A NOVEL TECHNIQUE FOR ENERGY SAVING IN FREEZING TECHNOLOGY

BY

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DECLARATION

I hereby certify that this material, which I now submit for assessment on the programme of study leading to the award of Master of Engineering (M.Eng) is entirely my own work and has not been taken from the work of others save and to the extent that such work has been cited and acknowledged within the text of my work.

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CONTENTS

ABSTRACT

In the freezing process, a single evaporating temperature is normally used in the food industry. This work proposes a new technique which uses two evaporating temperatures in freezing process. Based on the Carnot principle, the bigger the temperature difference of refrigeration, the greater the energy costs. The new technique uses a higher evaporating temperature in the first period of the new freezing process. It allows most of the refrigeration capacity to be obtained for less temperature difference. Then the second period employs a lower evaporating temperature which makes the object further cooled to the final temperature required. In this way, the energy saving can be obtained by proper operations and adjustments to refrigerating plants.

The parameters of the novel freezing technique are investigated and optimized using a personal computer. The simulation program is written in Quick Basic. Optimum parameters of the new process are studied for three refrigerants (R22, R502 and R717) and different schemes of industrial refrigeration plants and various objects to be frozen. The industrial implementation of the new technique is discussed. The results show that the novel freezing technique is feasible for industrial application, and the potential of its energy saving is significant.

INDEX IN FIGURES

NOTATIONS

A	Area of heat transfer, m^2
C_f	Specific heat of foods, kJ/kg.°C
DQ	Percentage of change of heat transferred, %
DG	Percentage of change of the refrigerant flow rate, %;
DN	Percentage of change of compressor power in the first period (%);
E	Energy consumption of freezing process, kW.H
E_s	Percentage of energy saving of the new process.
G	Mass flow rate of the refrigerant, kg/s.
H	Enthalpy of refrigerant or food, KJ/kg
HOU	Freezing time (running time of refrigerating plants), hour
HOU_s	Percentage of saving freezing time in the new process.
ΔH_w	Work required by a kilogram of refrigerant in the isentropic compression, kJ/kg
N	Adiabatic power consumed in a refrigeration cycle, kW
P	Pressure of refrigerant (Pa)
Q	Heat transferred or refrigerating capacity, kW.
ΔT	Temperature difference of heat transfer, °C
U	Overall heat-transfer coefficient, W/m^2 K
U_f	Average freezing speed, cm/h.
V	Volume flow rate of the refrigerant, m^3/s
V_E	Volumetric efficiency of compressors.
W	Net work input in a refrigeration process, kJ
c_i	Fitted constants.
k	Ratio of the specific heat of refrigerants.
q	Refrigeration effect or specific heat load, kJ/kg.
t	Temperature (°C) or T (k)
v_1	Specific volume of the refrigerant vapour at the suction condition, m^3/kg
δ	Thickness of the frozen layer, cm.

CHAPTER 1. INTRODUCTION

1.1 Refrigeration and food industry

The origins of the application of low temperature as a means of food preservation are uncertain. Records show that the Chinese are the first known civilization to harvest winter's ice and store it(1), in special cellars or packed in straw or dried weeds, for use in the summer months. Specially constructed cellars were also found in the Mediterranean, Minoan and elsewhere, before 2000 years ago(2). From 1805 to the end of the 19th Century, there was an important trade in natural ice in the USA and in Europe. For example, 225,000 tonnes of natural ice were shipped from North America to other places in 1872. But the lowest temperature could only reach 0°C by using ice. In 1842 a patent was granted to H.Benjamin in England for his method of freezing foods by immersion in a mixture of ice and brine(3), which can obtain a temperature under 0°C.

With the development of mechanical refrigeration, extensive application of the cold preservation of foods became possible. Professor Linde of Munich pioneered the application of thermodynamic principles to refrigeration(4) which led to the building of an ammonia compression machine in 1873. The first example of the use of refrigeration machinery for the transportation of food occurred in 1879. An air compression machine designed by Coleman of

Glasgow was fitted on the liner "Circassia" which brought successfully a cargo of chilled beef from America. The Russians were probably first to freeze fish by mechanical refrigeration, in 1888 at Astrakhan. Compressors were installed on a barge which was towed up and down the Volga, and on ships fishing in the Caspian Sea.

It is common knowledge that production of raw food is seasonal in nature. Generally speaking, food has to be transported because there is usually a distance between the production point and the place of processing as well as consumption. Major food, such as meat, fish, dairy products, fruit and vegetable, are easy to spoil. But these food must be available for consumers all year round. As a result, it is necessary to have some measure to store and keep the raw materials safely so that the raw materials are continuously available in sufficient quantity and quality. Refrigeration enables the processing of food to operate on a planned year-round basis. In this way, the products can be supplied to consumers all year round.

Food being stored may become spoiled in three ways:

(a) Living organism (e.g. vermin, insects, fungi or bacteria) may feed on the food and contaminate it.

(b) Biochemical activity within the food itself (e.g. respiration, staling, browning and rancidity development) may in time diminish its quality and usefulness.

(c) Physical process (e.g. bursting and spillage of the contents of package or recrystallisation phenomena in sugar confectionery, fats and frozen products) may have the same effect.

The three main factors of the storage environment which influence the storage life of a particular commodity are the temperature, humidity and the composition of the store atmosphere. The most important factor of them is temperature.

It is well known that temperature controls the rate of all physicochemical reactions and as a consequence temperature has a profound effect on the biological system[5]. The rate of growth of bacteria is reduced as the temperature falls. Under the low temperature storage (particularly frozen storage), bactericidal effect will be arrested or retarded. Fungi like bacteria, have a temperature range over which the growth is possible at a given water activity and an optimum growth temperature within this range. This optimum is nearer the upper end of the range and at, or above, the normal ambient temperature. Near the limiting temperatures, the growth takes place only very slowly. Figure 1-1 shows this feature. So again it may be said that lowering the temperature of a product will reduce its rate of deterioration. Most insect activity is inhibited below about 4°C, although some insect species and insect eggs are capable of surviving long exposures to these temperatures.

In addition, the rate at which biochemical reactions occur in food increases with increasing temperature. The relationship observed between the reaction rate and temperature is frequently similar to the one between the growth of bacterial spores and temperature. That is, the logarithm of the reaction rate is a linear function of the temperature. Hoff[6] found that the reaction rate approximately doubled for each 10°C temperature rise for many chemical reactions. Thus the lower the storage temperature, the more slowly do foods suffer degradation by those biochemical spoilage reactions mentioned in the above.

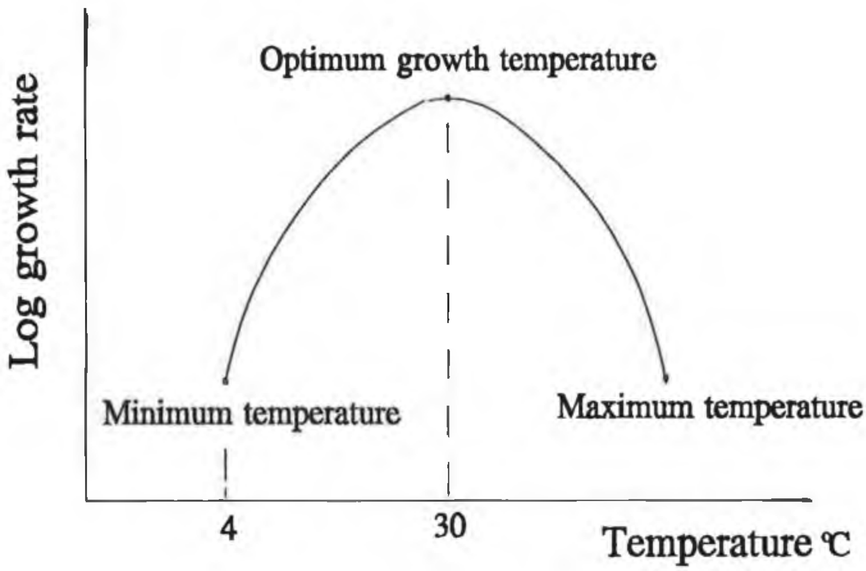


Fig.1-1 temperature and bacteria growth rate

Refrigeration enables us to create and maintain a temperature which is under that of ambience. It extends the period of consumption for seasonal perishable foods and simplifies their distribution. All the processing methods of food will change the appearance and flavour of food except refrigeration. Therefore, refrigeration is widely used in the food industry all over the world. For example, chill temperature, a temperature near the freezing point of food (-1.1 to -1.7°C), can be obtained by means of refrigeration. Chill temperature greatly reduces the growth of all microbes. This permits substantial increase of the stored time and many foods can be kept without spoilage. This is especially useful for those processed foods that will not withstand the high temperature required for sterilization - fluid milk, cream, sour-cream, butter milk, yogurt, cottage cheese, red meat, chicken, turkey, fish, shellfish, and certain cured meats. Although the chill temperature provides only limited shelf life for these products, it is sufficiently long (a few days to a few weeks) and the cost is sufficiently reasonable that the consumer can get these highly acceptable foods all the year round.

The major application of refrigeration historically has been in the area of food preservation, but there are also important uses in air conditioning, in industrial processing and in heat pump technology. During the past 100 years, refrigeration has become indispensable to our society. Nowadays not only is food preserved in our homes, but commercial preservation of food is one of the most important current applications of refrigeration. Commercial preservation and transport of food using refrigeration is so common that it would be difficult to imagine an un-refrigerated modern world. For example, more than three-fourths of the food that appears on American tables every day is produced, packaged, shipped, stored, and preserved by refrigeration[1]. In the UK, only about one-fifth of the food bought by the consumer is fresh with simply packaging, the four-fifth is processed and the majority of them need to be refrigerated[7, 8].

It is safe to say that present standards of living are as dependent on the availability of refrigeration as they are on the availability of pure water and abundant electrical power. The applications of refrigeration as well as its world markets are experiencing rapid growth, and they will continue to expand as all the countries in the world are improving their standard of living and engage in more industrial development.

1.2 Energy saving and refrigeration

As a result of the wide application of refrigeration, a great amount of energy is required by the food industry. A study[9] reveals that food industry has been an important section of energy consumption in many countries, while the food chain consumes a very substantial proportion of total primary energy requirement for a country. For example, the proportion is 15% for Australia, 28.6% for Ireland, 15.8% for the UK and 12-15% for the USA[10]. Therefore, energy savings on refrigeration has been an important topic in energy studies, ever since the energy crisis took place in 1970s.

In fact, the energy problem has the dimensions of a crisis and has come about for two reasons: one is the growing demand for energy because of the world economic development , and the other is the growing world population (it will probably be close to 6.5×10^9 people by the year 2000 if the present rate of growth continues[11]).

On the other hand, at the present rate of consumption, known oil and gas reserves will not last beyond 50 years from now[12], while coal has an expected usage of some 250 years left. Therefore, energy saving (energy efficiency) and the development of new energy sources are the only solutions to this serious problem. By means of energy saving, reducing energy consumption does not necessarily imply reducing the quality of life. An increase in the efficiency of use can be equivalent to an increase in supply.

In the UK in 1987 the installed refrigeration capacity is estimated to be 1366 MW, and the total electrical energy consumption of refrigeration system is estimated to be about 118 million GJ/year (368 million GJ/year primary energy). And according to a report from Energy Technology Support Unit[13], approximately 20-25 per cent of the total energy consumption of refrigeration plant in Britain could be saved. There are more than 250 MT of foods needed to be refrigerated every year in the world. Therefore, there is a great potential for energy savings in refrigeration.

There are various methods available for reducing refrigeration energy consumption in the food industry. According to the study reported by Energy Technology Support Unit, the electricity consumption for refrigeration on the food industry is 1030 million kwh in Britain every year. This means some 200-250 million kwh of electricity per annum could be saved[13]. Potential methods of energy saving by technique could be arranged in three categories:

-- Improvement in refrigeration machinery and equipment design.
-- Improvement in the design of refrigeration systems.
-- Maintenance and operation of refrigeration plants.

(a) Improvement in the machinery and equipment design

That means that efficient machines and equipment are developed as the new techniques and studies are developed. Clearly, the duty is carried out by the manufacturer. This will not be dealt with in this study.

(b) Improvement in the design of refrigeration system

Many purchasers of refrigeration plants base their buying decision largely on capital cost rather than taking into account the total life cycle costs. some of the more important design options which can improve energy efficiency are as follows:
-- Variable speed compressor, in preference to a system of unloading certain

cylinders on a single speed compressor.

-- Integrated plant layout, rather than modular layout, so that the use of available heat exchanger surfaces can be maximised.

-- Subcooling to increase the effective capacity of the plants, preferably where the subcooling effect can be obtained at no extra cost.

-- Plant sizing to ensure that the lower capital cost of smaller evaporators and condensers is balanced against the higher running costs incurred with reduced heat exchange surfaces.

-- Compressor selection to ensure that the most appropriate compressor type is selected for the size of refrigeration unit and its application.

-- Using electronic devices which can control refrigeration plants more effectively and more efficiently. There are now a number of devices available , such as maximum demand control and temperature control devices.

-- Microprocessor control system, which automatically assesses how much compressor power is required at any time, using information fed back from different temperature sensors.

Generally speaking, the above features offer payback of under five years if incorporated in the original design specification. Incorporation on a retrofit basis usually leads to longer payback.

(c) Maintenance and operation of refrigeration plants

Regular maintenance of refrigeration plant plays a major part in ensuring that the plant continues to function as efficiently as it was designed to. The most important items to check in routine maintenance are as follows:

-- Purging of the system to remove air and other non-condensable gases.

-- Draining of oil from evaporators and condensers to protect heat exchange surfaces from oil fouling.

-- Clearing away fouling of condensers.

-- Removing ice build-up from evaporator coils. An optimised defrost schedule is also dependent on proper design.

Proper maintenance of refrigeration plants can achieve energy saving at little or no capital cost. It is very clear that a fundamental prerequisite for effective plant maintenance is the education and training of maintenance staff and others responsible for operating refrigeration plants.

1.3 Analysis of energy consumption on cold chains

From the point of view of operation, the methods which are mentioned above are practical. But they can not display which link is of more potential for energy saving. In order to evaluate the opportunities for improvement of process efficiency, it is necessary to examine the stages of refrigeration processes, so that the stage can be identified at which there is greater potential for energy saving.

The whole process of refrigeration of foods from raw materials at the point of production to consumers could be divided into four stages - cooling, freezing, refrigerated storage, and refrigerated transport. We will inspect each link and its energy consumption in the following sections. It should be mentioned that some foods may not undergo all the four stages.

1.3.1 Cooling of foods

The cooling of food is defined as a process(14) which makes the temperature of foods to decrease to a temperature just above the freezing point of foods. Cooling can prolong the storage duration of the foods. It is carried out by following methods:

-- Air blast in tunnels or cold rooms. It is used for fruits, vegetables, meat, poultry and fish, and seldom for liquid food.
-- Cooling simply by broken ice. It is used for vegetables and fish.
-- Cooling by spraying cold water or by immersing in cold water. It is used for some vegetables and nuts, also used for poultry and fish.
-- Vacuum cooling. It is used mainly for vegetables.
-- Liquid / liquid heat exchange. It is applied to liquid foods, such as milk and drink.

Among all the methods, air blast is most commonly used(15, 16).

It is easy to understand that the cooling process involves the removal of thermal energy from a food product in an effort to reduce the temperature to some level below that of ambience as well as to reduce the associated deterioration reaction occurring in the product. It should be noted that the heat removed from the foods during a cooling process is only in a form of sensible heat. After cooling, products can be placed in refrigerated storage in order to ensure that the temperature is maintained at some desired level. The cooling process occurs during the first stage of thermal energy removal as described by the temperature-history curve in Figure 1-2. As illustrated, the sensible heat is removed during the linear portion of the temperature-history curve, with the product specific heat dictating the slope of the curve[17].

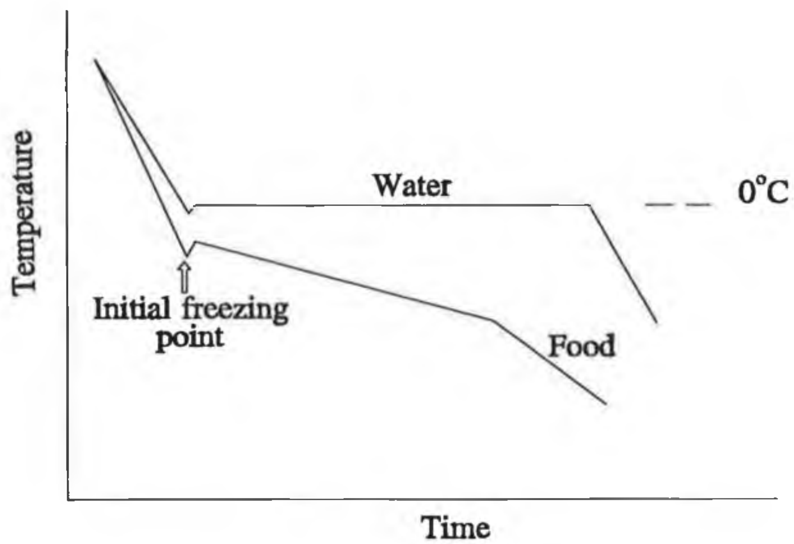


Fig.1-2 Comparison of freezing curves for pure water and food

The enthalpy is a term which is commonly used in refrigeration engineering. It means the heat content per unit mass[18]. The enthalpy change in moving from one temperature to another is the quantity of heat to be removed (or added if the temperature is raised) to effect this same change in temperature of the food. Enthalpy of some foods[19] is shown in Table 1.

Table 1
Enthalpy of food under different temperatures

Temperature °C	Enthalpy KJ/Kg			
	Beef	pork	Lamb	fish
-18	4.6	4.6	4.6	5.0
-16	10.0	9.6	9.6	10.9
-14	15.9	15.1	15.5	17.4
-12	22.2	21.4	21.8	24.5
-10	30.2	28.9	29.7	33.1
-8	39.4	37.3	38.5	42.9
-6	50.7	47.3	49.4	55.5
-5	57.4	54.5	55.7	67.9
-4	66.2	62.0	64.5	76
-3	75.4	73.7	77.1	87.4
-2	98.9	91.8	96.0	114.6
-1	186	170.1	179.8	206
0	232.5	212.0	224.2	258
5	248.5	227.1	240.1	265
7	255.2	233.4	246.4	282
10	264.8	242.2	255.6	292
20	297.1	272.8	287.0	327
35	345.4			

According to Table 1, it is easy to determine the stage at which energy saving has more potential. Suppose that some beef needs to be refrigerated from 20°C to -18°C. The temperature of the beef drops from 20°C to 0°C during a cooling process. From Table 1, the heat rejected in the cooling process is

$$297.1 - 232.5 = 64.6 \quad \text{kJ/kg}$$

and the heat rejected in the whole process from 20°C to -18°C is

$$297.1 - 4.6 = 292.5 \text{ kJ/kg}.$$

Therefore, the heat rejected (or the refrigerating capacity required) during the cooling process accounts for

$$64.6/292.5 \,\% = 22\%$$

of the total heat rejected in the whole refrigeration process from 20°C to -18°C.

1.3.2 Freezing of foods

Most fresh foods of animal or vegetable origin possess a cellular structure, and the cells usually contain a large amount of water together with a complex mixture of organic and inorganic substances. When the foodstuff is cooled and sufficient thermal energy is removed to reach the initial freezing temperature, ice crystals will form within the product. As illustrated in

Figure 1, the initial freezing temperature for a food product is always lower than the freezing temperature of water. This depression of the freezing temperature for water is the result of soluble solutes in the product liquid phase. In addition, the characteristic shape of the temperature-history curve during freezing is the result of continuous concentration of the soluble solutes within the product. Since the concentration of solutes occurs as the freezing process continues, the formation of ice crystals and the corresponding removal of latent heat occurs over a range of temperatures. During typical food freezing, this gradual removal of latent heat will continue until the freezing process is completed and a small fraction of unfrozen water remains in the product. The temperature-history curve during product freezing is illustrated by the gradual and non-linear temperature decline to the final product temperature.

In other words, freezing commences when the temperature reaches just below 0°C, when the liquid in food will become solid, and a typical phase change takes place. It is well known that a large amount of energy exchange must accompany a phase change of materials, hence a large amount of latent heat of fusion in addition to sensible heat must be removed in order to produce further cooling.

The amount of heat extracted during a process of phase change is much greater compared with a process of sensible heat change. For example, only 34 kJ of heat is extracted when the temperature drops from 10°C to 0°C for one kilogram of fish, but 170.6 KJ of heat is rejected when the temperature decreases from 0°C to -3°C for the same quantity of fish. Let us use the same example at section 1.3.1, the temperature of beef drops from 20°C to -18°C, the heat rejected from the beef during the freezing process (from 0°C to -18°C) accounts for

$$[(232.5 - 4.6)/292.5]\% = 78\%$$

of the total heat rejected in the whole refrigeration process from 20°C to -18°C.

It is clear that there is a great difference between the freezing and cooling of food according to the example. The cooling only cools the food to a temperature above its freezing point. There is no latent heat change during the cooling of food. A greater refrigeration capacity must be employed when the freezing is required. From Table 1, three times refrigeration capacity is needed in order to freeze beef from 35°C to -18°C, compared with cooling beef from 35°C to 0°C.

Up to the present, it is easy to understand that the energy consumption in a freezing process of food accounts for a major proportion in cool processing of foods since a large amount of refrigeration capacity is employed. In some plants the energy consumption in freezing process accounts for 80% of the total energy consumption[20]. Therefore, there is a great potential for energy saving in the process of freezing of food. Particular attention must be paid to the freezing of food.

1.3.3 Refrigerated storage

Refrigerated storage is a process which maintains a temperature below the ambience in a space (cold store). The temperature depends on foods as well as countries. The range of temperature is 1°C to 5°C for cooled foods and -18°C to -30°C for frozen foods[15,21].

Refrigeration capacity required by decreasing temperature of foods is defined as product load. The product load may consist of one or more of the following:

(a) Sensible heat, when the foods only have to be lowered in temperature.
(b) Latent heat, when crystallization or freezing takes place.
(c) Respiration heat, when living matter like fruit, plants and vegetables are involved.

For refrigerated storage of frozen foods, there will be in fact no product load if the temperature of frozen foods entering a storage room is the same as that of the storage room. In other words, refrigeration capacity of the refrigeration plant in this case is only used to resist the following heat:

(a) Heat entering the room by heat transfer through walls, ceiling and floor.
(b) Heat of air infiltration, it results from fresh air entering from the outside of cold rooms.
(c) Miscellaneous heat, which is brought into the cold store by people, electric

motors in cold store and electric lights.

Heat energy by thermal transmission is dependent on insulation materials of the cold store. The heat will not be too much if proper insulation is applied to the cold store. Heat of air infiltration could be limited if attention is paid to the open time of doors. Generally, one may assume that roughly 2% of the total heat load comes from miscellaneous heat[(2)].

Therefore, it is easy to infer that refrigerating capacity required in the refrigerated storage is far less than that of freezing of foods. It should be mentioned that attention should still be paid to the storage, since refrigerated storage involves a rather long duration (it may last for one year for some foods) and it could consume a great amount of energy.

1.3.4 Refrigerated transport

The energy consumption in refrigerated transport consists of two sources:

-- Energy consumption by transport, the energy consumption for refrigerated transport is the same with that for common transport. It will not be discussed in this study.

-- Calculation shows that energy consumption due to refrigeration of goods during transport part is only 10-15% of transport energy[(22)].

The purpose of refrigeration during transport of goods is to maintain a low temperature for foods. In fact, the equipment in refrigerated transport (refrigerated vehicles or ships) are movable cold storage. Therefore, refrigeration capacity required in refrigerated transport is similar to that in common refrigerated storage of foods. But the energy consumption of refrigeration at this stage is smaller, since the duration of transport is fairly short, generally from several hours to few days.

According to the discussion above, a conclusion appears - the freezing of foods requires major refrigerating capacity among the four refrigerated stages. As a result, a large amount of energy is consumed in this operation. Therefore, it may offer unique opportunity for process efficiency improvement.

1.4 The new idea and layout of this project

As stated above, refrigeration has been and is being widely employed in food industry all over the world, especially in the developed countries. As a result, a large amount of energy is consumed in refrigeration. Therefore, one challenge comes to refrigeration engineers - how to save energy consumption on refrigeration.

Energy saving on refrigeration is not only a link of world's energy saving under energy crisis, but also a great benefit for the owner of refrigeration plants as well as the consumer of frozen foods, because the costs of energy will be reflected directly in the cost of refrigerated or frozen food.

The objective of the present study is to improve the process efficiency and reduce the energy consumption in refrigeration of foods. At the same time, attention is also paid to the quality of product, because any improvements to process efficiency must be accomplished in a manner that results in product quality improvement. At minimum, optimization techniques should be utilized to assure maximum process improvements without reduction in product quality.

From the analysis in the last section, it can be seen that energy saving has a great potential in freezing processing. In any application two temperatures are fixed - the temperature of the refrigeration application (the refrigerant must evaporate at a lower temperature), and the condenser coolant inlet temperature (refrigerant must condense at a higher temperature). Single evaporating temperature is normally applied in industrial refrigeration plant. The evaporating temperature chosen depends on the final temperature of frozen food. But the initial food temperature is generally very high (ambient temperature or higher). This means that refrigeration cycles run over a large temperature difference and all heat is moved over this large temperature difference. According to the Carnot principle, the bigger the temperature difference of refrigeration cycle, the greater the energy costs. Therefore, it is possible to use two evaporating temperatures in the freezing process of foods.

At the start of the freezing process, a higher evaporating temperature can be used, thus a great part of the refrigeration capacity can be achieved at a smaller working temperature difference. Then the second evaporating temperature is employed, and the food is further cooled to the final temperature required. In this way, the energy saving can be obtained by proper operation and adjustment to refrigerating plants.

In order to achieve this goal it is necessary to understand the mechanical refrigeration and current common freezing process of foods. Therefore, basic laws and schemes on mechanical refrigeration are reviewed in Chapter 2. The common freezing processes of foods are also introduced.

Chapter 3 gives a detailed analysis of the new idea which employs two evaporating temperatures in freezing process. The following problems are discussed: the principle of energy saving using two evaporating temperatures, the feasibility of using two evaporating temperatures in industrial refrigeration plants, heat load during freezing of foods, and the quality of product in the new freezing process.

There are many different schemes in vapour compression refrigeration cycle in practice. Chapter 4 describes different schemes in industrial refrigeration plants. The features and arrangements of equipment of the schemes are also discussed.

Chapter 5 proposes the mathematical model of refrigerants, and equations for simulation. Three refrigerants (R22, R502 and R717) are discussed. This Chapter also reports the determination of fundamental parameters on refrigeration cycle. Finally, a flow chart and brief statement of the simulation program are proposed. A IBM PC is employed in the simulation.

One of the main objectives in this study is to determine the optimum evaporating temperature t_{e1} in the first period of the new freezing process. Chapter 6 discusses the optimum t_{e1} in different cases based on the results which are obtained from the simulation programs. The cases discussed are as follows: optimum t_{e1} with the variation of condensing and evaporating temperatures, optimum t_{e1} with the variation of different refrigerants, optimum t_{e1} with the variation of different foods.

One of the important features on the new freezing process is that the new process utilizes the same refrigeration plant of the common freezing process. The performance of the refrigeration plant will vary with t_{e1}. Chapter 7 analyses the remaining problems of refrigeration system under the new process. The problems discussed are as follows: the influence of the condensing load on the condenser performance, the influence of refrigerating capacity on evaporator performance, and the influence of the refrigerant flow rate on the performance of expansion devices. The performance of compressors is also investigated in this Chapter.

Chapter 8 outlines the conclusion of the study. It also presents some general considerations on the implementation of the new freezing process in practice. Some suggestions for future work are also proposed.

CHAPTER 2. REFRIGERATION METHODS AND FREEZING OF FOODS

2.1 Mechanical refrigeration and thermodynamics

Refrigeration, commonly spoken of as a cooling process, is more correctly defined as the removal of heat from a substance to bring it to or keep it at a desirable low temperature, which is below the temperature of the surroundings. Thermodynamics is the science of the relationship between heat, work, and the properties of systems. It is concerned with the means and efficiency of conversion between heat energy and work. The principles of refrigeration are based on thermodynamics. The laws of thermodynamics are natural hypotheses based on observation of the world in which we live. It is observed that heat and work are two mutually convertible forms of energy, and this is the basis of the First Law of thermodynamics. It was developed by scientists in the early part of the nineteenth century, became known as the principle of the conservation of energy. It is also stated in the following way -- energy can be neither created nor destroyed.

It is also observed that heat never flows unaided from an object at a low temperature to one at a high temperature, in the same way that a river never flows unaided uphill. In 1824, Carnot published a short pamphlet entitled 'Reflections on the motive force of fire'[23]. The principle expounded was later recognized as the Second Law of thermodynamics, which shows that a heat engine cannot convert all the heat supplied to it into mechanical work but must always reject some heat at a lower temperature. The importance of Carnot's concepts was not fully appreciated until later in the nineteenth century. In fact, the efforts in energy conservation have sometimes been haphazard, partly due to a lack of understanding of the Second Law of thermodynamics. Nowadays great use is made of the First and Second Law of thermodynamics in refrigeration engineering.

For a vapour compression refrigeration plant, it takes in heat from fluids or solids to be refrigerated, it takes in work via the shaft of the compressor, and it gives out heat in the condenser to the water or air. The block diagram from a work-operated refrigeration plant is shown in Figure 2-1. Disregarding incidental heat gains and losses from the pipes, the principal transfers are : (1) acceptance of heat Q_1 , from bodies or fluids to be refrigerated at a relatively lower temperature t_1; (2) acceptance of work W, to drive the plant, either mechanical or electrical depending on whether the motor is supposed to be enclosed in the box or not; and (3) rejection of heat Q_2, to cooling water or air at a temperature t_2 which is higher than t_1.

Where t_1 or T_1 is the temperature of a hot sink, t_2 or T_2 is the temperature of a cold source. Heat Q_1 is taken from bodies or fluids at a low temperature t_1. Work W is taken and heat Q_2 is given out at the higher temperature t_2. For a refrigerating plant, t_2 is the temperature of the environment, cooling air or water, to which heat can be rejected. For a heat pump, t_1 may be the temperature of the environment, air or water, from which heat can be accepted.

For a continuously operating system of this kind, the First Law of thermodynamics states that if the heat and work transfers are measured over a period of time for which the contents of the control space have exactly the same state at the beginning and the end[3, 24], then:

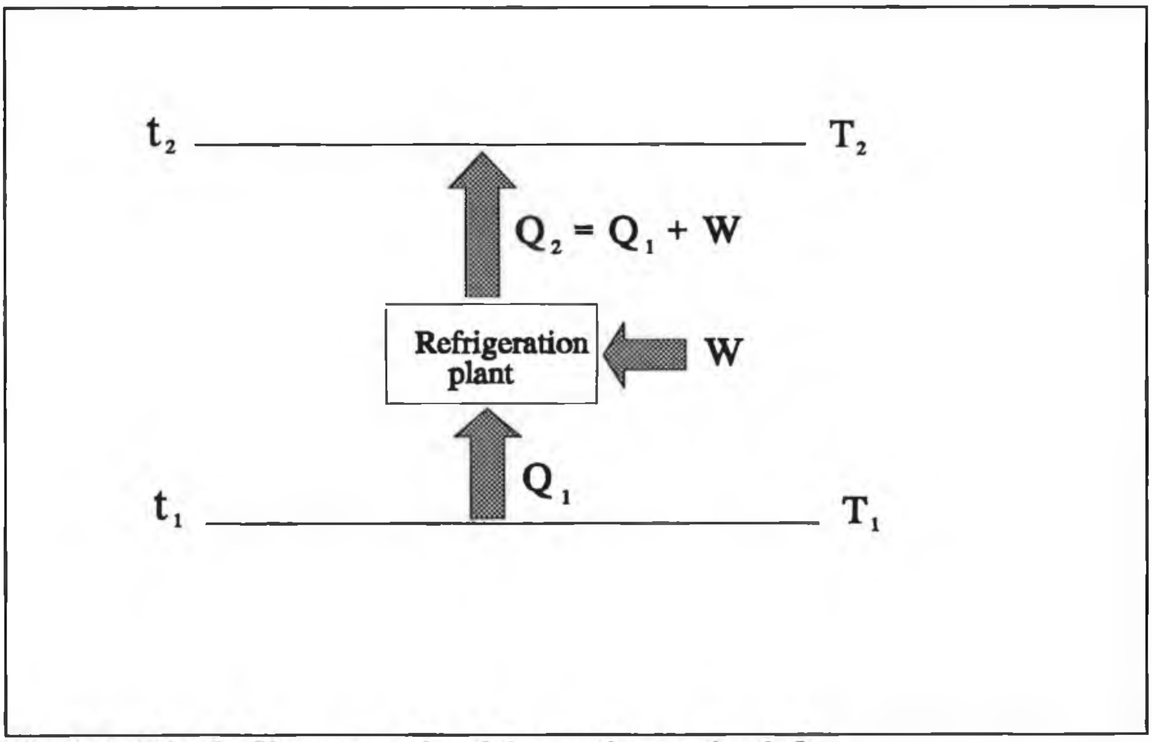


Fig.2-1 Block diagram of refrigerating principle

$$Q_1+W=Q_2 \tag{2-1}$$

The Second Law of thermodynamics, in the form due to Clausius[4], states essentially that the work W cannot be zero. But it goes further than this and shows that there is a certain maximum coefficient of performance, Q_1/W, which cannot possibly be exceeded without contravening the Law. Carnot showed that the maximum possible efficiency of an engine working between two temperatures depends only on these temperatures, and by a similar argument it can be shown that the maximum possible coefficient of performance of the work-operated refrigeration plant depends only on the temperatures t_1 and t_2. It is independent of the manner in which it works and the material used as refrigerants. Consequently this ideal refrigeration plant can act as a thermometer and tell us something about the relation of one temperature to another. For practical reasons the absolute thermodynamic temperature scale, on which temperatures are denoted by upper case T, is defined by the equation:

$$\frac{Q_1}{T_1}=\frac{Q_2}{T_2} \tag{2-2}$$

where Q_1 and Q_2 are the heat taken in and rejected at absolute temperature T_1 and T_2, respectively, by this ideal machine.

The performance of a refrigeration plant is defined by means of the coefficient of performance (COP), which is given by[25]:

$$COP=\frac{Q_1}{W} \tag{2-3}$$

where Q_1 is the heat taken in (or refrigeration capacity), and W is the net work input. The COP provides a measure of the energy efficiency of the system. Because one always wishes to have the greatest refrigeration capacity with the smallest expenditure of power, the largest practical value of the COP is desirable. The definition of absolute temperature immediately gives a value for the maximum possible coefficient of performance of a refrigeration plant operating between T_1 and T_2. Combining Equations 2-1 and 2-3, the COP is :

$$COP=\frac{Q_1}{W}=\frac{Q_1}{Q_2-Q_1} \tag{2-4}$$

and in the ideal case Q_1 and Q_2 are related by equation (2-2). Thus the maximum possible coefficient of performance is :

$$COP_C=\frac{T_1}{T_2-T_1}=\frac{1}{\frac{T_2}{T_1}-1} \tag{2-5}$$

The subscript "c" for Carnot is used here to denote the maximum possible value of COP, because Carnot proposed one hypothetical cycle of operations whereby this maximum value could be realised in principle.

The temperatures t_1 and t_2 or absolute temperature T_1 and T_2, are the temperatures of the body being refrigerated and of the medium to which heat can be rejected respectively. The temperatures of the refrigerant itself of course have to be different from these. The evaporating temperature t_e, for instance, has to be lower than t_1, and the condensing temperature t_c has to be higher than t_2. Consequently, the COP of a real refrigeration plant will always be less than COP_c. Nevertheless, Equation (2-5) is of fundamental importance for understanding the work requirements of refrigeration plants because the coefficient of performance of a real plant varies with the temperature in a similar manner.

2.2 Common refrigeration method - vapour compression cycle

Means of producing refrigeration with machinery, called mechanical refrigeration, began to be developed in the 1850's[3]. Today the refrigeration industry forms a vast and an essential part of any technological society. The methods of mechanical refrigeration can be classified as :

(a) refrigeration by means of phase change of materials (including vapour compression, absorption and steam jet refrigeration).

(b) Refrigeration by means of expansion of gas.

(c) Semiconductor refrigeration.

(d) Vortex-tube refrigeration.

Each method of refrigeration has its special feature. The most widespread method of mechanical refrigeration is the vapour compression system. Other methods are used only in special applications and their functioning will not be deliberated here. According to a survey[13] in 1985, over 99 per cent of UK refrigeration capacity, from domestic refrigerators to large industrial systems, operates on the vapour compression cycle and is electrically driven. Therefore, the investigation will be focused on vapour compression refrigeration in this project.

2.2.1 Simple vapour compression refrigeration cycle

The principles of vapour compression refrigeration can be explained as follows. First, every liquid exerts a vapour pressure, i.e. it has a tendency to turn into vapour. A liquid in a container empty of all gases but its own vapour comes to equilibrium with the vapour at a value of the pressure known as its saturated vapour pressure, or vapour pressure for short. The vapour pressure increases with temperature. If the vapour is removed from the container, some of the liquid will evaporate to replace it. The second principle comes into operation at this point. A liquid when it turns into vapour needs heat, called 'latent heat'. If enough heat is not forth coming from external sources, it must be obtained from the part of the liquid which does not evaporate, and the temperature of this part falls.

By increasing the pressure of the vapour, i.e. by compressing it, the vapour can be made to condense at a higher temperature, and in fact at a temperature high enough for it to give out heat to the ambient air, or to water at about the ambient temperature. The function of the compressor in a refrigeration system is to draw vapour from the evaporator, so maintaining a low pressure therein, at which the refrigerant can keep boiling at the desired temperature; and to raise the pressure of the vapour and deliver it to a condenser where this vapour can be condensed by the water or air.

In vapour compression refrigeration systems, fluids which absorb heat from cold source and release it to hot sink are called refrigerants. These fluids, in their liquid form, under a reduced pressure, absorb heat in the evaporator and, in absorbing heat, change step by step to a vapour. In their vapour form, the fluids are taken into the compressor where the temperature and pressure are increased, so that the heat absorbed in the evaporator can be released in the condenser, and the refrigerant is then returned to a liquid form for another cycle.

Figure 2-2 shows the arrangement of equipment and interconnecting piping for a simple vapour compression system. Typical operating conditions have been selected in order to make the discussion more practical.

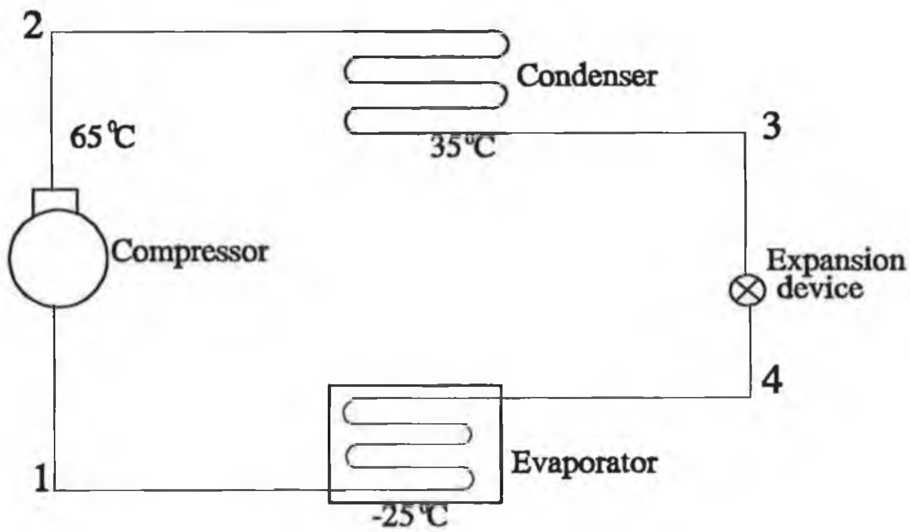


Fig.2-2 A simple vapour compression system with example of possible operating conditions

The refrigeration cycle in Figure 2-2 consists of four processes:

The process 3-4 through the flow control device. The fluid in this system is refrigerant R12. Liquid refrigerant R12 at 8.5 bar and 35°C, at point 3, enters the expansion device. The expansion device has a narrow opening, which results in a large pressure loss as the refrigerant flows through it. The refrigerant leaves at point 4 at 1.24 bar. This pressure is below the saturated pressure at point 3 due to the pressure loss. Some of the liquid refrigerant immediately flashes to vapour. The portion of the liquid takes the latent heat required for its evaporation from the flowing mixture, thus cooling it. The refrigerant leaves the valve as a liquid-vapour mixture in the saturated state. The saturation temperature for R12 at 1.24 bar is -25°C.

The process 4-1 through the evaporator. The temperature of substance to be cooled, say -15°C, is higher than that of refrigerant in the evaporator. Therefore heat will flow from it through the tube wall of evaporator to the refrigerant. Because the liquid refrigerant in the evaporator is at its boiling point(saturated temperature), its temperature does not change while it gains heat, just as water boils at 100°C.

The process 1-2 through the compressor. The compressor draws the vapour from the evaporator to its suction side and then compresses it to a suitable high pressure for condensing, 8.5 bar in this case. Work is required to compress the gas, coming from a motor or engine that drives the compressor. This work is converted into an increase in stored energy of the compressed vapour, resulting in a rise in its temperature. The refrigerant leaves the compressor at 65°C in this example, at point 2, in a superheated condition.

The process 2-3 through the condenser. The high pressure gas discharged from the compressor flows through the condenser. Heat will flow from the higher temperature refrigerant through the tube walls of the condenser to water or air at ambient temperature. Since the refrigerant is superheated when it enters the condenser, it will first be cooled until it reaches its saturated temperature, which is 35°C in this example. Further removal of heat results in gradual condensation of the refrigerant, until it is all liquidized. Then the liquid refrigerant leaves the condenser as a saturated liquid at 35°C, at point 3. As the processes repeat, refrigeration will continue.

2.2.2 Two stage compression cycle in industrial freezers

In many applications of refrigerating plants such as air conditioning, ice making, chilled storage of foods, etc. the difference between the temperatures in the condenser and the evaporator is not greater than about 40°C, and the simple (basic) vapour compression system is usually adequate for this purpose. There are other circumstances, however, in which a much greater temperature difference is desirable. These are mainly of two kinds: either because an unusually low temperature is required in the evaporator, or because an unusually high temperature is required in the condenser. Examples of the first kind occur in food freezing, where an evaporating temperature of -40°C is common with a condensing temperature of 30-40°C; and in chemical industry, where temperatures down to -100°C are often desired. Examples of the second kind occur when the refrigeration plant is used as a heat pump with a condensing temperature of about 70°C.

As the difference between temperatures in the condenser and the evaporator increases, several factors cause reduction in the refrigerating capacity and in the COP which eventually become unacceptable. For example, if the ambient temperature is 25°C and the evaporating temperature is -10°C, the COP_c of the cycle is 7.5, while if the condensing temperature is the same but the evaporating temperature drops to -40°C, the COP_c of the cycle is only 3.58, according to equation (2-5). Moreover, as the temperature difference between t_c and t_e increases, so does the temperature of the hot gas leaving the compressor. This is because more energy is required to compress the gas over a greater pressure range (temperature difference). This increases the heat of compressed refrigerant vapour. Therefore, the working condition of refrigeration compressors will deteriorate as the evaporating temperature decreases. When the temperature difference is over some limit, compressors will simply not be able to operate. For example, when t_c is 30°C and t_e is -40°C, the temperature of the discharge from the compressor will be about 170°C for ammonia refrigerant. The lubrication oil in the refrigeration compressor will be burned at this temperature. According to the typical properties of refrigeration oils(25), discharge temperatures of 150-165°C are considered the top limit.

The answer to these problems is the adoption of one of the two types of systems - multi-stage compression system and cascade system. The cascade system is only used at an evaporating temperature below -60°C(26), and it will not be considered here. In theory, one can select many stages in multi-stage compression. But a refrigeration plant will get complicated as the number of stages increases. Therefore, only two stage compression is used in industrial freezing plants[27,28,29].

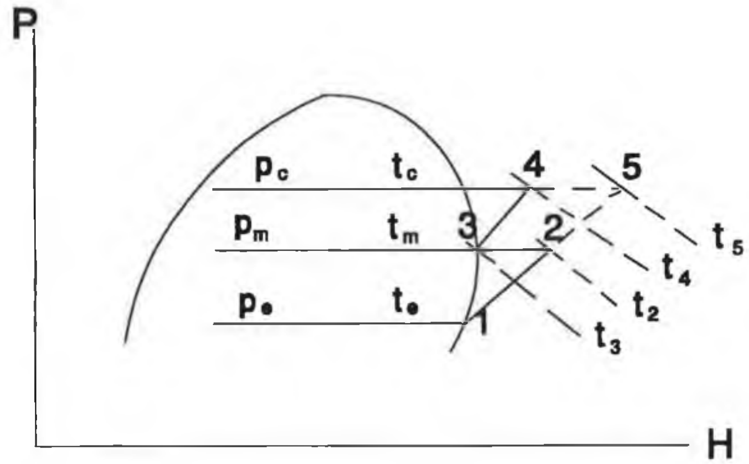


Fig.2-3 Pressure-enthalpy diagram of two-stage compression

A two stage compression system means that the suction vapour at evaporating pressure p_e is first compressed to a certain intermediate pressure p_m with low stage compressors. Then it is followed by cooling at a constant pressure as close as possible down to the corresponding saturation temperature t_m and then further compressed to the condensing pressure with high stage compressors. Figure 2-3 is the pressure-enthalpy diagram of a two stage system. It shows clearly that because of vapour interstage cooling, not only the low pressure discharge temperature t_2, but also the final discharge temperature t_4 of the high pressure stage are lower than the corresponding temperature t_5 for single stage compression under same conditions.

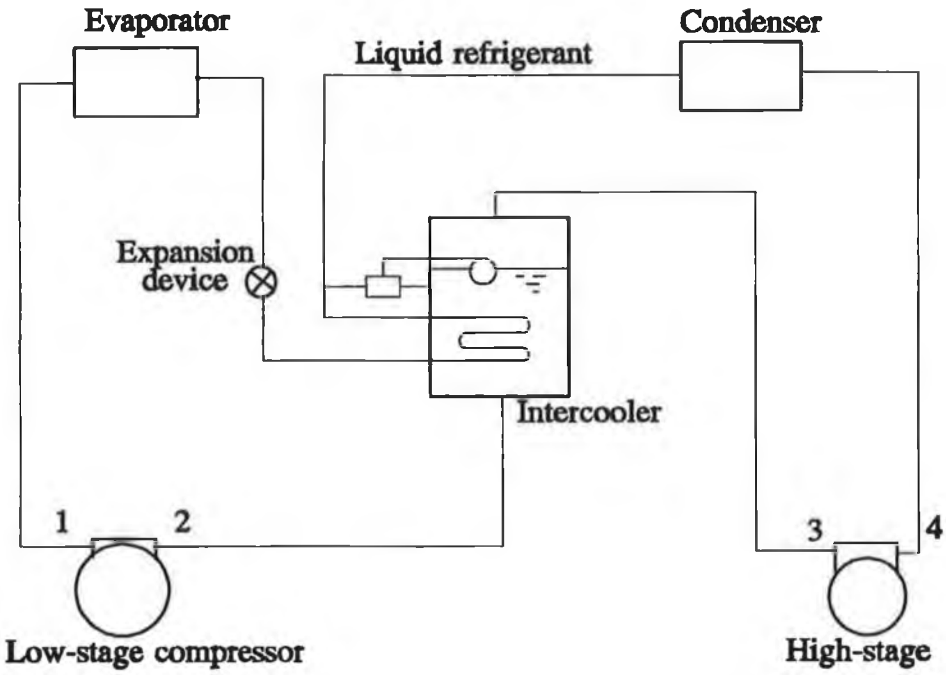


Fig.2-4 The system arrangement of a typical two stage compression refrigeration cycle with a closed type intercooler

Compression in stages may be performed in physically distinct machines working in series, so that each stage compresses the vapour over a part of the overall pressure ratio and provides only a part of the overall specific work of compression. The compressors are often called the low pressure stage (the first stage) and the high pressure stage (the second stage), respectively. In refrigerating plant practice the first stage machine is often called a 'booster' from the fact that at one time many old single stage plants were converted to two stage operation by the addition of a low stage compressor to deal with additional duties such as freezing of foodstuffs at a lower temperature.

Staged compression may also be performed in one machine, with some of the cylinders used for the first stage, others for the second. Figure 2-4 is the system arrangement of a typical two stage compression refrigeration plant.

2.3 Fundamental aspects of frozen foods

Freezing of foods is a process which converts liquid in foods to solid. Most fresh foods of animal or vegetable origin possess a cellular structure, the cells usually contain a large amount of water together with complex mixture of organic and inorganic substances. While the foodstuff is cooled, freezing may occur when the temperature reaches just below 0°C, and the liberated latent heat retards the rate of fall in temperature. The formation of ice within the tissue markedly changes its physical properties, and usually leads to pronounced structural changes with the tissue. The stability and chemical properties of macromolecules within the cell are very dependent on the interaction of their constituent groups with the surrounding aqueous phase. Freezing gives rise to a structural change in the water and consequently these interactions are altered. A little understanding of the mode of formation of ice crystals within the tissue will assist an appreciation of the changes which accompany freezing.

When the temperature of a biological system is reduced to below 0°C, the solutions it contains first supercool. Ultimately the formation of ice crystals occurs, and much of the water in the system is converted to ice. These conversions include nucleation, crystal growth and size.

Nucleation

Freezing of water or a solution will not occur until nuclei are present to initiate crystallization. Two types of nucleations are possible: homogeneous and heterogeneous. Water, if exceedingly pure, is limited to nucleation of the homogeneous type. A homogeneous nucleus forms by chance orientation of a suitable number of molecules into a tiny ordered particle. Homogeneous nucleation is impossible at 0°C and does not become probable until the temperatures are reduced many degrees below 0°C, particularly if the sample is very small. Homogeneous nucleation is of no concern with most practical situation.

Heterogeneous or catalytic nucleation is the type that occurs in foods and living specimens. Heterogeneous nucleation involves the formation of nuclei adjacent to suspended foreign particles, surface film, or on the walls of containers. Although heterogeneous nucleation necessitates some supercooling, it is more probable to occur than homogeneous nucleation at any given temperature and fixed sample. As the temperature is lowered to some critical value characteristic of the sample, nucleation begins, and further decreases in temperature result in an abrupt increase of rate.

Crystal growth

Growth of a crystal nucleus constitutes the second step in the crystallization process. Unlike nucleation, crystal growth can occur at a temperature just below the melting point of the system. At temperatures near the melting point, water molecules add to existing nuclei (if present) in preference to forming new nuclei[30].

The rate of crystallization in a complex aqueous system is governed by the rates of mass and heat transfer. During the course of crystallization, water molecules must move from the liquid phase to a stable site on the crystal surface, and solute molecules must diffuse away from the crystal. Since water molecules are small, highly mobile, and usually present in abundance, in most instances movement of water molecules is unlikely to limit the rate at which ice crystals grow.

Foods usually contain an abundance of dissolved solutes, suspended matters and sometimes cellular components. Small solutes are able to migrate short distances and eventually are concentrated among the growing ice crystals or between segments of a single crystal. Since suspended matter and cellular components are less able or are unable to migrate, ice crystals either form around them or push them aside. Therefore, it can be concluded that the rate of ice crystal growth in high moisture samples (foods) frozen at commercial realistic rates is generally not limited by mass transfer processes.

It follows that heat transfer is the process which usually limits the rate of crystallization. This is reasonable considering water's large latent heat of crystallization. Tissues and other non-fluid aqueous samples are especially troublesome in this respect since heat transfer must occur primarily by conduction. Growth rates of ice crystals increase greatly as the rate of heat removal is increased.

A lot of work has been done[5,21,31] in order to study the quality of frozen foods. The results show that the size and location of ice crystal in the frozen food is the key factor to the quality of frozen foods. And the rate of cooling is a decisive factor in determining the size and location of ice crystals in a frozen tissue. This fact appeared to have been reported first by Pland et al.(1916). It has been proved by many subsequent studies[32,33]. That is, if the rapid freezing is employed, the frozen foods will contain ice crystals that are small and numerous, whereas similar samples which have been slowly frozen contain ice crystals that are large and few in number.

2.4 The change of foodstuffs during freezing

Foodstuffs are composed of intricate and complex systems containing water, solute and macromolecules. The removal of a large part of the water as ice during freezing must upset the delicate balance of the system, and can be expected to produce marked changes in the structure and characteristics of some of the components. The most important aspects of the effects of freezing on foodstuffs are as follows.

Volume changes

Volume changes during freezing result because pure water at 0°C expands by 8.6% when water is transformed into ice at the same temperature. Most foods also expand on freezing but to a lesser extent than pure water, 6%, because only part of the water present is frozen and some foods contain air spaces[(21)]. The major point to keep in mind with respect to volume change during freezing is that ice crystals are relatively pure even when they originate from a complex system[(5)]. Since most other constituents contract as the temperature is lowered, it is apparent that the volume change is not uniform throughout the system. There are localized areas of expansion (ice crystals) and localized areas of contraction, with a likelihood of stresses and a possibility of mechanical damage.

Concentration of non-aqueous constituents

During freezing of aqueous solution, cellular suspensions, or tissues, water from solution is transferred into ice crystals of a variable but rather high degree of purity. Nearly all of the non-aqueous constituents are therefore concentrated in a diminished quantity of unfrozen water. The net effect is similar to conventional dehydration except in this instance the temperature is lower and the separated water is deposited locally in the form of ice. As a result the unfrozen phase changes significantly in such properties as PH, titratable acidity, ionic strength, viscosity, freezing point (and all other colligative properties).

As the freeze concentration process progresses, various solutes eventually reach or exceed their respective saturation concentrations and the simultaneous crystallization of ice and solute becomes possible. The temperature at which a crystallized solute can exist in equilibrium with ice and the unfrozen phase is known as the "eutectic point" of the solute. This temperature is often referred to as the "final eutectic temperature. For meat, it is between -50 and -60°C and for bread, it is apparently near -70°C[34]. Other natural foods can be expected to have final eutectic temperatures in the same general range. Consequently, it is safe to say that nearly all frozen foods stored at conventional commercial temperature contain some unfrozen and potentially freezable water.

2.5 Freezing curves of foods

Familiarity with the pattern of temperature changes that food materials undergo during freezing is basic to an understanding of the freezing process. Figure 2-5 shows typical freezing curves of foods, i.e. the relationship between the time and the temperature in foods during freezing. The most striking feature of the curves in Figure 2-5 is their simplicity.

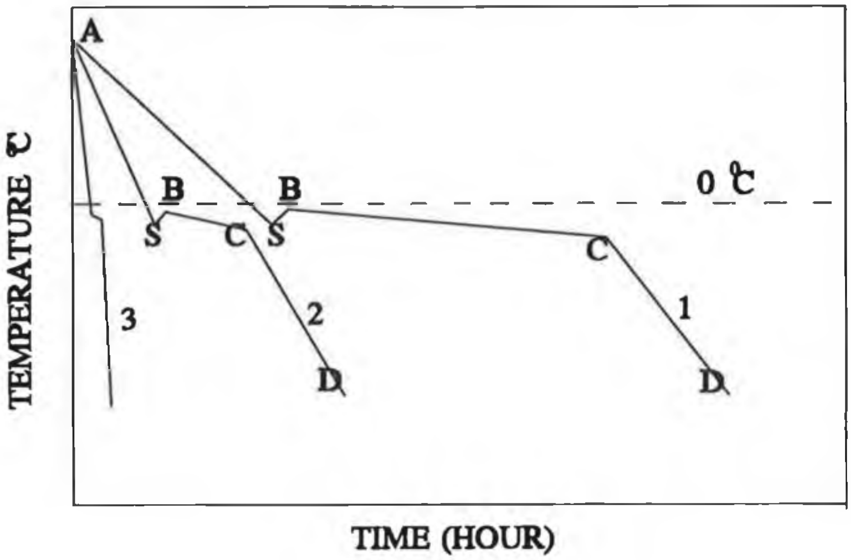


Fig.2-5 Typical freezing curves of foods

Curve 1 in Figure 2-5 is a freezing curve of slow freezing. The portion A-S represents simple cooling (removal of sensible heat) with no water-ice transformation. Point S represents supercooling and this event is believed to occur to some extent (usually not more than 10°C) in all specimens, although it is not always apparent. Detection of supercooling depends on the sensitivity, response time, etc.

Following the onset of crystallization at point S, the released heat of crystallization causes the temperature to rise promptly to the apparent initial freezing point of the sample, point B. The apparent initial freezing point as determined by this means is always somewhat lower than the true initial freezing point, especially if supercooling exceeds by more than 1-2°C[4]. The initial freezing point, which is determined by the number of dissolved particles in solution (molecules, ions, dimers, etc.), differs surprisingly little among various types of natural foods. Data illustrating this point are shown in Table 2.1[35].

Table 2.1 Initial freezing points of various classes of foods

Class of food	Freezing points °C
Nineteen common vegetables	-0.8 to -2.8
Twenty-two common fruits	-0.9 to -2.7
Fresh pork, beef, and lamb	-1.7 to -2.2
Milk and eggs	-0.5

Section B-C of curve 1 represents the period during which a major portion (often three-fourths) of the water is crystallized. Since heat which is generated from ice formation constitutes the bulk of the heat energy which must be removed during freezing, and since the ice formed during section B-C results in only a moderate increase in the concentration of solutes in the unfrozen phase (slight depression of freezing point), section B-C exists as a long plateau with a slight negative slope. During the early stages of section

B-C water separates as pure or nearly pure ice crystals, whereas during the late stages of section B-C and beyond, the possibility exists for formation of eutectic mixtures and other types of complex solids of largely unknown structures and compositions.

Each successive gram of ice formed beyond point C is responsible for a substantial and successively larger increase in molality (lower freezing point) of the unfrozen phase. Furthermore, since the specimen at point C contains far less freezable water than it had initially, removal of a given amount of heat energy can effect a much greater reduction in specimen temperature during section C-D than during B-C.

Curve 2 in Figure 2-5 is a rapid freezing process. It is clear that the section B-C is far shorter in the rapid freezing than in the slow freezing. This means that the time in which temperature of food drops from B to C is shorter in the freezing process. Consequently, the total freezing time is shorter. This type of freezing is employed in most modern freezing processes.

Curve 3 is a super rapid freezing process. It is carried out with liquified carbon dioxide or nitrogen. Although the quality of the frozen food by this way is excellent, this freezing method is not widely used, because about 20 times as much energy is required to produce a kilowatt of refrigeration at the temperature of liquid nitrogen (-196°C at normal pressure) as by conventional mechanical refrigeration plants at -30°C.

Figure 2-6 shows the typical relationship between the temperature and enthalpy in foods. It can further explain why the section B-C takes a longer time in freezing process.

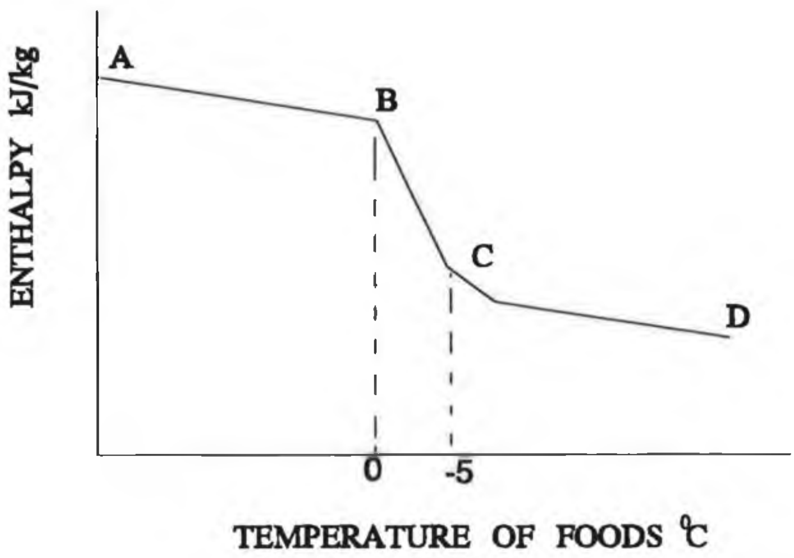


Figure 2.6. Variation of enthalpy of foods with temperatures

In the portion above 0°C of the curve in Figure 2-6, the curve declines slowly. This means that the enthalpy of foods decreases slowly as the temperature drops. In other words, heat rejected from foods during the period is small as the temperature of foods drops, because only the sensible heat is removed in this period. For example, the heat rejected from beef in this period is 3.1 kJ/kg[25] as the temperature drops by 1°C. In the period between 0°C to -5°C, the majority of water in foods turns into ice, and the so called "latent heat" is rejected. It is so called "zone of maximum ice crystal formation". The heat rejected for the beef is 9 to 87 kJ/kg as the temperature drops by 1°C in this period (section B-C). This is why section B-C declines sharply. In section C-D most water in foods turns into ice, and again most heat rejected from food is sensible heat. Therefore, the section C-D of the curve declines slowly.

CHAPTER 3. PRACTICAL ANALYSIS ON THE NEW FREEZING TECHNIQUE

3.1 The principle of using two evaporating temperatures

As briefly mentioned in section 1.4, the key point in this project is using two evaporating temperatures during the freezing process of food. In other words, a higher t_e is employed in the first period of freezing process. Then the second t_e (a lower t_e or final t_e) is applied in the second period of freezing process. As a result, the evaporating pressure p_e of refrigerant is controlled at a higher pressure in the first period of freezing process. Consequently, foods are first cooled to some temperature which is higher than the final temperature of frozen foods. Then, they are continuously cooled to the required final temperature in the second period.

As is stated, refrigeration means a process which moves heat from lower temperature source to higher temperature sink. During the process of freezing the foods, heat is moved from the foods to the ambience. The temperature of foods drops to a final required temperature. One important fact that should be kept in mind is that the initial temperature of raw materials to be frozen is rather higher than the final temperature of frozen foods. For animal carcasses without precooling, the temperature before freezing is 39°C[(25)]. Even when precooling is employed, the temperature of carcass before frozen is still much higher than its final temperature.

According to the standard of European Economic Community[36], the temperature (after precooling) is 7°C. The final temperature of frozen food is dependent on foods as well as different countries. In general, the final temperature is in the range of -15 to -30°C[37,38].

From Carnot equation (2-5) of efficiency, the work consumed in a refrigeration cycle depends on the working temperature difference between the low temperature source and high temperature sink. As can be seen from equation (2-5), the coefficient is quite low when the temperature difference is large, indicating that a large amount of power is needed to give a certain refrigeration capacity under these conditions. Conversely, higher COPs can be achieved when the temperature difference is smaller. In other words, for any given temperature T_2, the COP increases rapidly as T_1 approaches it (see Fig.2-1 as reference). In practical industrial refrigeration plants, the ambient temperature (t_2) can be considered to be constant, since it can not be controlled. The cold source temperature t_1 can be controlled. In this new freezing process, refrigeration is performed at a smaller temperature difference since a higher evaporating temperature t_e is selected. In this way, a part of the refrigeration capacity during the freezing process is obtained at a higher COP or higher efficiency.

In the second period of the new freezing process, the working condition of refrigeration plants is similar to that of the common freezing process. Therefore, it is clear that the efficiency (COP) of refrigeration plants is the same as that of a refrigeration plant in the common freezing process. Therefore, it is easy to understand that energy saving can be achieved by using the proposed freezing process. Because a part of the total refrigeration capacity (the total in new process is the same as in common freezing process) is obtained with the same efficiency, and the other part is obtained with a higher efficiency - in the first period of the freezing process, therefore, the total work consumed in the proposed freezing process must be less than that in the common freezing process. In other words, the running cost (electrical bill) is smaller when the proposed freezing process is employed.

Moreover, for a refrigeration plant, not only its efficiency is improved, but also its refrigeration capacity will increase when the work temperature difference of the plant gets smaller. The capacity of a refrigeration plant can be calculated by[3]:

$$Q_0 = \frac{V_H * V_E * q_0}{\nu_1} \tag{3-1}$$

where V_H is the displacement of refrigeration compressors, m^3/s.

V_E is the volume efficiency of the compressors.

q_0 is the refrigeration effect, kJ/kg.

v_1 is the specific volume of vapour, m^3/kg.

Q_0 is the refrigerating capacity of a plant, kW.

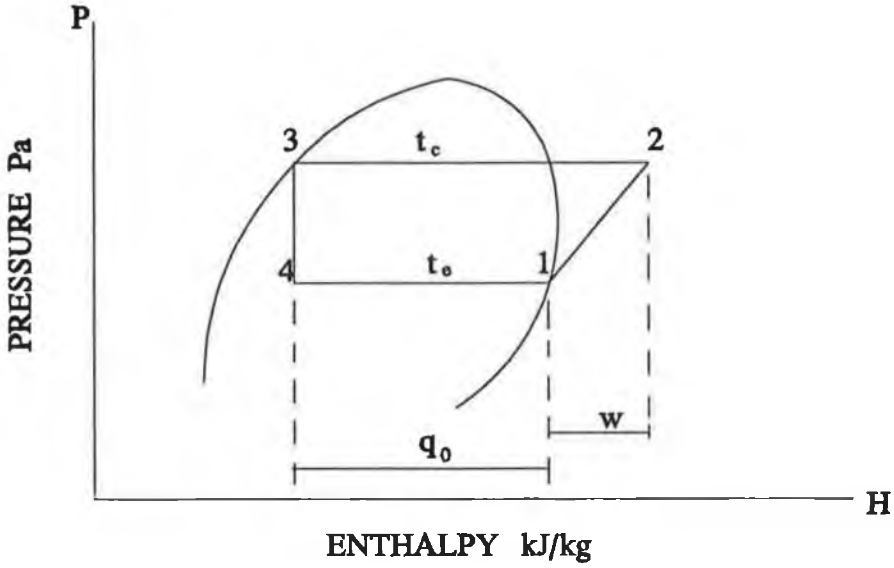


Fig.3-1 A simple refrigeration cycle depicted on a P-H chart

The graph shown in Figure 3-1 is commonly called a pressure-heat chart. It is also called a pressure-enthalpy chart. It is very useful for the calculation and the understanding of a refrigeration cycle. Note in the pressure-heat chart that as the refrigerant vaporizes at the lower constant pressure, it passes horizontally from point 4-1. This line indicates the vaporization of a unit refrigerant in the evaporator. It is commonly called the refrigeration effect q_0, the heat absorbed by a unit of refrigerant when it turns from liquid state into vapour. The enthalpy difference between point 1 and 2 represents the work which the compressor consumes to a unit refrigerant.

According to Figure 3-1, q_0 will increase and specific volume of vapour v_1 will decrease when the evaporating temperature rises. And the volume efficiency

V_E will increase as the work temperature difference declines[39]. For a given refrigeration plant, the displacement of the compressors V_H is a constant. Therefore, it is evident from Equation (3-1) that the refrigeration capacity will increase for a given refrigeration plant when the work temperature difference gets smaller.

3.2 Feasibility of using two evaporating temperatures

The purpose of freezing foodstuffs is to lower the temperature below their freezing point. During this process, heat is rejected from the foodstuffs and liquid (most of it is water) in the foodstuffs changes into solid. In general, most available equipments and technologies have evolved in response to the desired improvements in the product quality and the throughput capacities. Among the current freezing technologies, air blast and plate freezing systems are well established and accepted. Air blast is the most common method of freezing. Majority of energy in freezing is consumed by air blast freezers in the UK[13]. Therefore, this project will take the air blast freezer as the main target for discussion.

Before presenting the discussion of the feasibility of the new freezing process, a brief introduction of the common freezing in the current industrial refrigeration plants will be given below.

A typical freezing process consists of two parts - prefreezing treatments and freezing. Raw materials are treated before freezing. The procedure of treatment depends on the type of foods. For mammalian meat, slaughtered animal carcasses may be chilled promptly to a relatively low nonfreezing temperature for a period involving at least 20 hour so that glycolysis occurs slowly prior to freezing in some countries, such as EEC countries. In other countries, carcasses may be directly frozen without precooling.

After prefreezing treatment, raw material is sent to the air blast freezer. Packaged or un-packaged non-fluid foods will be frozen in air at temperatures ranging from -18 to -40°C[40,41,37]. If slow freezing is employed, air is circulated slowly or not at all and the rate of freezing can be exceedingly slow (3-72 hours or more depending on the conditions and the size of the product). This method was used until 1940's[41] and is uncommon in modern freezing operations. Vigorous circulation of cold air enables freezing to proceed at a moderately rapid rate and this common method is referred to as "air blast freezing". It is usually accomplished by placing the product on trays or on a mesh belt and passing it slowly through an insulated tunnel containing air at -18°C to -34°C or lower. The product is exposed to an air velocity of approximately 2 or 3 m/s. The final temperature of frozen food varies with the different foods as well as countries. The range is from -15 to -30°C.

The time of freezing process is a complex problem. It depends on the refrigeration capacity of the freezer, classes of food, the temperature and the velocity of the air circulated, etc. In general, the shorter the freezing time, the higher the initial cost of freezers. For current freezing process of meat, 20-36 hours of freezing time is well accepted[21,37,42].

Irrespective of that parameters selected in the current freezing process, a fact should be kept in mind: only one evaporating temperature is used in all cases. This means that the refrigerant evaporates at a single temperature from the beginning to the end of freezing process. For example, if the temperature of raw materials is 30°C and the final temperature of the product demanded is -20°C, clearly, the evaporating temperature of the refrigerant must be lower than that of the product as well as the cold air. Suppose that a difference of 10°C betweem the food and evaporating temperature is selected, then the evaporating temperature t_e will be -30°C in the whole process of freezing for the food temperature of -20°C. Consequently food will be cooled by this evaporating temperature t_e from its initial to final temperature.

If the proposed new freezing process is used in the same example, the first t_e (which is higher than -30°C) may be -15°C, which will work in the first period of freezing process. The foods may be cooled to -5°C at the end of the first period. Then the second t_e, -30°C, will be employed in the second period of freezing process, which enables food to be cooled to the same final temperature -20°C. In other words, the refrigeration plant in the new freezing process will work under the same conditions as the common freezing process during the second period.

Now that both the common and the new freezing processes have been explained, the feasibility of the new freezing process can be discussed. If the new freezing process replaces the common process, three aspects should be examined, i.e. the temperature and technical demand of frozen foods, the initial cost (capital cost), and the operating cost.

Considering the first point, it is clear that the difference between the new and common process is only the temperature of air in freezers. More accurately, the difference is only in the first period of the new freezing process. During the first period, food is cooled by an air temperature which is higher than that of the second period. The technical demands of freezing process depend on the temperature of frozen foods in industrial practice. For the frozen food both the new and common process have the same initial temperature as well as the final temperature. Therefore, there is no difference in the temperature and technical demands between the new process and the common process.

The initial cost is an important factor for the application of any new technology. Many new technologies are unable to be carried out due to this factor. Fortunately, this factor is not a barrier for the new freezing process, because all the facilities required in the new freezing process are the same as the common freezing process. In other words, the new freezing process can use the same freezing plant, including all the refrigeration compressors, equipment and freezing chambers or tunnels. The only difference in the new freezing process is its working parameters. Therefore, the initial cost will be the same as the common freezing process when the new freezing process is employed.

In general, the operating cost consists of the labour cost and the cost of energy consumption. The labour cost can be divided into two aspects, the cost in treating frozen foods and the cost of operating refrigeration plant. It is clear that there is no difference in the labour cost for the new process and the common process, because the operating procedures in two cases are entirely identical. The raw materials are sent into freezers, then they will be moved out of the freezers after freezing. For the labour cost of operating refrigeration plants, the new process will increase some work for the operators of the refrigeration plant. But the work is only in changing the operating parameters of the refrigeration plant. It need not lead to increase the labour cost.

For the cost of energy consumption, according to the discussion in section 3.1, the refrigeration cycle in the new process is of higher efficiency (COP). In other words, the new process will require less energy under the same refrigeration capacity. This means that the electricity bill of the new freezing process will be smaller compared to the common process because all the refrigeration plants in the freezing of food are driven by electrical power.

According to these discussions, it is safe to conclude that the new freezing process is feasible, and that there is no major obstacle in the technical demands, the initial capital cost and the operating cost.

3.3 Analysis of the heat loads of freezing process

In freezing process, food to be frozen releases heat and refrigeration plants must transfer the heat from the food (cold source) to the ambience (hot sink). It is clear that refrigerating capacity of a refrigeration plant should match the heat load rejected from foods if the freezing process is to continue properly. If the total heat rejected from foods is Q and the total freezing time is τ, then the total heat load can be calculated by:

$$Q=\int_0^{\tau}\frac{dQ}{d\tau}d\tau \qquad \textbf{(3-2)}$$

But the heat load of foods is variable during freezing process according to section 2.5. When the temperature of foods is above 0°C, the sensible heat is released. During the period the temperature drops from 0°C to -5°C, a large amount of heat is released since most water in the foods turns into ice. In the late period of a freezing process, the section C-D in Figure 2-6, most heat rejected from foods is again sensible heat. According to the law of heat transfer, heat transferred in a process can be calculated by[43]:

$$Q=U*A*\Delta T \tag{3-3}$$

where U is the overall heat-transfer coefficient, $W/m^2\ K$

A is the area of heat transfer, m^2

ΔT is the temperature difference of heat transfer

Q is heat transferred, kW.

For a given freezing process, U and A can be considered as approximately constants. The ΔT is the temperature difference between the food and cold air in the air blast freezer. Clearly, the ΔT is far greater in the early period of freezing than in the late period, since the temperature of food declines as the freezing process continues and it will get smaller and smaller as the freezing process goes on.

Therefore, the heat rejected from foods in the early period of freezing is much greater. The heat load is usually called peak load[44]. In practice, it is very difficult to calculate accurate peak load, since many parameters such as the classes, character and temperature of food, can not be determined precisely. In other words, the refrigeration capacity of plants can not match the heat load well in practice. On the other hand, it is unreasonable to design a refrigeration plant according to the peak load. Because many refrigeration machines have to switch off after the peak load. And the capital cost must be much higher since a much larger refrigeration plant has to be installed in this case.

The answer to the problem is that an average heat load is commonly used in the practical design of refrigeration plants[44]. This means that the total heat is divided by the total operating time of the refrigeration plant, i.e. average load or hour load.

Consequently, the refrigerating capacity of a common refrigeration plant is always less than the peak load, in the early period of the freezing process. Therefore, the common refrigeration plant is unable to match the heat load. And there is no doubt that this will affect the efficiency of the plant. It should be kept in mind from section 2.5 that same refrigeration plant has larger refrigeration capacity in the first period of the new proposed freezing process, because its temperature difference of work is smaller. Therefore, the refrigeration capacity of refrigeration plants will match the trend required by the peak load when the new freezing process is employed. And consequently, the new freezing process has a higher efficiency.

3.4 The quality of frozen food in the new freezing process

Any improvements to process efficiency must be accomplished in a manner that results in product quality improvement. At least, optimization techniques should be utilized to assure maximum process improvements without reductions in product quality. Therefore, the quality of frozen food in the new process should be discussed.

Now that we have some background information to understand the behaviour of frozen food products during freezing, a conclusion appears consequently - that the most important element in the evaluation of the quality of freezing products is the average freezing speed. The average freezing speed is the speed at which the ice-front moves forward through a product. Suppose the freezing time is τ (hour), and the thickness of the frozen layer is δ (cm), then the freezing speed U_f will be[2]

$$U_f = \frac{\delta}{\tau} \tag{3-4}$$

The further the ice-front moves forward in a product, the higher the freezing speed will be, because the thermal conduction coefficient λ of the frozen layer is higher than that of the water-bearing layer. It should be noted that λ is not constant but is dependent on the temperature. Furthermore, the freezing point is not constant, but depends on the degree of concentration. This is why an average freezing time between the outer layers and the core of the product is considered.

The freezing speed must be high and consequently the freezing time τ short if one wants to maintain the quality of the product. When freezing is slow the product will be damaged because as the concentration of minerals in the protoplasm between the cells is higher than inside the cells and the vacuoles, the freezing point there is lower than inside the cells. So when freezing starts there, the liquid is subcooled and water from inside the cells has a tendency to migrate by osmosis through the cell walls to the space in-between the cells. The ice crystals formed there will grow and become very large; the bigger they are the more water they attract and they can become so big that they perforate and damage the cell wall. Ice has 8.6% more volume than water, so it can be imagined what happens in the small space between the cells.

In the meantime, the concentration of minerals and other elements in the remaining water in the cells will increase the PH value and the salt concentration. So the cell structure of the product is totally changed and can never be restored. In addition, this dehydration causes changes in the enzymatic activity as in the case of freeze damage to fresh fruit.

It is impossible to reverse the freezing process completely by defrosting. Quick freezing is necessary because structure changes of frozen food must be prevented if possible. At a high freezing speed the ice crystals will form simultaneously throughout the product, in-between as well as inside the cells, but they will stay very small. Therefore, the ice crystals will not burst the cell membranes. Consequently the juices and the taste of frozen product are not lost during defrosting and the structure of the product remains unaltered.

The speed required for achieving a good quality final product varies with different foods. With vegetables frozen at the rate of 18 cm/h, as much as 90% of the ice crystals remain within the cells. For fish we need a speed of between 0.6 and 5 cm/h; for meat a speed of between 1 and 4 cm/h is satisfactory[2].

In the second period of the new freezing process, the operating parameters of the refrigeration plant is the same as the common freezing process. Therefore, the freezing speed will simply be the same as the common process. In the first period, on the other hand, the situation is a little complex. The temperature difference of heat transfer between the food and the refrigerant is smaller than that in the common process since a higher t_e is used in this period. In the meantime, the efficiency of the refrigeration plant is improved because the difference of working temperature gets smaller, which has been discussed in section 2.3. With the same refrigeration plant, the refrigeration capacity of the plant in the first period will increase greatly. Therefore, the freezing speed in the first period will be higher than that in the common process.

Consequently, the average freezing speed will be higher in the new freezing process than in the common freezing process. Thus it can be concluded that the quality of frozen food in the new freezing process will be better than that in the common process.

CHAPTER 4. DIFFERENT SCHEMES OF THE NEW FREEZING PROCESS

As stated earlier, vapour compression refrigeration is a refrigeration method which is most widely used in the world, and the same is true for the refrigerating plant used in food freezing. In industrial freezing plants, two stage compression is commonly employed because a lower refrigeration temperature (evaporating temperature) is required, and the ratio of pressures in which compressors operate is usually over the limit of the single stage refrigeration compressors. In this case, simple compression (single compression) is unable to work, as has been stated in section 2.2.

There are many different schemes[45-48] in the two stage compression refrigeration cycle. For example, two stage compression with an open flash interstage cooling, two stage compression with a closed flash interstage cooling, etc. It is difficult to say which scheme is the best because each scheme has its own features which suits a given case. Therefore, three typical schemes are selected and discussed in this project. In other words, refrigeration cycle using the new freezing process will be discussed under three typical schemes.

4.1 Scheme 1 - Two stage compression with open flash intercoolers

The two stage compression with an open flash intercooler is a typical scheme employed in industry[48,49]. Figure 4-1 shows the arrangement of equipment and piping connection, and the corresponding P-H chart is presented in Figure 4-2.

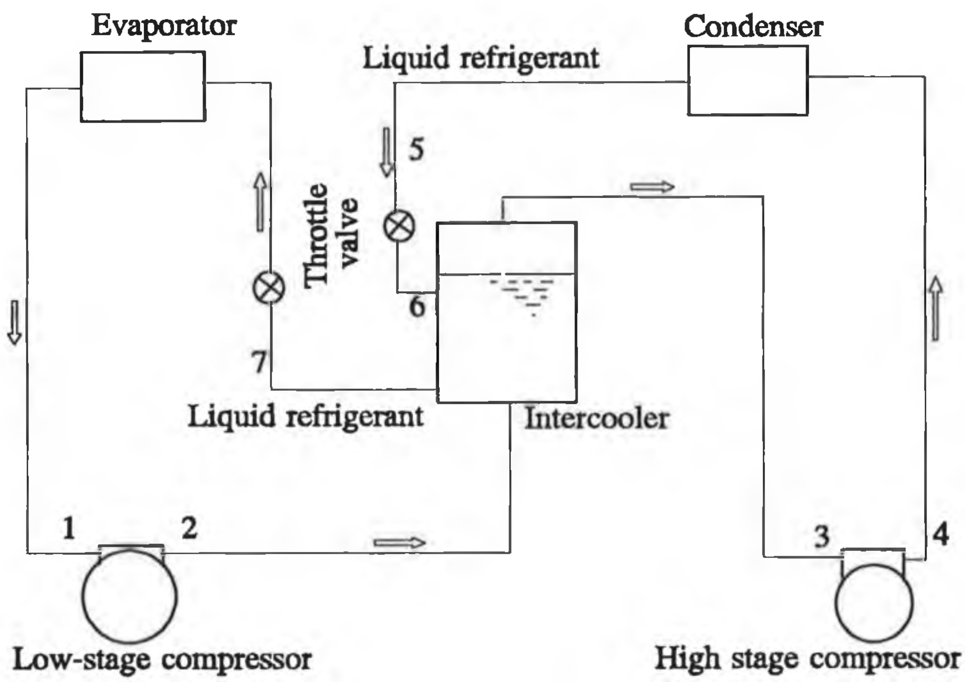


Fig.4-1 System arrangement of a two stage compression cycle with an open flash intercooler

In Figure 4-1, the interstage cooling takes place by passing full hot discharge gas flow from the low pressure stage through a bath of liquid refrigerant inside a vessel, called an open flash interstage cooler, which is aspirated at intermediate pressure by the high pressure stage. At the same time, the full liquid refrigerant flow is passed through this vessel, thereby undergoing a double expansion. It first condenses to intermediate pressure via a level controlled throttling valve. Then the saturated liquid at intermediate pressure is fed from the vessel to the evaporator via a second throttling control valve. It is also called two-throttling scheme.

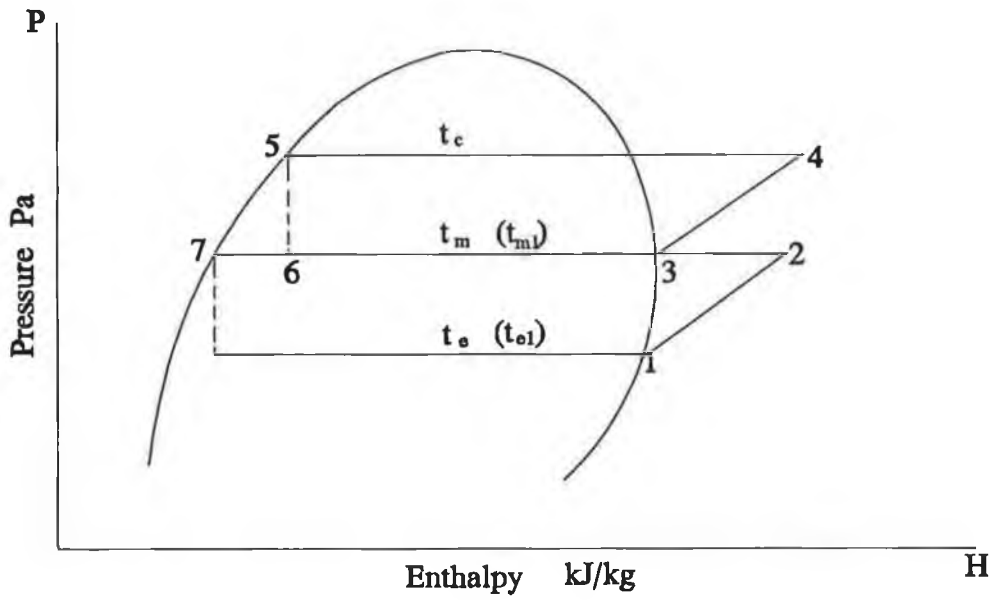


Fig.4-2 The P-H diagram of the cycle in Figure 4-1

The advantages of the scheme are following. (a) Enthalpy difference over the evaporator has maximum possible value for given operating conditions. Therefore, there is maximum refrigerating capacity for a given compressor with minimum specific power consumption (maximum COP) and minimum compressor price per unit of capacity obtained. (b) Minimum operating costs assuming a sufficiently large number of running hours per year are obtained. The disadvantages are: (a)Interstage cooling section is rather complicated and expensive due to the necessity of shut-off values, an oil separator in the low pressure discharge line, and a suction strainer in the high pressure suction line. (b) Installation is less convenient, and qualified operators are necessary. (c) The flash interstage cooler contains a considerable volume of liquid refrigerant and traps oil coming from the low pressure stage. Therefore, it is less suitable for R22. (d) Pressure difference across the throttling control valve for evaporator feeding is low. Hence it is less suitable for sending liquid refrigerant to a longer distance or to a higher position.

When this scheme is employed by the new freezing process, both the first period and the second period use the scheme. In the first period of the new freezing process, a higher evaporating temperature t_{e1} (compared with the second period) is selected and operated. In the second period, the refrigeration plant works under the final evaporating temperature, which is originally required in the common freezing process.

Clearly, the whole system (equipment and piping connection) is the same as the common process. The only difference is in the operating parameters, or more specifically, only in the evaporating temperatures. It is easy to understand that the performance of the refrigeration plant will vary as the selected evaporating temperature t_{e1} differs. Therefore, we should find the relationship between the t_{e1} and the performance of the plant so that a suitable t_{e1} can be decided.

4.2 Scheme 2 - Two stage compression with closed-type intercoolers

Two stage compression with closed-type intercooler is another typical scheme used in industry[50,51]. Figure 4.3 presents its arrangement of equipment and piping connection, the P-H diagram of the scheme is shown as Figure 4.4.

In this scheme the interstage gas cooling takes place in a similar manner. However, the liquid refrigerant flows under condensing pressure via a closed cooling coil in the interstage vessel to the throttle control valve of the evaporator. Therefore, single expansion of the liquid occurs. The only function of the level controlled throttling valve for feeding the interstage vessel is to inject just the amount of liquid required for the interstage gas cooling. In the coil, the liquid is subcooled down to a certain temperature, which is usually about 5°C higher than the saturation intermediate temperature[48,49].
The advantages of the scheme are as follows. The full pressure difference between the condenser and the evaporator is available to operate the throttling control valve properly for evaporator feeding. Also, it is very unlikely that flash gas bubbles will develop in the liquid line from interstage vessel to evaporator.

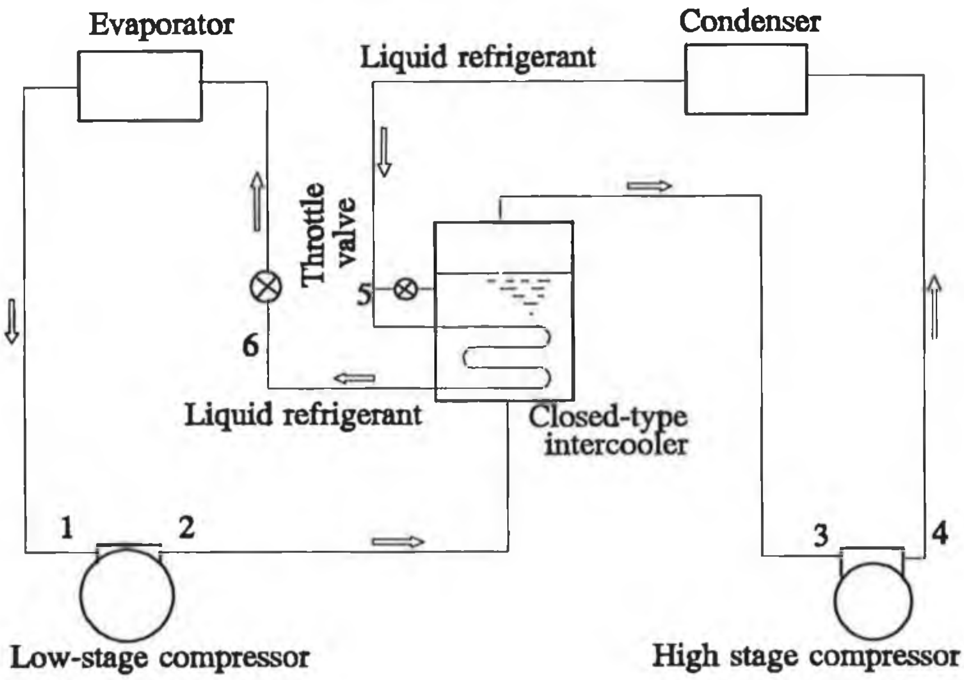


Fig.4-3 System arrangement of a two stage compression cycle with a closed type intercooler

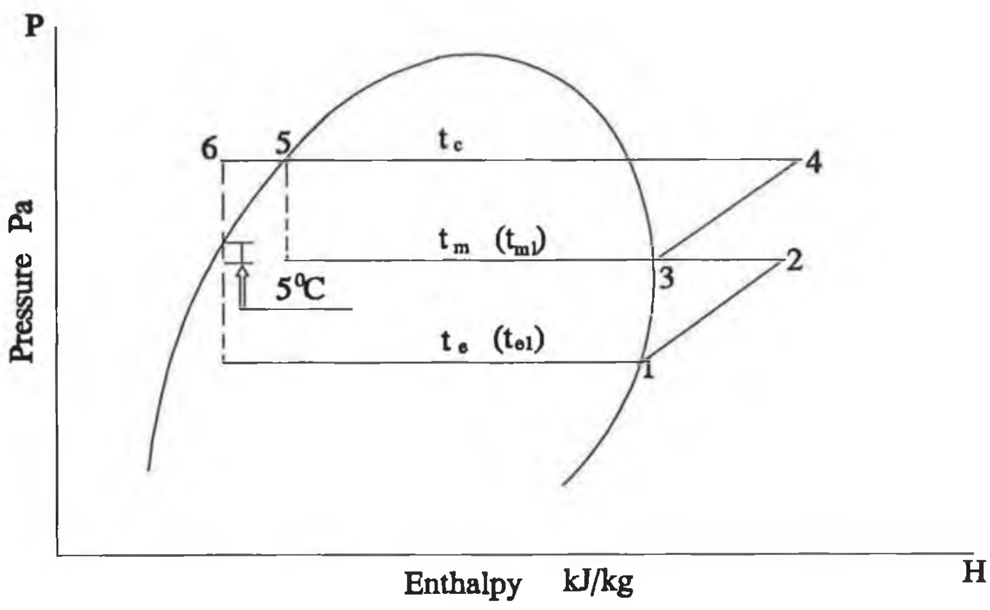


Fig.4-4 The P-H diagram of the cycle in Figure 4-3

The disadvantages of the scheme are: (a) Refrigerating capacity at given conditions is somewhat lower than that of the open intercooler scheme (approximately 3.5 percent, due to higher enthalpy of liquid/vapour mixture at evaporator inlet). Therefore, specific power consumption and compressor price per unit of capacity are slightly higher. (b) Interstage cooler is slightly more expensive than that of the scheme at Figure 4-1 due to the built-in coil. (c) Total operating costs are somewhat higher than those of system at Figure 4-1.

Although this scheme is not the optimum solution, it is more universal one. The scheme has no restrictions regarding the incorporation of different components with respect to one another. When the new freezing process uses this scheme, the system, both the first period and the second period, is the same except the operating parameters. Just as mentioned in section 4.1, it is necessary to find the relationship between the t_{e1} and performance of the plant so that a suitable t_{e1} can be decided and used in the first period of the new freezing process.

4.3 Scheme 3 - With single stage compression in the first period

As is stated earlier, a higher evaporating temperature t_{e1} is selected for the first period of the new freezing process. Because any evaporating temperature for a refrigerant corresponds to a relative saturated pressure and the relation between them is directly proportional. In other words, the higher the temperature, the greater the pressure. Therefore, the pressure ratio in which the refrigeration compressors operate gets smaller and smaller as t_e increases, since the condensing temperature t_c is dependent on the ambience which can not be changed artificially.

Consequently, a refrigeration plant can be operated in a single stage compression refrigeration cycle if the evaporating temperature selected is high enough. There is an obvious advantage when the single stage compression is used in the first period of freezing. That is, the refrigerating capacity during the first period will be much greater than that of above two schemes for a given refrigeration plant, because all the compressors in a refrigerating plant come into operation (refrigeration). As a result, the displacement (V_H) of the operating compressors increases compared with the two stage schemes above. According to equation 3-1, it is clear that refrigerating capacity Q_0 will increase as V_H rises.

In a two stage compression cycle, on the other hand, not all the compressors are for refrigeration effect. Some compressors are used as low stage compressors and the other compressor are used as high stage compressors. And only low stage compressors have refrigerating effect for a two stage compression plant.

In scheme 3, all the refrigeration compressors operate as single stage compression in the first period, at a higher evaporating temperature t_{e1}. Then the refrigeration plant is changed to the common two stage compression in the second period, supposing that the scheme of Figure 4-3 is used in the second period. Figure 4-5 shows the arrangement of equipment and piping connection for this scheme, and Figure 4-6 is the P-H chart of the scheme in the first period. Clearly, the P-H chart of the second period is the same as in Figure 4-4.

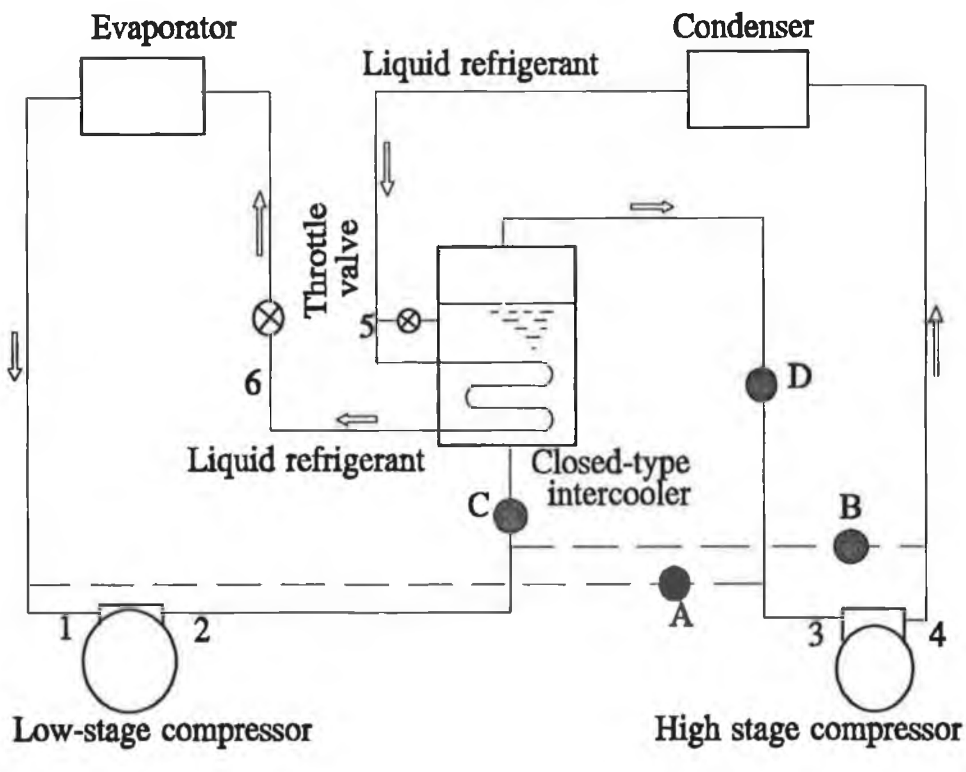


Fig.4-5 System arrangement of using single compression in the first period of the new freezing process

In Figure 4-5, valves A and B are kept open and valves C and D are kept closed during the first period of the new freezing process. In this way, the high stage compressor can draw refrigerant vapour directly from the evaporator, and the low stage compressor can displace refrigerant vapour directly to the condenser without passing the intercooler. When valves A and B are closed, and valves C and D are kept open, the system becomes a two stage compression cycle, in the second period. By means of the valves and relative piping, the plant can be switched easily from the single stage compression to the two stage compression, or from the two stage compression to the single stage compression. The compressors in the scheme have double functions.

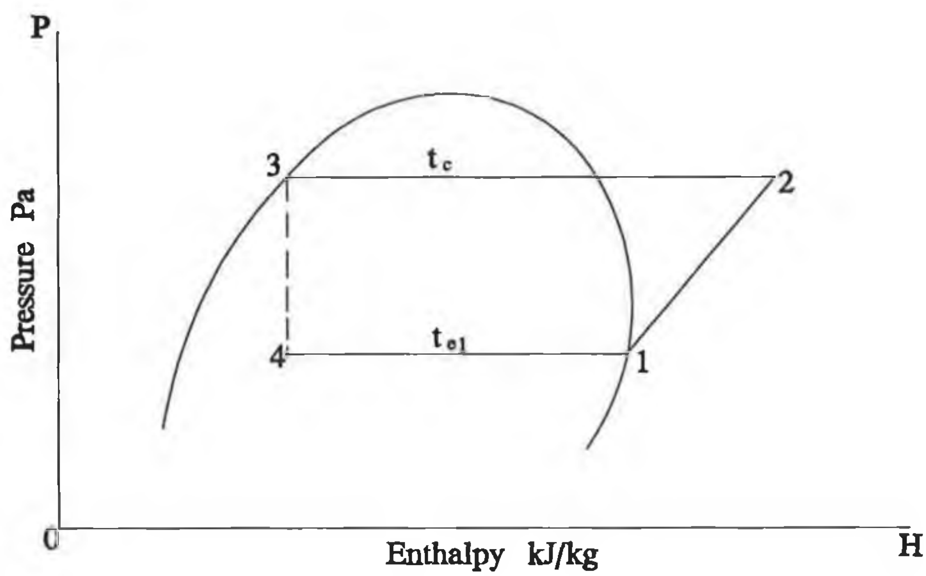


Fig.4-6 The P-H diagram of a single compression

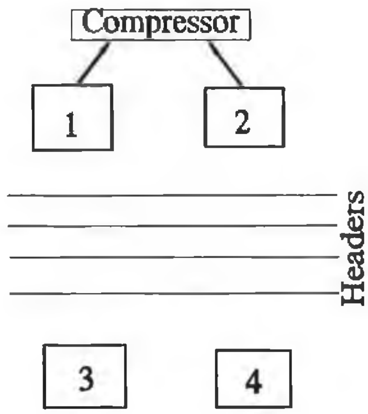
Consequently, more capital cost is required when the scheme is used, in order to have double functions for the compressors.

Fortunately, the increase of capital cost is quite small. There are two reasons for this. Firstly, only one additional valve is needed for each compressor. In Figure 4-5, valves C and D are necessary for a common two stage compression cycle in practice, since each compressor must be able to switch away from a system so that the maintenance and repair work can be carried out.

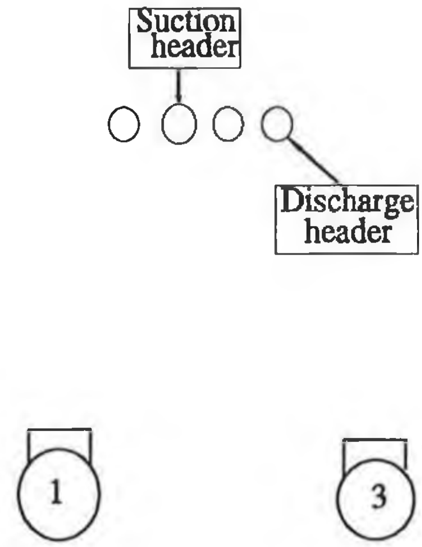
Therefore, only valve A is added for the low stage compressor, and valve B for the high stage compressor. If there are many compressors for a large plant, each compressor increases only one valve.

Secondly, the increase in pipe length in this scheme is very small. In an industrial refrigeration plant, the typical layout is that all the compressors are installed in a machine room[21,25]. Figure 4-7 shows a typical layout of a machine room. The headers for connecting compressors are arranged under the roof above the compressors. Therefore, the added pipe in which valves A and B are installed are very short. Usually each one is no more than 5 meters. Therefore, when a refrigeration plant is retrofitted to the new scheme, the capital cost is very small and can be nearly ignored. If an original design uses the scheme, the capital cost increase can be totally ignored.

For this scheme, the performance of refrigerating plant also varies as the selected t_{e1} differs. Therefore, it is necessary to find the relationship between the t_{e1} and the performance of the plant so that an optimal suitable t_{e1} can be decided and used in the first period of the new freezing process.



a. Plan of a machine room



b. Elevation of the machine room

Fig.4-7 Typical plan and elevation of machinery rooms

CHAPTER 5 MATHEMATICAL MODEL AND SIMULATION PROGRAM

5.1 Mathematical models for the simulation of refrigeration cycle

As has been discussed in chapter 4, there are several schemes in the two stage compression refrigeration. Two typical schemes and a new scheme (single stage compression is employed in the first period of freezing process) are selected in this project, in order to carry out the new freezing process. There is the same fundamental question for all three schemes - which evaporating temperature t_{e1} is optimal for the first period of the new freezing process ?

The answer to the question is to calculate these schemes so that the performances of these schemes with the evaporating temperature of the first period (t_{e1}) can be evaluated. With the changes of temperature conditions and other parameters, it is easy to imagine that there are a considerable amount of calculations relevant to this project, and manual calculation will be enormous and tedious. Fortunately, more and more equations of refrigerant thermodynamic properties are well developed(52-57) so that a personal computer can be employed in the calculation relevant to this project. And a number of works have also been reported on the simulation of refrigeration cycle(58-64).

Cleland[65] proposed some useful equations for evaluation of the thermodynamic properties of refrigerant. The pressure and saturation temperature of refrigerant vapour can be calculated by the following equations:

$$P=\exp(a_1 - a_2 /(t + a_3)) \tag{5-1}$$

and

$$t=-a_2 /(\ln(P) - a_1) - a_3 \tag{5-2}$$

where P is the pressure (Pa) and t is the temperature (°C), and a_1, a_2, a_3 are constants. Table 5.1 presents the value of these constants for three different refrigerants[65].

Table 5.1 Coefficients in the equation (5-1) and (5-2).

	a_1	a_2	a_3
R22	21.25384	2025.4518	248.94
R502	21.00668	1924.9516	248.46
R717	22.11874	2233.8226	244.20

The following two equations are used for the calculation of refrigerant enthalpy of saturated liquid and vapour:

$$H_L = a_4 + a_5 t + a_6 t^2 + a_7 t^3 \tag{5-3}$$

$$H_V = a_8 + a_9 t + a_{10} t^2 + a_{11} t^3 \qquad (5\text{-}4)$$

where H_L is the enthalpy of liquid refrigerant, KJ/kg

H_V is the enthalpy of vapour refrigerant, KJ/kg

t is the temperature of refrigerant liquid or vapour,°C.

The values of a_4, a_5, a_6, and a_7 are given in Table 5.2 and those of a_8, a_9, a_{10}, and a_{11} are given in Table 5.3. The value of a_4 depends on whether the ASHRAE (American Society of Heating, Refrigeration and Air-Conditioning Engineers) standard (H_L = 0 kJ/kg at -40°C) or the IIR (International Institute of Refrigeration) standard (H_L = 200 KJ/kg at 0°C) is used. The ASHRAE standard is used in this project, since the ASHRAE HANDBOOK is widely available.

Table 5.2 Coefficients in equation (5-3)

	a_4	a_5	a_6	$a_7 \times 10^{-3}$
R22	44518	1170.36	1.68674	5.2703
R502	41103	1114.60	2.12743	-1.7679
R717	184311	4751.63	2.04493	-37.875

The specific volume (m^3/kg) of the saturated vapour of refrigerant at a temperature is given by equation (5-5), and the corresponding coefficients are listed in Table 5.4.

$$v = \exp(a_{12} + a_{13}/(t+273.15)) * (a_{14} + a_{15}t + a_{16}t^2 + a_{17}t^3) \quad (5\text{-}5)$$

Table 5.3 Coefficients in equation (5-4)

	a_8	a_9	a_{10}	$a_{11} \times 10^{-3}$
R22	250027	367.265	-1.84133	-11.4556
R502	187890	406.454	-1.59402	-13.6010
R717	1441467	920.154	-10.20556	-26.5126

Table 5.4 Coefficients in equation (5-5)

	a_{12}	a_{13}	a_{14}	$a_{15} \times 10^{-4}$	$a_{16} \times 10^{-6}$	$a_{17} \times 10^{-7}$
R22	-11.82344	2390.321	1.01859	5.094	-14.846	-2.49547
R502	-12.03131	2327.862	1.03208	5.578	-25.501	-2.86511
R717	-11.09867	2691.680	0.99675	4.022	2.6417	-1.75152

The specific enthalpy increase during the isentropic compression is given by:

$$\Delta H_w = \frac{c_i}{(c_i - 1)} P_1 v_1 ((\frac{P_2}{P_1})^{\frac{c_i - 1}{c_i}} - 1) \qquad (5\text{-}6)$$

where P_1 and P_2 are the suction and discharge pressures, v_1 is the specific volume of the vapour at the suction condition and c_i is the fitted constant. By use of Equation (5-2) the saturation temperature corresponding to P_1 and P_2 can be found (t_{sat1} and t_{sat2}, respectively):

$$\Delta t = t_{sat1} - t_{sat2} \qquad (5\text{-}7)$$

and

$$\begin{aligned} c_i = a_{18} &+ a_{19}(t_{sat1}) + a_{20}(t_{sat1})^2 + a_{21}(t_{sat1})(\Delta t) \\ &+ a_{22}(t_{sat1})^2(\Delta t) + a_{23}(t_{sat1})(\Delta t)^2 \\ &+ a_{24}(t_{sat1})^2(\Delta t)^2 + a_{25}(\Delta t) \end{aligned} \qquad (5\text{-}8)$$

The values of the coefficients a_{18}-a_{25} from reference 65 are given in Table 5.5.

Table 5.5 Coefficients in equation (5-8)

	a_{18}	$a_{19}\times10^{-3}$	$a_{20}\times10^{-6}$	$a_{21}\times10^{-6}$
R22	1.137423	-1.50914	-5.59643	-8.74677
R502	1.050613	2.42242*	-12.0401	-2.80193
R717	1.325798	0.24520	3.10683	-11.3335
	$a_{22}\times10^{-7}$	$a_{23}\times10^{-8}$	$a_{24}\times10^{-9}$	$a_{25}\times10^{-4}$
R22	-1.49547	5.97029	1.41458	-4.52580
R502	0.05957	-2.95399	-0.13106	-6.69841
R717	-1.42736	6.35817	0.95979	-3.82295

All the equations are simple and easy to use in a personal computer. Moreover, all the parameters in general refrigeration design and simulation can be obtained by means of these equations.

5.2 Parameter determinations and equations

5.2.1 Calculation on single stage compression

For a vapour compression refrigeration cycle, one of the important features is its refrigeration effect q_0, i.e. heat absorbed by one kilogram of refrigerant in the evaporator (KJ/kg). It is given by

$$q_0 = H_{eb} - H_{ea} \tag{5-9}$$

where H_{ea} is the enthalpy of the refrigerant entering evaporators, kJ/kg

H_{eb} is the enthalpy of the refrigerant leaving evaporators, kJ/kg

q_0 is the refrigeration effect, kJ/kg.

If Figure 3.1 is taken as an example, then

$$q_0 = H_1 - H_4 \tag{5-10}$$

The mass flow rate of the refrigerant is:

$$G = \frac{Q_0}{q_0} \tag{5-11}$$

where Q_0 is the refrigerating capacity of a refrigeration plant, kW

G is the mass flow rate of the refrigerant, kg/s.

The volume flow rate of the refrigerant is given by:

$$V_f = G * v_1 \tag{5-12}$$

where V_f is the volume flow rate of the refrigerant, m^3/s

v_1 is the specific volume of the refrigerant vapour, m^3/kg

The size of a refrigerating compressor is dependent on its displacement. The theoretical displacement of a refrigeration compressor is given by:

$$V_H = \frac{V_f}{V_E} \tag{5-13}$$

where V_H is the theoretical displacement of compressors, m^3/s

V_E is the volumetric efficiency of compressors.

The volumetric efficiency V_E of a refrigeration compressor is the ratio of the actual suction volume (actual volume rate) and the ideal suction volume (swept volume of the compressor) of a compressor. There are many equations(2,3,24,66-68) for calculating V_E, but none is satisfactory. When the ratio of the compressor's clearance volume and cylinder volume is within 4% to 6%, the following equation is acceptable[69],

$$V_E = 0.94 - 0.85*((\frac{P_c}{P_e})^{\frac{1}{k}} - 1) \tag{5-14}$$

where P_c is the condensing pressure of the refrigerant, Pa

P_e is the evaporating pressure of the refrigerant, Pa

k is the ratio of the specific heat of refrigerants.

The power consumption of refrigerating compressors is another critical parameter for the calculation of a refrigeration cycle. For a piston compressor, the power consumed during an isentropic compression process is given by

$$N = G * \Delta H_w \tag{5-15}$$

where N is the adiabatic power consumed in a refrigeration cycle, kW

G is the mass flow rate of refrigerants, kg/s

ΔH_w is the work required by a kilogram of refrigerant in the isentropic compression

The work required by a kilogram of refrigerant during the isentropic compression is given by (see Fig.3-1 for reference)

$$\Delta H_w = H_2 - H_1 \quad (5\text{-}16)$$

where H_2 is the enthalpy leaving compressors, kJ/kg

H_1 is the enthalpy entering compressors, kJ/kg

5.2.2 The calculation on two stage compression

As discussed earlier in section 2.2.2, a two stage compression cycle is employed when the working temperature difference of a refrigeration plant, i.e. the difference between the condensing temperature and the evaporating temperature, is over the limit of a single stage refrigeration compressor. In the two stage cycle, in fact, the whole working temperature difference is divided into two parts in a two stage compression. One part is taken over by the high stage compressors, and the other part by the low stage compressors. It is very easy to understand the principle of two stage compression by means of Figure 5-1. Actually, each part is similar to a single stage compression. Therefore, a two stage compression can be calculated by the basic equations for the single stage compression after the two questions are solved.

The first question is the determination of the intermediate pressure. Clearly, both working temperature differences (for high stage and low stage) are determined when the intermediate pressure (or temperature) is decided upon. It has been proved that the performance of a two stage compression cycle varies with the intermediate pressure. Therefore, there is an optimum intermediate pressure (or temperature). The optimum intermediate pressure

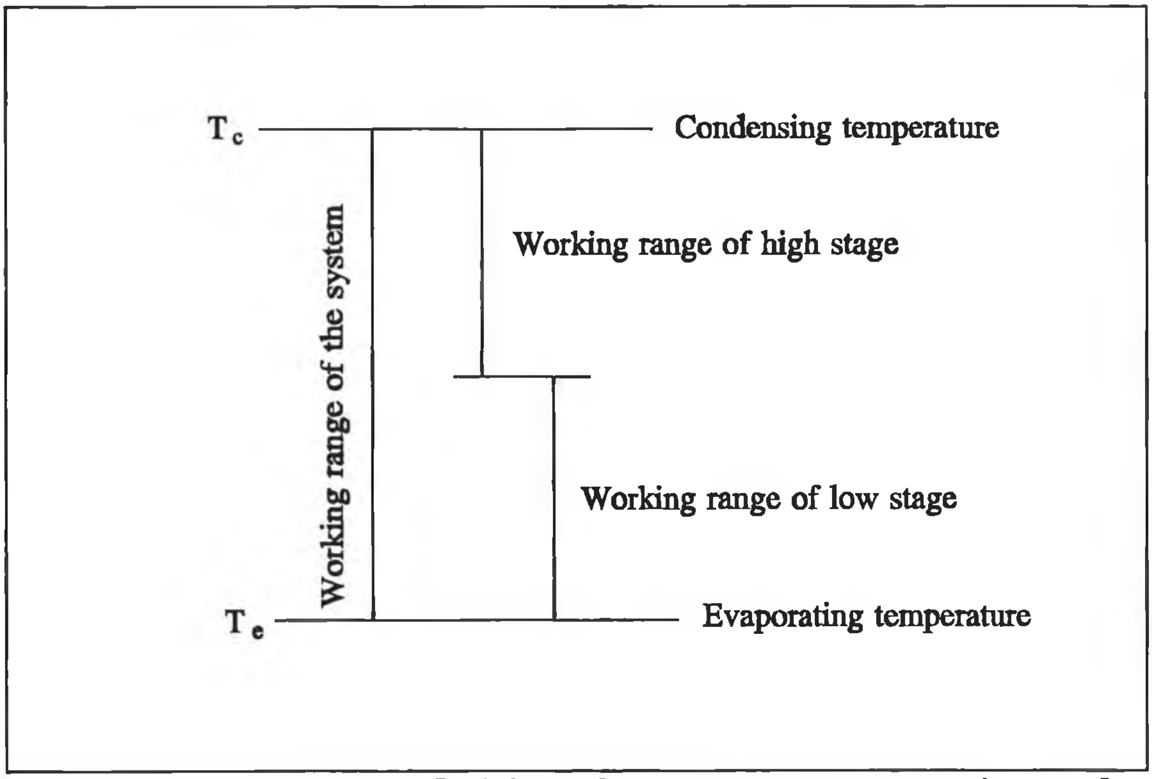


Fig.5-1 The temperature division of a two stage compression cycle

is dependent on refrigerants and schemes of refrigeration cycle. Many equations[68,70,71] have been published in this field. Fortunately, the performance of two stage compression refrigeration cycle around the optimum intermediate pressure does not vary significantly. In other words, there is no obvious difference for the results of different equations. In order to make the calculation more universal, the most basic and simple formula is selected in this project:

$$P_m = (P_c * P_e)^{\frac{1}{2}} \tag{5-17}$$

where

P_m is the intermediate pressure, Pa

P_c is the condensing pressure, Pa

P_e is the evaporating pressure, Pa

The other problem in the calculation of a two stage compression cycle is the mass flow rate of high stage compressors. The mass flow rate of the high stage compressors is greater than that of the low stage compressors, since the interstage cooling is usually employed, as has been discussed in section 2.2.2. The mass flow rate of the high stage compressor varies with the schemes of cycles. According to the energy balance of the interstage cooler, the energy entering an intercooler equals the energy leaving the intercooler, i.e.

$$Energy_I = Energy_O \tag{5-18}$$

where $Energy_I$ is the energy entering an intercooler

$Energy_O$ is the energy leaving the intercooler.

Figure 5-2 implies

$$G_L{*}H_2 + G_h{*}H_5 = G_L{*}H_6 + G_h{*}H_3 \tag{5-19}$$

where G_h is the mass flow rate of high stage compressors, kg/s

G_L is the mass flow rate of low stage compressors, kg/s

H_2, H_3, H_5 and H_6 are the enthalpy at the corresponding points shown in Figure 5-2, kJ/kg

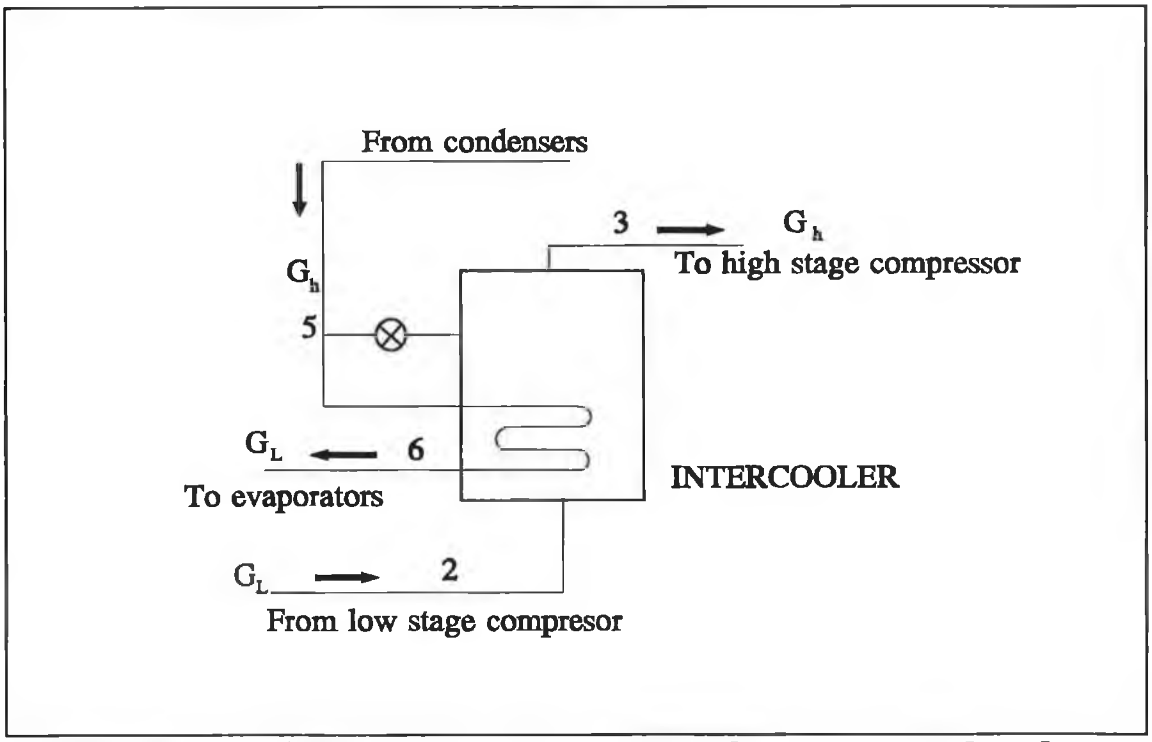


Fig.5-2 Schematic diagram of energy balance on a closed type intercooler

From equation (5-19), the mass flow rate of the high stage compressors is given by

$$G_H = G_L*(\frac{H_2-H_6}{H_3-H_5}) \tag{5-20}$$

It should be noted that equation (5-19) can only be used for a scheme like the one in Figure 5.2. But the principle expressed in equation (5-18) is universal for any scheme, and the mass flow rate of the high stage compressors can be obtained from it, although the state points in equation (5-19) may vary.

It should be mentioned that only the low stage compressors refrigerate to evaporator in a two stage compression refrigeration cycle, i.e. only the low stage compressors are of refrigeration capacity. But the power consumed by a two stage cycle is the sum of the power consumed by the low stage and the high stage compressors, i.e

$$N_{two} = N_L + N_H \tag{5-21}$$

where N_{two} is power consumed by a two stage refrigerating plant, kW

N_L is power consumed by the low stage compressors, kW

N_H is power consumed by the high stage compressors, kW

As mentioned previously, both the high stage and the low stage are single compression. Therefore, N_L and N_H can be calculated by Equation (5-15).

5.2.3 The calculation on the new freezing process

In order to examine the new freezing process, the new process should be compared with a typical common process under the same condition. There are four steps in the calculations.

(a) Choice of basic parameters for calculations. The essential parameters chosen should be from actual industrial conditions so that the result of calculations will be close to industrial practice.

(b) Calculation of the common freezing process. The purpose is to obtain energy consumption for the common process, and compressor size (capacity) of the refrigeration plant.

(c) Calculation of the new freezing process. The aim is to obtain the energy consumption and the freezing time of the new process under the same condition. In the computation, the capacity of the refrigeration compressors, which is obtained in the previous step, is directly used.

(d) Comparison of computed results. It is a comparison between the new freezing process and the common freezing process. The main features to be compared are the energy consumption and the freezing time.

As is stated earlier, air blast freezing is widely used in industrial freezing plants. A typical freezing room is taken as the object for discussion, and the load capacity of the room is assumed to be 20,000 kg for meats. Suppose the food to be frozen is beef. According to literature[25], the initial temperature for beef to be frozen is 39°C. The temperature for EEC standard is 7°C, since the precooling is carried out before freezing. The final temperature -18°C is a commonly used temperature in most countries. The specific heat of beef above 0°C is 3.1 kJ/kg. The total heat load to be removed is given by

$$Q_f = G_f * C_f * (t_{f1} - 0) + G_f * (H_{f0} - H_{ff}) \tag{5-22}$$

where Q_f is the total heat rejected from foods in freezing, kJ

G_f is the mass of the meat to be frozen, kg

C_f is the specific heat of foods, kJ/kg.°C

t_{f1} is the initial temperature of foods, °C

H_{f0} is the enthalpy of foods at 0°C, kJ/kg

H_{ff} is the enthalpy of foods at the final temperature, kJ/kg

For an air blast freezer, the temperature difference between the evaporating temperature t_e and the food temperature is usually taken as 10°C, i.e. t_e is 10°C lower than the food temperature. Therefore, t_e = -18 - 10 = -28°C.

The refrigeration plant moves heat from foods to ambience. The condensing temperature t_c must be higher than the ambient temperature. The temperature difference is dependent on the types of condensers. Also the ambient temperature varies with areas and seasons. Therefore, the condenser temperature of a refrigeration plant is dependent on the types of condensers as well as the environment in which the refrigeration plant operates. In this project, the condensing temperature is set as 35°C, which is a typical design condition in the subtropical area.

Freezing time is set as 20 hours, two hours for loading foods, the other two hours for unloading. So that foods can be frozen in a turn of one day (24 hours).

The refrigerating capacity required by a freezer is given by

$$Q_{OH} = \frac{Q_f}{HOU*3600} \qquad \textbf{(5-23)}$$

where Q_{OH} is the refrigerating capacity of the refrigeration plant, kW

HOU is the freezing time (running time of compressors), hour

As is stated earlier, the performance of the new freezing process varies with the evaporating temperature t_{e1} in the first period. Therefore, we have to calculate the performance of the new process under various t_{e1}, in order to discover the relation between the performance and t_{e1}. From the basic equations listed above, it is clear that the amount of calculation is considerable. Fortunately, a personal computer can take over the task. The only thing which needs to be done is to determine the variation range of t_{e1}.

Because the evaporating temperature in the second period of the new process has been set to -28°C, and the evaporating temperature t_{e1} in the first period should be higher than that in the second period, it is reasonable to choose -25°C as the lower bound of variation range of t_{e1}.

The upper bound of the variation range of t_{e1} can be set at -5°C, since there is a temperature difference of 10°C between the refrigerants and the foods. The evaporating temperature at -5°C means that the temperature of the foods cooled is 5°C. It is high enough from a point of view of practice.

For a common freezing process, the energy consumption is given by

$$E_c = N * HOU \tag{5-24}$$

where E_c is energy consumption in a common process, kW.H

N and HOU are the same as above.

For the new freezing process, it consists of two periods. The energy consumed in the first period is given by

$$E_1 = N_1 * HOU_1 \qquad (5\text{-}25)$$

where E_1 is the energy consumed in the first period, kW.H

N_1 is the power required in the first period, kW

HOU_1 is the running time in the first period, hour

The running time of the refrigeration plant in the first period can be obtained from

$$HOU_1 = \frac{Q_{f1}}{Q_{01} * 3600} \qquad (5\text{-}26)$$

where Q_{f1} is the heat rejected from foods during the first period, kJ

Q_{01} is the refrigerating capacity of the refrigeration plant in the first period, kW

When t_{e1} is set up, the temperature of foods is determined, thus the heat Q_{f1} rejected from foods can be obtained by equation (5-22).

The energy consumed in the second period is given by

$$E_2 = N_2 * HOU_2 \qquad (5\text{-}27)$$

where E_2 is the energy consumed in the second period, kW.H

N_2 is the power required in the second period, kW

HOU_2 is the running time in the second period, hour

The running time HOU_2 is given by

$$HOU_2 = \frac{Q_{f2}}{Q_{02} * 3600} \tag{5-28}$$

where Q_{02} is the refrigerating capacity in the second period, kW

Q_{f2} is the heat rejected from foods during the second period, kJ

The heat rejected from foods during the second period can be obtained directly from

$$Q_{f2} = Q_f - Q_{f1} \tag{5-29}$$

Then the energy consumption in the new freezing process is

$$E_n = E_1 + E_2 \tag{5-30}$$

where E_n is the energy consumption in the new freezing process, kW.H

The comparison of energy consumption between the new process and the common process is given by

$$E_s = \frac{E_c - E_n}{E_c} * \% \tag{5-31}$$

where E_s is the percent of energy saving of the new process compared to the common process.

The comparison of the freezing time required in the new process and the common process is given by

$$HOU_s = \frac{HOU - (HOU_1 + HOU_2)}{HOU} * \% \tag{5-32}$$

where HOU_s is the percentage of saving freezing time for the new process to the common process.

The time required by the new freezing process is

$$Time_n = HOU_1 + HOU_2 \tag{5-33}$$

where $Time_n$ is the total freezing time in the new freezing process, hour.

5.3 Computer simulation program

There are many computer languages nowadays, each has its features. Quick Basic is employed in this project, since it is a compilation language which has more functions and is more flexible compared to some other languages.

According to the above discussion, there are three schemes in the calculation. Each should be computed for its performance of the common freezing process and its performance of the new freezing process under different t_{e1}, then the comparison can be carried out. Figure 5-3 is the flow chart of the whole program. It consists of a main program and three sub-programs (for the three schemes). When the main program calls the sub-programs, it passes variables of the refrigerants and initial temperatures of foods to the sub-programs.

Figure 5-4(a and b) is the flow chart of steps of the main simulation program. A FOR...NEXT loop is used in it. The control variable is t_{e1}, the initial value of t_{e1} is -25°C and the limiting value is -5°C. The step value is 1, since it is accurate enough in industrial refrigeration plants.

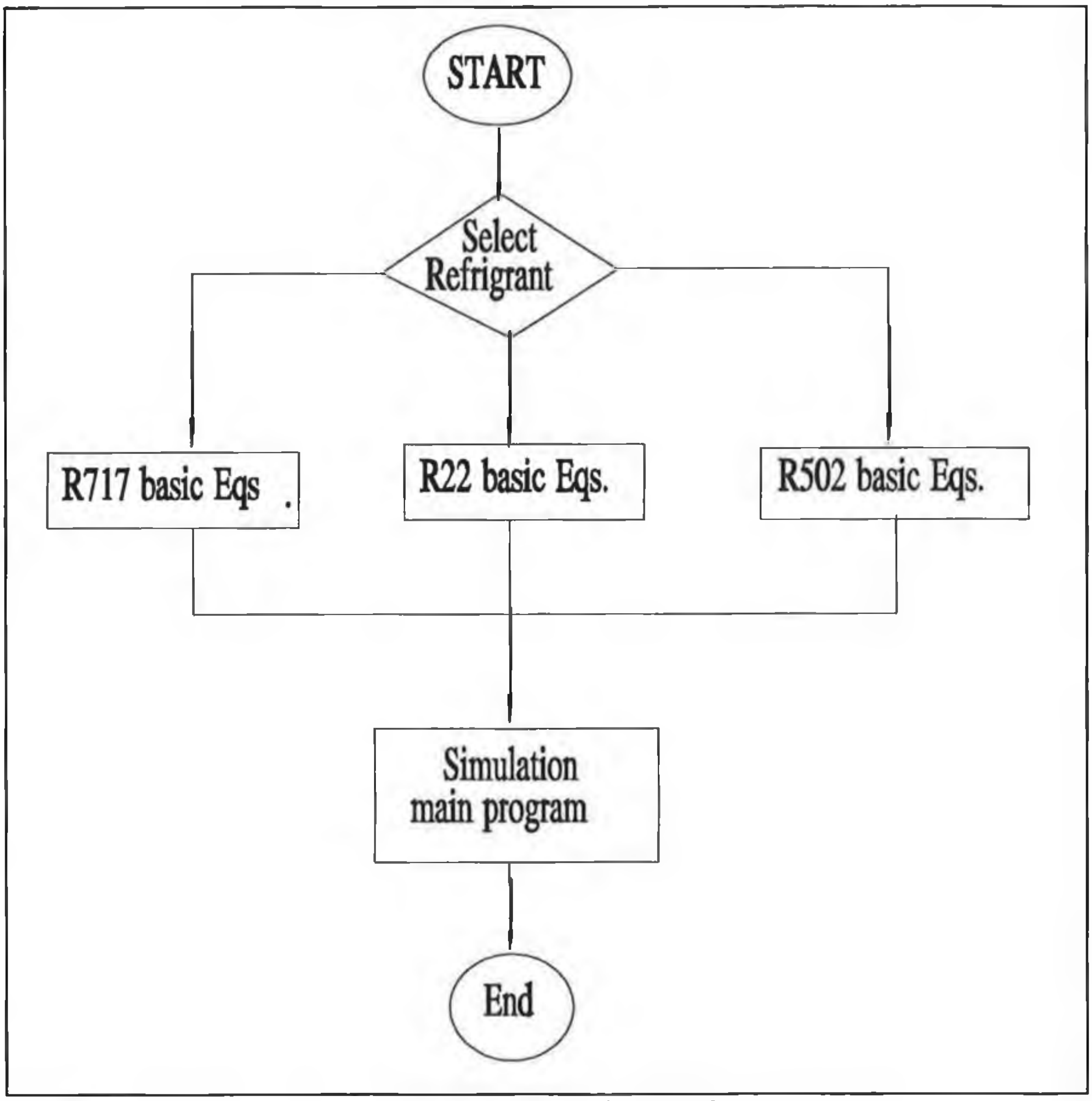


Fig.5-3 The flow chart of the whole simulation program

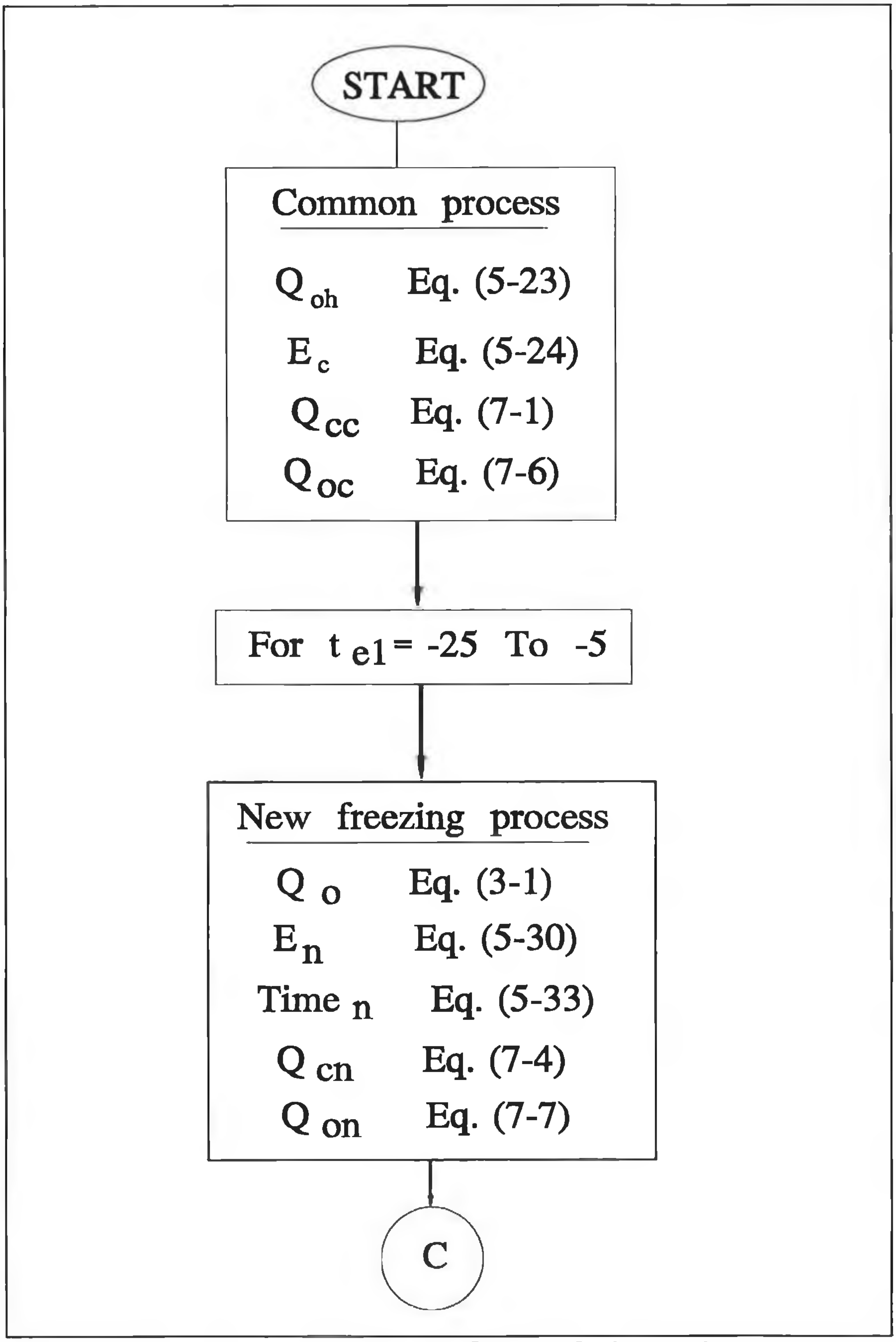


Fig.5-4a The flow chart of steps of the simulation main program

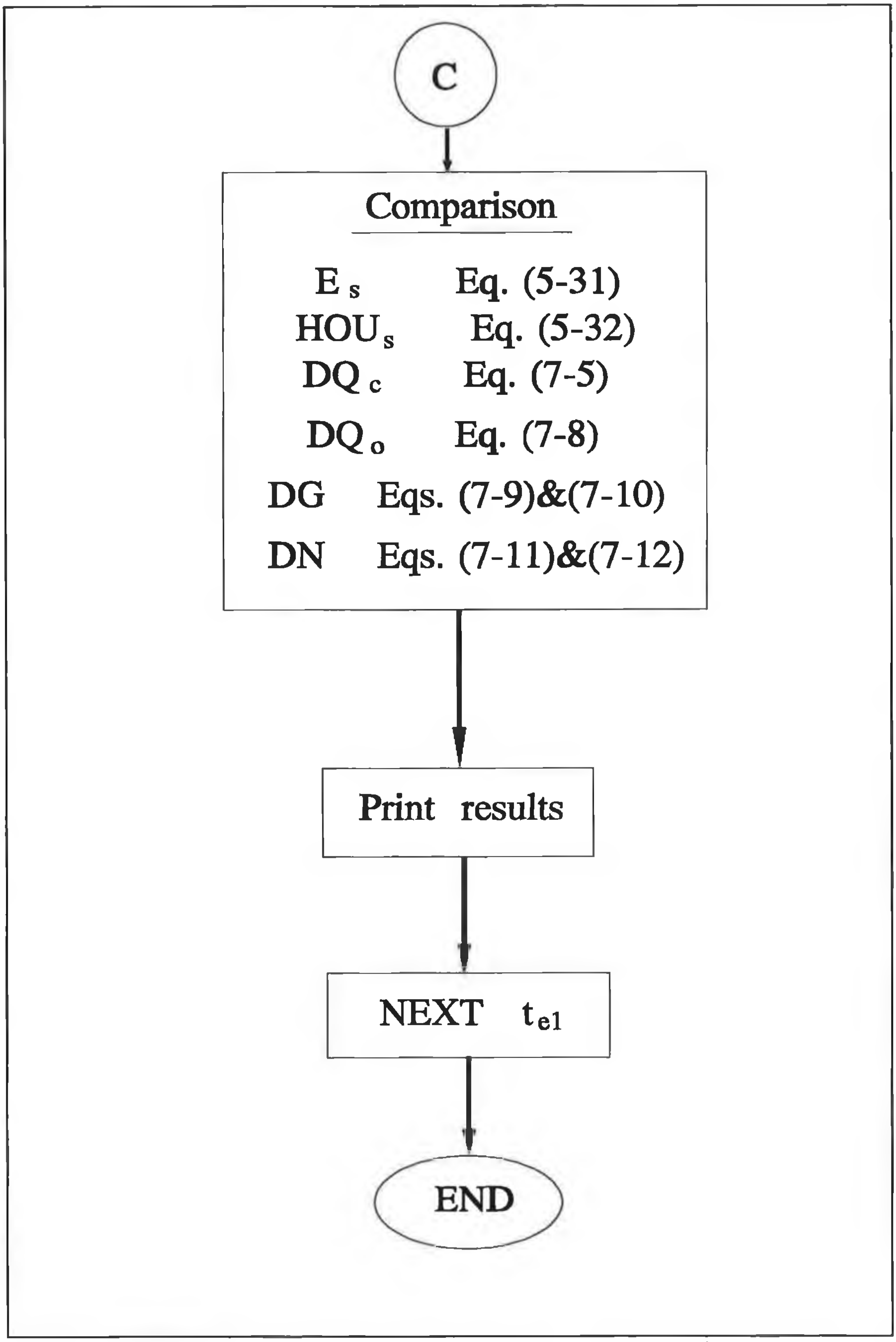


Fig.5-4b The flow chart of steps of the simulation main program

CHAPTER 6 RESULTS AND DISCUSSIONS OF THE OPTIMUM EVAPORATING TEMPERATURE t_{e1}

As stated earlier, the main purpose of this study is to investigate the possibility of saving energy (or energy cost) in food refrigeration. When the new freezing process is employed, the performance of a refrigeration plant varies with the evaporating temperature of the first period. There is an optimal evaporating temperature at which the energy saving of the new freezing process is maximum. This temperature is defined in this study as the optimum temperature t_{e1}. For a practical plant, many parameters of its operation vary with different cases. As a result, the optimum evaporating temperature t_{e1} is not a constant. The variations of t_{e1} under various typical cases are discussed in this chapter, based on the results of the simulation program.

6.1 Optimum t_{e1} for different refrigerants

Many different refrigerants have been used since the early days of mechanical refrigeration. There are several dozens of refrigerants nowadays. The choice of a refrigerant for a particular application frequently depends on properties not related to its ability to remove heat, for example, it toxicity, flammability, density, viscosity, availability and price.

In fact, mechanical compression refrigeration is facing a crisis in relations to refrigerants[72]. Chlorofluorocarbons (CFCs) are widely used as refrigerants in the world, but it is now clear that some of them are causing serious damage to the environment. An international protocol has been reached which limits the future use of some of these compounds[73]. Based on the situation and current industrial freezing plants[74], three refrigerants, R717, R22 and R502, are selected and discussed in this study.

Refrigerant R717

Refrigerant R717, i.e. ammonia, is one of the old steady refrigerants. It was used in some of the earliest equipment, and its use continues in the larger commercial and industrial plants, especially in the developing countries. It is likely to be used more widely in the world due to the problem of CFCs.

R717 is a chemical compound of nitrogen and hydrogen (NH_3) and under ordinary conditions is a colourless gas. Its boiling temperature at atmospheric pressure is -33°C and its melting point from the solid is -78°C. The low boiling point makes it possible to have refrigeration at temperatures considerably below zero without using pressure below atmospheric in the evaporators.

R717 is not only of excellent thermodynamic properties, but also of lower price and of availability. Therefore, it is commonly used in the industry although its toxicity is a shortcoming. As a result, R717 is taken as a main object among the three refrigerants in this study. Figure 6-1 to 6-3 are the results of the simulation program on this project.

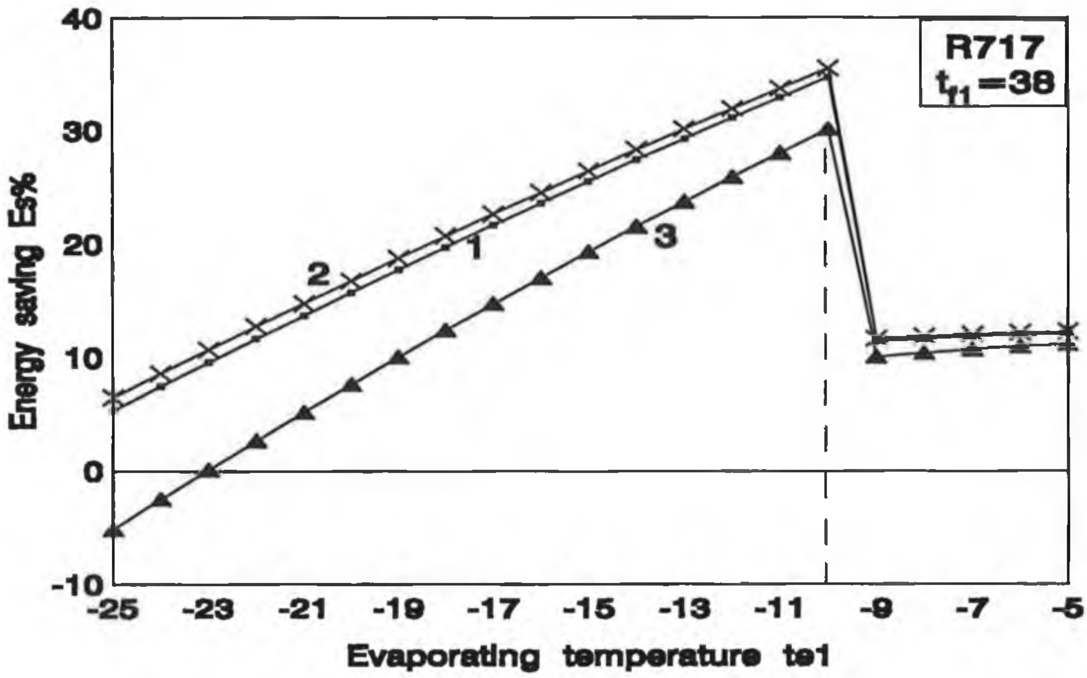


Figure 6-1. Energy saving of three schemes of the new freezing process with R717

Figure 6-1 shows the energy saving (E_s%) of the new freezing process compared with the common freezing process. Curves 1, 2 and 3 refer to the schemes 1, 2 and 3 described in Chapter 4, and this labelling will be used throughout this chapter for all the figures. From Figure 6-1, the E_s of scheme 2 is always greater than that of the scheme 1. But the difference is not considerable. The E_s of scheme 3 is obviously much lower than those of schemes 1 and 2. The three curves reach maximum values when the evaporating temperature t_{e1} is at -10°C.

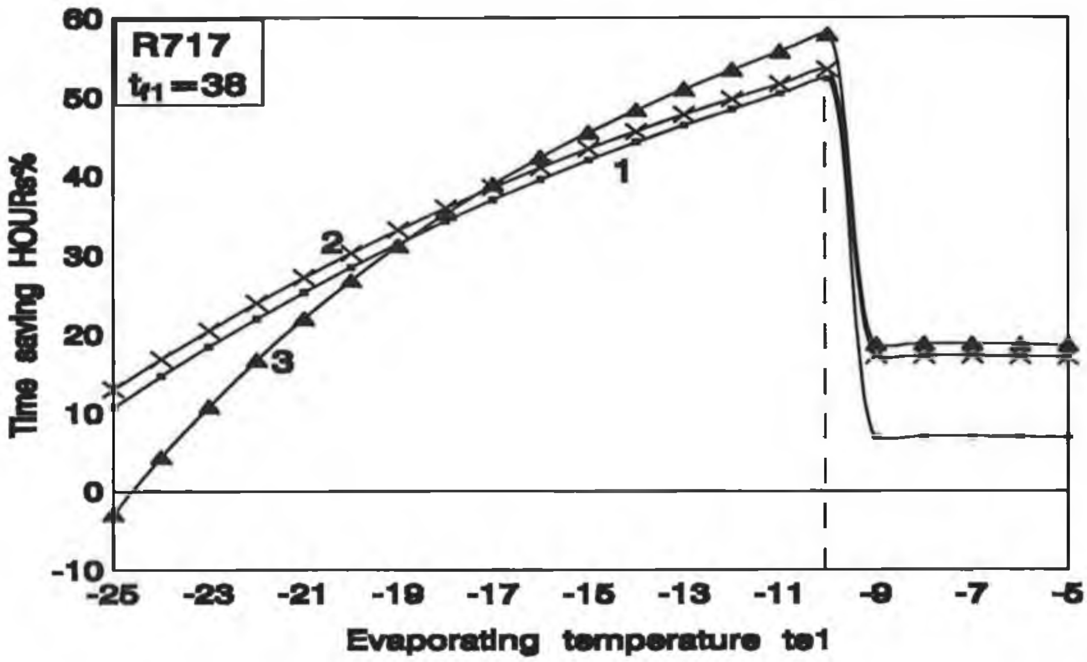


Fig. 6-2 Time saving of three schemes of the new freezing process with R717

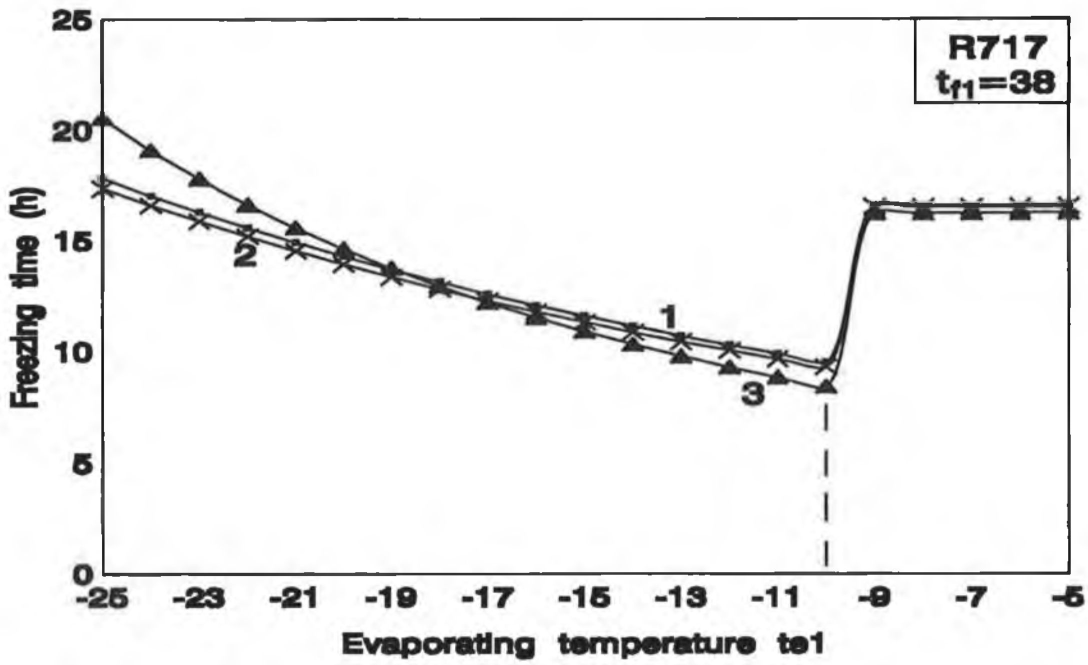


Fig.6-3 Freezing time of three schemes of the new freezing process with R717

The time saving ($HOUR_s\%$) of the new process versus the common process is shown in Figure 6-2. When the evaporating temperature t_{e1} is at -10°C, all three curves reach their maximum. But the values for scheme 3 are always minimum among the three curves as the evaporating temperature t_{e1} varies.

The variations of the freezing time on the three schemes are shown in Figure 6-3. At the point of optimum t_{e1} (t_{e1} = -10°C), all the schemes have minimum freezing time. The value of scheme 3 is minimum among the three curves. But the differences among the three schemes are not significant.

Refrigerant 22

Refrigerant 22 is a synthetic, man-made, refrigerant developed for refrigeration installations that need a low evaporating temperature. It is referred to as "monochlorodifluoromethane" and also as "chlorodifluoromethane". R22 is stable and is nontoxic, noncorrosive, non-irritating and nonflammable. Moreover, R22 has a much lower ozone depleting potential than either R11 or R12, and is seen as a transitional substance which can be used during the next 30 years or so until a more suitable refrigerant can be developed. R22 has a boiling point of -40.8°C at atmospheric pressure. Freezing plants are one of its important applications. Figures 6-4 to 6-6 are the results of calculation when R22 is employed.

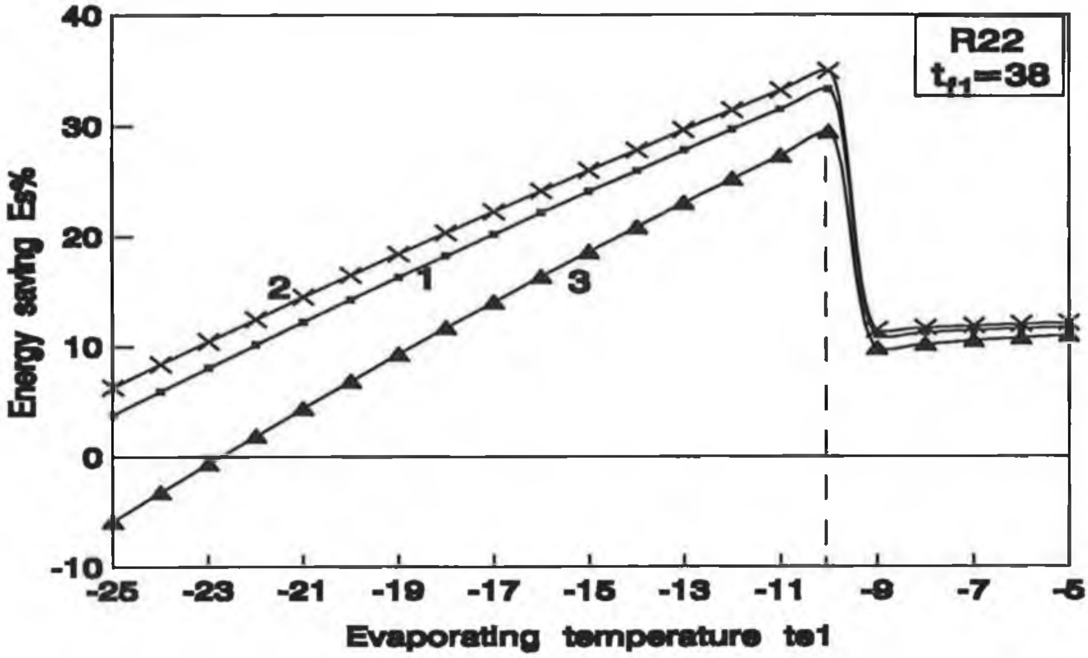


Fig.6-4 Energy saving of three schemes of the new freezing process with R22

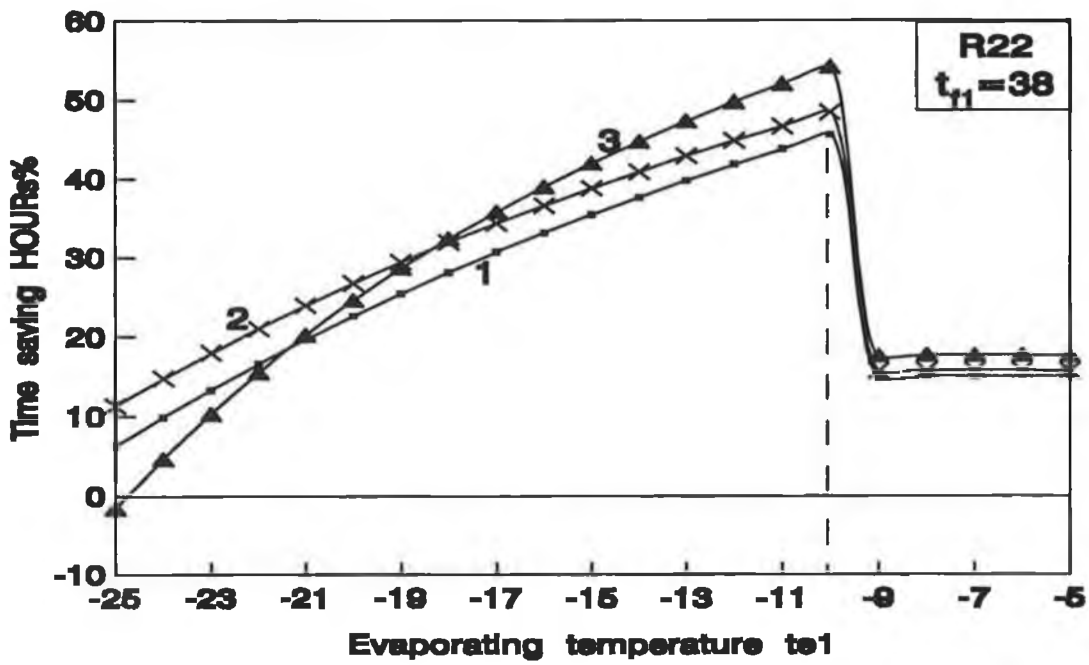


Fig.6-5 Time saving of three schemes of the new freezing process with R22

Figure 6-4 shows the energy saving (E_s%) of the new freezing process compared with the common freezing process. From Figure 6-4, the E_s of the scheme 2 is always greater than that of the scheme 1 even though the difference is very small. The E_s of scheme 3 is obviously smaller than those of schemes 1 and 2. The three curves reach their maximum value when the evaporating temperature t_{e1} is at -10°C.

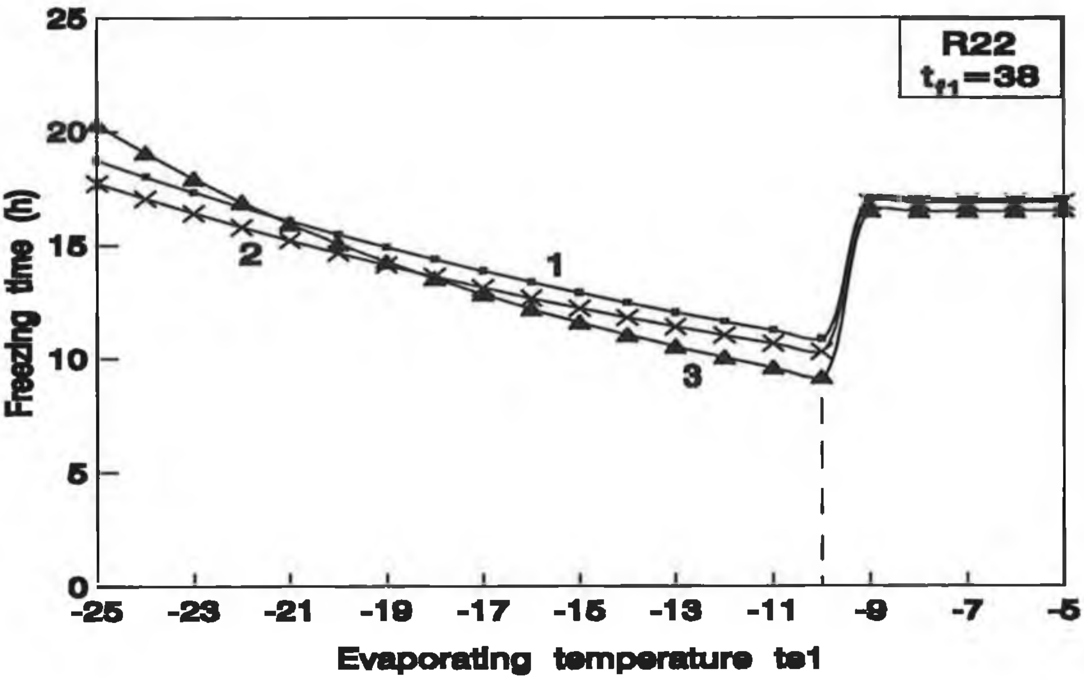


Fig.6-6 Freezing time of three schemes of the new freezing process with R22

The time saving (HOUs%) of the new process verses the common process is shown in Figure 6-5. When the evaporating temperature t_{e1} is at -10°C, all three curves reach their maximum. But the values of the scheme 3 are always minimum as the evaporating temperature t_{e1} varies.

The variations of the freezing time on the three schemes are shown in Figure 6-6. At the point of optimum t_{e1} (t_{e1} = -10°C), all the schemes have minimum freezing time. The value of the scheme 3 is minimum. But the differences among three schemes are small.

Refrigerant R502

Refrigerant R502 is an azeotropic mixture of 48.8 percent R22 and 51.2 percent R115. It has been used since 1961. It is a nonflammable, non corrosive, practically nontoxic liquid, and a good refrigerant for obtaining medium and low temperature (from -18°C to -51°C). R502 contains the CFC based R115, and will probably be unavailable after the year 2000.

R502 is often used in frozen food lockers, frozen food processing plants, frozen food display cases and storage units for frozen foods and ice cream. Its boiling point is -45.6°C at atmospheric pressure. Figure 6-7 to 6-9 are the results of the simulation program when R502 is used.

Figure 6-7 shows the energy saving (E_s%) of the new freezing process compared with the common freezing process. From Figure 6-7, the E_s of scheme 2 is always greater than that of scheme 1. But the differences of the values are not considerable. The E_s of scheme 3 is obviously smaller than those of schemes 1 and 2. The three curves reach their maximum value when the evaporating temperature t_{e1} is at -10°C.

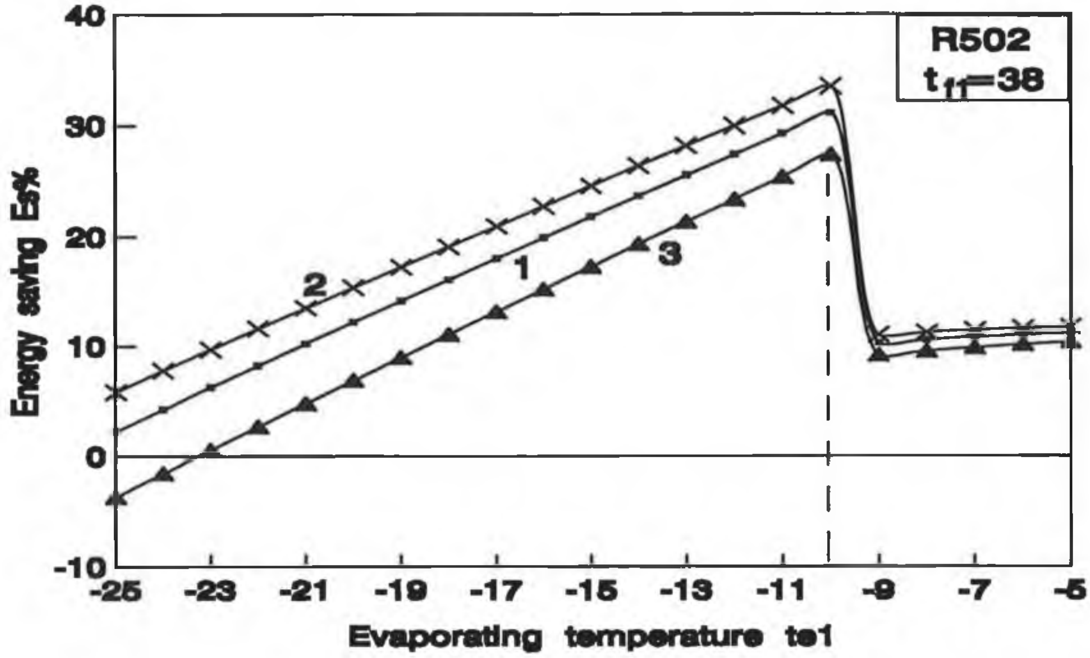


Fig.6-7 Energy saving of three schemes of the new freezing process with R502

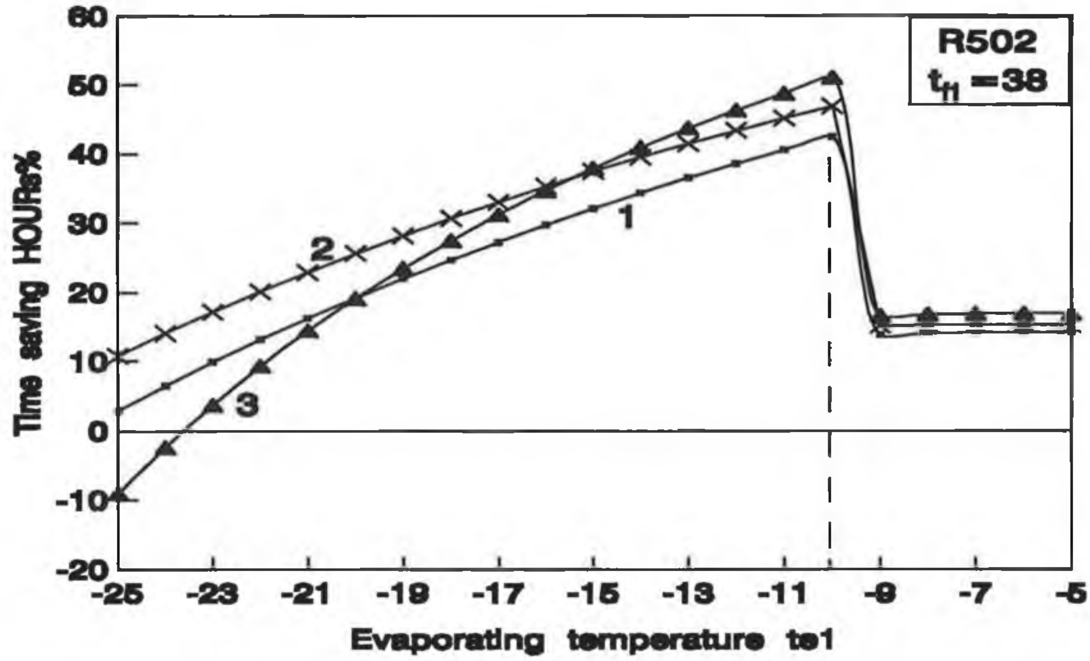


Fig.6-8 Time saving of three schemes of the new freezing process with R502

The time saving ($HOU_s\%$) of the new process verses the common process is shown in Figure 6-8. When the evaporating temperature t_{e1} is at -10°C, all three curves reach their maximum. But the values of scheme 3 are always minimum as the evaporating temperature t_{e1} varies.

The variations of the freezing time on the three schemes are shown in Figure 6-9. At the point of optimum t_{e1} (t_{e1} = -10°C), all the schemes have minimum freezing time. The value of scheme 3 is minimum. But the differences among three schemes are not so significant.

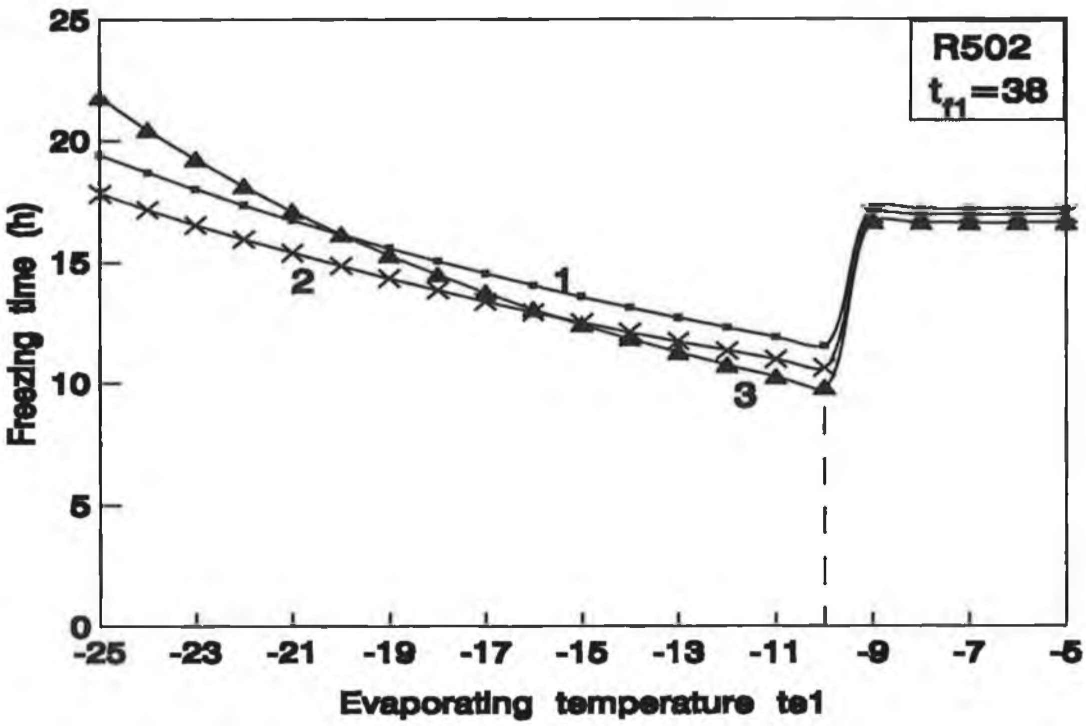


Fig.6-9 Freezing time of three schemes of the new freezing process with R502

Compared with the results of three refrigerants, it can be seen that there is a same optimum evaporating temperature t_{e1} (-10°C) and there is no obvious difference on the values of energy saving for different refrigerants. But the different schemes have evident effects to the values of energy saving. At the optimum t_{e1}, scheme 2 has a maximum for energy saving among the three schemes, and scheme 3 has a maximum for the saving of the freezing time.

6.2 Optimum t_{e1} with the variation of the t_c

For an industrial freezing plant, the evaporating temperature of refrigerant depends on the requirement for frozen food. In other words, the evaporating temperature will not change after a freezing plant and frozen foods are determined. On the other hand, the condensing temperature of a refrigerating plant depends on the ambient temperature. It is well known that ambient temperature is always changeable. As a result, the condensing temperature of a refrigerant plant is always variable. Therefore, it is necessary to discuss the optimum evaporating temperature t_{e1} with the variation of the condensing temperature.

The results of the computer program show that the optimum evaporating temperature t_{e1} does not vary with the condensing temperature of refrigerant.

All the three refrigerants have a same optimum evaporating temperature t_{e1} = -10°C while the condensing temperature varies. But the energy saving (E_s%) of the new freezing process varies with the condensing temperature. The variations are shown in Figure 6-10 to 6-12.

Figure 6-10 shows that the energy saving (E_s%) of R717 varies with the condensing temperature at the optimum evaporating temperature t_{e1} (-10°C). All three curves (scheme 1, 2 and 3) vary inversely as the condensing temperature, but the rates of variation are small.

Figures 6-11 and 6-12 show that the energy saving (E_s%) of R22 and R502 varies with the condensing temperature at the optimum evaporating temperature t_{e1} (-10°C). All three curves (schemes) also vary inversely as the condensing temperature, the rates of variation are still small. The values of scheme 3 are smallest among the three schemes.

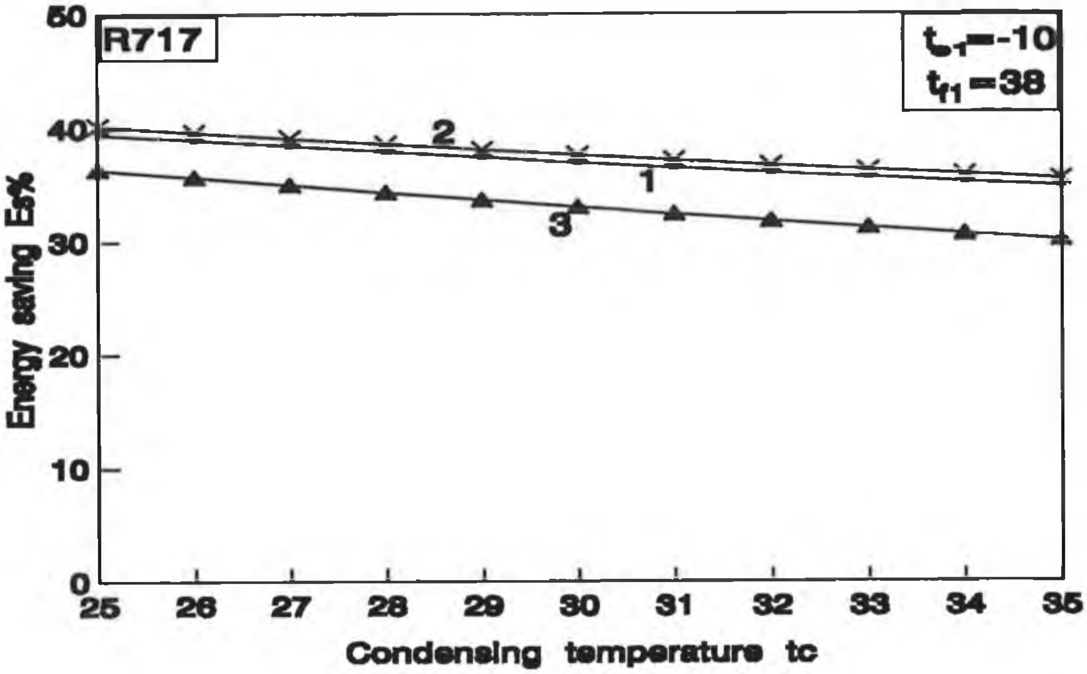


Fig.6-10 Energy saving at optimum evaporating temperature t_{e1} under different condensing temperature with R717

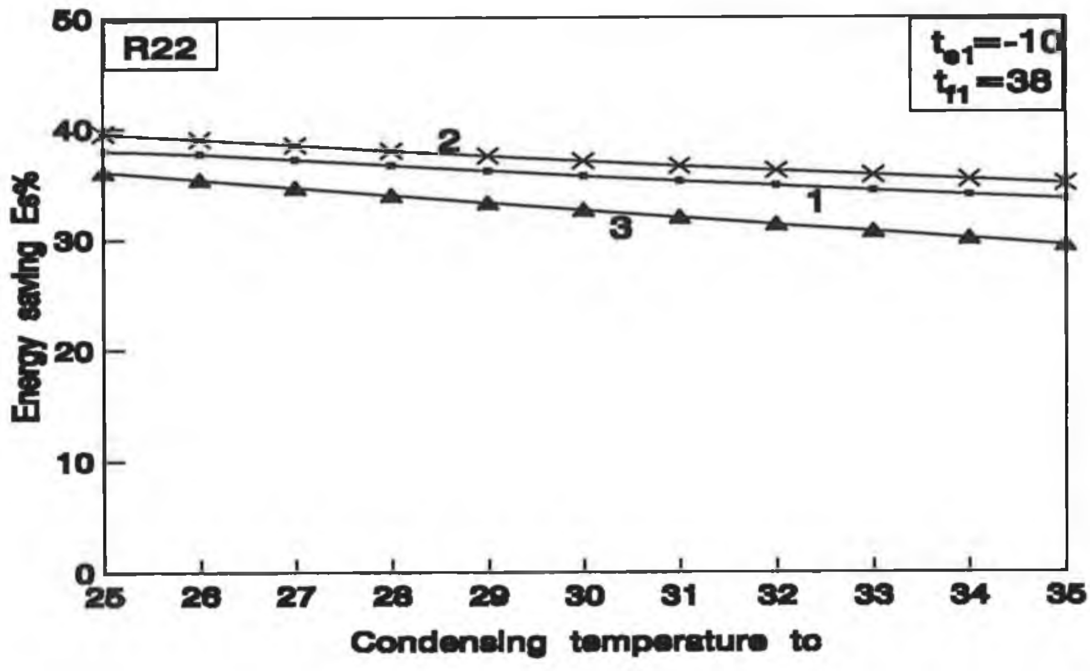


Fig.11 Energy saving at optimum evaporating temperature t_{e1} under different condensing temperatures with R22

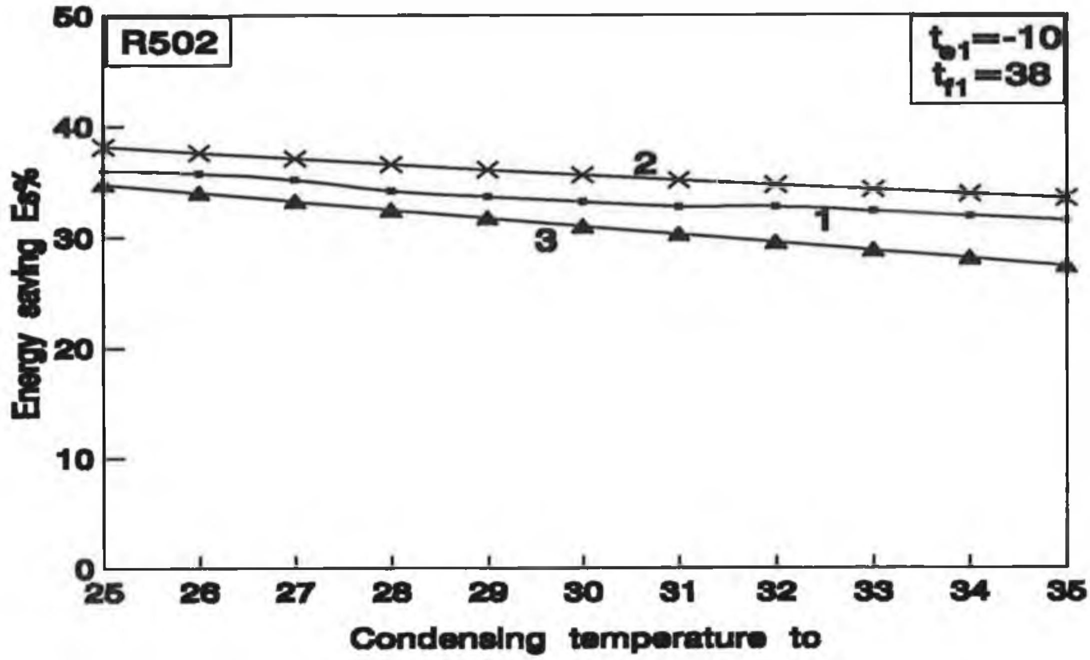


Fig.6-12 Energy saving at optimum evaporating temperature under different condensing temperatures with R502

6.3 Optimum t_{e1} with the variation of the initial temperature t_{f1} of foods

As stated in section 2.3, precooling procedure to meat to be frozen is adopted before freezing in many countries, such as EEC countries. According to the standard of EEC[(36)], meat should be chilled to 7°C before it is frozen. Therefore, it is a practical problem to discuss the optimum evaporating temperature t_{e1} with the variation of the initial temperature t_{f1} of foods to be frozen.

The initial temperature t_{f1} = 7°C is taken for the calculation. The results show that the optimum evaporating temperature t_{e1} does not vary when the initial temperature t_{f1} of foods varies.

Figure 6-13 shows the energy saving $E_s\%$ when the initial temperature t_{f1} is at 7°C and the refrigerant is R717. The values of energy saving ($E_s\%$) at the optimum evaporating temperature t_{e1} have a very slight variation. The variation is not over 1% compared with the initial temperature of food t_{f1} = 38°C (see Figure 6-1).

Figures 6-14 and 6-15 show the energy saving $E_s\%$ when the initial temperature t_{f1} is at 7°C and the refrigerants are R22 and R502, respectively. The values of energy saving ($E_s\%$) at the optimum evaporating temperature t_{e1} also have a very slight variation. The variation is less than 1% compared with the initial temperature of food t_{f1} = 38°C (see Figures 6-4 and 6-7). From the view of industrial practice, the difference can be ignored. Therefore, it is safe to say that the optimum evaporating temperature t_{e1} of the new freezing process does not vary with the initial temperature (t_{f1}) of food to be frozen.

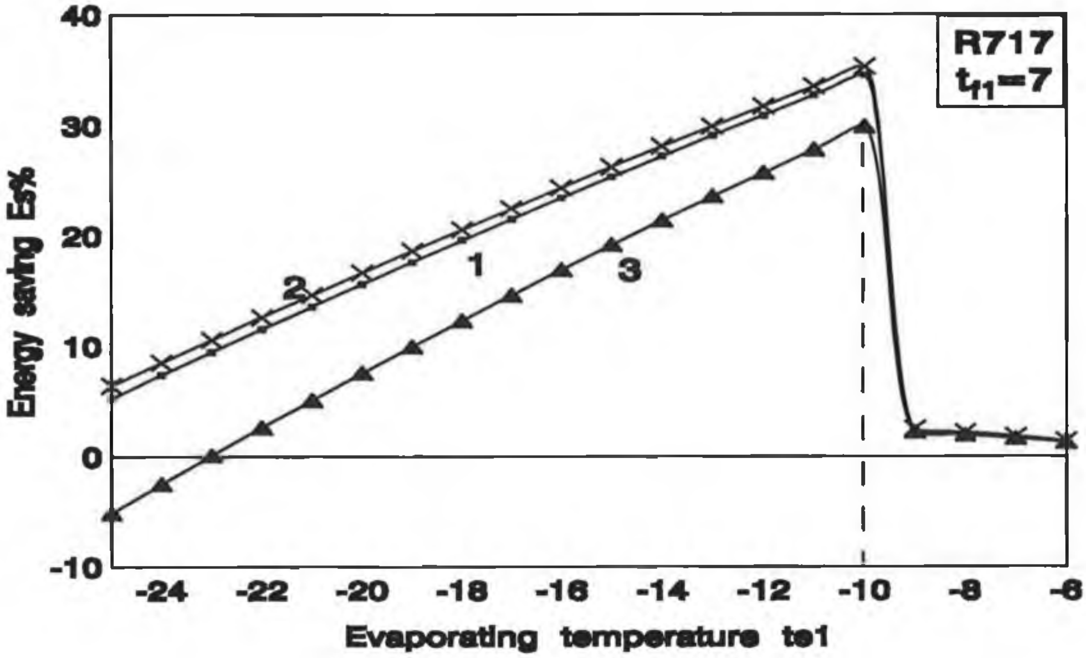


Fig.6-13 Energy saving with R717 when the initial temperature of food is at 7°C

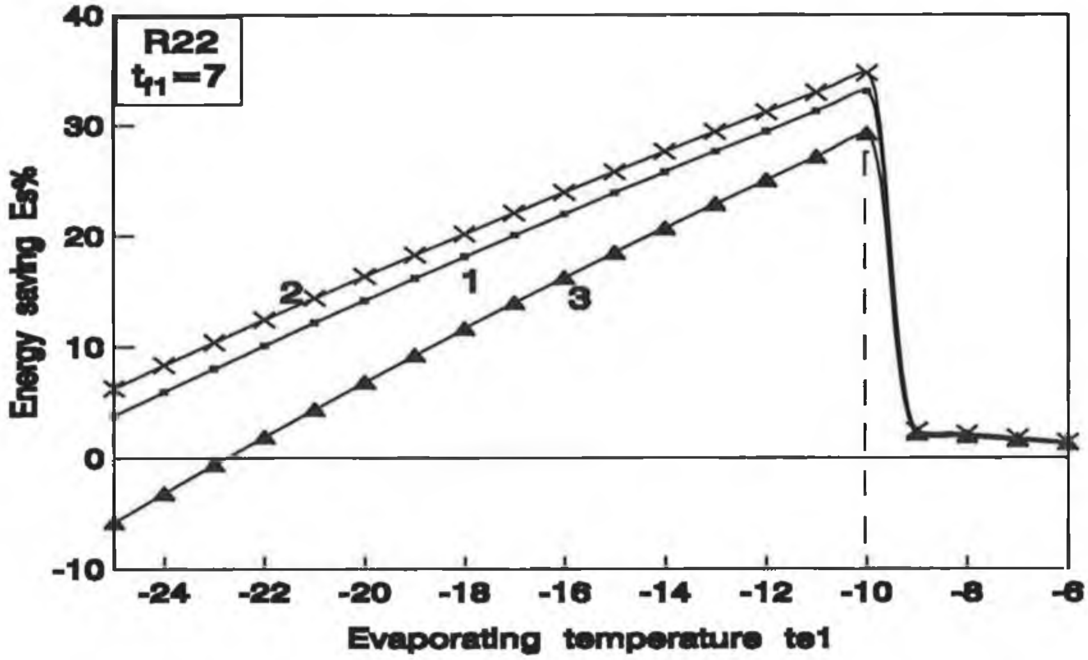


Fig.6-14 Energy saving with R22 when the initial temperature of food is at 7°C

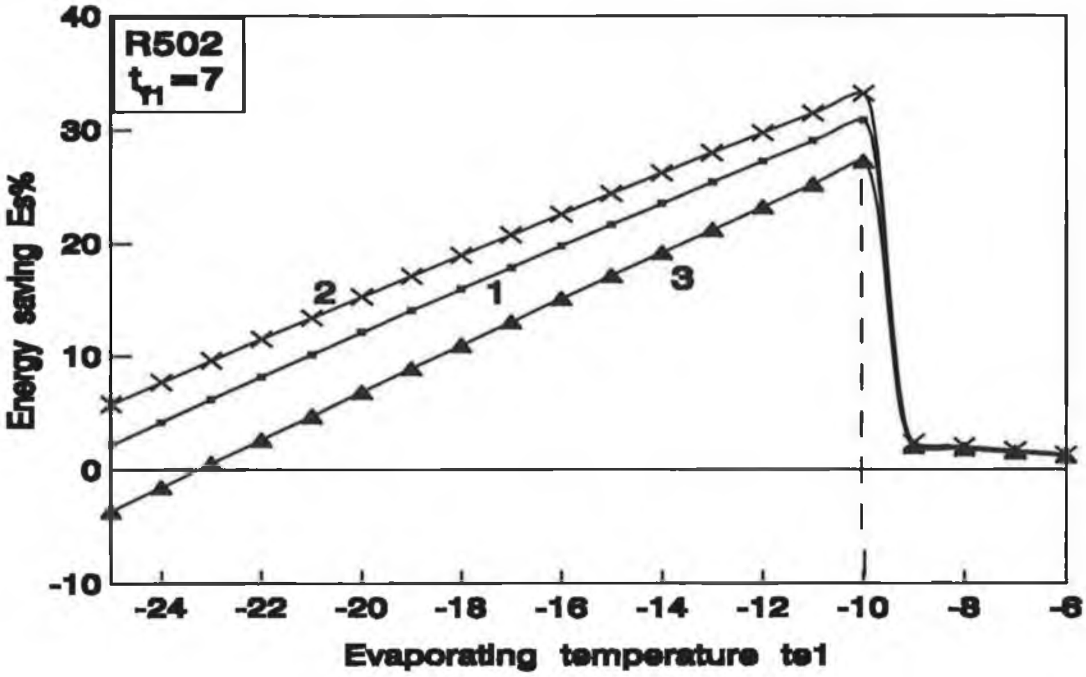


Fig.6-15 Energy saving with R502 when the initial temperature of food is at 7°C

6.4 Optimum t_{e1} with the variation of different foods

The frozen food discussed above is beef. Clearly, many foods are frozen in industrial freezers. Therefore, it is necessary to examine the optimum evaporating temperature t_{e1} in relation to different foods. Figure 6-16 shows the results of the calculation when the frozen food is cod and the refrigerant is R717.

Comparing Figure 6-16 with Figure 6-1, it is clear that the results are nearly the same. Further analysis shows that the increase of energy saving (E_s%) is about 2% more under the three schemes when beef is replaced by cod.

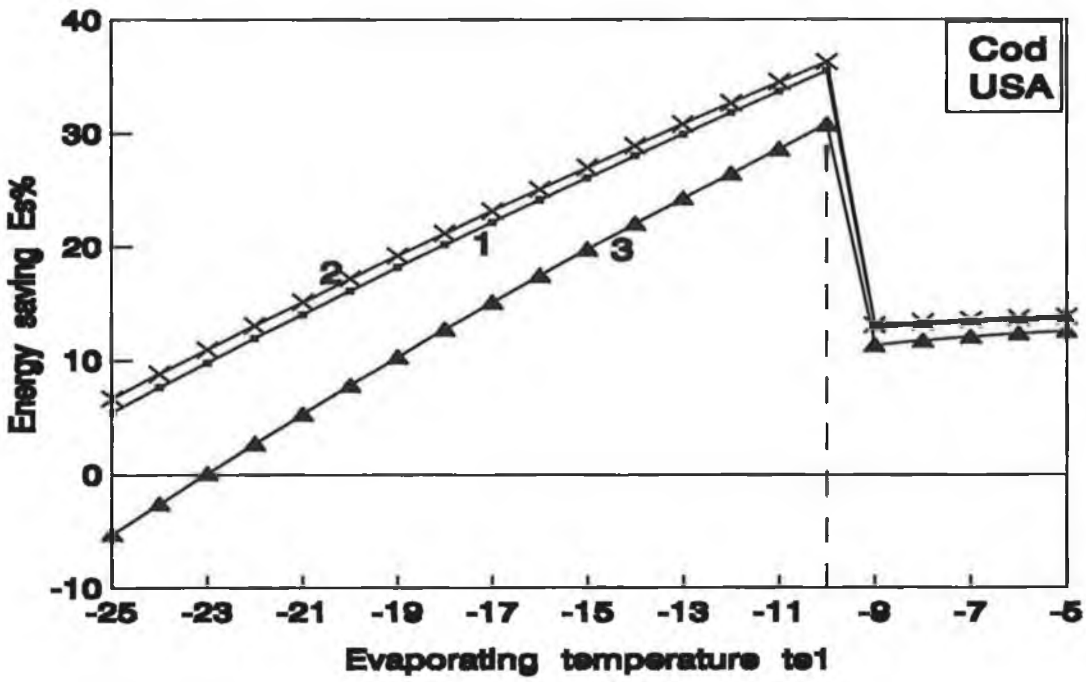


Fig.6-16 Energy saving with R717 when frozen food is cod (USA datums)

Comparing the data of food properties between ASHRAE handbook[75] and the recommendations[21] of the International Institute of Refrigeration (IIR), there is a considerable difference between the two datums. Therefore, the data from IIR are adopted for further examinations. Figure 6-17 to 6-19 are the results of these calculations.

Figure 6-17 shows the results when the frozen food is beef, based on the datums of IIR. Comparing Figure 6-17 with Figure 6-1, it can be seen that the optimum evaporating temperatures under the three schemes do not vary and is still at -10°C. But the values of energy saving (E_s%) increase considerably.

Figure 6-18 shows the results when the frozen food is fish (average values from IIR are adopted in the calculation). Comparing Figure 6-18 with Figure 6-16, it can also be seen that the optimum evaporating temperatures t_{e1} under the three schemes do not vary and is still at -10°C. But the values of energy saving (E_s%) increase considerably.

Figure 6-19 shows the results when the frozen food is pork. Comparing Figure 6-17, 6-18 and 6-19 with Figure 6-16 and 6-17, it is easy to observe that the results are greater under the IIR datums than under the ASHRAE datums, while the optimum evaporating temperatures t_{e1} under tall three schemes is always at -10°C. The rates of variation of curves in the two datums are the same.

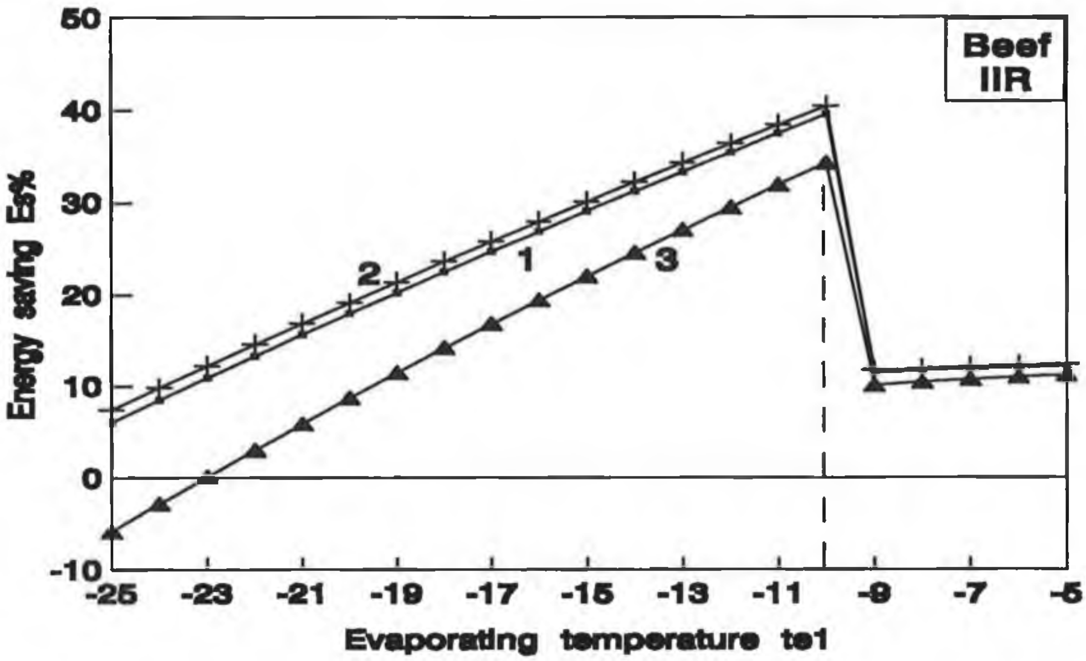


Fig.6-17 Energy saving with R717 when frozen food is beef (IIR datums)

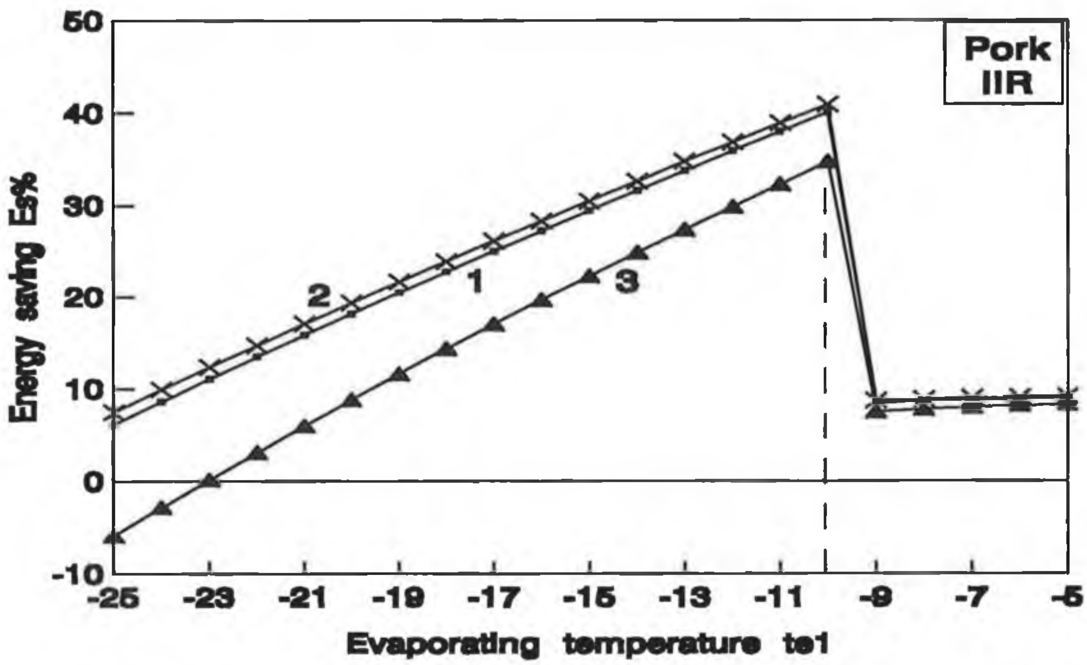


Fig.6-18 Energy saving with R717 when frozen food is pork (IIR datums)

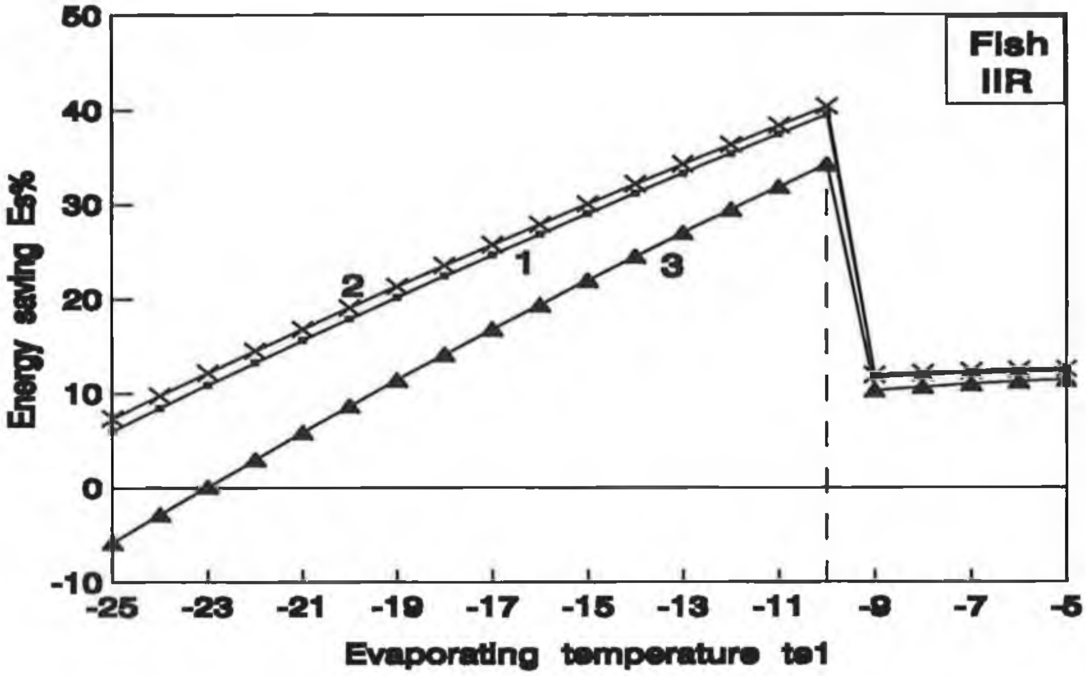


Fig.6-19 Energy saving with R717 when frozen food is fish (IIR datums)

CHAPTER 7. SOME PRACTICAL CONSIDERATIONS

One of the important features on this new technique is that the new freezing process utilizes the same refrigerating compressors of the common freezing process. As mentioned previously, new parameters are selected and operated in the refrigerating plant during the first period of the new freezing process. As a result, the performance of each individual component will change. In addition to this fact, when the components operate as a system, each has an effect on the performance of the other.

As stated in section 2.2, a compression refrigeration plant consists of four major components, i.e. compressor, condenser, expansion device and evaporator. The performance of each component and their match with each other in the new freezing process are discussed in this chapter.

7.1 The performance of condensers

The condenser is one of the major components in a compression refrigeration plant. As a heat exchanger, it removes the condensation heat from the refrigerant vapour and makes the vapour changed into liquid refrigerant.

In a common refrigerating plant, the condenser heat load is given by

$$Q_{cc} = q_{cc} * G \tag{7-1}$$

where Q_{cc} is the condenser heat load in the common freezing process (kW), i.e. heat rejected in condensers from a refrigerant in a refrigerating cycle. G_c is the mass flow rate of refrigerant (kg/s). And q_{cc} is the specific heat load of condensers, i.e. heat rejected by a unit of refrigerant in condensers (kJ/kg). It is given by

$$q_{cc} = H_{ca} - H_{cb} \tag{7-2}$$

where:

H_{ca} is the enthalpy of refrigerant entering condensers, kJ/kg;

H_{cb} is the enthalpy of refrigerant leaving condensers, kJ/kg.

As an example in figure 3.1,

$$q_{cc} = H_2 - H_3 \tag{7-3}$$

In the second period of the new freezing process, the operating condition of the refrigerating plant is nearly the same as the common freezing process. Therefore, only the first period of the new freezing process needs to be discussed.

In the first period of the new freezing process, the refrigerating capacity of the refrigeration plant increases since the temperature difference (temperature range) of operation of the refrigerating plant is smaller. That means that more heat is transferred from low temperature source to high temperature sink, i.e. from evaporators to condensers. The condenser heat load in this case is given by

$$Q_{cn} = q_{cn} * G_n \tag{7-4}$$

where Q_{cn} is the condenser heat load in the new freezing process (kW), i.e. heat rejected in the condensers from the refrigerant in the refrigerating process. G_n is the mass flow rate of refrigerant (kg/s). And q_{cn} is the specific heat load of the condensers in the new process, i.e. heat rejected by a unit of refrigerant in the condensers (kJ/kg). The q_{cn} can be obtained with equation (7-2), although the enthalpy value of each state point may vary.

In the first period of the new freezing process, the increase of the condenser heat load compared with the common process is given by

$$DQ_c = \frac{Q_{cn} - Q_{cc}}{Q_{cc}} * 100\% \tag{7-5}$$

where DQ_c is the percentage of the increase of the condenser heat load compared with the common process. Equation (7-5) is calculated using a personal computer. The results under different conditions are shown in Figures 7-1, 7-2 and 7-3.

Figure 7-1 shows the result when R717 is employed. Curves 1, 2 and 3 correspond to schemes 1, 2 and 3 respectively which are proposed in chapter 4. This labelling will be used throughout this chapter. According to Figure 7-1, it is clear that the condenser heat load increases with a rise of the evaporating temperature t_{e1}. Curves 1 and 2 have a same rate of increase. The rate of increase for curve 3 is the greatest among the three curves. At the optimum evaporating temperature t_{e1}, the increase of the condenser heat load is about 100% for schemes 1 and 2, and 127% for scheme 3.

Figure 7-2 shows the result when the refrigerant is R22. It is clear that the condenser heat load increases with a rise of the evaporating temperature t_{e1}. Curves 1 and 2 still have a same rate of increase, but the actual increase for curve 2 is greater. The rate of increase for curve 3 is the greatest among the three curves. At the optimum evaporating temperature t_{e1}, the increase of the condenser heat load is 72.8% for scheme 1, 81.4% for scheme 2, and 108% for scheme 3.

Figure 7-3 shows the result when the refrigerant is R502. It is clear that the results are similar to those in figure 7-2. At the optimum evaporating temperature t_{e1}, the increase of the condenser heat load is 64% for scheme 1, 76% for scheme 2, and 95% for scheme 3.

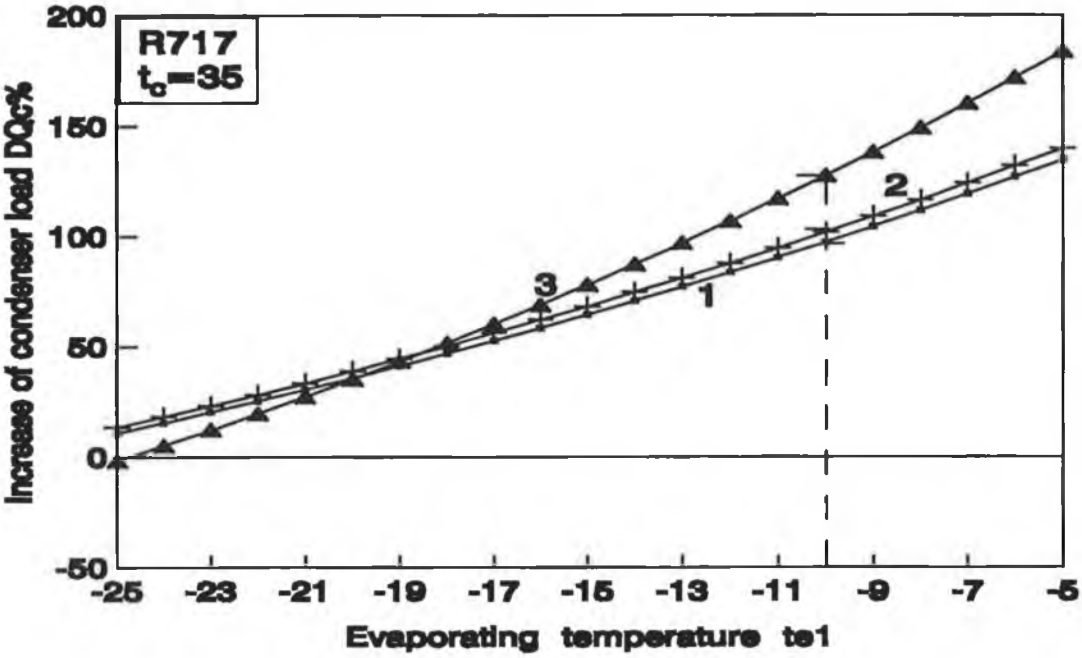


Fig.7-1 The increase of the condenser heat load with R717

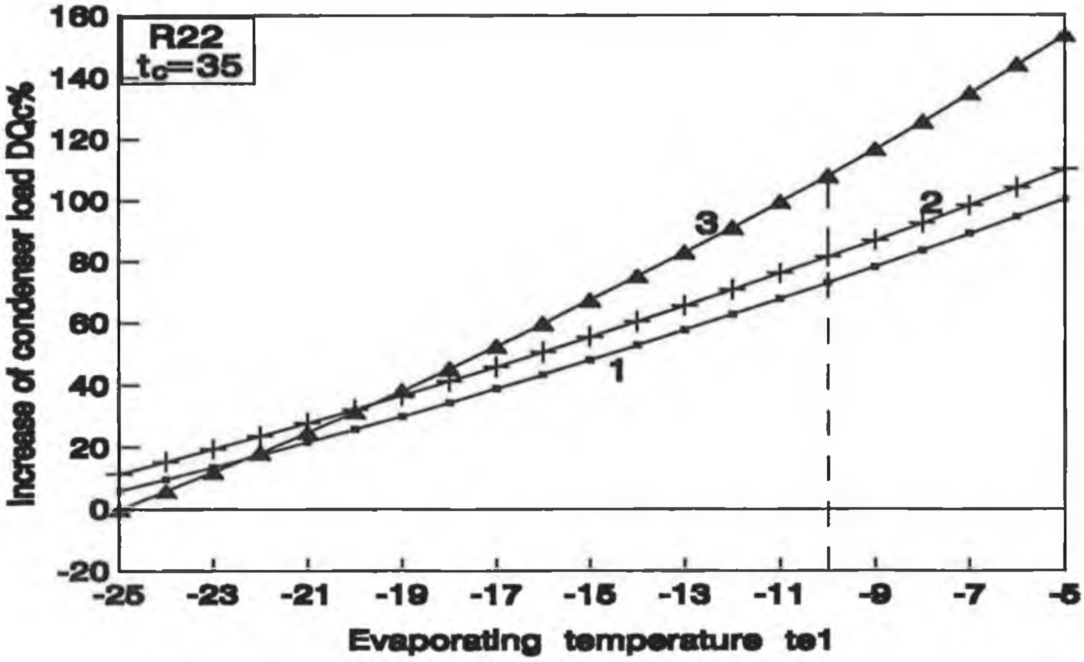


Fig.7-2 The increase of the condenser heat load with R22

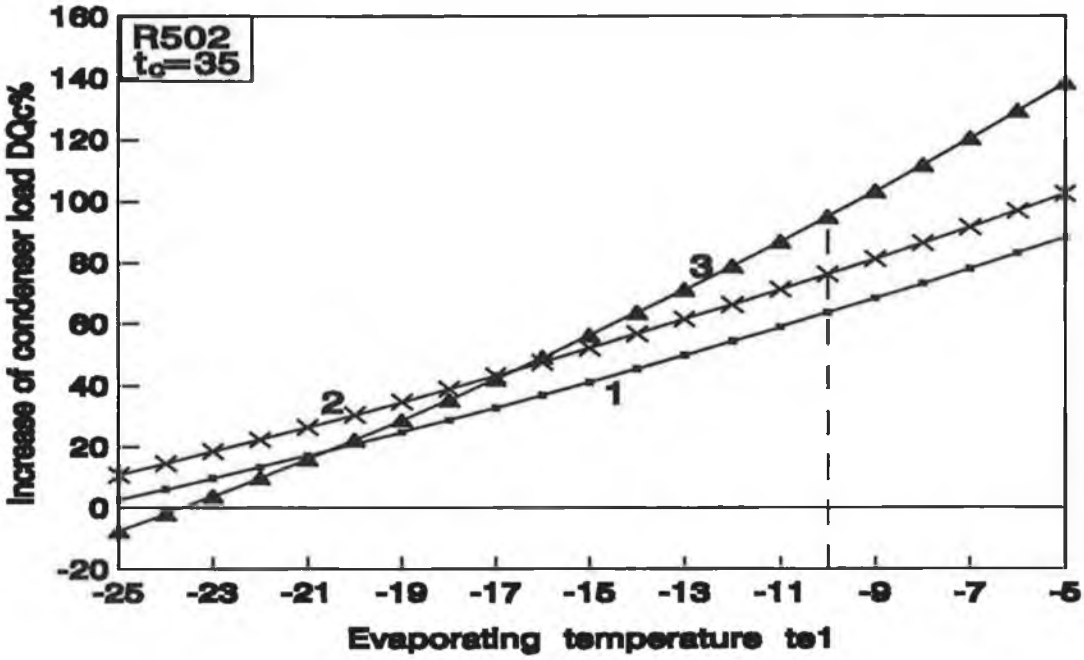


Fig.7-3 The increase of the condenser heat load with R502

7.2 The performance of evaporators

Evaporator is another major component in a compression refrigeration plant in which the refrigerant is converted from a liquid to a vapour through the process of evaporation. This takes place as the heat from the objects (such as frozen foods) is absorbed by the refrigerant in the evaporator.

In a common refrigerating plant, the capacity of evaporators for the common freezing process is given by

$$Q_{oc} = q_{oc} * G_c \tag{7-6}$$

where Q_{oc} is the capacity of evaporator in the common freezing process (kW), i.e. heat absorbed from objects to be refrigerated in a refrigerating process. G_c is the mass flow rate of refrigerant in the common process, kg/s. It can be calculated by basic equation (5-11) since the refrigerant cycles in the system and the mass flow rates in each component are the same. Also q_{oc} is the refrigerating effect, i.e. heat absorbed by a unit of refrigerant in the evaporator (kJ/kg) which can be obtained by equation (5-9).

As stated in the last section, the second period of the new freezing process do not need discussion. In the first period of the new freezing process, the capacity of evaporators is given by

$$Q_{on} = q_{on} * G_n \tag{7-7}$$

where Q_{on} is the capacity of evaporators in the new freezing process (kW), i.e. heat absorbed by refrigerant in a refrigerating process. G_n is the mass flow rate of the refrigerant in the new freezing process (kg/s), and q_{on} is the refrigerating effect in the new process, i.e. heat absorbed by a unit of refrigerant in evaporators (kJ/kg). Equation (5-9) is still valid in this case.

In the first period of the new freezing process, the increase of the capacity of evaporators compared with the common freezing process is given by

$$DQ_o = \frac{Q_{on} - Q_{oc}}{Q_{oc}} * 100\% \qquad (7\text{-}8)$$

Where DQ_0 is the percentage increase of the capacity of evaporators compared with the common process. Equation (7-8) is calculated using a personal computer. The results under different conditions are shown in Figures 7-4, 7-5 and 7-6.

Figure 7-4 shows the result when R717 is employed. According to the figure, it is clear that the capacity of evaporators increases with a rise of the evaporating temperature t_{e1}. Curves 1 and 2 have the same rate of increase. The rate of increase for curve 3 is the greatest among the three curves. At the optimum evaporating temperature t_{e1}, the increase of the capacity of evaporators is 115% for scheme 1, 120 for scheme 2, and 145% for scheme 3.

Figure 7-5 shows the result when R22 is used. It is clear that the capacity of evaporators still increases with a rise of the evaporating temperature t_{e1}. Curves 1 and 2 have the same rate of increase, the value of curve 2 is greater. The rate of increase for curve 3 is the greatest among the three curves. At the optimum evaporating temperature t_{e1}, the increase of the capacity of evaporators is 87% for scheme 1, 98% for scheme 2, and 124% for scheme 3.

Figure 7-6 displays the result when the refrigerant is R502. It is clear that the results are similar to those in figure 7-5. At the optimum evaporating temperature t_{e1}, the increase of the capacity of evaporators is 76% for scheme 1, 92% for scheme 2, and 109% for scheme 3.

Both the condenser and evaporator are heat exchanger, their function is to transfer heat and both of them follow the basic law of heat transfer. Therefore, the effects of their performance can be discussed together.

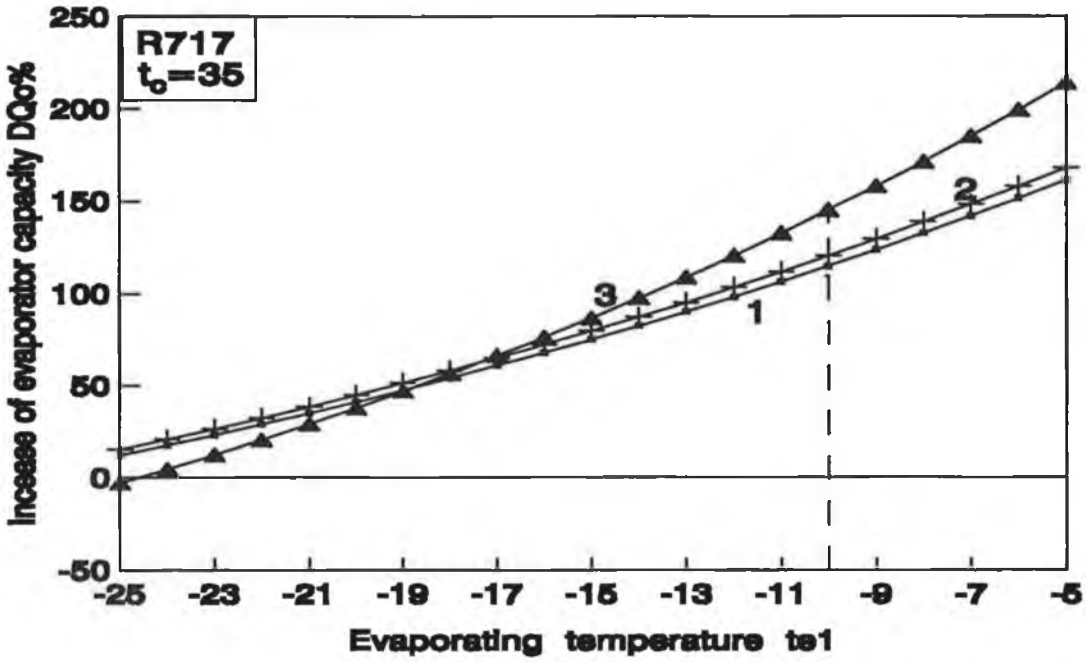


Fig.7-4 The increase of the capacity of evaporators with R717

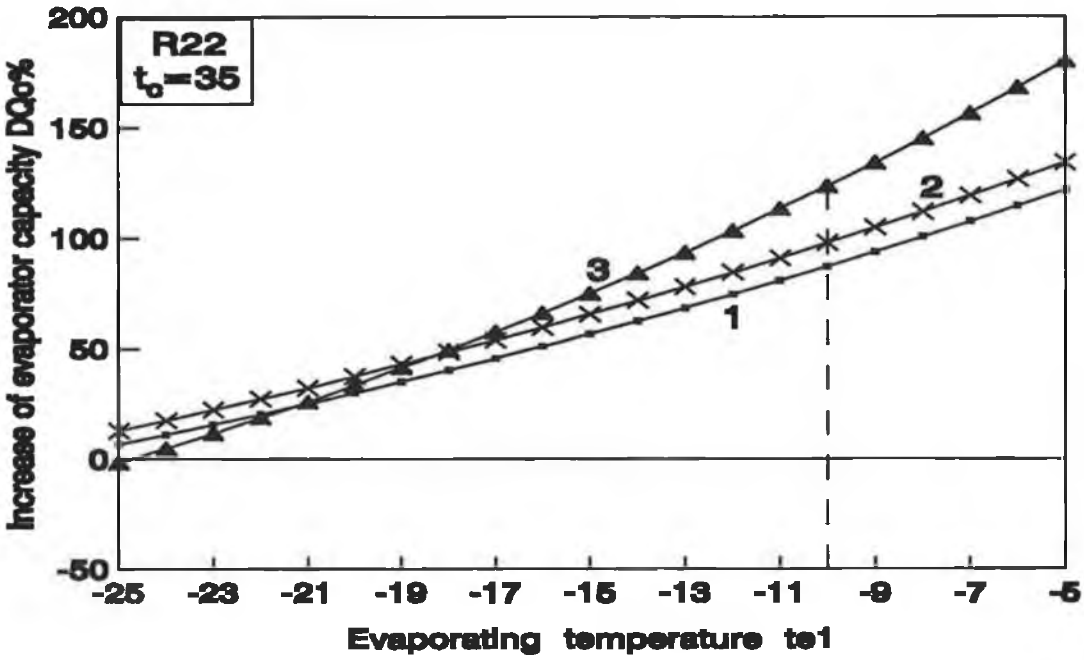


Fig.7-5 The increase of the capacity of evaporators with R22

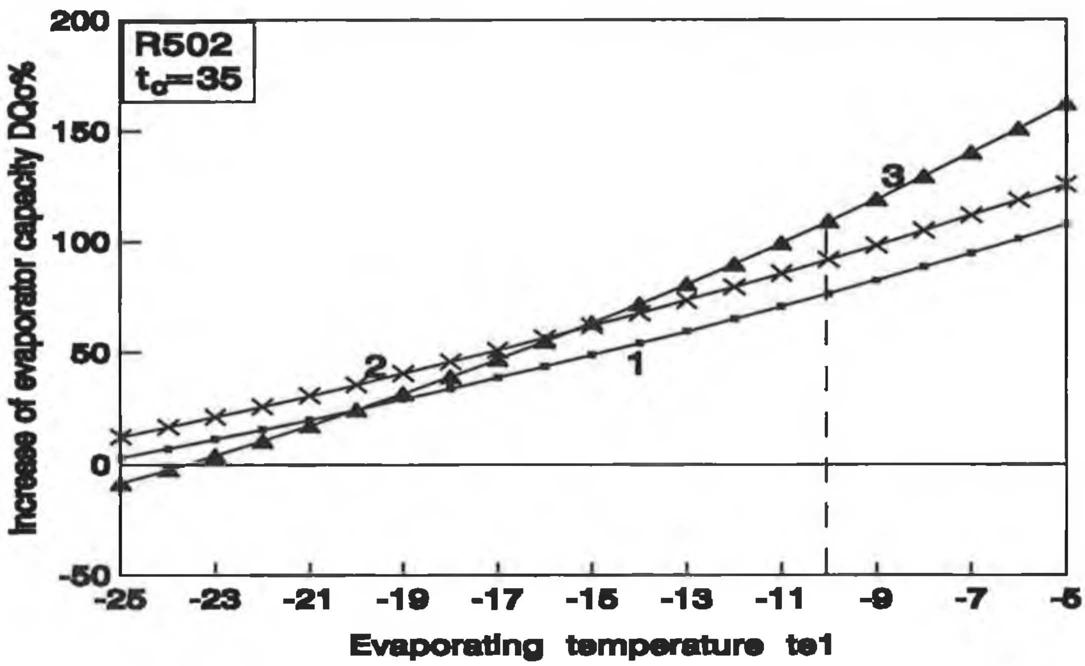


Fig.7-6 The increase of the capacity of evaporators with R502

From Figures (7-1) to (7-6), it is evident that the capacities of the condensers and evaporators in the first period of the new process are much greater than that in the common process. In a given refrigeration plant, the area (A) of the heat exchanger (condensers or evaporators) is a constant. The overall heat transfer coefficient (U) is also constant at a certain condition. According to equation (3-3), in this case, the temperature difference (ΔT) of heat transfer has to increase when the heat Q transferred increases.

For the condenser, the ΔT is the temperature difference between ambient temperature and the condensing temperature of a refrigerant. When ambient temperature is constant, the increase of ΔT means that the condensing temperature of the refrigerant rises. As a result, the efficiency of the refrigeration plant declines and the refrigeration plant will automatically operate at a new balance point.

For the evaporator, ΔT is the temperature difference between the air temperature in the cold space (such as freezer) and the evaporating temperature of the refrigerant. For a given air temperature in a freezer, the increase of ΔT means that the evaporating temperature of the refrigerant drops. As a result, the efficiency of the refrigeration plant declines and the refrigeration plant will automatically operate at a new balance point.

7.3 The performance of the expansion devices

As stated in section 2.2.1, the refrigerant evaporates and absorbs heat in the evaporators for a compression refrigerating plant. The pressure of the refrigerant must be low enough so that a corresponding low temperature will be obtained. The function of an expansion device is to control the pressure of the refrigerant in the evaporator. At the same time, it controls the mass flow rate of the refrigerant entering the evaporator.

If the flow rate is too high, the liquid refrigerant will flow out of the evaporator before it is completely evaporated. If the liquid refrigerant enters the reciprocating compressor which is widely used in current refrigerating plants, damage could result. If the flow rate is too low, the evaporator will be starved and lose capacity, the suction pressure to the compressor will drop, the compressor will work harder at a lower efficiency, and the operating cost will rise. Therefore, it is necessary to examine the variation of flow rate of the refrigerant and the performance of the expansion device in the new freezing process.

In schemes 1 and 2, compared with the common freezing process, the variations of flow rate in the first period of the new process are given by

$$DG_L = \frac{G_{L1} - G_{LC}}{G_{LC}} * 100$$

$$DG_H = \frac{G_{H1} - G_{HC}}{G_{HC}} * 100 \qquad (7\text{-}9)$$

where:

DG_L is the increase of flow rate in the low stage, %;

DG_H is the increase of flow rate in the high stage, %;

G_{L1} is the flow rate of the low stage in the first period, kg/s;

G_{LC} is the flow rate of the low stage in the common process, kg/s

G_{H1} is the flow rate of the high stage in the first period, kg/s

G_{HC} is the flow rate of the high stage in the common process, kg/s.

In scheme 3, the single stage compression is employed in the first period of the new freezing process. Compared with the common freezing process, the variation of flow rate in the first period of the new process is given by

$$DG = \frac{G_S - G_{LC}}{G_{LC}} * 100 \qquad (7\text{-}10)$$

where DG is the increase of flow rate in the scheme 3, %;

G_s is the flow rate of single stage compression, kg/s.

Figures 7-7, 7-8 and 7-9 show the results of equation (7-9) and (7-10) when the refrigerant is R717, R22 and R502, respectively. The increase of the flow rate is directly proportion to the rise of the evaporating temperature t_{e1}. The increase in scheme 3 is the greatest among the three schemes for all the three refrigerants. The increase of the flow rate of the low stage in scheme 1 is the same as that in scheme 2. But the flow rates of the high stages are different in schemes 1 and 2.

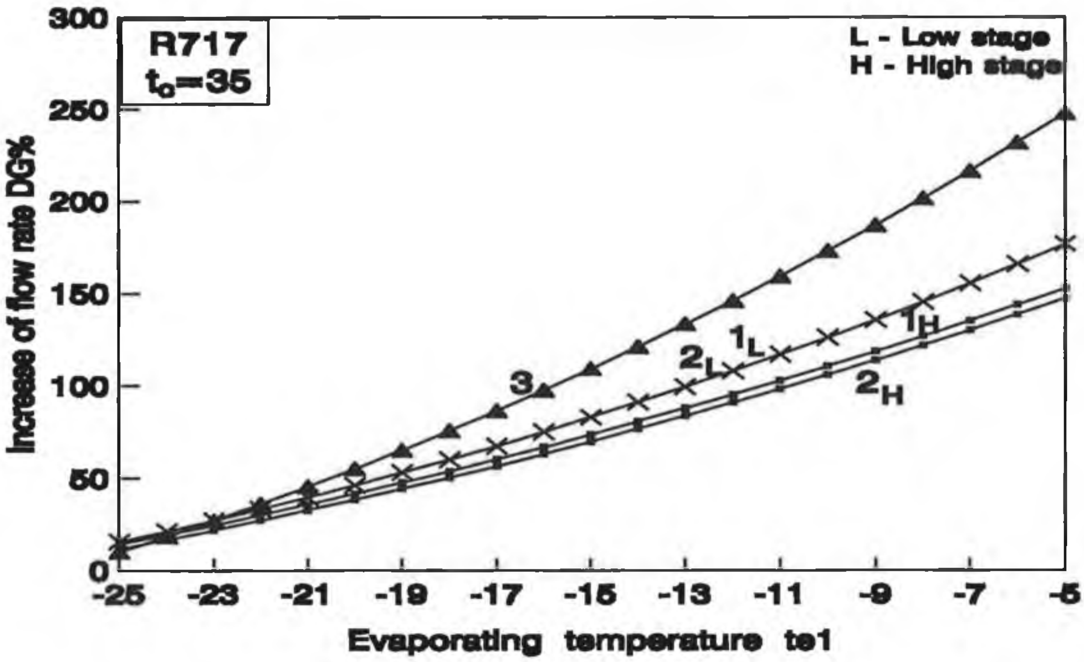


Fig.7-7 The variation of the mass flow rate of R717

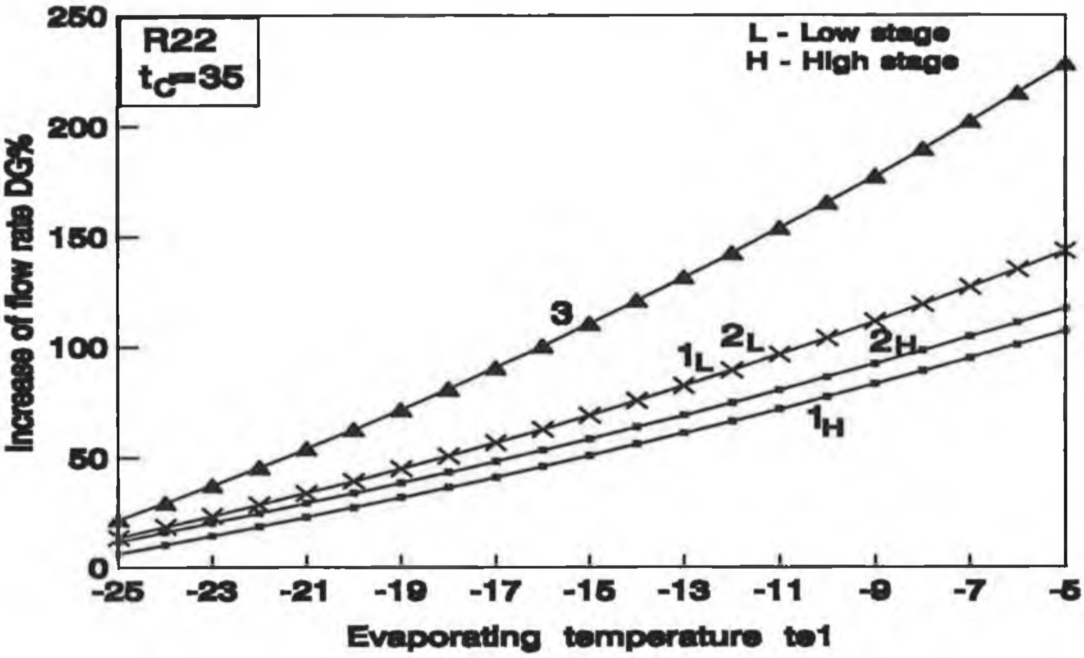


Fig.7-8 The variation of the mass flow rate of R22

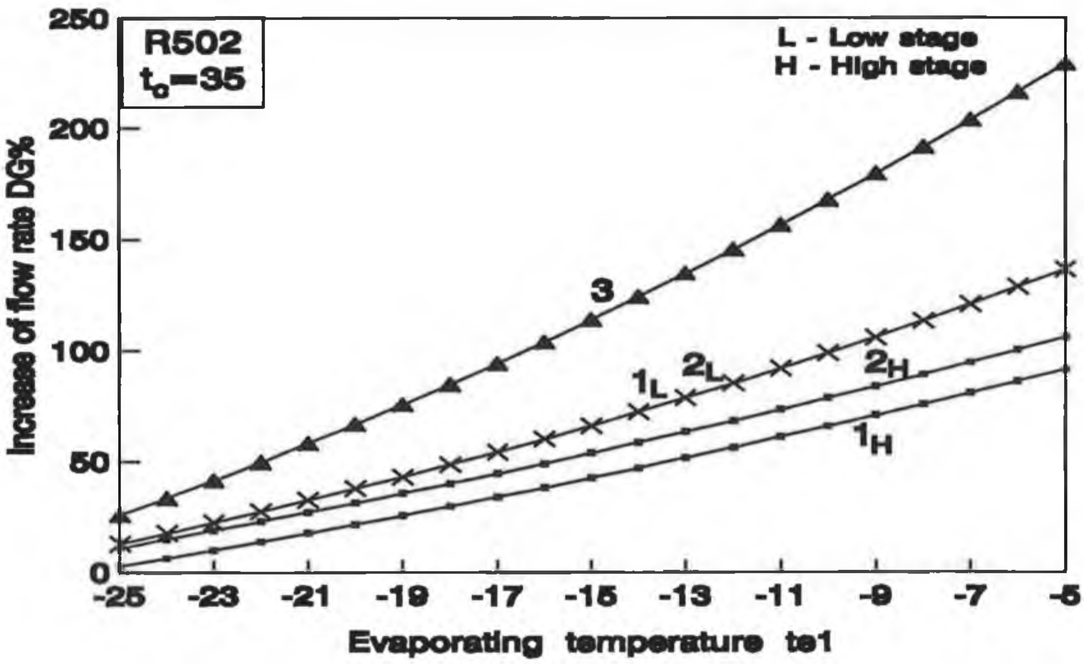


Fig.7-9 The variation of the mass flow rate of R502

According to these results, the increase of flow rate at the optimum evaporating temperature t_{e1} is very obvious. There are three common types of evaporator and expansion valve combinations:

(1) The direct expansion evaporator for which the device controlling refrigerant entry is normally a thermostatic expansion valve.

(2) The flooded evaporator for which the device controlling refrigerant entry is normally a level control device maintaining a constant liquid refrigerant level in the associated surge drum.

(3) The pump-circulated evaporator for which liquid refrigerant is deliberately oversupplied by pump from a liquid/vapour separator vessel. There may be a throttling valve prior to the evaporator for the purpose of ensuring even refrigerant distribution. The pressure drop created by this valve is overcome by the pump system. A level control valve modulates entry of high pressure liquid refrigerant to the separator vessel.

For types (2) and (3), the performance of the evaporator and expansion valve combinations will be affected when the flow rate of refrigerant increases obviously. But they can still work normally. For type (1), the thermostatic expansion valve will not work well when the flow rate passing through it varies greatly.

Fortunately, types (2) and (3) are commonly used in the industrial freezing plants. Nevertheless, it is necessary to examine the limit of the expansion device before the new freezing process is put into operation.

7.4 The performance of compressors

The refrigeration compressor is a motor-driven device which moves the heat-laden vapour refrigerant from the evaporator and compresses it into a small volume and to a high pressure. The compressor maintains the pressure in the evaporator and makes refrigerant cycle in the refrigeration plant. Therefore, it is often called the heart of a compression refrigeration system.

The performance of compressors varies with the condensing temperature and the evaporating temperature. For a given compressor operating at a certain speed, the compressor capacity will increase with the increase of the evaporating pressure (or evaporating temperature). At the same time, the power required by the compressor will increase.

Generally speaking, there is no problem for the compressor itself when the evaporating temperature rises, because there is a wide range of operating parameters when a compressor is designed. But it is not the same for the motor driving the compressor. In order to operate at a high efficiency, the range of operating parameters of a motor is limited. As a result, if the operating parameters of the motor is beyond the limit, damage could result. Therefore, the variation of power required by the compressor should be inspected.

For schemes 1 and 2, compared with the common freezing process, the variation of compressor power in the first period of the new freezing process is given by

$$DN_{two} = \frac{N_1 - N_C}{N_C} * 100 \tag{7-11}$$

where:

DN_{two} is the increase of compressor power in the first period (%)

N_1 is the compressor power in the first period, kW;

N_c is the compressor power in the common process, kW.

For scheme 3, the variation of compressor power in the first period of the new freezing process is given by

$$DN = \frac{N_s - N_C}{N_C} * 100 \tag{7-12}$$

where :

DN is the increase of power in the first period for the scheme 3 (%);

N_s is the power required in the first period (single stage compression), kW;

N_c is the power required in the common process (two stage compression), kW.

The powers in different cases in equations (7-11) and (7-12) can be calculated by equations (5-15) and (5-21).

Figure 7-10, 7-11 and 7-12 show the results of equations (7-11) and (7-12) when the refrigerant is R717, R22 and R502, respectively. These results show that the increases of the compressor power in the first period are directly proportional to the rise of the evaporating temperature t_{e1}. The rate of increase in scheme 3 is the greatest among the three schemes in all the three refrigerants. The increase of compressor power in scheme 2 is slightly greater than that in scheme 1.

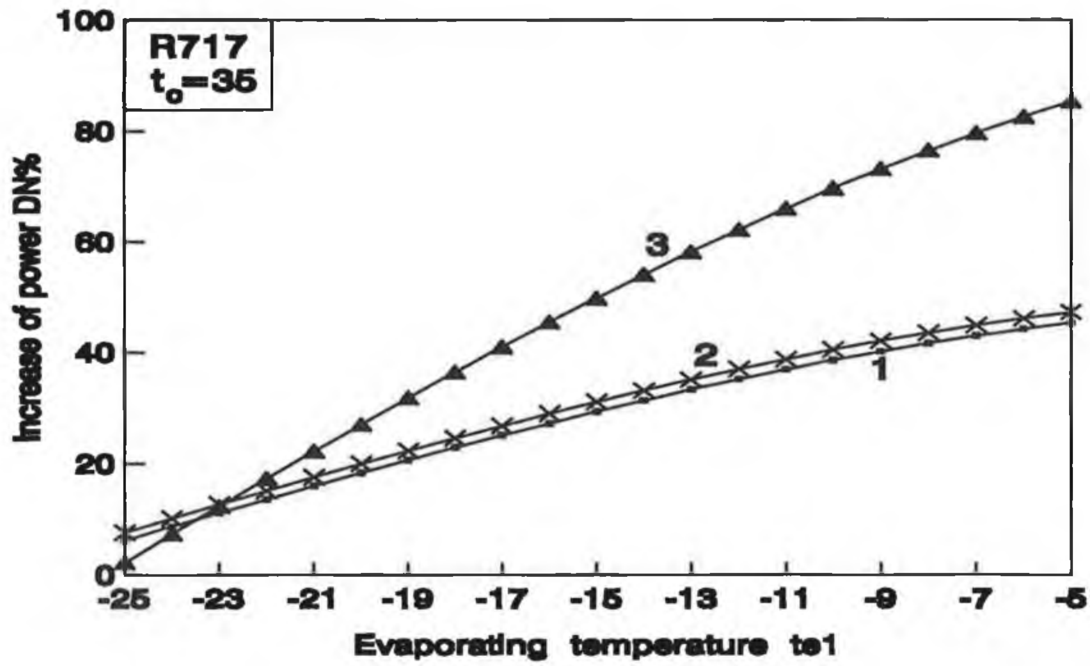


Fig.7-10 The variation of the power in the first period with R717

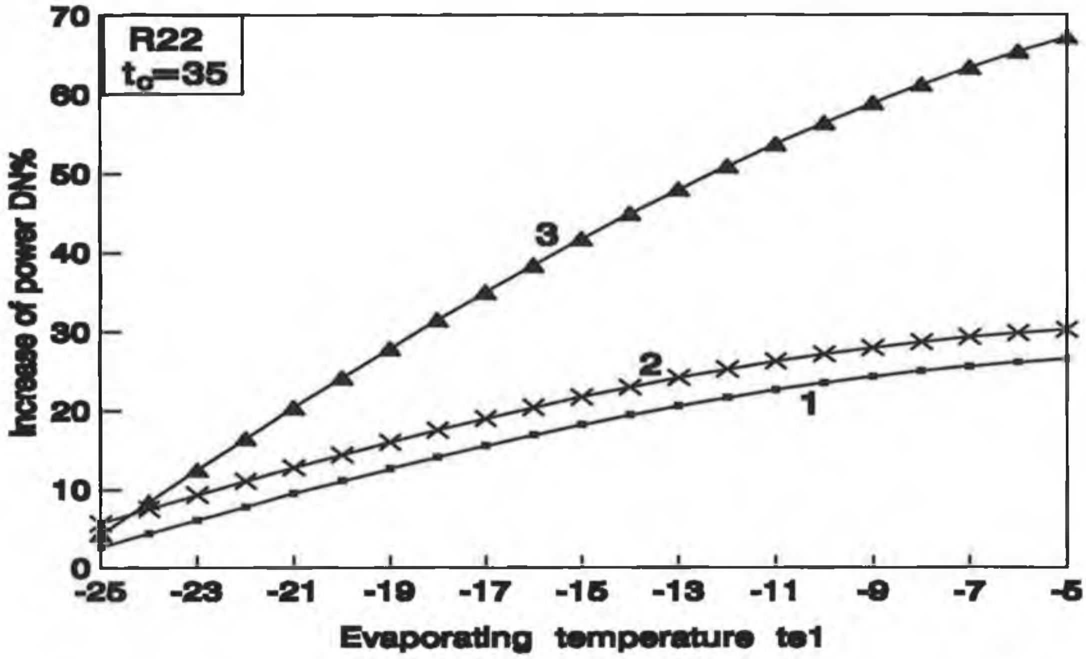


Fig.7-11 The variation of the power in the first period with R22

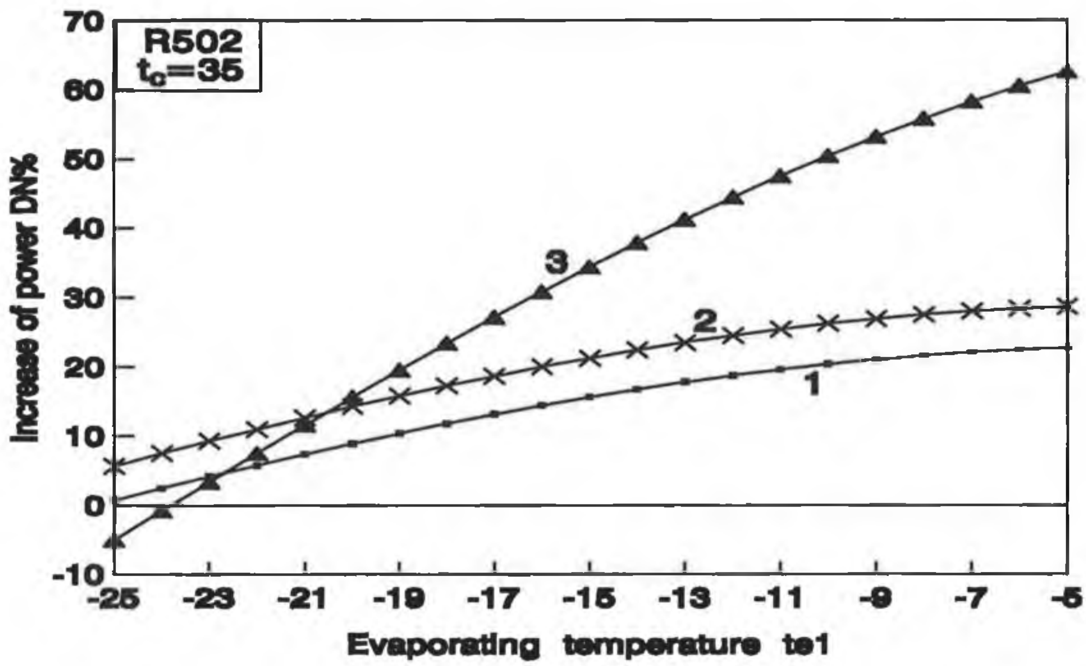


Fig.7-12 The variation of the power in the first period with R502

According to these results, the increase of the flow rate at the optimum evaporating temperature t_{e1} is obvious. Two type of compressors are adopted in a two stage refrigerating system of industrial freezers. One is compound compressor, with different numbers of cylinders devoted to the stages according to the conditions. Alternatively, two different compressors are used in which the low stage compressor is termed a booster compressor.

For a compound compressor, the range of working parameters of its motor is more limited when the motor is designed, because the compressor is only used as two stage compression. Therefore, the power of compressor motor should be examined before the parameters of the new freezing process is decided if the compound compressor will be (or has been) adopted in the refrigeration plant.

If different compressors are adopted for the low stage and the high stage, generally speaking, the motor matching the respective compressor has a wide range of operating parameters. But it is safer to deal with the problem before the parameters of the new freezing process are decided.

CHAPTER 8 CONCLUSIONS AND REMARKS

8.1 Conclusions

Based on the principle of Carnot, the idea of the new freezing process is investigated. According to the analysis presented, the new freezing technique in industrial plants appears to be feasible. With the help of a personal computer, the results under various conditions are obtained and presented.

Based on the results of the simulation program the following conclusions may be made:

1. Compared with the common freezing process, energy savings can be obtained when the new freezing process is employed. There is a maximum of the energy saving (E_s) curves in all the cases. The maximin of E_s can reach 35.6% for refrigerant R717, 35% for R22 and 33% for R502.

2. There is an optimum evaporating temperature, t_{e1opt} for the first period of the new freezing process, at which the energy saving E_s is maximum. The optimum evaporating temperature t_{e1opt} is not affected for different refrigerants (R717, R22 and R502), or for various schemes of refrigeration cycle, the condensing temperature of refrigerants, and the initial temperature of frozen foods. In all the cases, the optimum evaporating temperature, t_{e1opt} is always -10°C.

3. The values of E_s vary with the evaporating temperature, t_{e1} of the first period and the schemes of refrigeration cycles. The values of E_s in scheme 2 are greatest among the three schemes, while the value in scheme 3 is smallest.

4. The freezing time of the new freezing process is always less than that of the common freezing process except at two temperatures (around -25°C) in scheme 3. Compared with the common freezing process, the saving of the freezing time can reach 57.5% for refrigerant R717, 54.1% for R22 and 51% for R502 at the optimum evaporating temperature t_{e1opt}, when scheme 3 is used. As a result, the quality of frozen food can be improved when the new process is adopted.

5. The output of frozen foods for the original refrigeration plant will increase when the new technique is adopted, because the freezing time in the new process is shorter than that in a common process.

6. When the frozen food is changed from beef to cod, there are slight differences (around 2%) for all the results of the simulation program. From the point of view of industrial practice, it can be considered that the kinds of frozen foods is not a factor to affect the results of the new freezing technique.

7. The results of the simulation program suggest that the value of energy

saving E_s is around 10% greater with the data of IIR than the data of ASHRAE.

8. During the first period of the new freezing process, heat flow transferred by the heat exchangers (condensers and evaporators) increases with increasing of the evaporating temperature t_{e1}. Compared with the common freezing process, the increase of heat load for the heat exchangers can reach around 100%, or even over 100% at the optimum evaporating temperature t_{e1opt} in the first period under different cases.

9. The simulation program suggests that When the new freezing process is put into operation, the mass flow rate of refrigerant in the first period increases with increasing of the evaporating temperature t_{e1}. The increase of the flow rate is very obvious. At the optimum evaporating temperature t_{e1opt}, the increase of flow rate can reach 173% for refrigerant R717, 165% for R22 and 167% for R502 when scheme 3 is adopted. Although the level control devices (such as float switches) are commonly used in industrial freezing plants, which have a wide range of operation, it is necessary to examine their capacity before the new freezing process is used.

10. The power required by the compressors increases with increasing of the evaporating temperature t_{e1} during the first period. At the optimum evaporating temperature t_{e1opt}, the increase of the power will reach 69.6% for refrigerant R717, 53% for R22, and 50% for R502 when scheme 3 is

employed.

If scheme 1 or 2 is used, at the optimum evaporating temperature t_{e1opt}, power required increases 38-40% for R717, 20-27% for R22 and R502. It is clear that properly matched motor with the compound compressor will cope well if the compound compressors are employed in the original refrigeration plant.

Based on the facts above, the new freezing process is feasible in refrigeration plants of industrial freezers. But in most cases the optimum evaporating temperature t_{e1opt} is unlikely to be used because the balance point of operation of the plant will leave too much from the design point, which could make the components in the plant overloaded and cause trouble.

However, a suitable evaporating temperature t_{e1} to an actual refrigeration plant can be selected according to the simulation results which should give corresponding energy saving or improvement of efficiency even though such improvement may not be the maximum.

8.2 Remark for further work

The discussions above is based on the original refrigeration plant for a common freezing process. As stated in chapter 7, some limits rise due to the balance between the components of the plant. As a result, the maximum improvement of efficiency could not be obtained when the new freezing process is adopted in a common refrigeration plant of industrial freezers. Although the energy saving could not reach the maximum of the new freezing process, one important feature is that the capital cost need not increase when the new freezing process is adopted.

For a new refrigeration plant being designed, the optimum evaporating temperature t_{e1opt} can be selected and operated in the first period of the new freezing process. In this way, the maximum improvement of efficiency could be obtained. In this case, each component in the refrigeration plant should be selected in a way which is suitable to operate in the new freezing process. For example, the areas of the heat exchanger can be enlarged.

As a result, the capital cost for the plant will increase. But based on the results in chapter 6, the energy saving can reach around 35%, it can be estimated that the increase of the capital cost will be recovered from the saving of the operation cost (energy bill) in a short period. Further research is needed for this case.

Another plan, which could implement the new freezing process, is to retrofit the original refrigeration plant. In other words, some components in an original refrigeration plant are changed or/and increased in order for the plant to operate in the optimum parameters of the new freezing process. In this way, the efficiency of the plant is greatly improved and the operating cost declines, more benefits may be obtained although some capital cost has to be paid off for the rebuilding of the plant.

In reference (76) the scheme 3 is adopted experimentally. According to the experiment without computer simulation, -20°C is selected as the evaporating temperature, t_{e1}, in the first period of freezing fish. Some single stage compressors in the original single compression system are connected to the existing two stage compression system. The results of this experiment show that the payback period of rebuilding is very short, benefit could be obtained especially when the output of freezing foods is large.

Although only scheme 3 is adopted and optimum parameters have not been calculated and selected in this experiment, the principle of the idea is the same. It has been proved by this experiment that the new freezing process is feasible and has great potential.

Although the potential advantages of the new technique are considerable, it is no doubt that more experiments to this project need to be carried out to evaluate its commercial application. It is better to do the experiments in an industrial refrigeration plant instead of in a laboratory, because it can make use of the actual plant and material. Moreover, the result will be entirely relevant.

Great efforts have been made in the discussion of this project in order to make the result to be more practical. But some conditions still have to base on supposition. The following points should be inspected in further work.

1. Only product heat load is considered in the calculation of this project. In fact, the total heat load of a practical freezing process consists of thermal transmission from ambience to freezer (or cold room), product load, infiltration air load and miscellaneous heat loads. Therefore, it should be further discussed that the total heat load instead of only product load affects the result of the simulation.

2. The compression process of refrigerants is assumed as isothermal compression in the simulation. In other words, only the isothermal work of compression process is calculated in this project. Generally speaking, the actual compression is not an isothermal process. Moreover, there is a factor of the mechanical efficiency to an actual compressor. More work is required to take account of these factors.

3. When the new freezing process is adopted, the original refrigeration system will operate at a corresponding balance point to each evaporating temperature t_{e1} of the first period. At the same time, each component in the refrigeration plant will have a certain capacity required. A universal table or diagram of curves for the new freezing process can offer great help for the implementation of the new freezing process. From the point of view of

practice, this is a matter of great significance.

The whole discussion on the new freezing technique in this study has been based on industrial food freezing because food freezing is the major application in freezing technique. It should be emphasized that the novel freezing technique can also be used in other freezing processes except foods. For example, it can also be used in ice making, freezing drying, ice rink, freezing of subsurface soils and biomedical applications. It goes without saying that further study is needed for these different applications.

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APPENDIX

APPENDIX A : Computer program

```
'     ************************************************
'     *         COMPUTER SIMULATION PROGRAM          *
'     *       FOR A NOVEL FREEZING TECHNIQUE         *
'     ************************************************

'                       BY ZHANG, JIANYI

'NOTE:
'This program uses refrigerant R22, R502 and R717 only.
'After ONE OF THE THREE REFRIGERANTS IS SELECTED, THREE
'SCHEMES ARE SIMULATED AND CALCULATED. THE RESULTS ARE SENT
'TO DATA FILES FOR CREATING FIGURES.

'R = The code of refrigerant used

INPUT R

IF R = R717 THEN R717-EQUATIONS
ELSEIF R = R22 THEN R22-EQUATIONS
ELSEIF R = R502 THEN R502-EQUATIONS
ELSE
PRINT "ONLY R22, R502 AND R717 ARE SELECTED IN THIS PROGRAM"
END IF

' P--Absolute pressure ( Pa )
' t--Temperature  ( 'c )
' HL--Liquid enthalpy  ( KJ/kg )
' HV-- Saturated vapour enthalpy  ( KJ/kg )
' VS-- Ssturated vapour specific volume  ( M^3/kg )
' Ci-- The coefficients calculating enthalpy change

R22-EQUATIONS:

DEF FNP (t) = EXP(21.25384 - 2025.4518# / (t + 248.94))

DEF FNT (P) = -2025.4518# / (LOG(P) - 21.25384) - 248.94

DEF FNHL (t) = (44518 + 1170.36 * t + 1.68674 * t ^ 2 -
.0052703 * t ^ 3) * .001

DEF FNHV (t) = (250027 + 367.265 * t - 1.84133 * t ^ 2 -
.0114556# * t ^ 3) * .001

DEF FNVS (t) = EXP(-11.82344 + 2390.321 / (t + 273.15)) *
(1.01859 + 5.09433E-04 * t - 1.48464E-05 * t ^ 2 -
2.49547E-07 * t ^ 3)

DEF FNCi (t) = 1.137423 - 1.50914E-03 * t - 5.59643E-06 * t
^ 2 - 8.74677E-06 * t * DT - 1.49547E-07 * t ^ 2 * DT +
5.97029E-08 * t * DT ^ 2 + 1.41458E-09 * DT ^ 2 * t ^ 2 -
4.5258E-04 * DT
```

```
OPEN "C:\HG3\DATA\R22P.RES" FOR OUTPUT AS #1

PRINT #1, "SUB1R22.RES38"
PRINT #1, "  "
CALL TWO2.bas(38, 1.18)

PRINT #1, "  "
PRINT #1, "SUB1R22.RES7"
PRINT #1, "  "
CALL TWO2.bas(7, 1.18)


PRINT #1, "  "
PRINT #1, "SUB2R22.RES38"
PRINT #1, "  "
CALL TWO1.bas(38, 1.18)

PRINT #1, "  "
PRINT #1, "SUB2R22.RES7"
PRINT #1, "  "
CALL TWO1.bas(7, 1.18)

PRINT #1, "  "
PRINT #1, "SUB3R22.RES38"
PRINT #1, "  "
CALL SINGLE.bas(38, 1.18)

PRINT #1, "  "
PRINT #1, "SUB3R22.RES7"
PRINT #1, "  "
CALL SINGLE.bas(7, 1.18)
END

GOTO MAINPROGRAM


R502-EQUATIONS:

DEF FNP (t) = EXP(21.00668 - 1924.9516# / (t + 248.46))
DEF FNt (P) = -1924.9516# / (LOG(P) - 21.00668) - 248.46


DEF FNHL (t) = (41103 + 1114.6 * t + 2.12743 * t ^ 2 +
.0017679 * t ^ 3) * .001

DEF FNHV (t) = (187890 + 406.454 * t - 1.59402 * t ^ 2 -
.013601# * t ^ 3) * .001

DEF FNVS (t) = EXP(-12.03131 + 2327.862 / (t + 273.15)) *
(1.03208 + 5.57865E-04 * t - 2.55008E-05 * t ^ 2 -
2.86511E-07 * t ^ 3)

DEF FNCi (t) = 1.050613 + 2.42242E-03 * t - 1.20401E-05 * t
^ 2 - 2.80193E-06 * t * DT + 5.957E-09 * t ^ 2 * DT -
```

```
2.95399E-08 * t * DT ^ 2 - 1.3106E-10 * DT ^ 2 * t ^ 2 -
6.69841E-04 * DT

OPEN "C:\HG3\DATA\R502P.RES" FOR OUTPUT AS #1

PRINT #1, "SUB1R502.RES38"
PRINT #1, "  "
CALL TWO2.bas(38, 1.133)

PRINT #1, "  "
PRINT #1, "SUB1R502.RES7"
PRINT #1, "  "
CALL TWO2.bas(7, 1.133)

PRINT #1, "  "
PRINT #1, "SUB2R502.RES38"
PRINT #1, "  "
CALL TWO1.bas(38, 1.133)

PRINT #1, "  "
PRINT #1, "SUB2R502.RES7"
PRINT #1, "  "
CALL TWO1.bas(7, 1.133)

PRINT #1, "  "
PRINT #1, "SUB3R502.RES38"
PRINT #1, "  "
CALL SINGLE.bas(38, 1.133)

PRINT #1, "  "
PRINT #1, "SUB3R502.RES7"
PRINT #1, "  "
CALL SINGLE.bas(7, 1.133)
END

GOTO MAIMPROGRAM

R717-EQUATIONS:

DEF FNP (t) = EXP(22.11874 - 2233.8226# / (t + 244.2))

DEF FNt (P) = -2233.8226# / (LOG(P) - 22.11874) - 244.2

DEF FNHL (t) = (184311 + 4751.63 * t + 2.04493 * t ^ 2 -
.037875 * t ^ 3) * .001


DEF FNHV (t) = (1441467 + 920.154 * t - 10.20556 * t ^ 2 -
.0265126# * t ^ 3) * .001

DEF FNVS (t) = EXP(-11.09867 + 2691.68 / (t + 273.15)) *
(.99675 + 4.02288E-04 * t + 2.6417E-06 * t ^ 2 - 1.75152E-07
* t ^ 3)
```

```
DEF FNCi (t) = 1.325798 + .0002452 * t + 3.10683E-06 * t ^
2 - 1.13335E-05 * t * DT - 1.42736E-07 * t ^ 2 * DT +
6.35817E-08 * t * DT ^ 2 + 9.5979E-10 * DT ^ 2 * t ^ 2 -
3.82295E-04 * DT

OPEN "C:\HG3\DATA\R717P.RES" FOR OUTPUT AS #1

PRINT #1, "SUB1R717.RES38"
PRINT #1, "  "
CALL TWO2.bas(38, 1.28)

PRINT #1, "  "
PRINT #1, "SUB1R717.RES7"
PRINT #1, "  "
CALL TWO2.bas (7, 1.28)

PRINT #1, "  "
PRINT #1, "SUB2R717.RES38"
PRINT #1, "  "
CALL TWO1.bas(38, 1.28)

PRINT #1, "  "
PRINT #1, "SUB2R717.RES7"
PRINT #1, "  "
CALL TWO1.bas(7, 1.28)

PRINT #1, "  "
PRINT #1, "SUB3R717.RES38"
PRINT #1, "  "
CALL SINGLE.bas(38, 1.28)

PRINT #1, "  "
PRINT #1, "SUB3R717.RES7"
PRINT #1, "  "
CALL SINGLE.bas(7, 1.28)
END

GOTO MAINPROGRAM



MAINPROGRAM:

DATA 55,58,61.5,65,68.5,72,76,81,88,95,
105,113,138,180,285,304,0,0,0,0,0

DECLARE SUB TWO2.bas (X1!, X2!)
DECLARE SUB TWO1.bas (X1!, X2!)
DECLARE SUB SINGLE.bas (X1!, X2!)


SUB TWO2.bas (X1, X2)

'NOTE: Sub-program TWO1.bas refers to the scheme 1 of
refrigerating cycle -'- <Two stage compression & two time
```

```
expansions>

'    < BASIC PARAMETER >

' Gf-- Mass of frozen food    ( kg )
' Cf-- Specific heat of food above freezing   ( KJ/kg.'C )
' tf1--Initial temperature of food    ( 'C )
' tf2-- Final temperature of food    ( 'C )
' Hfi--the enthalpy of food at 0 C   ( KJ/kg )
' Hff-- Enthalpy of food at final temperature    (KJ/kg )
' Qf-- Total heat during freezing dood    ( KJ )

Gf = 20000
Cf = 3.1
tf1 = X1
tf2 = -18
Hfi = 304
Hff = 47

Qf = Gf * Cf * (tf1 - 0) + Gf * (Hfi - Hff)

' HOU-- compressor running time of the common freezing
process   (hour )
' Qhl-- hourly load    ( KW )
' tc-- Condensing temperation    ( 'C )
' te-- Evaporating temperation    ( 'C )

HOU = 20
Qhl = Qf / HOU / 3600      'KW
te = -28
tc = 35

'    < SIMULATION OF THE COMMON FREEZING METHOD >

' Pc--Condensing pressure    ( Pa )
' Pe--Evaporating pressure   ( Pa )
' Pm--Internal pressure      ( Pa )
' tm--Internal temperature, °C

Pc = FNP(tc)
Pe = FNP(te)

Pm = (Pe * Pc) ^ (1 / 2)
tm = FNT(Pm)

t3 = tm
t7 = tm
H1 = FNHV(t1)
H7 = FNHL(t7)
H3 = FNHV(t3)

t5 = tc
t1 = te
q0 = H1 - H7
```

```
' q0-- Refrigeration effect     ( KJ/kg )
' GLc--Mass flow rate of low step     (kg/s )
' DT--Temperature difference       ( 'C )
' DH-- Enthalpy difference of compression    ( KJ/kg )
' V1 is special volume of the low stage compressors, m^3/kg
' V3 is special volume of the high stage compressors, m^3/kg


GLc = Qhl / q0

V1 = FNVS(t1)
V3 = FNVS(t3)

DT = tm - te

Ci12 = FNCi(t1)

DH12 = (Ci12 / (Ci12 - 1) * Pe * V1 * ((Pm / Pe) ^ ((Ci12 -
1) / Ci12) - 1)) * .001

DT = tc - tm
Ci34 = FNCi(t3)

DH34 = (Ci34 / (Ci34 - 1) * Pm * V3 * ((Pc / Pm) ^ ((Ci34 -
1) / Ci34) - 1)) * .001

H2 = H1 + DH12
H4 = H3 + DH34

H5 = FNHL(tc)


' GHc--Mass flow rate of high step compressor in the common
process, kg/s
' k-- Ratio of specific heats of refrigerants
' Ev--Volumetric efficiency of compressors

GHc = GLc * (H2 - H7) / (H3 - H5)
k = X2
Ev = .94 - .085 * ((Pc / Pm) ^ (1 / k) - 1)
Qcc = GHc * (H4 - H5)

'Qcc - Concending load of the common freezing process, kW


' NLc -- Power consumed by the low step compressors in the
common process, ( kw )
' NHc -- Power consumed by the high step compressors in the
common process, ( kw )

NLc = GLc * DH12
NHc = GHc * DH34


' VHL -- Displacement of low step compressor,  m^3/s
```

```
' VHH -- Displacement of the high compressors, m^3/s
' VH  -- Displacement of total compressors,  m^3/s

VHL = GLc * V1 / Ev
VHH = GHc * V3 / Ev

Nc = NLc + NHc
VH = VHH + VHL
 Ec = Nc * HOU

' Ec -- Total work consumed in the common freezing process,
   ( KW.H )


 '    < SIMULATION OF THE NEW FREEZING PROCESS >


'    < The first period--the two stage compression >

'te1--EVAPRATING TEMPERATURE OF THE FIRST period
'tfM--FINAL TEMPERATURE OF FOOD AFTER THE FIRST period
' HtfM-- Enthalpy of food in different temperations    (
KJ/kg )

PRINT #1, "   te1   "; "   ENERGYs %   "; "    HOURs %   "; "
  TIME (h)  "; "    DQc%  "; "        DQ0%  "; DGH%; DGL%;
DN
PRINT #1, "
      "

FOR te1 = -25 TO -5
tfM = te1 + 10
READ HtfM
RESTORE


IF te1 <= -10 THEN
  Qf1 = Gf * Cf * (tf1 - 0) + Gf * (Hfi - HtfM)
ELSE
  Qf1 = Gf * Cf * (tf1 - (te1 + 10))
END IF

Pc = FNP(tc)
Pe1 = FNP(te1)


Pm = (Pc * Pe1) ^ (1 / 2)
tm = FNT(Pm)
t7 = tm
t1 = te1
H1 = FNHV(t1)
H7 = FNHL(t7)

q01 = H1 - H7
```

```
Ev = .94 - .085 * ((Pc / Pm) ^ (1 / k) - 1)

V1 = FNVS(t1)
t3 = tm
V3 = FNVS(t3)

GL1 = VHL * Ev / V1
Q01ST = GL1 * q01
HOU1 = Qf1 / (Q01ST * 3600)

' HOU1 -- Running time of compressions in the first period,
 ( hour )


DT = tm - te1

Ci12 = FNCi(te1)

' DH12 -- Enthalpy diffenerce( copression work ) of low
stage, ( KJ/kg )
'DH34 -- Enthalpy diffenerce( copression work ) of high
stage, ( KJ/kg )

' q01 -- Refrigeration effect at the first period, KJ/kg

DH12 = (Ci12 / (Ci12 - 1) * Pe1 * V1 * ((Pm / Pe1) ^ ((Ci12
- 1) / Ci12) - 1)) * .001
H2 = H1 + DH12
H3 = FNHV(t3)

DT = tc - tm
Ci34 = FNCi(tm)
DH34 = (Ci34 / (Ci34 - 1) * Pm * V3 * ((Pc / Pm) ^ ((Ci34 -
1) / Ci34) - 1)) * .001
H5 = FNHL(tc)
GH1 = GL1 * (H2 - H7) / (H3 - H5)

NL1 = GL1 * DH12
NH1 = GH1 * DH34

N1 = NL1 + NH1
E1 = N1 * HOU1


' E1 -- Work consumed in the first period     ( KW.H )


 '   < THE SECOND PERIOD -- THE SAME AS THE COMMON FREEZING
PROCESS >

' Qf2 -- Total heat load of the second period, ( KJ )
' HOU2 -- Running time of refrigeration plant in the second
PERIOD ( hour )
' E2   --  Work consumed in the second PERIOD    (KW.h )
```

```
Q02nd = Qhl
Qf2 = Qf - Qf1
 HOU2 = Qf2 / Qhl / 3600

E2 = Nc * HOU2

En = E1 + E2

'En - Total work consumed in the new freezing process, KW.H

' ENERGYs -- Percentage of energy saving of the new freezing
process compared with the common freezing process, %
' HOURs -- Percentage of the freezing time decrease of the
new freezing process vs the common freezing process, %
' timeN -- The freezing time of foods in the new freezing
process, hour

ENERGYs = (Ec- En) / Ec * 100
HOURs = (HOU - HOU1 - HOU2) / HOU * 100
timeN = HOU1 + HOU2

' Qc1 -- The condensing load of the first period, kW
' DQc -- The increase of condensing load of the first period
compared with the common freezing process, %
' DQ0 -- The increase of refrigerating capacity of the first
period compared with the common freezing process, %

H4 = H3 + DH34
Qc1 = GH1 * (H4 - H5)
DQc = (Qc1 - Qcc) / Qcc * 100

DGL = (GL1 - GLc)/GLc %
DGH = (GH1 - GHc)/GHc %
DN = (N1 - Nc)/Nc %
DQ0 = (Q01ST - Qhl) / Qhl %

PRINT #1, "     "; te1; "      "; ENERGYs; "      "; HOURs; "
"; timeN; "     "; DQc; "      "; DQ0; DGH; DGL; DN


NEXT te1

END SUB


SUB TWO1.bas (X1, X2)

'NOTE: SUB-PROGRAM TWO1.bas refers to the scheme 2 of
refrigerating cycle - <Two stage compression & single
expansion >

'    < BASIC PARAMETER >

' Gf-- Mass of frozen food   ( kg )
' Cf-- Specific heat of food above freezing  ( KJ/kg.'C )
```

```
' tf1--Initial temperature of food   ( 'C )
' tf2-- Final temperature of food   ( 'C )
' Hfi--the enthalpy of food at 0 C  ( KJ/kg )
' Hff-- Enthalpy of food at final temperature   (KJ/kg )
' Qf-- Total heat during freezing dood   ( KJ )
'
Gf = 20000
Cf = 3.1
tf1 = X1
tf2 = -18
Hfi = 304
Hff = 47

Qf = Gf * Cf * (tf1 - 0) + Gf * (Hfi - Hff)

' HOU-- Running time of compressors in the common freezing
process, hour
' Qhl-- hourly load   ( KW )
' tc-- Condensing temperation   ( °C )
' te-- Evaporating temperation   ( °C )

HOU = 20
Qhl = Qf / HOU / 3600      'KW
te = -28
tc = 35

' <SIMULATION AND CALCULATION OF COMMON FREEZING METHOD>

' Pc--Condensing pressure,   ( Pa )
' Pe--Evaporating pressure,  ( Pa )
' Pm--Internal pressure of two stage compression,   ( Pa )
' tm--Internal temperature of two stage compression, °C

Pc = FNP(tc)
Pe = FNP(te)

Pm = (Pe * Pc) ^ (1 / 2)
tm = FNT(Pm)

t3 = tm

t6 = tm + 5
t5 = tc
t1 = te
H1 = FNHV(t1)
H6 = FNHL(t6)

q0 = H1 - H6

' q0-- Refrigeration effect   ( KJ/kg )
' GLc--Mass flow rate of the low step in the common process,
(kg/s )
' DT--Temperature difference      (°C)
' DH-- Enthalpy difference of compression   ( KJ/kg )
' V1 is special volume of the low stage compressors, m^3/kg
```

```
' V3 is special volume of the high stage compressors, m^3/kg

GLc = Qhl / q0

V1 = FNVS(t1)
V3 = FNVS(t3)

DT = tm - te

Ci12 = FNCi(t1)

DH12 = (Ci12 / (Ci12 - 1) * Pe * V1 * ((Pm / Pe) ^ ((Ci12 -
1) / Ci12) - 1)) * .001

DT = tc - tm
Ci34 = FNCi(t3)

DH34 = (Ci34 / (Ci34 - 1) * Pm * V3 * ((Pc / Pm) ^ ((Ci34 -
1) / Ci34) - 1)) * .001

H3 = FNHV(t3)
H2 = H1 + DH12
H4 = H3 + DH34

H5 = FNHL(tc)

' GHc--Mass flow rate of the high step compressor in the
common freezing process, ( kg/s )

' k-- Ratio of specific heats of refrigerants
' Ev--Volumetric efficiency of compressors

GHc = GLc * (H2 - H6) / (H3 - H5)
k = X2
Ev = .94 - .085 * ((Pc / Pm) ^ (1 / k) - 1)
Qcc = GHc * (H4 - H5)

'Qcc - CONDENSING LOAD of the common freezing process, kW

' NLc -- Power consumed in low step compressors,   ( kw )
' NHc -- Power consumed in high step compressors,  ( kw )

NLc = GLc * DH12
NHc = GHc * DH34


' VHL -- Displacement of the low step compressors, (m^3/s)
' VHH -- Displacement of the high step compressors, (m^3/s)
' VH  -- Displacement of total compressors,  (m^3/s)

VHL = GLc * V1 / Ev
VHH = GHc * V3 / Ev

Nc = NLc + NHc
VH = VHH + VHL
```

```
Ec = Nc * HOU

' Nc -- Power required by a common freezing plant, kW
' Ec -- Total work consumed in the common freezing process,
KW.H


 '    < SIMULATION OF THE NEWE FREEZING PROCESS >


'    < The frist period -- still  two stage compression >

'te1--Evaprating temperature of the first period, °C
'tfM--Final temperature of food after the first period, °C

' HtfM-- Enthalpy of food in different temperations    (
KJ/kg )
'Qf1 -- Heat rejected by foods in the frist period of the
new freezing process, kJ

PRINT #1, "   te1  "; "   ENERGYs %  "; "    HOURs %  "; "
  TIME (h)  "; "    DQc%  "; "        DQ0%  "; DGH%; DGL%;
DN%
PRINT #1, "
        "

FOR te1 = -25 TO -5
tfM = te1 + 10
READ HtfM
RESTORE

IF te1 <= -10 THEN
  Qf1 = Gf * Cf * (tf1 - 0) + Gf * (Hfi - HtfM)
ELSE
  Qf1 = Gf * Cf * (tf1 - (te1 + 10))
END IF

Pc = FNP(tc)
Pe1 = FNP(te1)


Pm = (Pc * Pe1) ^ (1 / 2)
tm = FNT(Pm)
t6 = tm + 5
t1 = te1
H1 = FNHV(t1)
H6 = FNHL(t6)

q01 = H1 - H6
k = X2
Ev = .94 - .085 * ((Pc / Pm) ^ (1 / k) - 1)

V1 = FNVS(t1)
t3 = tm
```

```
V3 = FNVS(t3)

GL1 = VHL * Ev / V1
Q01ST = GL1 * q01
HOU1 = Qf1 / (Q01ST * 3600)


DT = tm - te1

Ci12 = FNCi(te1)

' DH12 -- Enthalpy diffenerce( copression work ) of low
stage, ( KJ/kg )
'DH34 -- Enthalpy diffenerce( copression work ) of high
stage, ( KJ/kg )
' q01 -- Refrigeration effect at the first period, ( KJ/kg
)

DH12 = (Ci12 / (Ci12 - 1) * Pe1 * V1 * ((Pm / Pe1) ^ ((Ci12
- 1) / Ci12) - 1)) * .001
H2 = H1 + DH12
H3 = FNHV(t3)

DT = tc - tm
Ci34 = FNCi(tm)

DH34 = (Ci34 / (Ci34 - 1) * Pm * V3 * ((Pc / Pm) ^ ((Ci34 -
1) / Ci34) - 1)) * .001

H5 = FNHL(tc)
GH1 = GL1 * (H2 - H6) / (H3 - H5)

NL1 = GL1 * DH12
NH1 = GH1 * DH34
N1 = NL1 + NH1

E1 = N1 * HOU1


' HOU1 -- Running time of compressors in the first period,
 ( hour )
' N1 -- Power consumed in the first period,     ( KW )
' E1 -- Work consumed in the first period,     ( KW.H )



 '  < THE SECOND PERIOD -- THE SAME AS THE COMMON TREEZING
PROCESS >

' Qf2 -- Total heat load of the second periond,    ( KJ )
' HOU2 -- Running time of refrigeration plant in the second
PERIOD ( hour )
' E2   --  Work consumed in the second period,    (KW.h )
```

```
' En -- Total work consumed in the new freezing process, kW.H

 Qf2 = Qf - Qf1
 HOU2 = Qf2 / Qhl / 3600

E2 = Nc * HOU2

En = E1 + E2

' ENERGYs -- Percentage of energy saving of the new freezing process compared with the common freezing process, %
' HOURs -- Percentage of the freezing time decrease of the new freezing process vs the common freezing process, %
' timeN -- The freezing time of foods in the new freezing process, hour

ENERGYs = (Ec - En) / Ec * 100
HOURs = (HOU - HOU1 - HOU2) / HOU * 100
timeN = HOU1 + HOU2

' Qc1 -- The condensing load of the first period, kW
' DQc -- The increase of condensing load of the first period compared with the common freezing process, %
' DQ0 -- The increase of refrigerating capacity of the first period compared with the common freezing process, %

H4 = H3 + DH34
Qc1 = GH1 * (H4 - H5)
DQc = (Qc1 - Qcc) / Qcc * 100

DGL = (GL1 - GLc)/GLc %
DGH = (GH1 - GHc)/GHc %
DN = (N1 - Nc)/Nc %
DQ0 = (Q01ST - Qhl) / Qhl %

PRINT #1, "    "; te1; "     "; ENERGYs; "     "; HOURs; "
"; timeN; "     "; DQc; "     "; DQ0; DGH; DGL; DN

NEXT te1

END SUB



SUB SINGLE.bas (X1, X2)

'NOTE: Sub-program SINGLE.bas refers to the scheme 3 of refrigerating cycle -- <Single compression in the first period of the new freezing process>

'    < BASIC PARAMETER >

' Gf-- Mass of frozen food    ( kg )
```

```
' Cf-- Specific heat of food above freezing  ( KJ/kg.'C )
' tf1--Initial temperature of food   ( 'C )
' tf2-- Final temperature of food   ( 'C )
' Hfi--Initial enthalpy of food   ( KJ/kg )
' Hff--Final enthalpy of food    (KJ/kg )
' Qf-- Total heat during the process of freezing dood, KJ

Gf = 20000
Cf = 3.1
tf1 = X1
tf2 = -18
Hfi = 304
Hff = 47

Qf = Gf * Cf * (tf1 - 0) + Gf * (Hfi - Hff)

' HOU--Compressor running time in common freezing process
(hour )
' Qhl--Hourly load    ( KW )
' tc-- Condensing temperation    ( 'C )
' te-- Evaporating temperation   ( 'C )

HOU = 20
Qhl = Qf / HOU / 3600      'KW
te = -28
tc = 35


'   < SIMULATION OF THE COMMON FREEZING METHOD >

' Pc--Condensing pressure    ( Pa )
' Pe--Evaporating pressure   ( Pa )
' Pm--Internal pressure      ( Pa )

Pc = FNP(tc)
Pe = FNP(te)

Pm = (Pe * Pc) ^ (1 / 2)
tm = FNT(Pm)

t3 = tm

t6 = tm + 5
t5 = tc
t1 = te
q0 = FNHV(t1) - FNHL(t6)

' q0-- Refrigeration effect    ( KJ/kg )
' GLc--Mass flow rate of low step    (kg/s )
' DT--Temperature difference       ( 'C )
' DH-- Enthalpy difference of compression    ( KJ/kg )


GLc = Qhl / q0
```

```
V1 = FNVS(t1)
V3 = FNVS(t3)

DT = tm - te
Ci12 = FNCi(t1)

DH12 = (Ci12 / (Ci12 - 1) * Pe * V1 * ((Pm / Pe) ^ ((Ci12 -
1) / Ci12) - 1)) * .001

DT = tc - tm
Ci34 = FNCi(t3)

DH34 = (Ci34 / (Ci34 - 1) * Pm * V3 * ((Pc / Pm) ^ ((Ci34 -
1) / Ci34) - 1)) * .001

H2 = FNHV(t1) + DH12
H4 = FNHV(t3) + DH34
H3 = FNHV(t3)
H1 = FNHV(t1)
H5 = FNHL(tc)
H6 = FNHL(t6)

' GHc--Mass flow rate of high step compressor     ( kg/s )
' k-- Ratio of specific heats of refrigerants
' Ev--Volumetric efficiency of compressors

GHc = GLc * (H2 - H6) / (H3 - H5)
k = X2
Ev = .94 - .085 * ((Pc / Pm) ^ (1 / k) - 1)
Qcc = GHc * (H4 - H5)

 'Qcc -- Condensing load of the common freezing process, kw
' NLc -- Power consumed in low step compressors      ( kw )
' NHc -- Power consumed in the high step compressors, ( kw
)

NLc = GLc * DH12
NHc = GHc * DH34


' VHL -- Displacement of the low step compressor, m^3/s
' VHH -- Displacement of the high step compressors, m^3/s
' VH  -- Displacement of total compressors, m^3/s

VHL = GLc * V1 / Ev
VHH = GHc * V3 / Ev

Nc = NLc + NHc
VH = VHH + VHL
 Ec = Nc * HOU

' Ec -- Total work consumed in freezing food      ( KW.H )
' Nc -- Total power consumed in the common freezing process,
kW
```

```
'    < SIMULATION OF TWO EAPORATING TEMPERATURE FREEZING >

'    < THE FIRST PERIOD--SINGLE COMPRESSION >

'te1--EVAPRATING TEMPERATURE OF THE FIRST STEP
'tfM--FINAL TEMPERATURE OF FOOD AFTER THE FIRST STEP
' HtfM-- Enthalpy of food in different temperations    (
KJ/kg )

PRINT #1, "   te1   "; "   ENERGYs %   "; "    HOURs %   "; "
  TIME (h)  "; "    DQc%   "; "         DQ0%   "; DG; DN
PRINT #1, "
      "

FOR te1 = -25 TO -5
tfM = te1 + 10
READ HtfM
RESTORE


IF te1 <= -10 THEN
  Qf1 = Gf * Cf * (tf1 - 0) + Gf * (Hfi - HtfM)
ELSE
  Qf1 = Gf * Cf * (tf1 - (te1 + 10))
END IF

Pc = FNP(tc)
Pe1 = FNP(te1)
Hs1 = FNHV(te1)

DT = tc - te1

Cis = FNCi(te1)
VS1 = FNVS(te1)

' DHs -- Enthalpy diffenerce( copression work ) of single
step   ( KJ/kg )
' q01 -- Refrigeration effect of single step compression
 ( KJ/kg )

DHs = (Cis / (Cis - 1) * Pe1 * VS1 * ((Pc / Pe1) ^ ((Cis -
1) / Cis) - 1)) * .001
Hs2 = Hs1 + DHs
Hs3 = FNHL(tc)

q01 = Hs1 - Hs3
Ev1 = .94 - .085 * ((Pc / Pe1) ^ (1 / k) - 1)

Gs = VH * Ev1 / VS1

Q0s = Gs * q01
   'Q0s   --   Refrigeration   capacity   of   single   stage
compression. KW
' HOU1 -- Running time of compressions in the first period,
```

```
( hour )
' Ns -- Consumed power of compressors in the first period,
 ( KW )
' E1 -- Work consumed in the first period,     ( KW.H )

HOU1 = Qf1 / (Q0s * 3600)

 Ns = Gs * DHs
 E1 = Ns * HOU1



 '  < THE SECOND PERIOD - TWO STEP COMPRESSION (THE SAME AS
THE COMMON ONE) >

' Qf2 -- Total heat of the second period,      ( KJ )
' HOU2 -- Running time of refrigeration plant in the second
period, ( hour )
' E2   --  Work consumed in the second period,     (KW.H )
' En -- Work consumed in the new freezing process, (KW.H)

 Qf2 = Qf - Qf1
 HOU2 = Qf2 / Qhl / 3600

E2 = Nc * HOU2

En = E1 + E2

'En - Total work consumed in the new freezing process, KW.H

' ENERGYs -- Percentage of energy saving of the new freezing
process compared with the common freezing process, %
' HOURs -- Percentage of the freezing time decrease of the
new freezing process vs the common freezing process, %
' timeN -- The freezing time of foods in the new freezing
process, hour

ENERGYs = (Ec - En) / Ec * 100
HOURs = (HOU - HOU1 - HOU2) / HOU * 100
timeN = HOU1 + HOU2


Qc1 = Gs * (Hs2 - Hs3)
DQc = (Qc1 - Qcc) / Qcc * 100

DQ0 = (Q0s - Qhl) / Qhl * 100
DG = (Gs - GLc)/GLc %
DN = (Ns - Nc)/Nc %

PRINT #1, "    "; te1; "     "; ENERGYs; "     "; HOURs; "
"; timeN; "     "; DQc; "     "; DQ0; DG; DN

NEXT te1

END SUB
```

APPENDIX B : Paper published

1. Jianyi Zhang, W.Hu & M.S.J.Hashmi. A new cycle and its computer optimization on single stage ammonia compression refrigeration. Proceedings of 1992 international refrigeration conference, pp309-317. Purdue University, USA. July 14-17, 1992.

2. Jianyi Zhang, W.Hu & M.S.J.Hashmi. Super-subcooling with R12 and R22 refrigeration plants using reciprocating compressors. Applied Energy, Vol. 45 issue (2), 1993. UK. (In Press).