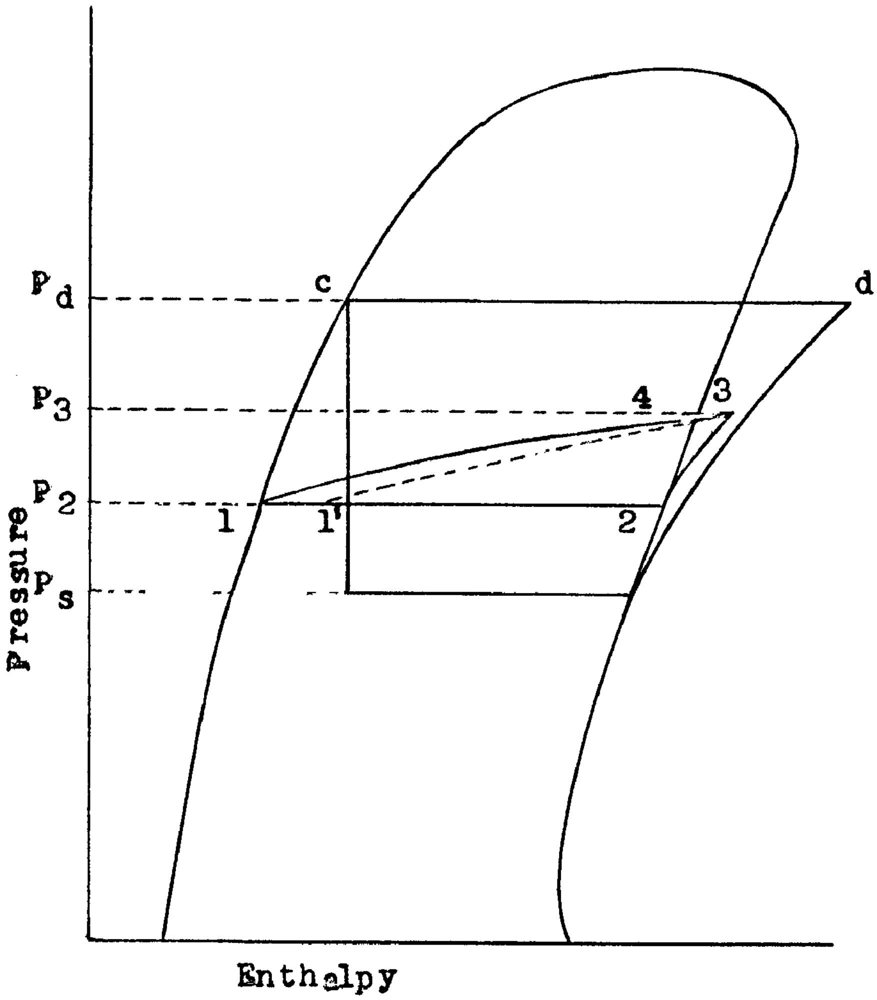


Fig. 3.53 P-h CHART FOR HOT GAS METHOD

Heat loss in the compressor = $w_c \times q_c$

Heat in the evaporator required for defrost $Q_d = w_c(h_4 - h_1)$ With the help of the procedure laid down earlier q_f and q_s can be found, the term heat required by refrigerant does not have any significance in this case. Since the defrost is quite rapid and the circulating fans, if any, are off during this period, the quantity q_a can be neglected. The equation 3.21 will reduce to

$$Q_d = q_f + q_s$$

The energy equation for this method may be written as follows

Heat added in the re-evaporator + Work supplied to the compressor = Heat rejected in the evaporator during defrost + Heat loss in the pipe line + Heat loss in the compressor.

Substituting for each term on the above expression we can get

$$(h_2 - h_1) + W_c/J = (h_3 - h_4) + (h_4 - h_1) + q_c \qquad (3.56)$$

whereas for the compressor

$$W_c/J = (h_3 - h_2) + Q_c$$

from which we can find the heat equivalent of net energy transferred to the compressor Q_w

$$Q_w = w_c\,(W_c/J - Q_c)$$

The bypass line losses are small and may be left out. Neglecting these the equation 3.56 can be rewritten as

$$Q_d = w_c\,(h_4 - h_1) = w_c\,(h_2 - h_1) + (W_c/J - Q_c)\,w_c$$

$$= Q_{re} + Q_w$$

Considering the term Q_{re}, we know that this is the amount of heat transferred across the walls of the re-evaporator.

$$Q_{re} = A_{re}\,U_{re}\,(t_o - t_1)\,T = w_c\,(h_2 - h_1) \qquad (3.57)$$

and

$$Q_d = Q_w + A_{re}\,U_{re}\,(t_o - t_1)\,T$$

yielding

$$T = \frac{Q_d - Q_w}{A_{re}\, U_{re}\, (t_o - t_1)} \tag{3.58}$$

An examination of the expression for T reveals that if the heat demand for defrost Q_d is fixed, the following points shall be fulfilled for minimum defrost period.

1. Heat source temperature t_o shall be substantially higher than the temperature of the re-evaporator t_1 .

2. The area of the re-evaporator shall be as large as may be partically feasible although this will mean an unnecessarily long suction line during normal refrigeration operation. A bypass line has been suggested as a possible solution. (Art. 2.72, fig. 2.74).

3. The heat of compression Q_c is increased by raising the compression ratio P_3 / P_2 between head and suction pressure during defrost.

4. Bypass losses are undesirable as they will tend to increase the time of defrost. (equations 3.56 and 3.58).

Considering each of the above points separately, the temperature of heat source when it is the ambient cannot be varied at will and shall change according to natural, seasonal and daily variations. An artificial source can be added which shall be capable of supplying the heat that may be required of it, during defrost periods.

To increase the ratio P_3 / P_2 , an orifice can be included in the line to the evaporator. The choice of the diameter of the orifice is quite important and critical. If it is too large it will build up no pressure difference, on the other hand if it is too small a throttling action will occur and the compressor discharge may be cooled to a temperature below 32°F. No defrosting of the evaporator will be possible.

3.55 REVERSE CYCLE DEFROST:

With the help of a four way valve or a combination of valves, the cycle of performance is reversed. The pressure-enthalpy diagram for the reverse cycle and the corresponding normal refrigeration cycle both are given in figure 3.54. The conventional refrigeration cycle is sdce. At the beginning of the reverse cycle the head pressure will drop and during the reverse operation, the stabilized cycle will look like 1234 with a lower head pressure. The head and suction pressures will depend upon the conditions during reverse operation. Process 12 of re-evaporation occurs in the usual condenser, 23 is compression, at 3 hot vapour enters the evaporator, reject heat to the walls and through it to the frost which will melt, loosen and drift away. Process 41 is throttling.

Energy analysis can be done on lines identical to the previous discussion.

$$Q_d = q_f + q_s$$

Neglecting all losses the energy balance for the system can be written as

$$w_c\ (h_3 - h_4) + w_c \times Q_c = w_c\ (h_2 - h_1) + w_c\ W_c\ /J \quad (3.59)$$

and considering the performance of the compressor alone

$$w_c\ (W_c\ /J - Q_c) = w_c\ (h_3 - h_2)$$

$$\text{or} \qquad w_c\ (h_3 - h_4) = w_c\ (h_2 - h_1) + w_c\ (h_3 - h_2) = Q_d$$

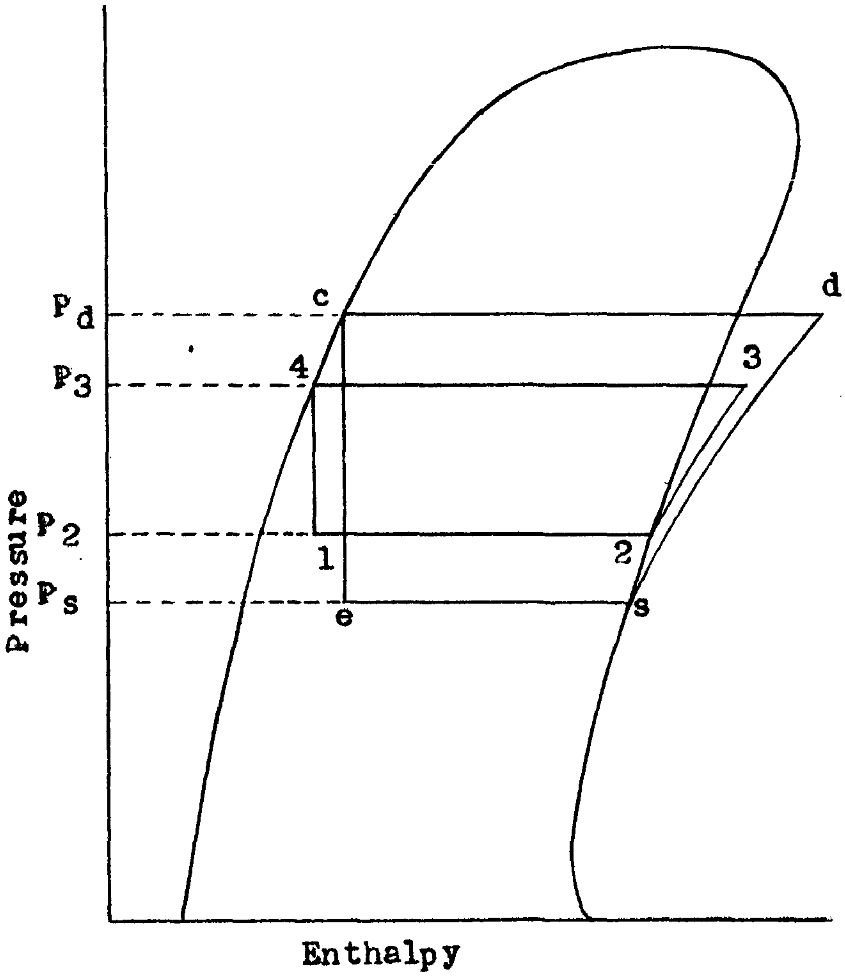


Fig. 3.54 P-h CHART FOR REVERSE CYCLE DEFROST

3.56 ELECTRIC DEFROST:

The electric motor is stopped and the heating is done electrically for sufficient time to allow the removal of frost. The analysis is exactly identical to that used for the off system with the difference that in this case heat is supplied in a different way and at a more rapid rate. Like that it is a non-flow constant volume process for the refrigerant but the final temperature attained at state 2 in fig. 3.51 will be more in this case as compared to the off system. The value q_r will, therefore, change. The energy equation can be written as

$$Q_d = q_f + q_s + q_a + q_r$$

where Q_d is electric energy in Btu to heaters.
Obviously the time period will be small if the wattage of the heaters is increased.

In case with electric heating, the air may take its share of heat as supplied from heaters, more so when the fans are not shut off during defrost and air is used as the heat carrier. The heat taken by air cannot be neglected for the later case, rather may be an important factor far outweighing the other items. A thorough treatment can be made with the help of subject matter presented here, and in Art. 3.2 and 3.51.

3.6 ECONOMICS OF DEFROST:

For each of the method described for defrosting certain expenditure is to be incurred with the exception of time defrost (Art. 3.2). The operating cost for water system will be for the amount of water consumed, for warm air defrost method will be for the power required to run the blower circulating air and if employed, for the electric heating of air. During hot gas, reverse cycle or electric defrost certain amount of electrical energy is consumed and this will constitute the operating cost.

In order to compare the expenditure incurred in different methods, we will consider the amount of heat available to remove the frost for the same investment. Let us define an arbitrary term "Operating Cost Index", which will be mathematically expressed as a ratio

$$\text{Operating Cost Index} = \frac{\text{Heat required to remove the frost}}{\text{Cost of defrost phenomenon}}$$

Obviously the greater will be this quantity the less will be the operating cost.

All these expenditures will depend upon the local rates of energy (methods 2.6, 2.7 and 2.9) and of water (method 2.4). It may be noted that the expenditures for water and warm air systems where ambient air is available at fairly high temperature for defrost, will be very small. However, with air systems that involve preheating, the expenditure will be increased as the heat for defrost is supplied by electric heaters; in addition the circulating air which acts as heat carrier needs large quantities of energy for its heating.

In electric defrost the energy cost will be more than that in case of hot gas or reverse cycle methods. This will be so because the total heat required in later two cases is only partially met by the electrical energy fed to the motor, while in electric defrosts it is to be wholly supplied in the form of electrical energy.

However, it must be pointed out here that the expenditure during defrost is generally a very small fraction of the expenditure during normal operation and need not be over emphasized. The selection of a particular method is governed by the first and installation costs of the different methods and other practical considerations.

3.7 <u>RECOMMENDATIONS FOR DESIGN</u>:

A good design of a refrigeration system must take into consideration the frost factor. The frost problem is more pronounced in smaller units and display cases where respectively the space occupied by evaporator is quite a large proportion of the refrigerator space or where air circulation is utilised to distribute the refrigerating effect properly. The importance of frost factor for display cases is so great that the design of every part of the cabinet is influenced largely by the considerations of amounts of frost deposit. In larger plants such as cold storages there is a varying opinion and feeling towards defrosting. The storage owners often request a plant that may not need defrosting but once in a storage season. These two classes offer certain basically different frost problems. Special care must be exercised in case of more demanding applications as display cases.

3.71 In any refrigeration system, following four points must be borne in mind. These factors are very important for display case design.

1. As far as possible, moist air shall be kept out of the refrigerated space or cabinet case.

2. The location of frost deposits and their amounts shall be

controlled.

3. Frost should be eliminated from objectionable places.

4. Defrosting shall be properly done to completion.

Obviously if the first factor could be controlled fully the other points will become unnecessary. However, in actual places they may often play an important role and need be considered.

The problem becomes more complex due to a number of unpredictable variables that influence the performance. In open top low temperature display cases the cooling is provided by means which are more or less exposed to existing seasonal humidity and would be affected by geographic location. A great loss to the refrigeration effect occurs owing to the sensible and latent cooling required to produce frost. Smaller shall be the top opening so that less fresh air may enter the case. This will however, result in reduced display area. A possible solution will be to provide slant front glass which will provide greater display area yet a small opening. It is used widely.

Air handling is quite important a factor and generally yields results to cut and try methods. Air must be evenly distributed along the full length of the case avoiding zones of high and low velocities. Distribution devices may include a high pressure plenum ahead of discharge grille and baffles ducting to the discharge. A more reliable and elaborate way is to use a number of fans spaced over the length of the case and blowing against a spreader baffle.

After the distribution of the proper amount of air transgressing the opening has been obtained, it is important to ensure a smooth flow of air across the top layers. It can be achieved by holding the upper air as a blanket while allowing to diffuse the lower portion of the air stream downwards onto the product. Air flowing in this manner will prevent much of air filtration because of the absence of the turbulence and in addition on the same account will keep the upper regions of the case free from frost. Splitters and baffles or a perforated baffle may be lowered from the top to allow the lower stream to diffuse downwards while the upper strata flows along.

The design of the coil is an important factor and will be discussed in section 3.72. The location of the coil is important. A coil shall be located either inside the refrigerated space or very near to the load. A minimum possible ducting shall be used. In display cases, it shall be located as close to the discharge grille as possible. In some cases the ducts may carry the cold air from the bottom of the case to the discharge at the top and it may involve a rise in temperature of cold air to the tune of 10°F. It is a serious drawback since it is a loss in a region of temperature which is most important. Care shall also be taken that if return ducts are used they shall not remove too much of moisture and get chocked.

The defrosting must be done at regular intervals and a defrost ought to be complete and rapid. The water of defrost must be removed from the cold areas as soon as possible so that no water may freeze again to form ice as soon as the refrigerator is restarted. It may be pointed out again that it is harder to remove solid ice than frost. For smooth removal of this water a gutter under a unit cooler or a drip tray under coils is used. Care must be taken in each case to ensure that the drain is sealed during regular operation to prevent warm air from penetrating into the room through it.

3.72 EVAPORATOR DESIGN:

As pointed out in section 3.71 evaporator is quite important to be considered since it is the coldest component in refrigeration system and is the logical place to collect almost all of the frost. The function of the coil is to provide adequate refrigerating effect and that it shall be capable of holding the products in the cold space at a proper temperature for long periods of time and even under adverse conditions of operation. The problem of frost is more important where air is pushed through the coils which are finned because the choking of air passage offers acute problem.

In one method extra evaporator surface may be allowed to balance for the loss in effectiveness of the system due to the frost accumulated on the coils. This requires large evaporator area. Another method consists of using two coils separated and arranged in such a way that one or both can be used in a recirculating manner. In this design a certain percentage of air is passed through the coil a second time to dehydrate and cool it further. The major disadvantage of this system is the large surface area that will be involved. An earlier method, more suited to freezer units or display cases, utilises proper fin-spacing-staging. It may also use one or more smaller coils of progressively closer fin spacing. Fins spaced wider apart are located at the entrance end and mainly dehydrate the return air. The wider spacing allow for greater frost collection without coil blockage. Progressively closer fins are used in the following stages of the evaporator coil and they sensibly cool the air. It has the advantage of incorporating more effective use of fins and the coil blockage will occur at larger intervals thereby reducing the number of defrosts required daily. The optimum combination for fin-space-staging will depend upon a number of unpredictable factors and will vary under different operating conditions.

The system shall neither be defrosted too soon nor too late. Determination of the optimum defrost frequency will have to be done on the field as the unpredictable and changing conditions of the field cannot be duplicated in laboratory testing. Defrosting must be done when the temperature of the refrigerated space tends to rise beyond the desired value and the system is not capable of handling the load even with a continuously operating compressor. Defrosting a bit too often is always on the safer side but it must not be too frequent to affect the performance adversely rather than contributing to it.

CHAPTER 4

HEAT TRANSFER UNDER FROSTING CONDITIONS

Heat transfer under frosting conditions is studied with particular reference to cylindrical surface.

ABBREVIATIONS

A = Area

C_p = Specific heat at constant pressure

D = Diameter of frosted cylinder

D_o = Diameter of bare cylinder

h = Film heat transfer coefficient

h_m = Mass transfer coefficient

k = Thermal conductivity

r = Radius

t = Temperature

T = Time

L = Latent heat of sublimation

U = Velocity

W = Specific humidity

W_a = Weight of air

i = Enthalpy of air

h_i = As defined by Art. 4.22

δ = Mass diffusivity

μ = Coefficient of viscosity

ρ = Density of air

Nu = Nusselt Number

Sh = Sherwood Number

Pr = Prandtl Number

Sc = Schmidt Number

Re = Reynolds Number

<u>Subscripts</u>

s = Surface

o = Free stream

f = Frost

x = means x direction

4.1 INTRODUCTION:

The phenomenon of heat and mass transfer from air in contact with a metal surface at a temperature below the freezing point is very complex. The local heat transfer and overall heat transfer were studied and it was noted that a quazi-steady state was reached after the initial transient period (25, 26, 27). During the transient period, the problem of analysis is very much complicated since the heat and mass transfer coefficients are not uniform over the whole surface and will undergo different modes of variation of conditions with time. An attempt has been made to consider this problem under certain simplifications. During the quaze-steady state the problem is one of solving boundary layer equations for heat and mass transfer, and the basic equations for heat transfer and diffusion, the solution for a cylindrical surface in cross flow of air has been discussed by Messrs Chung & Algeren (26),

4.2 HEAT TRANSFER FROM METAL UNDER FROSTING CONDITIONS:

The air in contact with a surface at sub freezing temperature is cooled by convective heat transfer across the air film at the frost air interface. The total heat transfer flux will then be given by the equation

$$\frac{dQ}{dA} = h(t_o - t) + h_m \rho (W_o - W) L \qquad (4.21)$$

The heat and mass transfers both contribute towards the total amount of heat transferred. Therefore, it will be influenced by temperature and partial pressure differences both. The results under frosting conditions may be most conveniently compared by using the coefficient h_i based upon enthalpy differences as the driving forces

$$h_i = \frac{(dQ/dA) \text{ at } x=0}{i_0 - i_s} \tag{4.22}$$

where the enthalpy i is given by

$$i = C_p t + L \times W$$

If the Lewis ration $\dfrac{h}{h_m \times C_p \times \rho}$ be assumed at unity we can write (See appendix 1)

$$\frac{dQ}{dA} = h_i (i_0 - i_s) = W_0 \frac{di}{dA} \tag{4.23}$$

The temperature at frost air interface t will be varying from time to time because of continuous build up of the frost on the metal surface. Due to this reason equation 4.23 cannot be integrated by ordinary methods.

If it is assumed that the heat delivered to the frost air interface passes through the frost layer and that the heat stored in the frost is small, then

$$h_i (i_0 - i_s) = k_f \frac{t - t_s}{x} \tag{4.24}$$

which on substitution for $i = C_p t + L \times W$, can be written as

$$t = \frac{h_i i_0 + \left(\frac{k_f}{x}\right) t_s - L h_i W}{k_f/x + C_p h_i} \tag{4.25}$$

Further, if it is assumed that the frost air interface is saturated and if the value of k_f is known, equation 4.24 can be solved for any value of frost thickness x. Once the value of t has been obtained the value of i can be found and the equation 4.23 can be step-wise integrated over the area.

It may be worthwhile to consider the variation of the total heat transfer rate with time. For such cases a relationship should be found for the variation of x with time T. Assuming that the frost forms uniformly over the surface and keeps cylindrical shape, the variation of x with T can be given by

$$x = h_m \rho \int_0^T \frac{W_0 - W}{\rho_f} dT \qquad (4.26)$$

Once the value of x has been determined t can be found for different T since the density of frost keeps on varying with changing conditions of frost deposit and since W also is dependent upon frost thickness, integration of equation 4.26 becomes virtually impossible unless a number of unwarranted assumptions are made. The average thickness $\bar{x}$ of frost accumulated over the whole surface A, may be expressed as

$$\bar{x} = \int_0^T \int_0^A \frac{h_m \rho}{\rho_f} (W_0 - W)\, dA\, dT \qquad (4.27)$$

This equation will again present similar integration difficulties and becomes further complicated because ρ is also a function of position along the length of the tube; obviously no general solution can be obtained from equations (4.26) and (4.27), yet the heat flux can be readily determined for two unique conditions representing the limits of operation. One corresponds to the beginning of operation when no frost exists and $t = t_s$. In the second case, if the dew point of air is above 32°F, a limit would be reached when t attains a value of 32°F. In each of the above situations i can be determined, i_a is constant, and the equation 4.23 can be integrated.

From equation 4.24 and for the case when t equals 32°F a relation 4.28 can be developed

$$\frac{x}{k_f} = \frac{32 - t_s}{h_i (i_o - 12.3)} \qquad (4.28)$$

since the value of i at 32°F = 12.3 Btu per pound of dry air. Note that x, the frost thickness, depends upon the value of k_f. When the frost surface attains a temperature of 32°F the water vapour will continue to condense and will be soaked by the porosity of frost and will freeze gradually. The value of thermal conductivity shall increase during this period, approaching the value of solid ice. As k_f will increase, so will increase x (equation 4.28) until k_f becomes maximum. Thus the frost thickness may increase even though the surface temperature is 32°F. When maximum x is reached dehumidification will still continue with liquid water draining off. An interesting situation will arise when the air will possess a dew point below 32°F. Dehumidification will first occur as usual but will stop as the frost surface temperature reaches the dew point. There will be no latent heat load then and heat transfer will be by sensible cooling alone to be given by expression (4.29) where t_d is dew point of air.

$$\frac{dQ}{dA} = h\,(t_o - t_d) \qquad (4.29)$$

The results obtained by theory did not very much agree with experimental results (25). The authors assigned the deviation to the assumption of Lewis number as unity. The data showed certain agreement with calculations based on the Colburn Analogy for calculating mass transfer coefficients.

4.3 VARIATION OF HEAT TRANSFER WITH TIME (CYLINDRICAL SURFACE):

A different method can be utilised to analyse the heat transfer without using Lewis Number. Two assumptions are made that the frost forms uniformly over the surface and maintains its cylindrical shape, and the thermal conductivity of the frost does not vary with time. Although both of the assumptions are not exact but they are valid to the extent as discussed by Messrs Chung and Algeren (26).

Let us consider the conditions after some frost has deposited, the temperature of air frost interface is t and the outer diameter of frosted cylinder is D. Neglecting the heat storage within the frost layer and the superheat of the vapour that is transferred, the rate of heat transfer Q can be given by equation (4.31)

$$Q = \frac{2\pi k_f}{\ln D/D_o} = h\pi D\,(t_o - t) + \left\{\pi D\, h_m\, \rho\,(W_o - W)\right\} L \qquad (4.31)$$

which can be rewritten as (4.32) by introducing dimensionless numbers

$$\frac{2K_f}{K}\frac{t-t_s}{\ln D/D_o} = \dot{Nu}(t_o-t)+Sh\left(\frac{Pr}{Sc}\right)\frac{L}{C_p}(W_o-W) \qquad (4.32)$$

By heat and mass transfer analogy

$$Nu \propto C_1 \times Sh\ )$$

while $$Sh = C_2\left(\frac{U_o D}{\mu\rho_f}\right)^n = C_2\, Re^n\,(D/D_o)^n$$

or $$Nu = C_3\, Re^n\,(D/D_o)^n$$

where $C_3 = C_1 \times C_2$

Writing $W_o - W = (W_o - W_s) - (W - W_s)$

and $t_o - t = (t_o - t_s) - (t - t_s)$

$t - t_s$ is generally small and can be assumed protional to $W-W_s$, and if the constant of proportionality is b

$$t - t_s = b\,(W - W_s)$$

The constant b depends upon the range of temperature. With those substitutions equation 4.32 will reduce to

$$\frac{W-W_s}{W_o-W_s} = \frac{C_3\dfrac{t_o-t_s}{W_o-W_s} + C_2\dfrac{Pr}{Sc}\times\dfrac{L}{C_p}}{2\dfrac{K_f}{K}\dfrac{b\,Re^{-n}}{(D/D_o)^n \ln D/D_o} + C_2\dfrac{Pr}{Sc}\times\dfrac{L}{C_p} + C_3 b} \qquad (4.33)$$

This equation can give the value of $W - W_o/W_o - W_s$ as a function of D/D_o for known values of $(t_o - t_s)\ /\ (W_o - W_s)$ i.e. for known states of free air stream and the surface conditions of bare cylinder.

For the mass transfer process

$$2\pi r \rho_s \, dr = h_m \, 2\pi r \rho \, (W_o - W) \, dr \, dT$$

which can be changed to equation 4.34 by substituing for h_m in terms of R_e and S_e

$$\frac{d(D/D_o)}{d(U_oT/D_o)} = \frac{2C_2 (W_o - W)}{Re^{1-n} \, Sc \, (\rho_s/\rho)(D/D_o)^{1-n}} \qquad (4.34)$$

$$\text{or} \int_1^{D/D_o} f(D/D_o) \, d(D/D_o) = (W_o - W_s) \frac{U_o T}{D_o} \qquad (4.35)$$

$$\text{where } f\left(\frac{D}{D_o}\right) = \frac{Re^{1-n} \, Sc \, (\rho_s/\rho)(D/D_o)^{1-n}}{2C_2 \left[1 - \frac{W - W_s}{W_o - W_s}\right]} \qquad (4.36)$$

We can substitute the value of $W-W_s$ / W_o-W_s in (4.35) from equation 4.33 in terms of D/D_o and can get a relation between D/D_o and U_oT/D_o. The integral in (4.35) must be evaluated numerically because the integrand (D/D_o) is quite complicated. Relations between D/D_o and $W-W_s/W_o-W_s$, and D/D_o and U_oT/D_o having been found out the relation between $W-W_s/W_o-W_s$ and U_oT/D_o can be obtained. The values of W and hence t can be obtained at different stages by this relation and expression (4.31) will determine the total heat transfer. The total mass transfer upto any time T can be obtained from the relation between D/D_o and U_oT/D_o .

The integral in expression 4.35 must be evaluated numerically because the integrand f (D/D_o) given by expression 4.36 is quite complicated. In certain applications, where the difference between temperature of free air stream t_o and that of bare cylinder t_s is small, the integration of f (D/D_o), for the duration of time when D/D_o is nearly equal to 1, becomes rather simple. In order to get value of $W-W_s/W_o-W_s$ from equation (4.33), certain simplifications can be made since the difference $t_o - t_s$ is small, it can be assumed that $t_o - t_s/W_o-W_s$ $= t - t_s/W_o - W_s = b$. Substituting this value in expression (4.33), we get

$$\frac{W-W_s}{W_o-W_s} = \frac{c_3 b + C_2 \frac{P_r}{Sc} * \frac{L}{C_p}}{2\frac{K_f}{K}\left\{\frac{b\,Re^{-n}}{(D/D_o)^n \ln(D/D_o)}\right\} + C_2 \frac{P_r}{Sc}\cdot\frac{L}{C_p} + C_3 b}$$

and if
$$\psi_o = C_3 b + c_2 \frac{P_r}{Sc}\cdot\frac{L}{C_p}$$

$$\frac{W-W_s}{W_o-W_s} = \frac{\psi_o}{2\frac{K_f}{K} \times \frac{b\,Re^{-n}}{(D/D_o)^n \ln(D/D_o)} + \psi_o} \qquad (4.37)$$

from expression (4.36), we can get

$$f\left(\frac{D}{D_o}\right) = \psi_1 \left(\frac{D}{D_o}\right)^{1-n} \left\{\psi_2 + \psi_o \left(\frac{D}{D_o}\right)^n \left(\frac{D}{D_o} - 1\right)\right. \qquad (4.38)$$

where
$$\psi_1 = \frac{Sc\ \ R_e\ (\rho_t/\rho)}{4\,c_2\,b\ \ k_f/k}$$

and
$$\psi_2 = 2\frac{K_f}{k}\ b\ Re^{-n}$$

Now the value of $f\,(D/D_0)$ from (4.38) can be substituted in expression (4.35) and integrated yielding equation (4.39)

4.4 DIFFUSION OF WATER VAPOUR IN FROST LAYER AND VARIATION OF THERMAL CONDUCTIVITY WITH TIME:

The properties of frost on evaporator coil vary with time. The frost possesses a porous structure and a concentration gradient for water vapour exists through it. The concentration gradient is created owing to a temperature gradient that exists in a frost layer. Because of the existence of concentration gradient, it is thought that the water vapour in the air will diffuse through the frost layer and may change its density and thermal conductivity.

An enlarged view of the cross-section of porous frost layer is shown in fig. 4.41. Air in the voids in frost can be considered to be saturated with water vapours at local temperature of the frost. Thus with the variation in temperature of frost from air frost interface to coil surface, the concentration will also vary. Consider the vapour to be driven across the plane at X, if all the area could be freely utilised for mass transfer.

The amount of mass diffused $\quad m_x = -\delta\left(\frac{\partial c}{\partial x}\right)$

But the frost layer does not provide a free surface for diffusion of water vapour as it cannot pass through solid particles of the frost and has to go through the gaps inbetween them. If ρ_s

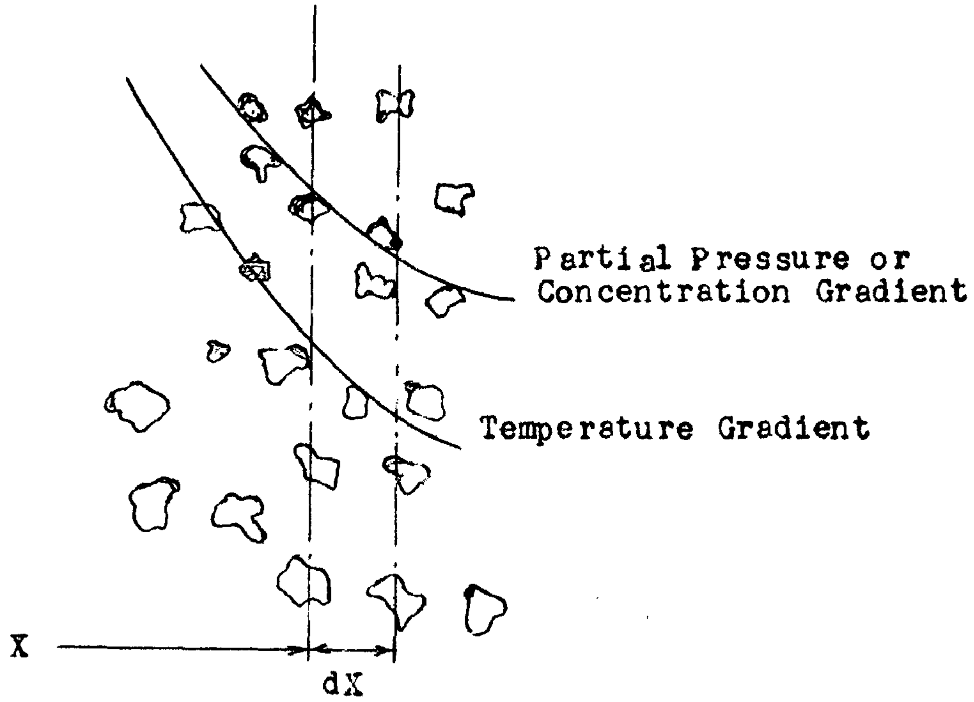


Fig. 4.41 REPRESENTATION OF ENLARGED CROSS-SECTION OF A POROUS FROST LAYER

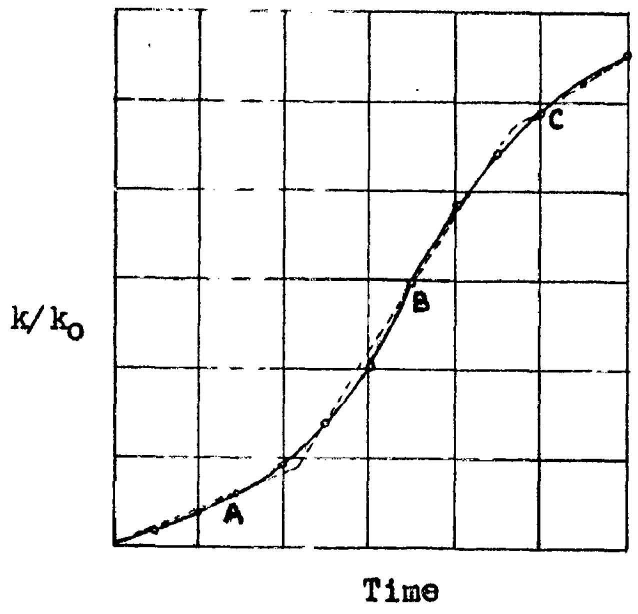


Fig. 4.42 VARIATION OF OVERALL CONDUCTIVITY WITH TIME

be the density of frost at section X and ρ_{fo} represent the density of frost for which no free area is available.

Actual amount of mass diffused $m_X = -\delta(1-\frac{\rho_f}{\rho_{fo}})(\frac{\partial C}{\partial X})$ (4.41)

where ρ_f and C are functions of X and T.

Since the total heat transfer through the frost layer occurs by conduction and by virtue of diffusion of water vapours that condense as snow.

The rate of total heat transfer at X at any given time T

$$= -K(\frac{\partial t}{\partial X}) - \delta L(1-\frac{\rho_f}{\rho_{fo}})(\frac{\partial C}{\partial X}) \quad (4.42)$$

The temperature t and concentration are interdependent and for the small range encountered with in the case they can be assumed to be linearly related i.e. $t = a+bC$.

Also the following relationship is very nearly true (29)

$$k = k_o \rho_f$$

where k_o is a constant. Substituting the values of t and k in (4.42) we get

$$Q_X = \left\{-k_o b \rho_f + \delta L(1-\frac{\rho_f}{\rho_{fo}})\right\}(\frac{\partial C}{\partial X}) \quad (4.43)$$

The net difference in total heat transfer at X and X + dX will be stored by the frost and thus

$$(k_o b - \frac{\delta L}{\rho_{fo}})\rho_f \frac{\partial^2 C}{\partial X^2} + \delta L \frac{\partial^2 C}{\partial X^2} + (k_o b - \frac{\delta L}{\rho_{fo}})(\frac{\partial C}{\partial X})(\frac{\partial \rho_f}{\partial X})$$

$$= c_{pf} \rho_f b \frac{\partial C}{\partial T} \quad (4.44)$$

Inexpression (4.44) the heat transfer due to the condensation of water vapour is small as compared to the heat conduction. Also that the weight of frost is small and therefore the heat stored in the frost is also small and can be neglected. Equation 4.44 will then reduce to

$$K_o b \rho_f \frac{\partial^2 c}{\partial X^2} + K_o b \left(\frac{\partial c}{\partial X}\right)\left(\frac{\partial \rho_f}{\partial X}\right) = 0 \qquad (4.45)$$

Similarly with the help of equation 4.41 the transient diffusion equation can be written as

$$\delta\left(\frac{\partial^2 c}{\partial X^2}\right) - \frac{\rho_f}{\rho_o}\delta\frac{\partial^2 c}{\partial X^2} - \frac{\delta}{\rho_{fo}}\frac{\partial c}{\partial X}\cdot\frac{\partial \rho_f}{\partial X} = \frac{\partial \rho_f}{\partial T} \qquad (4.46)$$

The equations 4.44 and 4.46 or 4.45 and 4.46 can be solved simultaneously by considering boundary conditions for as a function of X and T; and for k_f as function of ρ_f and thereby of X and T boundary conditions can be determined from the considerations of actual condition.

The variation of conductivity of the frost with time is shown in Fig. 4.42. It can be subdivided into three parts. During the first period the variation in the value of k is small. It is followed by period of rapid increase of conductivity and finally k approaches a limiting value. These periods are represented by A,B and C respectively upon the curve. Similar results have been reported by Beatty, finch and schoenburn (25).

It can, therefore, be summarised that during the earlier stages the conductivity of frost does not change appreciably by the diffusion of the vapour through the frost. During this period the time dependence of the conductivity and density of frost may be neglected.

For a cylinder equations 4.47 and 4.48 can be derived proceding on similar lines, for heat and mass transfers respectively

$$\left(k_0 b - \frac{\delta L}{\rho_{f0}}\right) \rho_f \frac{\partial^2 C}{\partial r^2} + \delta L \frac{\partial^2 C}{\partial r^2} + \left(k_0 b - \frac{\delta L}{\rho_{f0}}\right) \frac{\partial C}{\partial r} \cdot \frac{\partial \rho_f}{\partial r}$$

$$+ \left(k_0 b - \frac{\delta L}{\rho_{f0}}\right) \frac{\rho_f}{r} \cdot \frac{\partial C}{\partial r} + \frac{\delta L}{r} \frac{\partial C}{\partial r} = c_{pf} \rho_f b \frac{\partial C}{\partial T} \qquad (4.47)$$

and

$$\left(1 - \frac{\rho_f}{\rho_{f0}}\right) \frac{\partial^2 C}{\partial r^2} - \frac{1}{\rho_{f0}} \left(\frac{\partial \rho_f}{\partial r} \cdot \frac{\partial C}{\partial r}\right) + \left(1 - \frac{\rho_f}{\rho_{f0}}\right) \frac{1}{r} \frac{\partial C}{\partial r} = \frac{1}{\delta} \cdot \frac{\partial \rho_f}{\partial T} \qquad (4.48)$$

Neglecting heat transfer due to the condensation of water vapour and the heat stored in frost, equation 4.49 can be obtained from expression 4.47.

$$k_0 b \rho_f \frac{\partial^2 C}{\partial r^2} + k_0 b \frac{\partial C}{\partial r} \cdot \frac{\partial \rho_f}{\partial r} + k_0 b \frac{\rho_f}{r} \cdot \frac{\partial C}{\partial r} = 0 \qquad (4.49)$$

CHAPTER 5

EXPERIMENTAL SET UP

Experimental set up is described.

5.1 REFRIGERATION PLANT:

The evaporator used as the object of study was that of a R-12 cold storage plant located in laboratory building. It was cold diffuser type in which a blower was used to suck air and to circulate it through the refrigerated space, as is shown in figure 5.11. The specifications of different components were as follows.

Evaporator:

Coil and finned type	
Coil diameter	3/4 in. OD
Number of coils	96
Fin spacing	$3\frac{1}{2}$ per in.

Compressor:

Vertical reciprocating	2 cylinders
Bore and stroke	$2\frac{1}{2}$ in. x 3 in.
R.P.M.	455

Electric Motor Drive:

Induction motor	400/440 Volts
5 H.P.	7.5 Amp
3 Phase	50 c/s
R.P.M.	1450

Condenser:

Coil and finned type.

In the absence of an efficient air blower alternative arrangements were made to effectively cool the condenser by spraying water. Water from supply mains was used for circulation through condenser.

A line diagram for the refrigeration plant is drawn in

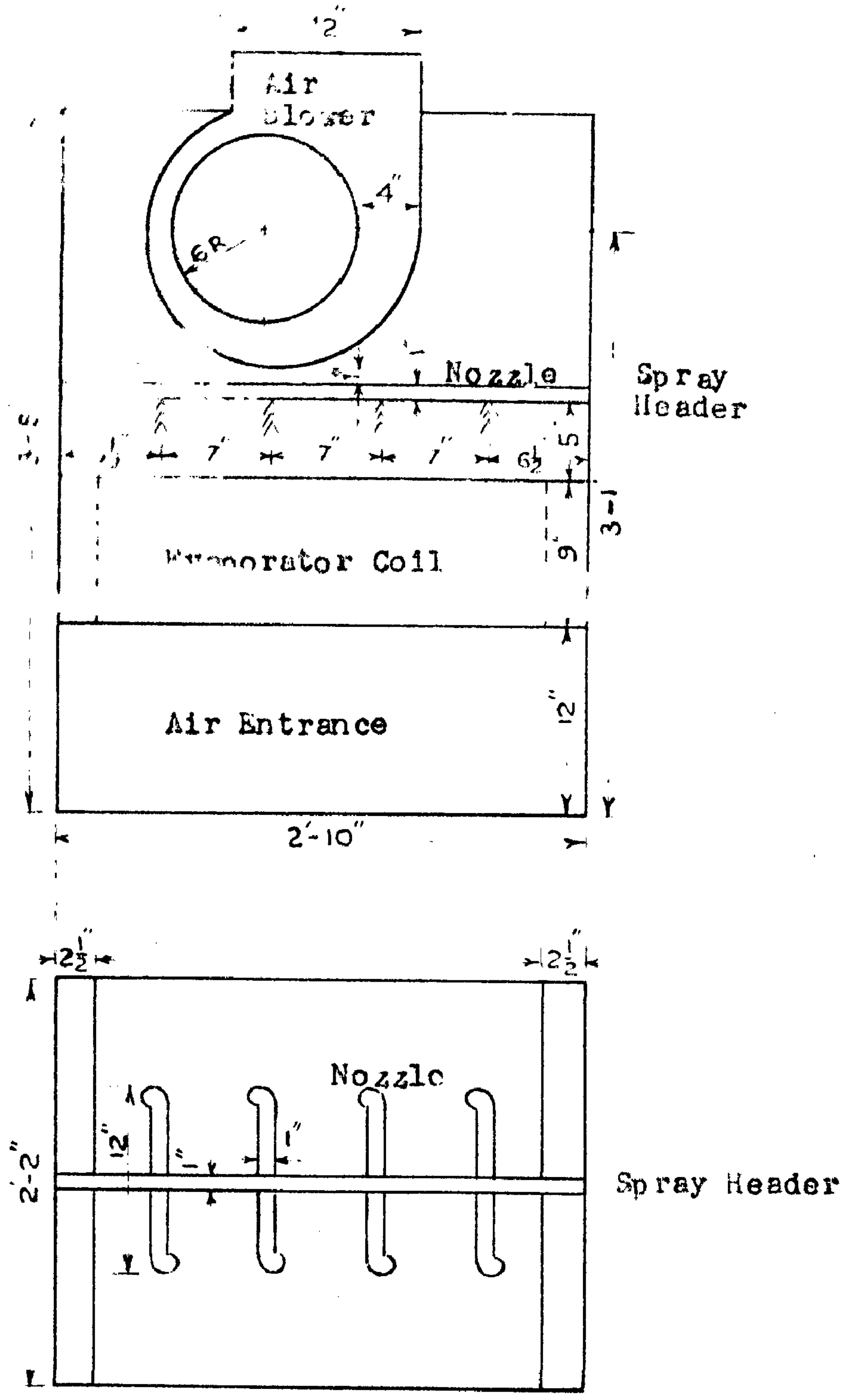


Fig. 5.11 COOLING COIL FRAME

fig. 5.12. Two suction-liquid heat exchangers were employed in series to subcool the liquid refrigerant from the condenser. The first heat exchanger was located outside the cold storage room while the other one was installed inside the same. The expansion valve was placed outside the room.

5.2 REFRIGERATED SPACE AND LOAD:

The evaporator was located in the cold storage room of which the outside dimensions were 12' x 12' x 6' 7". All the walls, the ceiling and the floor were insulated by 5 in. thick fibre wool (density 6 lb./cu.ft.) supported between 1/8 in. thick copper-zinc alloy plates. Thermocouples had been mounted at six spots viz. at the middle of four corner panels of the side walls, and two at the centre of the side ceiling panels. Each panel was 4 ft. wide, thus all the thermocouples were well. about 2 ft. removed from the corners of the room. The load on the plant was that due to transmission through the insulation. Different additional loads by strip heaters and electric bulbs were tried to give satisfactory operation to the plant. Ultimately an additional load of over 500 watts was provided by placing electric bulbs. This additional load could be measured by a KWH metre.

In order to impose a latent heat load on the plant water was filled in a container placed inside the cold storage, and was heated by means of immersed electric heaters. The water container was insulated with 1/2 inch asbestos coating on all sides and additional 4 in. cork on three sides. The electric power to the heater was fed through a variable transformer so

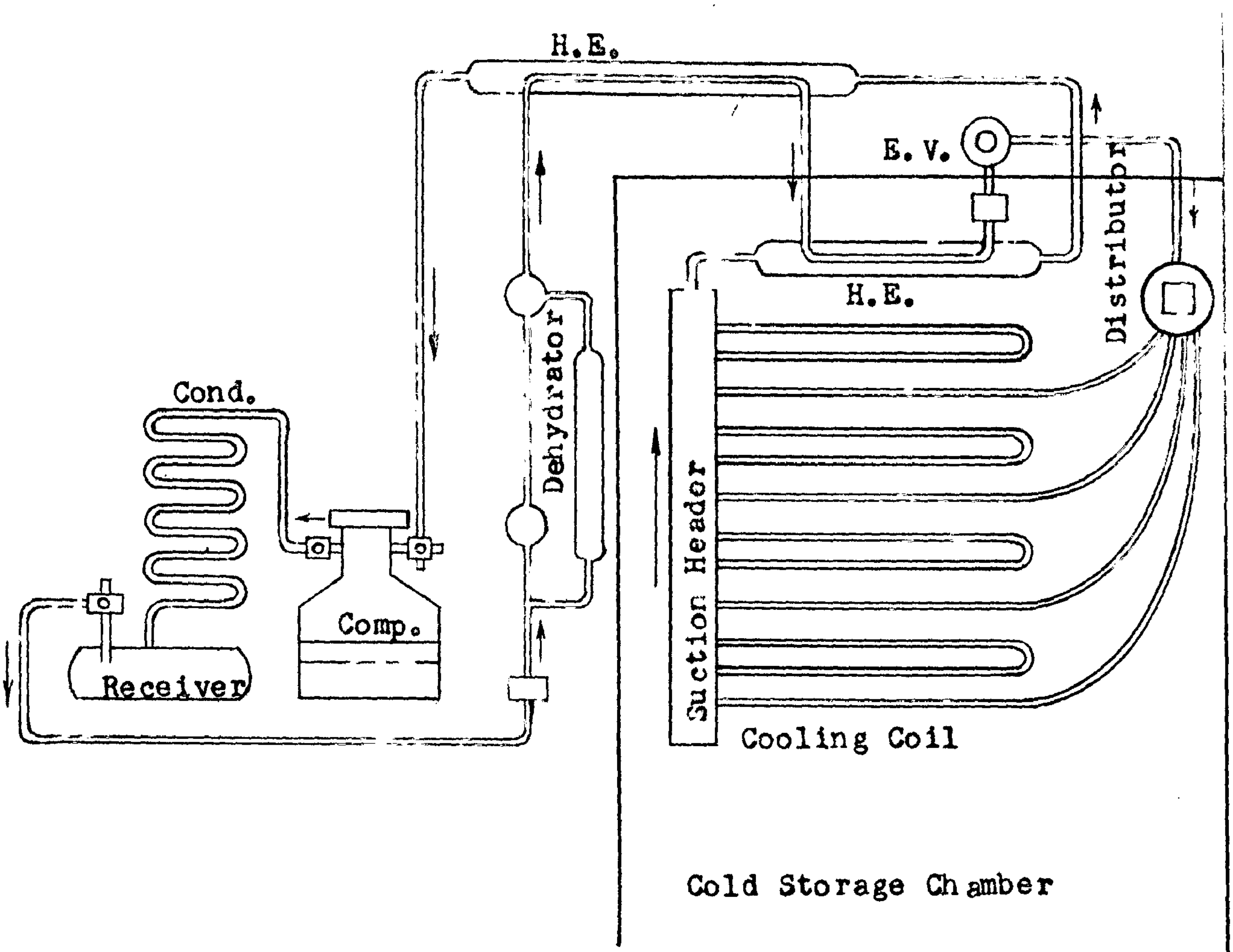
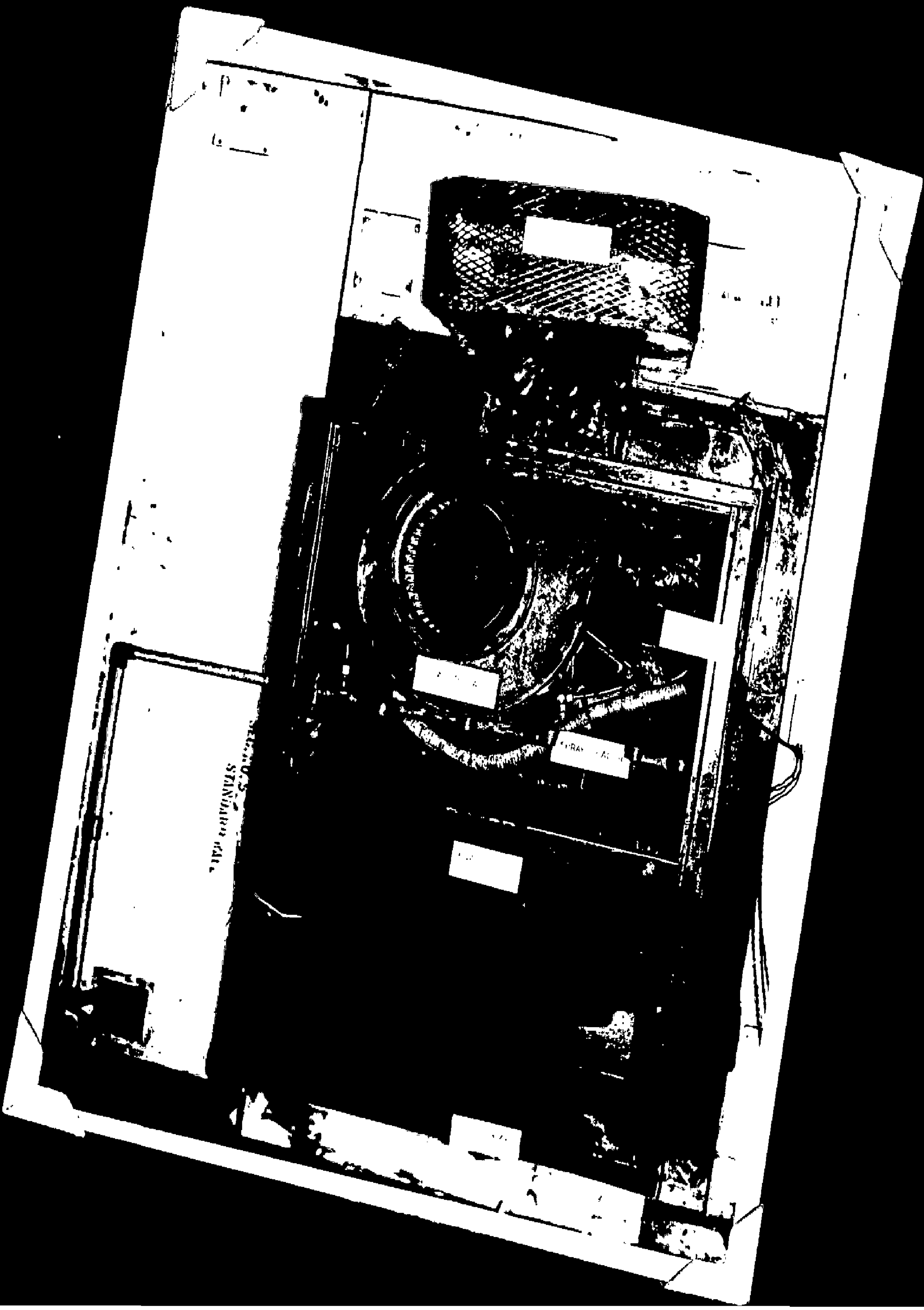


Fig. 5.12 LINE DIAGRAM OF COLD STORAGE PLANT




8 TON UNIVERSAL
COLD STORAGE
AUTOMATIC POTENTIOMETER
POTENTIOMETER
VARIABLE TRANSFORMER
M.E.D./5/102/R
M.E.D./5/102/R


KCB

that an adequate electric power could be supplied by regulating the transformer. The power being supplied to the heater, could be measured by a wattmeter connected to the immersion heater.

To record the temperature of the cold storage walls on the inside, the thermocouples, that were mounted, were connected to an automatic electronic potentiometer recorder. For recording the temperature of the outside, automatic temperature recorders were placed outside and adjacent to the cold-storage walls.

5.3 DEFROST ARRANGEMENT:

Water defrost as supplied with the system, was installed for the evaporator coils. Eight nozzles were used spaced at 7 inch distance apart and in double row to cover the whole free area of the evaporator, refer fig. 5.11. The water for defrost was obtained from water supply main, and separate supply and drain valves were used, refer fig. 2.42. The separate drain valve the function of which was to drain effectively the water remaining after defrost in the spray header and the pipe carrying water to it, was provided to allow for the effective draining. The horizontal portion of the pipe connecting it to the spray header was kept a bit inclined towards the exit to safeguard against any entrapping of water after defrost, which would otherwise freeze within the connecting pipe during refrigeration cycle and would block the passage of water to the header during the following defrost. A drip tray was located underneath the coil and the main drain was connected to that. Measurement of the rate of water flowing could be done by collecting the discharge in any vessel and noting the time for water flow with a stop watch.

CHAPTER 6

RESULTS & DISCUSSION

Results are discussed.

6.1 EXPERIMENTAL TECHNIQUE:

The object of study of the present experimentation was to determine for a particular plant, under certain operating conditions, the variation of rate of energy consumption and average power of the motor per ton of refrigeration with defrost interval for uniform refrigeration load. The plant was operated under similar load and surrounding conditions for different runs. Different defrost frequencies were chosen and the optimum was determined practically. The sensible heat load existed due to transmission gain through the cold storage walls, which was reinforced by electric lamps placed inside the room. Energy supplied to the immersion heater in the water container was considered to be latent heat load. The heat loss by conduction through the insulation was neglected. Whatsoever error it might involve in calculating the latent heat load, it did not alter the total load. Although this will result in greater ratio of latent heat load to total heat load than actually on the plant, but the loading on the plant remained of a uniform nature. Therefore, a comparison of plant performance under such operating conditions is fully justified.

6.2 EXPERIMENTAL RESULTS:

Runs were made for eight defrost intervals varying from four hourly to eleven hourly.

Observations recorded during the tests have been included in Appendix 2. Tables 6.21 and 6.22 show the results obtained from the test. Transmission and lamp loads, and thereby the sensible heat load, and the latent heat load were computed as

TABLE 6.21

LOAD CALCULATIONS

Defrost Interval (hrs.)	Defrost Period (Min.)	Transmission Load Btu.	Lamp Load Btu (KWH x 3,410)	Sensible Heat Load Btu	Latent heat Load (Btu) Watt x 3.4xT_o Wattage = 500	Total Heat Load Btu.
4	15	4,180	5,840	10,020	6,390	17,410
5	16	5,440	6,450	11,890	8,060	19,950
6	16	6,260	8,480	14,740	9,760	24,500
7	18	7,700	11,970	19,670	11,300	30,970
8	20	8,690	11,450	20,140	13,060	33,200
9	22	9,460	13,170	22,630	14,800	37,430
10	22	10,800	18,600	29,400	16,950	46,350
11	26	11,350	19,800	31,150	18,100	49,250

indicated above. The values of rate of energy consumption per ton and average power per ton (Art. 3.3 and 3.4) for different defrost intervals were computed. Curves were drawn between the total energy consumed by the motor during operating time and defrost interval, fig. 6.21, between rate of energy consumption and defrost interval, fig. 6.22, and between average power per ton and defrost interval fig. 6.23.

TABLE 6.22

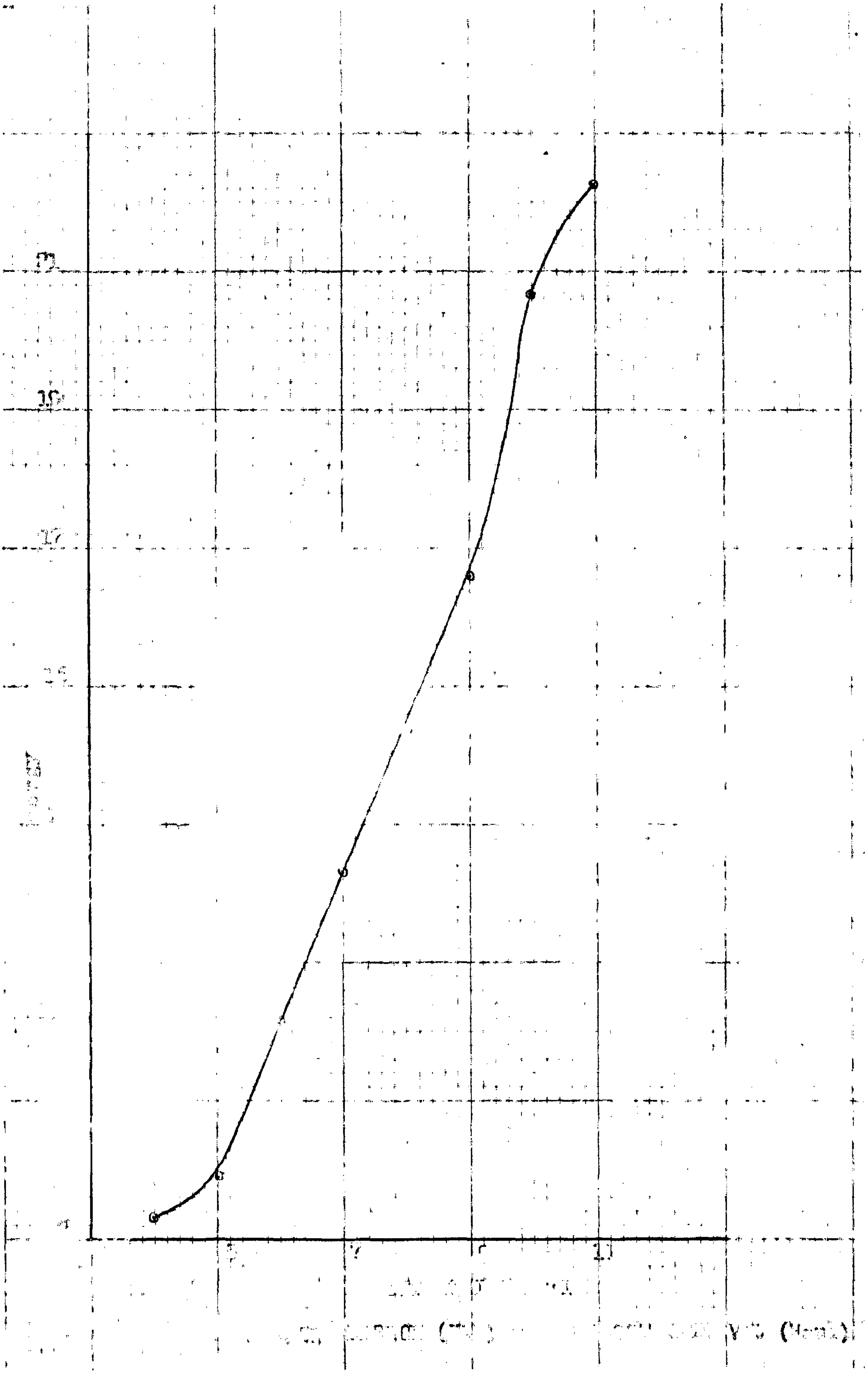
VARIATION OF ENERGY RATE AND AVERAGE POWER WITH DEFROST INTERVAL

Defrost Interval	Defrost period Min.	Energy Rate KWh/(Hr.)(Ton)	Average Power KW/Ton.
4	15	5.01	5.35
5	16	4.71	4.98
6	16	4.97	5.25
7	18	4.96	5.19
8	20	5.18	5.41
9	22	5.27	5.50
10	22	5.35	5.55
11	26	5.40	5.62

6.3 DISCUSSION:

The performance could not be studied in low defrost interval zone because many unpredictable factors influence it non-uniformly and any observation in this region cannot be generalised.

Fig. 6.21 reveals that the energy consumption varies



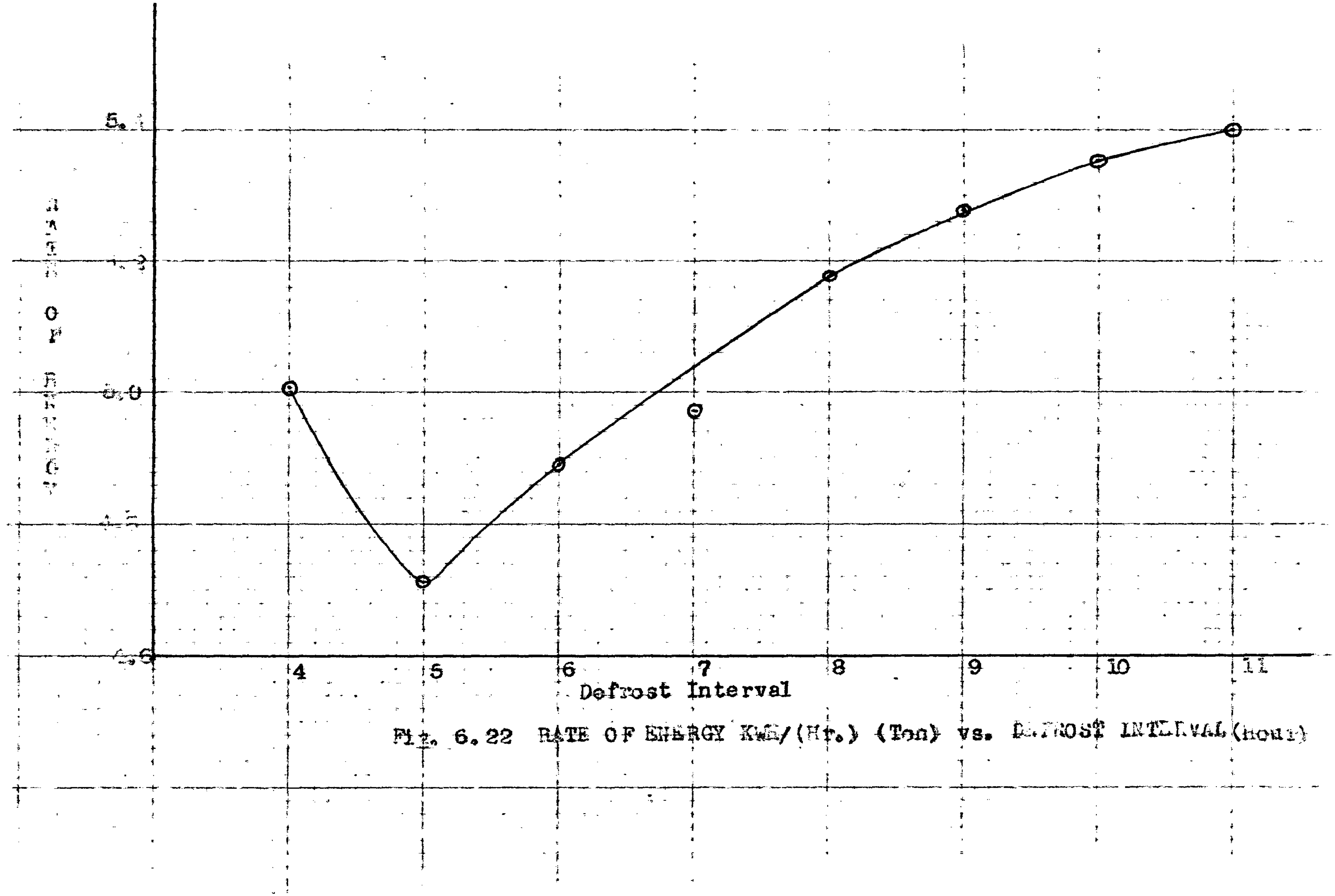
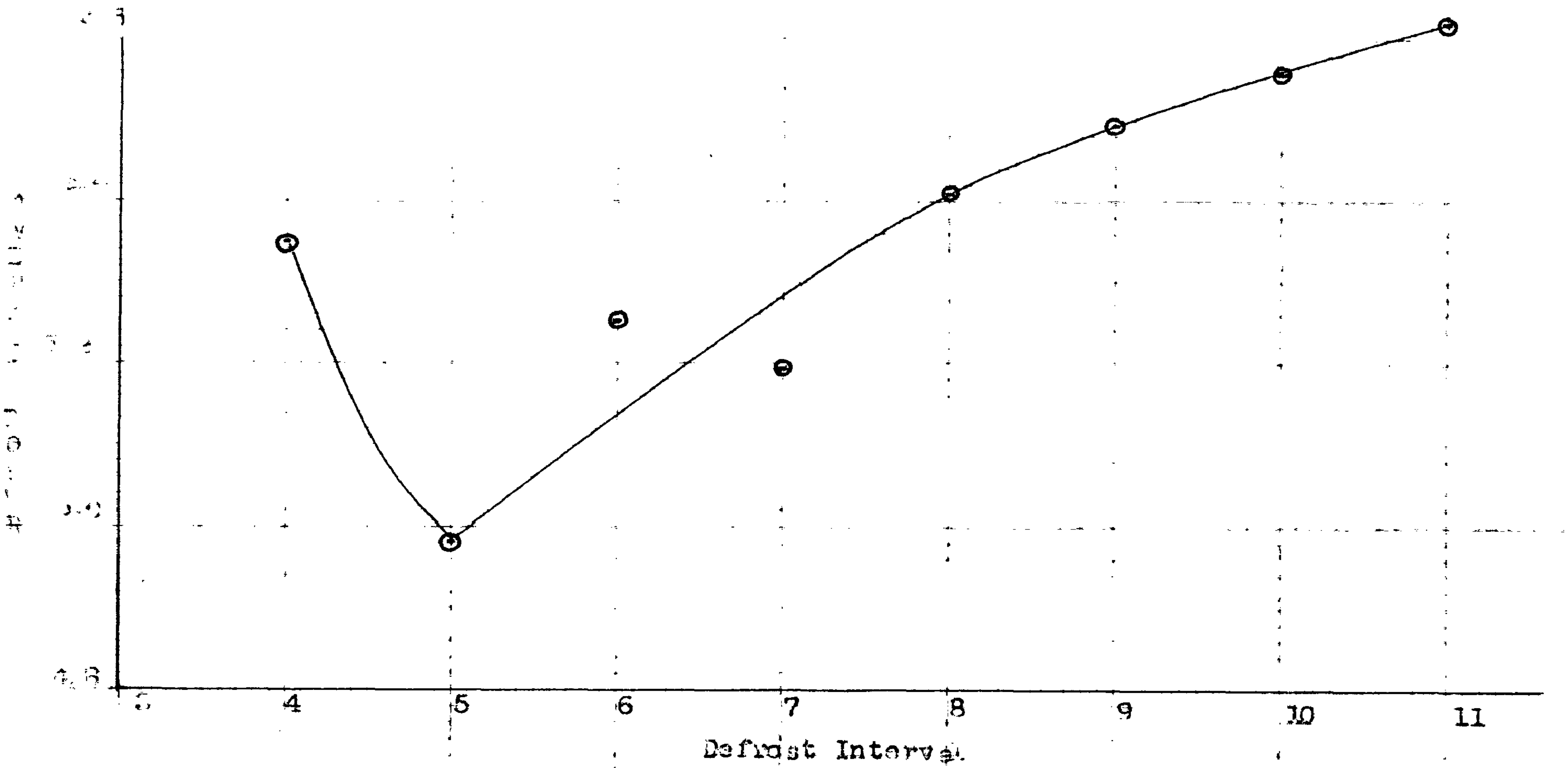


Fig. 6.22 RATE OF ENERGY KWH/(Hr.) (Ton) vs. DEFROST INTERVAL (hour)



3. 23 AVERAGE POWER KW/Ton vs. DEFROST INTERVAL (Hours)

OBSERVATION SHEET NO.4

TEMPERATURES (°F) AT DIFFERENT POINTS WITHIN THE ROOM

TIME	THERMOCOUPLE POINTS						
Hr. Mts.	1	2	3	4	5	6	
00.00		44.0	43.5	43.0	41.5	41.0	
00.30		43.0	44.0	43.0	48.0	40.5	
01.00		43.0	44.0	42.0	42.0	38.0	
01.50		43.0	43.0	41.5	40.0	36.5	Energy to Motor (KWH) = 12.79.
02.00		42.0	43.0	41.0	44.5	36.5	
02.30		42.0	43.0	41.5	48.0	37.5	
03.00		42.0	43.0	41.0	48.0	37.0	
03.30		42.0	43.0	40.5	48.0	36.5	
04.00	Average Value = 32.53	41.5	42.0	39.5	40.5	35.0	
04.30		41.5	42.0	39.5	38.5	35.0	
05.00		42.0	42.0	39.5	38.5	35.0	
05.30		42.0	42.0	40.0	39.0	35.0	
06.00		42.0	43.0	40.0	40.5	35.0	
06.30		42.0	43.0	40.5	42.0	35.5	
06.42		41.5	42.0	39.5	39.5	35.0	
06.45		42.0	42.0	40.5	42.0	36.0	
06.50		42.0	42.0	41.5	43.0	37.0	
06.55		43.0	43.0	42.0	43.0	38.5	
07.00		43.0	43.0	42.0	43.5	38.5	

Contd.....

proportional to time except near five hourly defrost. At this stage the curve has a lower gradient which tends to increase in this zone and is more or less maintained beyond it. Referring to fig. 6.22 and table 6.22, it can be observed that the rate of energy consumption per ton is minimum for the plant, under loading conditions of the experiment, at about five hourly defrost with sixteen minutes defrost period. On either side of it the variation is steep. However as the time interval between two defrosts is increased from this optimum, there is a small-range where the variation is steep, while if increased further the curve flattens. The increase in rate of energy consumption under most stringent conditions expedited, was .69 KWH/(hr)(ton) more than that for optimum i.e. an increase of 14.6 percent above optimum. While for four hourly defrost the rate of energy consumption is 5.01 KWH/(hr)(ton), it is 4.71 for five hourly defrost. Thus on decreasing the defrost interval by one hour below the optimum, the rate of energy consumption increases by 0.3 KWH/(hr.)(ton) or about 6.4 percent over the optimum. Thus for a lower defrost interval than the optimum the rate of energy consumption is significantly greater than optimum. On increasing the defrost interval, it can be observed that increasing the defrost interval from five hourly to eight hourly, increases the rate of energy consumption from 4.71 to 5.18 i.e. by 0.47 KWH/(ton)(hr.) or an increase of about 9.9 percent. On the other hand increasing the defrost period from eight hourly to eleven hourly, another increase of three hours in defrost interval, changes the rate of energy consumption from 5.18 to 5.40 i.e. by 0.22 KWH/(ton)(hr). This means further percentage increase of about 4.7.

Readings were not taken beyond eleven hourly defrost as it choked the coil excessively and increased the defrost period abruptly by a large amount.

The increase in the rate of energy consumption is more rapid near the zone of optimum performance. In a region of greater interval between two successive defrosts, defrosting is not very important from the point of view of energy considerations. Under the conditions of tests, from energy considerations, it was desirable to defrost five hourly. It also indicated that if it is not done at this interval and the time interval is increased, then after a certain zone other aspect viz. the resistance to flow of air may govern the frequency.

The nature of the variation of average power per ton with defrost interval is similar to the previous curve, as is shown by fig. 6.23. This may also be expected since the two do not differ by a large amount. The optimum condition for the average power per ton occurs at more or less the same time as that for rate of energy consumption.

6.31 GENERAL INFERENCE:

Although the performance of a particular plant was studied during the experiment but it verified the general behaviour of the variation of rate of energy consumption and average power of the motor, as theoretically discussed in Chapter 3 (Art. 3.3 and 3.4). This, however, cannot be predicted as to what would be the exact nature of the variation of the two curves near the optimum from energy considerations. In the case under study the variation was more pronounced at the optimum and it represented the desirability to operate at optimum, if the large frequency does not spoil the product. Under most stringent conditions of frost accumulation expedited, when the frequency was reduced by more than 50 percent below optimum, the rate of energy consumption increased by 14.6 percent and average power of the motor by 12.9 percent. In any other case these quantities shall vary. Even for the same plant under different load conditions, the performance may vary under different load and surrounding conditions. The latent heat load was excessive on the plant during the experiment under study. Under less demanding conditions the performance would vary. The optimum defrost interval would change and so should the variation of rate of energy consumed and average power of the motor.

In spite of the fact that performance of any new plant cannot be predicted by the present experiment, which expedited a particular plant and the variation in the magnitudes of its certain important characterstics, it indicated that from energy considerations an optimum frequency exists. On reducing the defrost interval below optimum there was a marked influence on the energy terms. As there is not much of frost in this period and the number of defrosts will increase increasing the cost of defrosting, no point will be gained by reducing the defrost interval below optimum. Therefore, it is not desirable.

Nonetheness, increasing the defrost interval above it shall not necessarily entail an increase in operating cost, because the smaller number of defrosts required will bring about a reduction in the cost of defrosts required. Before deciding upon the frequency of defrost all such factors should be considered. In addition the factors pointed out in Chapter 1 should be looked into.

<u>END</u>

A P P E N D I X 1

EXPLANATION OF EQUATION 4.23

Equation 4.23 is explained.

EXPLANATION OF EQUATION 4.23:

i_o = enthalpy of air/lb. dry air.

i = enthalpy of saturated air at temperature of frost air interface.

$$\frac{dq}{dA} = h(t_o - t) + h_m \rho (W_o - W) L$$

$$= h_m \rho \left\{ c_p (t_o - t) + L(W_o - W) \right\}$$

$$= h_m \rho (i_o - i) = h_i (i_o - i)$$

since $h_m \rho = h_i$

and if whole of heat transferred is assumed to occur because of cooling of air, it will be equal to total enthalpy change of air, thus

$$\frac{dQ}{dA} = -W_o \frac{di_o}{dA}$$

COMMENT ON h_i NUMERICALLY EQUAL TO $h_m \times \rho$

$$h_i = \frac{(dQ/dA)_{at\ x=0}}{i_o - i_s} \qquad \text{equation (13)}$$

$h_m \rho$ = Mass transfer coefficient lb./(hr)(sq. ft.)(H_g-H_f)

where x is the thickness of frost layer

Now $$\frac{dQ}{dA} = h_m \rho (i_o - i)$$

an expression already obtained when $x = 0$ $i_i = i_s$

$$\therefore \left(\frac{dQ}{dA}\right)_{x=0} = \left[h_m \rho (i_o - i)\right]_{x=0} = \left[h_m \rho (i_o - i_s)\right]_{x=0}$$

Comparing the two $h_m \rho = h_i$

APPENDIX 2

OBSERVATIONS

Observations during the tests are recorded.

OBSERVATION SHEET NO.1

TEMPERATURES (°F) AT DIFFERENT POINTS WITHIN THE ROOM

TIME Hr. Mts.	THERMOCOUPLE POINTS 1	2	3	4	5	6
00.00	43.5	43.5	43.0	43.5	44.0	43.0
00.30	35.0	39.5	40.0	42.0	42.0	41.5
01.00	34.5	39.5	44.5	40.5	41.5	39.5
01.30	33.5	39.5	45.5	41.0	42.0	39.0
02.00	33.5	40.0	44.0	41.5	41.5	38.5
02.30	33.0	39.5	43.5	41.5	41.5	38.0
03.00	31.5	36.5	43.5	39.0	40.5	37.5
03.30	30.0	38.0	38.5	38.0	39.0	37.5
03.45	30.0	38.0	38.5	38.0	38.5	37.5
03.50	31.5	39.0	39.0	39.5	39.5	38.0
03.55	32.0	39.5	39.0	40.0	40.0	39.5
04.00	32.5	39.5	39.5	40.0	40.0	39.5

Energy to Motor (KWH) = 7.27

Mean temperature for:

3 hrs. 30 min. = 40.1° F

next 15 min. = 36.8° F

(Defrost Cycle) last 15 min. = 37.8° F

Water spray period = 5 min.

Run-off period = 10 min.
Mean outside temperature = 79.5° F.

OBSERVATION SHEET NO. 2

TEMPERATURES (°F) AT DIFFERENT POINTS WITHIN THE ROOM

TIME	THERMOCOUPLE POINTS					
Hr. Mts.	1	2	3	4	5	6
00.00	34.0	40.5	43.0	41.5	41.0	41.0
00.30	35.0	41.0	42.0	42.0	42.0	40.0
01.00	31.0	38.0	42.0	39.5	40.5	38.0
01.30	31.5	38.0	38.0	38.5	40.5	38.5
02.00	32.0	38.0	38.5	39.0	40.0	38.5
02.30	33.0	39.0	40.0	40.0	41.5	38.5
03.00	33.5	39.0	41.5	41.0	41.0	38.5
03.30	33.5	39.5	43.0	41.0	41.0	38.5
04.00	33.5	39.5	45.0	40.0	40.0	38.0
04.30	32.5	37.5	38.0	38.0	39.5	38.0
04.44	32.0	39.0	44.5	39.5	39.5	37.5
04.49	33.5	40.0	44.0	41.0	41.0	38.0
04.54	35.0	41.5	48.0	42.0	42.0	39.0
05.00	35.0	42.0	48.0	42.0	42.0	39.0

Energy to Motor (KWH) = 7.84

Mean temperature for:

4 hrs. 30 min. = 38.4° F

next 14 min. = 37.9° F

(Defrost Cycle) last 16 min. = 37.8° F

Water spray period = 6 min.

Run-off period = 10 min.

Mean outside temperature = 79.7° F.

OBSERVATION SHEET NO.3

TEMPERATURES (°F) AT DIFFERENT POINTS WITHIN THE ROOM

TIME	THERMOCOUPLE POINTS					
Hr. Mts.	1	2	3	4	5	6
00.00	38.5	41.0	57.0	45.5	45.5	43.0
00.30	33.0	40.0	40.0	41.0	42.0	40.5
01.00	32.0	39.0	39.0	40.5	41.0	39.0
01.30	32.0	38.5	38.0	38.5	40.5	38.5
02.00	31.5	38.0	38.0	39.0	40.0	38.5
02.30	31.5	38.0	38.5	39.0	40.0	37.5
03.00	32.0	38.0	39.0	39.0	40.0	38.0
03.30	32.5	38.5	40.5	40.0	40.5	38.0
04.00	33.0	39.5	43.5	40.5	41.0	38.0
04.30	30.0	37.0	39.0	38.5	39.5	37.0
05.00	32.0	38.0	40.0	39.5	40.0	37.5
05.30	32.5	39.0	42.0	40.0	40.5	38.0
05.44	32.0	38.0	40.0	39.5	40.0	37.5
05.49	33.5	40.5	43.0	41.5	41.0	38.0
05.54	36.0	42.0	44.0	43.0	42.0	39.5
06.00	37.0	43.0	44.5	43.0	42.0	41.5

Energy to Motor (KWH) = 10.17

Mean temperature for:

5 hrs. 30 min. = 38.5° F

next 14 min. = 38.3° F

(Defrost Cycle) last 16 min. = 39.7° F

Water spray period = 6 min.

Run-off period = 10 min.

Mean outside temperature = 80.0° F.

OBSERVATION SHEET NO.4 (Contd.)

Mean temperature for:

	6 hrs.	30 min.	= 39.6°F
	next	12 min.	= 38.8°F
(Defrost Cycle)	last	18 min.	= 39.9°F
Water spray period			= 7 min.
Run-off period			= 11 min.
Mean outside temperature			= 81.5°F.

OBSERVATION SHEET NO.5

TEMPERATURES (°F) AT DIFFERENT POINTS WITHIN THE ROOM

TIME	THERMOCOUPLE POINTS					
Hr. Mts.	1	2	3	4	5	6
00.00	43.0	43.5	43.0	44.0	44.0	43.0
00.30	35.0	39.5	39.5	41.5	42.0	41.5
01.00	33.5	39.0	44.5	40.5	41.0	39.5
01.30	34.0	40.0	46.0	41.5	41.5	39.0
02.00	34.0	40.0	44.0	41.0	41.5	38.5
02.30	33.0	39.5	43.5	41.0	41.5	38.0
03.00	31.5	38.0	44.0	39.5	40.0	37.5
03.30	30.0	36.5	38.0	38.0	39.5	37.5
04.00	30.5	37.5	38.5	38.5	39.5	37.5
04.30	32.0	38.0	40.0	39.5	40.0	37.5
05.00	34.0	39.0	44.5	41.0	41.0	38.0
05.30	32.0	39.0	40.5	40.0	40.5	37.5
06.00	29.5	37.0	37.5	37.5	38.5	37.0
06.30	32.0	38.0	39.5	39.0	40.0	37.5
07.00	34.0	39.5	36.5	40.5	40.5	38.5
07.30	31.5	37.5	39.0	39.0	39.5	37.5
07.42	31.0	37.5	37.0	38.0	39.0	37.0
07.47	33.5	39.0	39.5	39.5	40.5	39.5
07.52	35.5	40.5	40.5	40.5	42.0	40.5
08.00	37.5	42.0	42.0	42.0	42.0	42.0

Energy to Motor (KWH) = 14.35

Contd...

OBSERVATION SHEET NO. 5 (Contd.)

Mean temperature for:

	7 hrs. 30 min.	= 38.7
	next 12 min.	= 36.8
(Defrost Cycle)	last 18 min.	= 39.1
Water spray period		= 7 min.
Run-off period		= 11 min.
Mean outside temperature		= 80° F.

OBSERVATION SHEET NO.6

TEMPERATURES (°F) AT DIFFERENT POINTS WITHIN THE ROOM

TIME	THERMOCOUPLE POINTS					
Hr. Mts.	1	2	3	4	5	6
00.00	35.0		41.5	41.0	41.0	40.0
00.30	34.5		43.0	41.0	42.0	41.5
01.00	34.5		49.0	41.5	43.0	40.5
01.30	34.0		46.5	41.5	42.0	40.0
02.00	32.0	Average Value = 37.5	43.0	41.5	41.5	39.0
02.30	30.5		42.0	40.5	41.5	39.0
03.00	30.0		42.0	40.5	41.5	38.5
03.30	29.0		42.0	41.0	41.5	38.0
04.00	29.0		40.3	39.5	40.5	38.0
04.30	29.0		45.0	39.5	40.5	37.5
05.00	32.0		45.0	41.0	41.0	38.0
05.30	29.5		46.0	40.5	40.5	38.0
06.00	26.5		39.0	39.0	39.5	37.5
06.30	32.0		40.0	38.0	39.5	38.5
07.00	35.0		43.0	42.0	42.0	40.5
07.30	33.0	35.5	43.0	40.5	41.5	40.0
08.00	31.5	37.5	38.0	38.5	39.5	38.5
08.30	32.5	38.5	39.5	39.0	40.0	38.0
08.38	30.0	37.5	41.5	37.5	39.0	37.0
08.43	30.5	37.5	42.0	38.0	38.5	37.0
08.48	33.5	40.0	42.5	40.0	40.0	38.0

Energy to Motor (KWH) = 16.48

Contd.....

OBSERVATION SHEET NO.6 (Contd.)

TEMPERATURES (°F) AT DIFFERENT POINTS WITHIN THE ROOM

TIME Hr. Mts.	1	2	3	4	5	6
08.53	35.0	41.5	43.0	41.5	41.5	39.5
08.58	37.0	43.0	43.5	43.0	42.0	41.0
09.00	37.5	43.0	43.5	43.0	42.0	41.5

Mean temperature for:

	8 hrs. 30 min.	= 39.55° F
	next 8 min.	= 37.40° F
(Defrost Cycle)	last 22 min.	= 39.10° F
Water spray period		= 10 min.
Run-off period		= 12 min.
Mean outside temperature		= 79.5° F.

OBSERVATION SHEET NO.7

TEMPERATURES (°F) AT DIFFERENT POINTS WITHIN THE ROOM

Energy to Motor (KWH) = 20.67

TIME Hr. Mts.	THERMOCOUPLE POINTS 1	2	3	4	5	6
00.00		43.5	44.0	43.0	44.5	39.5
00.30		43.0	43.0	41.0	39.5	36.0
01.00		43.0	43.5	41.0	41.5	36.0
01.30		43.0	43.5	41.5	42.0	36.0
02.00		43.0	43.5	42.0	45.0	36.5
02.30	32.53	42.0	43.0	39.5	44.0	35.0
03.00		42.0	43.0	39.5	39.5	34.5
03.30		42.0	43.0	40.0	41.5	35.0
04.00	"	43.0	43.5	41.0	43.0	36.0
04.30		42.0	43.0	39.5	45.0	35.0
05.00		42.0	43.0	39.5	40.0	34.5
05.30		43.0	43.5	41.0	42.0	36.0
06.00	VALUE	42.0	43.0	39.5	39.5	34.0
06.30		42.0	43.5	40.5	41.0	35.0
07.00		42.0	43.0	39.5	40.0	34.0
07.30		42.0	44.0	40.5	43.0	35.0
08.00	AVERAGE	42.0	43.5	39.5	41.5	34.5
08.30		41.5	43.5	39.5	41.0	34.0
09.00		41.5	43.0	39.5	42.0	34.0
09.30		41.5	43.5	39.5	42.0	34.0

Contd......

OBSERVATION SHEET NO.7 (Contd.)

TEMPERATURES (°F) AT DIFFERENT POINTS WITHIN THE ROOM

TIME	THERMOCOUPLE POINTS					
Hr. Mts.	1	2	3	4	5	6
09.38		41.5	43.5	39.5	42.0	34.0
09.43		42.0	43.5	39.5	43.0	34.5
09.48		43.0	44.0	41.0	43.0	35.0
09.53		43.0	44.5	41.5	44.0	35.0
10.00		44.0	45.0	42.0	44.0	36.0

Mean temperature for:

9 hrs. 30 min. = 39.2°F

next 8 min. = 36.3°F

(Defrost Cycle) last 22 min. = 39.9°F

Water spray period = 10 min.

Run-off period = 12 min.

Mean outside temperature = 80.5°F

OBSERVATION SHEET NO.8

TEMPERATURES (°F) AT DIFFERENT POINTS WITHIN THE ROOM

TIME	THERMOCOUPLE POINTS					
Hr. Mts.	1	2	3	4	5	6
00.00	43.5	43.0	43.0	43.0	44.0	43.5
00.30	35.0	40.0	39.0	42.0	41.0	41.0
01.00	34.0	45.0	45.0	41.0	41.0	39.5
01.30	34.0	45.0	45.5	41.5	41.5	39.5
02.00	34.0	45.0	44.0	41.0	41.5	38.0
02.30	32.5	43.0	44.0	41.0	41.5	38.5
03.00	32.0	43.5	44.0	40.5	41.0	37.0
03.30	30.0	38.0	38.0	38.5	39.5	37.5
04.00	30.5	38.5	38.5	38.5	39.5	37.5
04.30	31.5	40.0	39.5	40.0	41.0	37.5
05.00	33.5	44.5	45.0	41.0	40.0	37.5
05.30	33.0	37.5	40.5	40.0	40.0	37.5
06.00	30.0	40.5	37.5	37.5	39.0	37.5
06.30	31.5	39.5	39.5	39.5	40.5	38.5
07.00	34.0	36.5	36.5	40.0	40.0	37.5
07.30	31.5	39.0	39.0	39.0	39.5	37.5
08.00	30.5	36.5	38.0	38.0	39.0	37.0
08.30	30.5	36.5	39.0	37.5	39.0	36.5
09.00	30.5	36.5	39.0	38.5	39.5	36.0
09.30	30.5	36.5	39.0	37.5	39.0	36.5

Energy to Motor (KWH) = 22.21

Contd....

OBSERVATION SHEET NO.8 (Contd.)

TEMPERATURES (°F) AT DIFFERENT POINTS WITHIN THE ROOM

TIME	THERMOCOUPLE POINTS					
Hr. Mts.	1	2	3	4	5	6
10.00	31.0	37.5	41.5	38.5	40.0	36.5
10.30	32.0	38.5	43.0	39.5	41.0	36.5
10.34	32.0	38.5	43.0	39.5	41.0	37.0
10.39	33.5	39.5	43.0	40.5	42.0	38.5
10.44	34.0	39.5	43.5	41.0	42.0	40.0
10.49	35.0	40.0	43.5	42.0	42.0	40.5
10.54	35.5	40.0	43.5	42.0	42.0	41.0
11.00	36.5	40.5	43.5	42.0	42.0	41.0

Mean temperature for:

10 hrs. 30 min. = 38.7° F

next 4 min. = 38.5° F

(Defrost Cycle) last 26 min. = 39.8° F

Water spray period = 12 min.

Run-off period = 14 min.

Mean outside temperature = 78° F.

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