بسم الله الرحمن الرحيم

Palestine Polytechnic University

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Graduation Project

General Study and Redesign for Halhul's Refrigeration Station

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Abstract

This project comes as the fruit of five years of hard work in the field of refrigeration and air conditioning engineering.

In this study a general study and redesign will be proposed for Halhul's refrigeration station to rehabilitate it to work with high efficiency, enabling supplying the Palestinian market with the products all around the year.

The redesign includes the redesign of all refrigeration station systems.

The refrigeration redesign will take in consideration the variable cooling capacity for each cold storage, such that the same room can handle many types of products from fruit and vegetables.

To control the operation of this station will be based on the required room circumstances necessary for the preservation of the horticultural commodities, synchronizing and regulating the systems operation to achieve these desired conditions.

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List of Symbols

Q	Cooling load
Q^{o}	Steam flow rate
A	Area
COS Φ	Power factor of the electric motor
U	Overall heat transfer coefficient
Φ	Relative humidity
T	Temperature
V^{o}	Volumetric flow rate
A	Ampere
V	Voltage
v	Velocity
Cp	Specific heat at constant pressure
h	The convection heat transfer coefficient
I	Current
D	Diameter
P	Pressure
Z	Correction factor
m^{o}	Mass flow rate

Chapter One

INTRODUCTION

1.1 Introduction to the Refrigeration

The refrigeration has become one of the most important engineering branches that directly affect human being life. The applications of this engineering branch reached every house and organization. Refrigeration is used in many wide applications mainly in the food storage and processing, Chemical industry and other wide applications.

The refrigeration engineering is a branch of the thermo-mechanical engineering that is specialized in the engineering methodology of obtaining low temperatures at different circumstances. And applying this methodology on their designs and maintenance of the refrigeration systems.

1.2 General Overview of Project

In this project, the mechanical engineering principles and refrigeration engineering techniques will be applied in redesigning and evaluating the refrigeration station in Halhul's main market.

The station includes ten refrigerating rooms for the long term storage for different kinds of fruit and vegetables. The storage is for all around the year. Each room is about 60 m^2 and with height of 4 m. located in the basement of the central fruit market. Under the ground level and provided with two service elevators and modern lifting vehicle.

This station was first established in 1980s, but some errors have been occurred in the refrigeration station operation and handling caused the rot of the fruit and vegetables stored in it.

The main purposes of this project are:

- Full study of the refrigeration station, to warrant the fail in this station performance. Analyzing its design.
- Total redesign the refrigeration station systems taking in account the multivariable cooling loads. Enabling it storing various types and quantities of fruit and vegetables.

- Full evaluation and analyzing for the refrigeration station system and operation. Comparing it with the performance characteristics of the existing design of the refrigeration station

At the end of the project, new complete designs for the station will be proposed and evaluated.

1.3 Importance of Project

The Refrigeration is important in the general life aspects. The importance of this project comes from the direct relation between the refrigeration engineering and the human being life.

As known, the refrigeration is used for the product storage for many wide ranges of products. In this project, the importance came from cold storage application in fruit and vegetables storage in order to make many types of them available all around the year. This serves the financial situation in the west bank region by providing the Palestinian market with the desired products in all times with competitive prices.

1.4 Project Outline

Chapter One:

This chapter includes introduction to refrigeration systems, general overview about the project, the importance of the project, the time Table of the project, Project outline, and general overview for the refrigeration history, refrigeration systems and cold storage applications.

Chapter Two:

This chapter includes the horticultural commodities preservation techniques and their storage effecting factors, And the cold storage requirements for food preservation.

Chapter Three:

This chapter includes general field survey study for the refrigeration station under study, analyzing its civil, mechanical, and electrical wiring designs.

Chapter Four:

This chapter includes the main design parameters of the refrigeration station, the inside and outside condition estimation, the storage circumstances for the stored products.

Chapter Five:

This chapter includes the multivariable cooling load calculations for the refrigeration station rooms. Considering both storage and chilling variable demands.

Chapter Six:

This chapter includes the station system design. Including all the refrigeration station systems; the refrigeration system, humidification system, ventilation system, and the control requirements of the refrigeration station operation.

Chapter Seven:

This chapter includes the new designed system evaluation. And brief comparison between the new system operation characteristics and the existing design referring to the required cold storage requirements for food preservation.

1.5 Historical Background

In prehistoric times, man was trying to keep food for long periods in which food was not available if stored in the coolness of a cave or packed in snow. In China, before the first millennium, ice was harvested and stored.

Hebrews, Greeks, and Romans placed large amounts of snow into storage pits dug into the ground and insulated with wood and straw. The ancient Egyptians filled earthen jars with boiled water and put them on their roofs, thus exposing the jars to the night's cool air. In India, evaporative cooling was employed. When a liquid vaporizes rapidly, it expands quickly. The rising molecules of vapor abruptly increase their kinetic energy and this increase is drawn from the immediate surroundings of the vapor. These surroundings are therefore cooled.

The intermediate stage in the history of cooling foods was to add chemicals like sodium nitrate or potassium nitrate to water causing the temperature to fall. Cooling wine via this method was recorded in 1550, as were the words "to refrigerate".

Cooling drinks came into vogue by 1600 in France. Instead of cooling water at night, people rotated long-necked bottles in water in which saltpeter had been dissolved. This solution could be used to produce very low temperatures and to make ice. By the end of the 17th century, iced liquors and frozen juices were popular in French society.

The first known artificial refrigeration was demonstrated by William Cullen at the University of Glasgow in 1748. Cullen let ethyl ether boil into a partial vacuum; he did not, however, use the result to any practical purpose.

Ice was first shipped commercially out of Canal Street in New York City to Charleston, South Carolina in 1799. Unfortunately, there was not much ice left when the shipment arrived. New Englanders Frederick Tudor and Nathaniel Wyeth saw the potential for the ice business and revolutionized the industry through their efforts in the first half of the 1800s. Tudor, who became known as the "Ice King", focused on shipping ice to tropical climates. He experimented with insulating materials and built ice houses that decreased melting losses from 66 percent to less than 8 percent. Wyeth devised a method of quickly and cheaply cutting uniform blocks of ice that transformed the ice industry, making it possible to speed handling techniques in storage, transportation and distribution with less waste.

In 1805, an American inventor, Oliver Evans, designed the first refrigeration machine that used vapor instead of liquid. Evans never constructed his machine, but one similar to it was built by an American physician, John Gorrie.

In 1842, the American physician John Gorrie, to cool sickrooms in a Florida hospital, designed and built an air-cooling apparatus for treating yellow-fever patients. His basic principle that of compressing a gas, cooling it by sending it through radiating coils, and then expanding it to lower the temperature further--is the one most often used in refrigerators today. Giving up his medical practice to engage in time-consuming experimentation with ice making, he was granted the first U.S. patent for mechanical refrigeration in 1851.

Commercial refrigeration is believed to have been initiated by an American businessman, Alexander C. Twinning, in 1856. Shortly afterward, an Australian, James Harrison, examined the refrigerators used by Gorrie and Twinning and introduced vapor-compression refrigeration to the brewing and meatpacking industries.

Ferdinand Carré of France developed a somewhat more complex system in 1859. Unlike earlier compression-compression machines, which used air as a coolant, Carré's equipment contained rapidly expanding ammonia. (Ammonia liquefies at a much lower temperature than water and is thus able to absorb more heat.) Carré's refrigerators were widely used, and vapor compression refrigeration became, and still is, the most widely used method of cooling. However, the cost, size, and complexity of refrigeration systems of the time, coupled with the toxicity of their ammonia coolants, prevented the general use of mechanical refrigerators in the home. Most households used iceboxes that were supplied almost daily with blocks of ice from a local refrigeration plant.

Beginning in the 1840s, refrigerated cars were used to transport milk and butter. By 1860, refrigerated transport was limited to mostly seafood and dairy products. The refrigerated railroad car was patented by J.B. Sutherland of Detroit, Michigan in 1867. He designed an insulated car with ice bunkers in each end. Air came in on the top, passed through the bunkers, and circulated through the car by gravity, controlled by the use of hanging flaps that created differences in air temperature. The first refrigerated car to carry fresh fruit was built in 1867 by Parker Earle of Illinois, who shipped strawberries on the Illinois Central Railroad. Each chest contained 100 pounds of ice and 200 quarts of strawberries. It was not until 1949 that a refrigeration system made its way into the trucking industry by way of a roof-mounted cooling device, patented by Fred Jones.

Brewing was the first activity in the northern states to use mechanical refrigeration extensively, beginning with an absorption machine used by S. Liebmann's Sons Brewing Company in Brooklyn, New York in 1870. Commercial refrigeration was primarily directed at breweries in the 1870s and by 1891, nearly every brewery was equipped with refrigerating machines.

Natural ice supply became an industry unto itself. More companies entered the business, prices decreased, and refrigeration using ice became more accessible. By 1879, there were 35 commercial ice plants in America, more than 200 a decade later, and 2,000 by 1909. No pond was safe from scraping for ice production, not even Thoreau's Walden Pond, where 1,000 tons of ice was extracted each day in 1847.

However, as time went on, ice, as a refrigeration agent, became a health problem. Says Bern Nagengast, co-author of Heat and Cold: Mastering the Great Indoors (published by the American Society of Heating, Refrigeration and Air-conditioning Engineers), "Good sources were harder and harder to find. By the 1890's, natural ice became a problem because of pollution and sewage dumping." Signs of a problem were first evident in the brewing industry. Soon the meatpacking and dairy industries followed with their complaints. Refrigeration technology provided the solution: ice, mechanically manufactured, and giving birth to mechanical refrigeration.

Carl (Paul Gottfried) von Linde in 1895 set up a large-scale plant for the production of liquid air. Six years later he developed a method for separating pure liquid oxygen from liquid air that resulted in widespread industrial conversion to processes utilizing oxygen (e.g., in steel manufacture).

The Development of Air-Conditioning

Willis Carrier, a Cornell graduate with a degree in engineering, noticed at a Brooklyn plant that a magazine printing process worked less efficiently in summer time due to the temperature and humidity. In 1911, he delivered a paper to the American Society of Mechanical Engineers that articulated the theory of air-conditioning — the use of refrigeration for comfort cooling — that we know today. Going beyond the theoretical, Carrier devised a new centrifugal compressor for refrigeration in 1923. At first, air-conditioning was limited to office and industrial applications. Soon patrons found comfort in public buildings like theaters, hotels and restaurants.

Says Bern Nagengast, co-author of a definitive history of refrigeration for the American Society of Heating, Refrigeration and Air-conditioning Engineers (ASHRAE), "Without refrigeration, air-conditioning wouldn't have gotten off the ground. Mechanical refrigeration made it possible and it has drastically changed the way we live and work. Many geographic areas were virtually uninhabitable without air-conditioning."

Figure (1-1) Development of Refrigeration in Air Conditioning Applications

Though meat-packers were slower to adopt refrigeration than the breweries, they ultimately used refrigeration pervasively. By 1914, the machinery installed in almost all American packing plants was the ammonia compression system, which had a refrigeration capacity of well over 90,000 tons/day.

Despite the inherent advantages, refrigeration had its problems. Refrigerants like sulfur dioxide and methyl chloride were causing people to die. Ammonia had an equally serious toxic effect if it leaked. Refrigeration engineers searched for acceptable substitutes until the 1920s, when a number of synthetic refrigerants called halocarbons or CFCs (chlorofluorocarbons) were developed by Frigidaire. The best known of these substances was patented under the brand name of Freon. Chemically, Freon was created by the substitution of two chlorine and two fluorine atoms for the four hydrogen atoms in methane (CH_4); the result, dichlorodifluoromethane (CCl_2F_2), is odorless and is toxic only in extremely large doses.

Though ice, brewing, and meatpacking industries were refrigeration's major beneficiaries, many other industries found refrigeration a boon to their business.

In metalworking, for instance mechanically produced cold helped temper cutlery and tools. Iron production got a boost, as refrigeration removed moisture from the air delivered to blast furnaces, increasing production. Textile mills used refrigeration in mercerizing, bleaching, and dyeing. Oil refineries found it essential, as did the manufacturers of paper, drugs, soap, glue, shoe polish, perfume, celluloid, and photographic materials.

Fur and woolen goods storage could beat the moths by using refrigerated warehouses. Refrigeration also helped nurseries and florists, especially to meet seasonal needs since cut flowers could last longer. Moreover, there was the morbid application of preserving human bodies. Hospitality businesses including hotels, restaurants, saloons, and soda fountains, proved to be big markets for ice.

In WWI, refrigeration in ammunition factories provided the required strict control of temperatures and humidity. Allied fighting ships held carbon-dioxide machines to keep ammunition well below temperatures at which high explosives became unstable.

In 1973, Prof. James Lovelock reported finding trace amounts of refrigerant gases in the atmosphere. In 1974, Sherwood Rowland and Mario Molina predicted that chlorofluorocarbon refrigerant gases would reach the high stratosphere and there damage the protective mantle of the oxygen allotrope, ozone. In 1985 the "ozone hole" over the Antarctic had been discovered and by 1990 Rowland and Molina's prediction was proved correct.

The basic components of today's modern vapor-compression refrigeration system are a compressor; a condenser; an expansion device, which can be a valve, a capillary tube, an engine, or a turbine; and an evaporator. The gas coolant is first compressed, usually by a piston, and then pushed through a tube into the condenser. In the condenser, the winding tube containing the vapor is passed through either circulating air or a bath of water, which removes some of the heat energy of the compressed gas. The cooled vapor is passed through an expansion device to an area of much lower pressure; as the vapor expands, it draws the energy of its expansion from its surroundings or the medium in contact with it. Evaporators may directly cool a space by letting the vapor come into contact with the area to be chilled, or they may act indirectly, i.e. by cooling a secondary medium such as water. In most domestic refrigerators, the coil containing the evaporator directly contacts the air in the food compartment. At the end of the process, the warmed gas is drawn toward the compressor.

1.6 Refrigeration Systems Overview

Refrigeration is defined as the transfer of heat from a lower temperature region to a higher temperature one. Refrigeration devices that produce refrigeration operate using cycle, and the most frequently used is the vapor-compression cycle. Some examples of refrigeration devices are heat pumps, refrigerators, automotive air-conditioners, and residential or commercial air-conditioners. All of these devices have one thing in common, to maintain an enclosed environment at low temperature.

The ideal vapor-compression cycle uses refrigerant as the working fluid to absorb and reject heat energy. The energy transfer allows the vapor-compression cycle to reduce or cool a closed environment. The ideal vapor-compression cycles assumes that the system is perfect based on thermodynamic theory, therefore neglecting any loses associated to performance.

In the ideal vapor-compression cycle, refrigerant enters the compressor as a saturated vapor (Figure 1) at state 3. As the refrigerant is compressed, it increases in temperature and pressure in process 1-2. After the compressor the refrigerant passes through the condenser. Heat energy (Q_H) is exchanged with the surrounding environment causing the refrigerant to cool and become a saturated liquid as shown in the process from 2-3. Next, the refrigerant passes through the expansion valve causing the temperature and pressure to decreases in process 3-4. Because of the reduction in temperature and pressure, the refrigerant enters the evaporator as a saturated mixture. As the refrigerant passes through the evaporator, it absorbs heat energy (Q_L) from the environment that it is trying to cool. The refrigerant exits the evaporator as a saturated vapor and returns to the compressor to begin the process. All over again (points 4-1).

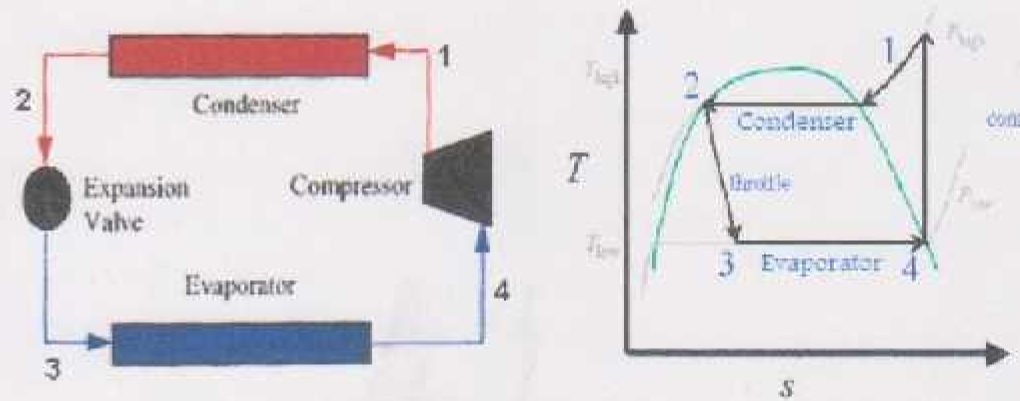


Figure (1-2) Ideal Vapor Compression Refrigeration Cycle

Due to fluid friction, heat transfer losses, and component inefficiency, the refrigeration cycle is unable to achieve complete thermodynamic saturation. The result of the inefficiencies and losses prevent the refrigeration cycle from operating at the optimum performance. This is called the actual vapor-compression refrigeration cycle. And that is which the nowadays refrigeration cycles are operating; these losses are compensated by the factor of safety for the system.

When designing a refrigeration system, it is important to understand how the refrigeration cycle works and the effects of component inefficiency on overall performance. Another important factor when designing a refrigeration system is the working fluid. The working fluid is commonly referred as refrigerant. It is the media used to absorb and reject heat energy. Chlorofluorocarbons (CFCs), ammonia, propane, ethane, ethylene, carbon dioxide, air and water are just some of the refrigerants used in refrigeration systems. The design of the system should include refrigerant that is nontoxic, non-corrosive, nonflammable, chemically stable, and inexpensive. In order to achieve sufficient heat transfer, the refrigerant and the surrounding environment must have a temperature difference at least 5°C to 10°C. As shown in Figure (1-3).

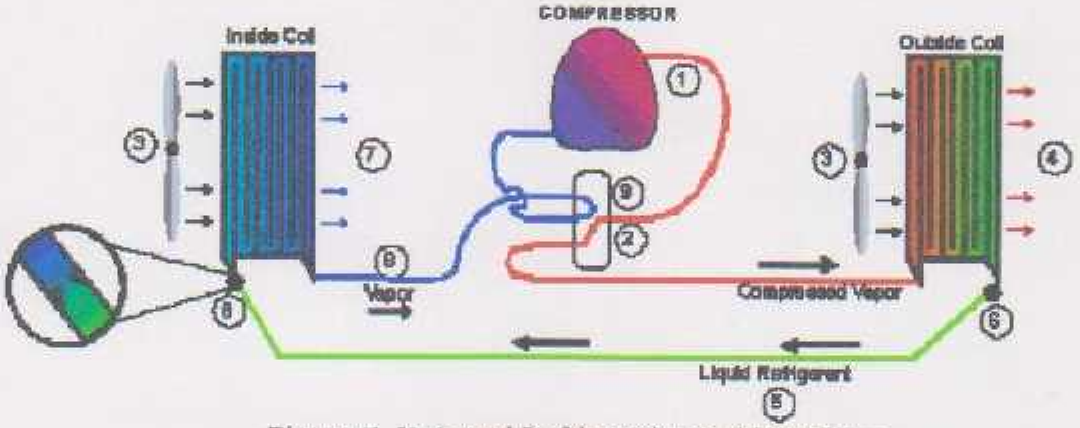


Figure (1-3) Actual Refrigeration and Heat Pump

1.7 Cold Storage of Fruit & Vegetables Requirements

Deterioration of fruits and vegetables during storage depends largely on temperature of the cold storage. One way to slow down this change and so increase the length of time fruits and vegetables can be stored, is by lowering the temperature to an appropriate level. It must be remembered that if the temperature is too low the produce will be damaged and also that as soon as the produce leaves the cold store, deterioration starts again and often at a faster rate.

1.7-1 Products Processing before Storage

A) Harvesting

It is essential that fruits and vegetables are not damaged during harvest and that they are kept clean. Damaged and bruised produce have much shorter storage lives and very poor appearance after storage. Dirty produce can introduce pests and moulds into the store.

B) Handling

It is important that the produce does not get dirty or damaged during handling. Careful handling should be the rule. The best option is for the produce to be prepared for storage in the field and placed carefully in the storage containers used in the cold store. This considerably reduces the amount of handling and will keep damage to a minimum. It is essential that the produce is handled and placed in the store as quickly as possible as delays between harvesting and cooling can substantially reduce storage life.

C) Preparation

If the produce is dirty it should be cleaned before storage. The sanitized water used has to be kept clean or fungus spores will be spread throughout the produce.

D) Preliminary cooling (Chilling)

Dipping the produce in cool water to remove field heat can reduce the energy requirements of the store. However, this can spread fungus spores throughout the produce. A suitable alternative is to pick the produce either early in the morning when it is cool or late in the evening and leave it overnight to cool down.

1.8-2 Storage Conditions

A) Temperature

All fruits and vegetables have a 'critical temperature' below which undesirable and irreversible reactions or 'chill damage' takes place. Carrots for example blacken and become soft, and the cell structure of potatoes is destroyed. The storage temperature always has to be above this critical temperature. One has to be careful that even though the thermostat is set at a temperature above the critical temperature, the thermostatic oscillation in temperature does not result in storage temperature falling below the critical temperature. Even 0.5°C below the critical temperature can result in chill damage.

B) Relative Humidity

For most produce, a high but not saturated, relative humidity is required, eg 85 – 95%. There is always some moisture loss during cold storage but excessive moisture loss is a problem. It is essential that the relative humidity is kept above 85%. This can be done by installing proper humidification system to achieve maintenance of high relative humidity in the refrigeration room.

Chapter Two

STORAGE REQUIRTEMENTS OF HORTICULTURAL COMMODITIES

2.1 Introduction

Many changes have been occurred in packing, handling, transportation, marketing and distribution of horticultural crops that directly influence cooling requirements for and results. The new preservation procedures made the product cooling more difficult. The current desire is to prolong the storage life, and market a product that better satisfies the consumers. These objectives require more uniform and precise cooling requirement.

2.2 The Need of Cooling

An understanding of the cooling requirements of the horticultural commodities begins with adequate knowledge of their biological responses. All fresh horticultural crops are living organisms; Carrying on the biological processes that are essential to the maintenance of life. They must remain alive and healthy until being processed or consumed. The energy that is needed for these life processes comes from the food reserves that accumulated while the commodities were still attached to the plant.

The process by which these food reserves are converted into energy is called respiration. In a complex series of steps, the stored food reserves (starches and sugars) are converted first to organic acids, then to more simple carbon compounds. Oxygen from the surrounding air is utilized in the process, and carbon dioxide is released. As in the following equation (see Figure 2-1):

$$C_6H_{12}O_6 + 6O_2 \rightarrow 6CO_2 + 6H_2O + \text{Energy (Respiration Heat)}$$

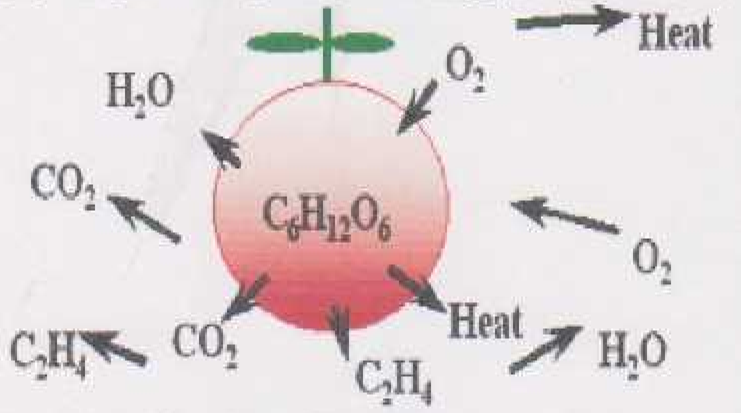


Figure (2-1) Normal Respiration of Products

If oxygen severely limited, anaerobic respiration will occur, aldehydes, alcohols and other undesirable materials will be produced, and the tissue will ultimately die. As in the following equation (see Figure 2-2):

$$C_6H_{12}O_6 \rightarrow CO_2 + \text{Alcohol} + \text{Aldehydes} + \text{Organic Acids}$$

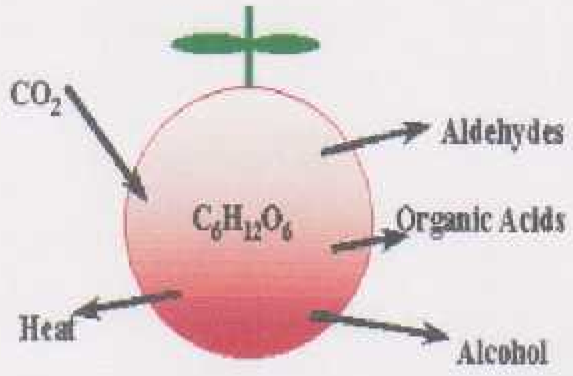


Figure (2-2) Anaerobic Respiration of Products

The horticultural commodities are not the tight, dense, closed tissues that they appear to be. Rather, they have an amazingly open structure, with a complete network of interconnecting air spaces (called intercellular spaces) throughout. This can be illustrated with many commodities by injecting air while the intact tissue is under water. Air bubbles appear almost instantly and simultaneously over the entire surface. The oxygen concentration in the intercellular spaces in the centre of many commodities is almost as high as in the surrounding atmosphere as a result of this structure.

Some of the energy that is produced through respiratory activity is utilized in maintaining the life process. Excess energy is released in the form of heat, called vital heat. The amount of vital heat varies with the type of product, variety, maturity or stage of repines, injures, temperature, and other stress related factors. This heat must be considered in any temperature management program.

Product temperature is a measure determinant of the rate of respiratory activity. Since the final result of this respiratory activity is product deterioration, it is normally desirable to achieve as low a respiratory rate as possible without danger of tissue injury or death. And exception would be during controlled ripening of fruit. Each 10 degree of temperature reduction reduces the respiratory activity by a factor of 2 - 4. For example, the respiratory rate of a product at 5 C would be only 1/4 to 1/16 of what it would be at 25 C. good cooling and temperature management practices are therefore critical to lowering the rate of physiological deterioration.

2.2-1 Speed of Cooling

Many products benefit from prompt, thorough cooling. For example, strawberries experiences increasing deterioration losses as delays between harvesting and cooling exceeds one hour. A similar pattern exists for sweets cherries when delays exceed about four hours. These fruits benefits from cooling even when rewarming will occur during subsequent handling; deterioration is proportional to the total exposure time to warm temperature, and not to the pattern of cooling and warming.

Some Bartlett pears should be cooled to – 0.5 C within 24 hours of harvest to reduce losses from an enzymatic breakdown problem called (watery breakdown). This problem is especially serious among fruits harvested in mid- and late season. The rapid cooling to low temperature also limits decay and ensures max storage life, and thus is generally recommended for all Bartlett pears that are to be stored.

There are exceptions to the need for prompt cooling after harvesting. For many years freshly-harvested freestone peaches in South Africa have been held at ambient temperature about 36 hours prior to cooling. This fruit is subject to chilling injury (dry and / or brown tissue), and the delayed cooling treatment was found to delay onset of the problem. A recent report suggests similar treatment may be useful for Honeydew melons if they are gassed with C_2H_4 to initiate repining prior to shipment at chilling temperature.

While most fruits would not be expect to be harm by moisture condensation during warming, some grapes have reportedly developed berry cracking (and subsequent rot) where condensation moisture remained in the undesirable unless low temperatures could be maintained. Thus, it is important to know the product and its temperature and marketing requirements, before selecting a cooling program.

The amount of heat in product is governed by the temperature around it. The temperature difference between newly harvested product and its optimum storage temperature is an indicator of field-heat. Rapidly lowering the temperature of harvested produce to near storage temperature is known as precooling, or removal of field-heat. Product is usually precooled to 78 or 88 % of the temperature difference. Additional cooling is limited by the time and energy required to reduce the produce temperature to the optimum storage temperature.

Many methods are available to precool produce. The one you choose depends on several factors:

1. What method can the produce tolerate?
2. How quickly must the produce be cooled?
3. To ensure a high-quality product?
4. Is the method energy efficient?
5. Is skilled labor required?
6. How expensive is the equipment?

What utilities are available on the site? Precooling is highly recommended and often required by processors. When marketing produce in wholesale or processing outlets, a grower has minimal or no control over produce storage and handling after it is sold. If the produce does not hold up in the distribution chain, then the grower is usually blamed for having poor handling practices. Precooling "buys" the grower shelf-life time that the wholesaler and retailer may reduce with poor handling procedures.

The most efficient way to precool and store the horticultural commodities is to use forced air cooling. Forced-air cooling is similar to room cooling in that produce is placed in a cold storage room. Forced-air cooling is designed to force cold air through produce containers instead of around them. Converting existing cooling facilities to forced-air cooling is practical and feasible. There are variations of forced-air cooling that fit specific container needs. The key to forced-air cooling is moving the cold air through the container and its contents. Important factors in container ventilation are location of container vents, stacking of containers, and size of the vents. Container vents should be aligned whether the containers are straight-stacked or cross stacked, to maximize air flow through the containers. If vents are too small or too few, air flow is slowed. If there are too many, the container may collapse. In this method, containers are stacked close together (tight). Five percent vent-hole space per side and/or end is best. Liners, bags, wrappers, or dividers can slow the flow of air through the container, so precooling produce is usually recommended prior to additional packing. The following are forced-air cooling alternatives. (See Figure 2-3)

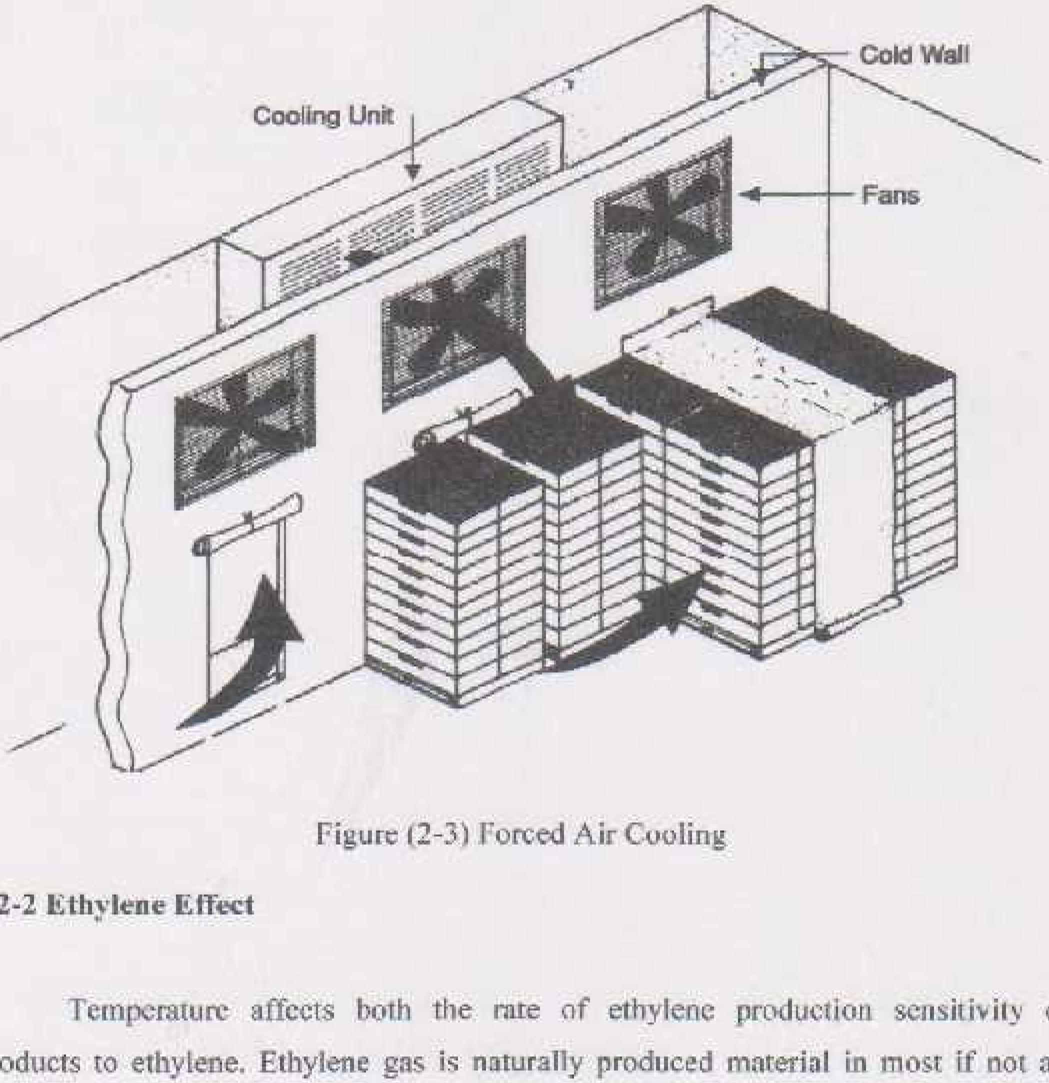


Figure (2-3) Forced Air Cooling

2.2-2 Ethylene Effect

Temperature affects both the rate of ethylene production sensitivity of products to ethylene. Ethylene gas is naturally produced material in most if not all plant tissue. This simple chemical compound is generally recognized as a fruit ripening hormone. It can have important beneficial or determinal effects on fresh commodities depending on the management needs.

For these ethylene effects to occur, a minimum concentration must accumulate within the internal atmosphere of the product, and the temperature must be above a minimum level. These threshold concentration and temperature levels are not well defined. However, since the rate of both production and action of ethylene are temperature-dependent, rapid cooling and good temperature management are vital if fruit ripening and are determination processes are to be delayed.

In terms of respiratory activity, fruits-including fruit-type, vegetables-have been grouped into two classifications, "climacteric" and "non-climacteric". Climacteric fruits are those which normally ripen after harvesting, during which time sugars normally increase and volatile constituents (flavor and odors) develop. Flesh softening accompanies ripening, but if such fruit are picked while immature they may soften without development of sweetness and flavors. Ethylene gas will initiate ripening, which is accompanied by a rapid rise in the respiratory activity in the fruit (called the climacteric rise). Included are apples, peaches, papayas, cantaloupes, tomatoes, and many other fruit.

The non-climacteric fruits, including citrus, grapes, strawberries, and others, don't ripen after harvest and exhibit no rise in respiratory activity. Even under optimum ripening conditions, dessert quality doesn't improve noticeably. Ethylene gas may effect color changes among theses fruits; for example the breakdown of the green chlorophyll 11 pigments causes orange to "color". However, the sugar, acid, and flavor of the fruit are not influenced by the treatment.

Ethylene gas is also implicated in a number of product injury and deterioration problems. Among these are russet spotting of lettuce, "sleepiness" in carnation, leaf abscissions, rind pitting in citrus, bitter flavor in carrots, yellowing of cucumbers, softening of kiwifruits, and others.

2.2-3 Moisture Loss

Fresh commodities constantly lose water to the surrounding environment After harvest this lost water cannot be replaced by the plant (with the exception of flowers), and weight loss will occur (see figure 2-4). Many products show visual shriveling or wilting after losing 3 to 5 % of their initial weight. They lose water as a result of a water vapor gradient between their essentially saturated internal atmosphere (within the intercellular spaces) and the less-saturated external atmosphere. Water vapor migrates in the direction of lower concentration, primarily through natural openings on the fruit surface, but also through surface injuries. The rate of migration is controlled by the vapor-pressure difference between the product and its environment, which is governed by temperature and relative humidity. Warm air can hold much more water-vapor than cold air. Relative humidity is a measure of the amount of water vapor in the air as a percent of the amount of air can hold at that temperature. At 25 C and 30 % relative humidity, the product will lose water 36 times faster than it would at 0 C and 90 % relative humidity. Thus, maintenance of low product temperature is essential in reducing water loss and subsequent product shriveling and wilting.

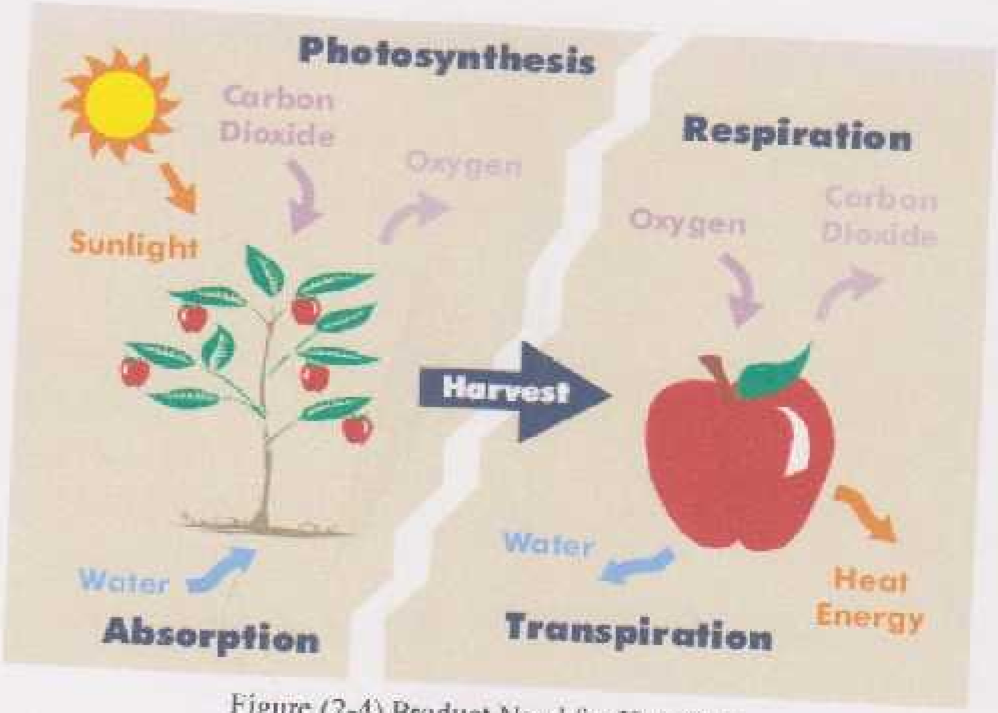


Figure (2-4) Product Need for Humidity

2.2-4 Rot Organisms

Post harvest management of fresh horticultural commodities actually involves two living systems- the product and the microorganisms that attack it. Of thousands of potential microbial pathogens, only a few cause problems. These few organisms, however, cause extensive direct loss of fresh fruits, vegetables, flowers and ornamentals.

Temperature affects the rate of growth and spread of these microorganisms in the same way that it affects the commodity- the lower the temperature the slower their life processes progress. Certain organisms that can cause severe losses will not grow at low storage temperatures. For example, Ruinous rot ceases growth at about 5 C, and germinating spores have been found to be chilling – sensitive, and to be killed about to day's exposure at 0 C. Other organisms will continue growth, but at a very slow rate at safe storage temperatures near 0 C. In studies with Botrytis rot of strawberries during a seven-day marketing period, at 2 C or below ,germinating spores would not penetrate into the fruit ,and at 0 C mycelium (the vegetative fungal growth) would not penetrate a healthy fruit from an adjacent invaded fruit. Good temperature management, thus, plays a vital role in reducing microbial loss problems.

2.2-5 Injuries

Physical injuries can result from abuses to fresh commodities at any temperature, but temperature affects the severity of the product response to those injuries .Bruises and other wounds because increased ethylene production, which may accelerate respiration, cause deterioration problems, or imitate fruit ripening. Bruising usually damages the natural barriers on the product surface, thus increasing the opportunity for water loss and for entry for rot organisms .Prompt cooling and maintenance of low temperature reduces the results of injuries by affecting all of these processes.

2.2-6 Low Temperature

Good temperature management is the single most important factor in delaying product deterioration .Prompt cooling and maintenance of proper temperatures are both essential parts of the temperature management system. For many products, this means maintaining as low a temperature as possible without danger of freezing. The freezing point will vary with soluble solids content; and for some products freezing point guides are available. How closely the freezing point can be approached will depend on the accuracy and sensitivity of the temperature controls.

Many horticultural crops are subject to "chilling" injury at temperatures considerably above their freezing point .Often these crops must be held at modified temperatures. Chilling injury symptoms include surface or internal browning, surface pitting, failure to degree, failure to ripen, increased susceptibility to micro-organisms, texture changes (mealy or woolly texture) and loss of flavor. Most crops of tropical and subtropical origin and even some deciduous fruits are subject to chilling injury. Guides to the lowest safe temperature for various commodities are available.

2.3 Changes Affecting Cooling

Cooling is affected by many different factors throughout the handling system from the field to the consumer. Some handling changes have increased the need for more rapid, through cooling. Included here are the desires to market more mature (or even ripe) products, to extent the storage period, to supply more distant markets, and to meet new consumer quality standards. Some other changes have made fast, thorough cooling more difficult to attain, and have caused the development of new cooling procedures. Included here are the changes from field lugs to large bins, new package designs, unitized handling, and changing transport equipment. The effects of these changes must be understood if improved products performance is to be achieved.

2.3-1 Extended Market life

Handlers want to extend the post harvest life of fresh horticultural commodities for a variety of reasons. Processors may store to accumulate product for processing, to protect quality when deliveries exceed processing capacity, to hold the product over weekend periods in order to avoid labor complication, or simply to extend the processing season. Storage may be for a few days to several months.

Packers may want to store the product in order to extend their marketing season, to accumulate a supply for holiday period, to facilitate orderly marketing, to attempt avoiding price declines during period of over-supply, or to develop export markets. Thus, longer shelf life may be required. Since deterioration is a function of time and temperature, faster cooling can significantly extend shelf life.

2.3-2 Consumer Demands

There is increasing evidence that consumer satisfaction with fresh commodities is related to certain "quality" aspects, especially appearance, flavor, maturity or ripeness, or ability to quickly ripen after purchase. With consumer purchasing larger quantities of fresh commodities, handlers are anxious to make the changes needed to sustain consumer demand. To meet these requirements fresh produce shippers are shipping slightly more mature products that have developed more of the characteristic aromas and flavors. Fast cooling and good temperature management are vital to protecting such commodities.

2.3-3 Transportation

Refrigerated transport equipment has changed during the past generation from the use of ice in bunkers to mechanical refrigeration system. The correct use of ice bunkers equipment, which included proper loading patterns, frequent re-icing and adequate air circulation capacity, generally allowed reasonable of most products during transport. Most mechanical refrigeration equipment in current use is designed to maintain temperature, but lacks the air flow and refrigeration capacity needed for rapid cooling. Further, packages and loading patterns have changed. High-density loads are used to minimize transportation costs. These factors all inhibit cooling during transit. Therefore, through produce cooling before loading and protection against warming are very important.

2.3-4 Packaging

In harvest operations, small filed lugs have generally been replaced by large pallet bins, often holding about one-half ton of product. Those filed lugs where separated by cleats, which facilitated air flow when cooling was needed. In contrast, the larger dimensions pallet bins results in much of the product being remote from the top or side air flow surfaces. This allows little opportunity for cold air to penetrate into the bin in normal room cooling operations.

2.3-5 Shipping Containers

Corrugated fiberboard has essentially replaced wood in the construction of shipping containers. Corrugated containers normally stack more tightly than wooden containers and lack cleats which on wooden containers facilitated air circulation during storage and transport. Further, they generally have much less ventilation than the wood containers they replaced. Internal packaging materials, such as warps, trays, liners, pads, and plastic bags further restrict heat removal. Despite these adverse properties, proper stacking and adequate spacing of corrugated shipping containers in storage pallets, together with provision for high velocity air flow, can provide reasonable cooling speed.

2.3-6 Unitization

Many horticultural products are now handled and shipped unitized on pallets. When corrugated containers are tightly stacked on a pallet for unitized handling, cooling problems become more servers. Containers vents, even if adequate sized, are too often blocked when containers are cross-stacked on pallets, and much of product has little access to cold air. Cooling of such units in conventional cold rooms is slow and irregular, and often inadequate without some modification in containers design and venting pallet stacking patterns, fan operation and/or cooling systems.

2.4 Meeting These Changing Needs

The conflict between the need to achieve more rapid cooling and the increasing difficultly of cooling appears almost impossible to resolve through the use of conventional room cooling procedure. However, rapid cooling systems are available which can achieve good results if properly used.

2.4-1 Cooling as Part of Handling System

The entire product handling system must be considered when cooling facilitates are planned because any change in the system may effect cooling rate, uniformity and requirements. The packing method will dictate how, and sometimes when, the product is presented for cooling. Packaging materials and design will affect access of the coolant to the product, and pallet stacking patterns will influence coolant flow through and around containers. Loading patterns, transport equipments, and marketing procedures all greatly affect cooling requirements. The maximum market life of many commodities may be approached when they are shipped for long distances by refrigerated ocean transport for export marketing.

2.4-2 Cooling and Storage are Two Separate Operations

The refrigeration capacity for fast cooling and for cold storage is quite different. For example, it takes about 100 times more refrigeration to cool pears in 24 hours than to cold store them for 24 hours. Even when the fruit is cooled over a six-day period, the daily refrigeration capacity during cooling is almost 25 times for cold storage. This ignores the refrigeration required to remove the heat entering the facility through walls, doors, fans, forklifts, etc.

There are other differences between cooling and cold storage that must be considered. During fast cooling, high air velocity will not increase the total water loss from the product if it occurs only while the product is being cooled. However, during subsequent cold storage high air velocity will desiccate horticultural commodities because of the long period of exposure to rapidly moving air. High relative humidity is essential to prevent excessive water loss during cold storage, but is not as important during the short cooling period.

Thus, cooling and storage are two separate operations that have vastly different requirements. The specific requirements for achieving fast, uniform cooling must be considered independently of the cold storage requirements.

2.5 Preparation before Cooling Process

The preservation of the horticultural products requires product processing before cooling it, since the cooling effect doesn't gives its desired objective of food preservation if the post-cooling of the products hasn't been performed.

There are many steps and processes that should be performed before initiating the cooling process of the products starts from the harvesting of the product, and ends with the cold storage preparation for preserving.

2.5-1 Product Cooling Delay Time

Cooling delays cause reduced product quality for three main reasons:

1- Allowing respiration and associated normal metabolism to continue at high rates, consuming sugars, acids, vitamins, and other constituents.

2- Fostering water loss.

3- Increasing decay.

Delays may also allow increased susceptibility to ethylene damage, but ethylene concentrations are usually low near cooling facilities and ethylene does not usually cause as much damage as the other three factors. Delays can also cause undesirable product curvature and growth, but this is a problem with only a few commodities such as asparagus and green onions.

2.5-2 Respiration

Fresh produce consumes photosynthetic that were stored in the product before harvest. Consumption rate depends on the respiratory activity of a particular commodity and its temperature. Commodities such as apples, cabbage, citrus, potatoes and Table grapes have low respiration rates compared with avocados, mushrooms, asparagus and sweet corn. At temperatures above 70° to 75°F (21 - 24°C), respiration is especially high. For example at 90°F (32°C), asparagus respires three times faster than at 60°F (16°C). Temperatures above 90°F (32°C) may cause ripening disorders in commodities such as tomatoes and most tree fruits. Exposure to high temperatures and to direct sunlight can cause sunburn and sun scald injury. Cooling products from high summer field temperatures to room temperature, 70° to 75°F (21 - 24°C), significantly slows respiration.

Increased respiration caused by a few hours of cooling delay rarely causes a noticeable effect on external product quality. The consumer may detect slightly poorer eating quality and appearance and the product may have lost some nutritional value, but it is often very salable. Several tests with cantaloupe melons have shown that cooling delays up to 12 hours, even with afternoon harvested product, cause no consistent loss of visual appearance or soluble solids. However delays do cause measurable effects on quality and many produce items may not be suitable for long-distance transport or long-term storage if exposed to significant cooling delays. For example, sweet corn begins losing sweetness immediately after harvest and the rate of loss increases with increasing temperature. If cooling is delayed sweet corn should be sold quickly. A few commodities such as peaches, nectarines, plums, Bartlett pears, and tomatoes are intentionally held at 60° to 75°F (16°- 24°C) after packing to promote ripening. If kept at a relative humidity above 85%, these commodities will actually improve in quality in a controlled ripening process. But they must be cooled after ripening has been started to slow further ripening during transport and handling. Lots prone to decay may experience increased decay losses because of ripening conditions. Although this can be considered an advantage in some situations because decayed product can be removed before shipping.

2.5-3 Water loss

Shriveling and the loss of fresh, glossy appearance are two of the most noticeable effects of cooling delays, particularly for commodities that lose water quickly and show visible symptoms at low levels of water loss, like most leafy vegetables. Air temperatures 70°-75°F (21- 24°C) have a particularly great effect on rate of water loss. For example, Thompson seedless grapes show visible symptoms of water loss (stem shriveling) at 3% or less. To prevent the consumer from seeing shrivel, moisture loss in the field should be kept less than 1%. At 90°F, a 1% loss can occur in about 2 hours, but at 70°F cooling can be delayed up to 12 hours. Moisture loss is slowed by holding produce in plastic liners. Liners should be vented to prevent temperature rise caused by product respiration and damaging levels of carbon dioxide. Solid bin covers also act as a moisture barrier. Some produce items can tolerate water contact, and spraying them with water slows product moisture loss and can even rehydrate slightly wilted produce. However water remaining on the surface tends to increase decay development and water should be clean and sanitized. Many leafy vegetables can completely recover from less than a few percent water loss by contact with hydrocooler water. If hydrocooling is not available, many vegetables can be sprayed with water before forced air or vacuum cooling to regain some water and reduce loss during subsequent cooling.

2.5-4 Decay

Cooling delays tend to increase decay losses, although decay becomes apparent many days after the cooling delay. Damage can be minimized by applying decay control treatments within a reasonable time after harvest. For example Table grapes should fumigate with sulfur dioxide within 12 hours after harvest. The process can be done with warm fruit and cooling delays may cause cold storage operators to set up facilities for fumigation separate from their forced air cooler. Another option for dealing with cooling delays in grapes to field-pack they in plastic liners and a sulfur dioxide generator pad. The liner will slow water loss and the pad will control Botrytis decay. Free water together with high humidity speeds decay development in tree fruits and berries. For example at room temperatures, only four hours of contact with free water allows brown rot to penetrate fruit tissues. Decay-prone fruits and vegetables should be protected from prolonged water contact and very high humidity during cooling delays.

2.5-5 Recommendations for minimizing damage caused by cooling delays

1. Protect produce from temperatures above 21 - 24°C. Start harvest as early as possible in the morning. On days that are predicted to be especially warm, harvesting may need to be halted when air temperatures exceed about 85°F (29°C).

2. Get as much product as possible to the cooler before the period of expected electrical interruptions. Trucks should make trips back to the cooler on a scheduled basis not just when they are full. If outages typically begin at 2 PM, all trucks should leave the field by 11 AM to noon so that the maximum amount of product can be in the cooler for at least some time before the cooler shuts down.

3. Protect product from moisture loss by using vented plastic liners, bin covers, or plastic containers. Some products like carrots can be sprinkled with water to reduce moisture loss during temporary holding at warm temperatures.

4. Begin decay control procedures within 8 to 12 hours after harvest. Field and packinghouse decay control procedures must be carefully applied with product exposed to cooling delays because of its increased potential for decay development.

5. Product destined for long-term storage or long distance transport should not be subjected to cooling delays.

6. Product subjected to long cooling delays should be marketed quickly.

7. Use the following Table (2-1) as a guide for acceptable time between harvest and the start of cooling. It is based on typical ambient conditions during normal harvest periods. Colder than normal air temperatures may allow longer

delays in cooling. If the delay time added to the typical time between harvest and the beginning of cooling is greater than the allowable delay, then backup generation or refrigeration may be needed.

Product	Allowable delay (hours)	Disadvantage of cooling delay	Advantage of cooling delay	Comments
Vegetables				
Artichoke	8	Water loss	None	
Asparagus	4	Increased toughness; Reduced shelf-life	None	
Broccoli	4	Water & firmness loss, reduced shelf-life	None	
Cauliflower	8	Water loss	None	
Carrot	8	Water loss, loss of crispness	Reduced cracking if carrots are cold and turgid	Carrots reabsorb water during hydrocooling
Cucumber	8	Water & chlorophyll loss	None	
Dry garlic	16	None	Curing & drying	
Dry onion	16	None	Curing & drying	
Green beans	8	Water loss	None	
Green Onion	8	Texture loss	Water loss during cooling delay may reduce curvature	
Mushroom	4	Water loss, decay	None	
Potato	16	Weight loss if low relative humidity	Cure harvest wounds w/ high RH	
Peppers	16	Water loss; loss of firmness, increased decay	Reduce mechanical injury if peppers cold and turgid	
Sweetpotato	16	Weight loss if low relative humidity	Cure harvest wounds w/ high RH	
Spinach	4	Water loss	None	Can be rehydrated
Summer squash	8	Water loss	None	
Sweet corn	4	Sugar loss	None	
Tomato	16	Increased decay; color development	Color development	Apply fungicide within 8 hr.; higher pulp temp. can cause decay from dump tank water absorption
Leafy Green vegetables	4 – 8	Loss of crispness, water loss	Product is subject to mechanical injury if cold & turgid	Greens reabsorb water during hydrocooling

Table (2-1) Allowable maximum cooling delay between harvest and the start of initial cooling for vegetables

2.6 Product Preparation

The product preparation is very important stage before the cooling process. The preparation process consists of three main steps:

- Classification of Product
- Washing, Sanitizing.
- Packaging.

2.6-1 Classification of Product

The product should be classified after the harvesting process. The harvested product includes injured fruit, immature fruits, and fanged fruits. These fruit causes the spraying of diseases among the product. So, before any step of preservation; the product classification must be accomplished. (See Figure 2-5)

Figure (2-5) Classification of Product

2.6-2 Washing and Sanitizing

After the product classification is performed, the washing and sanitizing is performed for the healthy product. The products is washed in hot water (about 40 C) mixed with special sanitizing materials. This process is done on an automatic conveyer for this process. As shown in Figure (2-6).

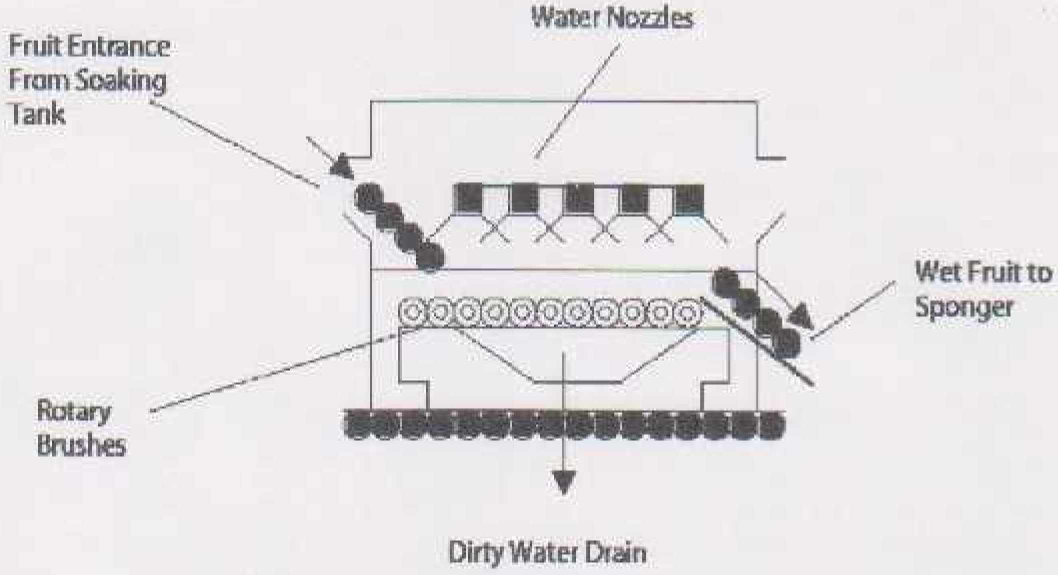


Figure (2-6) Washing and Sanitizing Unit

2.6-3 Packaging

After the processes mentioned above, the packaging process for each type of product is done depending on the product type. (See figure 2-7)

Figure (2-7) Packaging of Classified Product

Before the product is cooled in the refrigeration room, the room should be well-ventilated, cleaned, and sanitized. Then the product is cooled and stored.

2.7 Cold Storage Requirements

The cold storage design must satisfy all of the product cooling and storing requirements needed to keep the product with acceptable quality and marketable characteristics. The cold storage requirements are:

- Civil design.
- Refrigeration system.
- Humidification system.
- Air Handling system.
- Control and monitoring system.

2.7-1 Civil Design

The cold storage civil design must satisfying many aims, the main aim is to enable the cold storage to handle the desired a mount of product that satisfies the aimed markets, also the cooling rooms should be squared and wide to facilitate the product distribution and the uniformity of cooling process.

Also the insulation of the construction should be enough to lower the continuous heat wall gain, which forms a part of the cooling load of the refrigeration rooms. And the arrangement of rooms should facilitate the installation of the other systems and their maintenance, and should facilitate the transportation process of the product. And it's recommended to be under grounded.

2.7-2 Refrigeration System

The refrigeration system of the cold storage must be able to handle the cooling requirements for the harvested product. Based on the collected design parameters; the refrigeration system designed.

The refrigeration system must perform two main processes; chilling of the product, and the storing of the product. In the first process, the refrigeration system removes the field heat of the product rapidly in a limited time to prolong its life time and prevents moisture loss. And in the storing of product the refrigeration removes the wall gain, respiration, and other miscellaneous heat gains from the refrigerated space.

The refrigeration system recommended to be based on variable loading depending on type and amount of product to be cooled. Such that the system will operates under specific working circumstances depending on the loading level. And the system should ensure the uniformity of cooling of the product; Such that the product temperature of all the quantity varies uniformly with time.

2.7-3 Humidification System

The humidification system of the cold storage should be designed to compensate the loss of the moisture content in the refrigerated rooms, to ensure the keeping of relative humidity from 85-100% depending on the product type.

The relative humidity in the cold storage rooms decreases as a result to the frosting that is formed on the indoor cooling unit and the ventilation of the cold storage air. This affects badly on the product health; the decrement of the relative humidity in the cold storage air under the design relative humidity leads to the loss of water from the product and color changing and shriveling.

2.7-4 Air Handling System

Each room in the cold storage facility need to be ventilated, the ventilation main objective is to get rid of the alcoholic gases, ethylene, and the high concentrations of CO_2 from the refrigerated space, and inject a fresh air in place. This process depends on the volume of the refrigerated room.

The suction of these gases is performed from the bottom of the refrigerated space; because these gases distinct by their high molar mass, so they sinks to the bottom when the air circulation is stopped. The injected air should be well-filtered and distributed in the room.

2.7-5 Control and monitoring system

This system is required to monitor the previous systems and their performance. The control system performance considers each system set point, and leads the performance of each system to accomplish this set point characteristic within acceptable range.

For example, the set point for the inside relative humidity for grapes storing is 90%, the control system enables the working of the humidification system and control its related variables to accomplish this value ± 3%. And so on.

This system is considered to be the most important system, because it synchronizes the cold storage performance and provides significant monitoring for all cold storage systems and requirements.

Chapter Three

PRACTICAL SURVEY STUDY

FOR

HALHUL'S REFRIGERATION STATION

3.1 Introduction

Every structure consists of basic layers that combined together forming the whole foundation. These layers are the architectural design, the civil design, the mechanical design, and the electrical design, including the accessories of each design and its components.

Halhul's refrigeration station has all the previous components; The station system design will be studied and analyzed carefully, to get an overview for proposed designs of the refrigeration station system, and find out the problems associated with these systems and there performance in food preservation.

In this chapter we will consider the civil, refrigeration, and the electrical design for the refrigeration station by study, to obtain a complete overview about the designs of the refrigeration station.

3.2 Civil Design of the Refrigeration Station

The civil design consists of many branches that forming together the whole design, starting from the top plan of the refrigeration Station and ending with the structure of the walls of the refrigeration station.

The refrigeration station is located under Halhul's fruits and vegetables central market at the basement of the main building of the central market.

The civil construction includes walls, ceiling, floor, and partitions (All shown in the construction diagram). Starting from the walls in the refrigeration system is divided into two main parts:

- The outside walls.
- Partitions.

A) The Outside Walls

The outside walls form the outside perimeter for the refrigeration station. The major height of these walls is under the ground level (2.8), and the rest is over the ground level (1.2m).

A-1) <u>Higher Part</u>

The construction of the higher part of walls is as follows:

- Building stone 5 cm thickness.
- Concrete. 10 cm thickness.
- Asphalt, 5 mm thickness.
- Insulator "Seloncis", 2 cm thickness.
- Asphalt roles, 5 mm thickness.
- Reinforced concrete, 25 cm, from grid to grid 20 cm.
- Asphalt, 5 mm thickness.
- Polystyrene boards, 10 cm.
- Steel sheet, 2 mm thickness.

A-2) <u>Lower Part</u>

The construction of the lower part of walls is as follows:

- Asphalt, 5 mm thickness.
- Insulator "Seloncis", 2 cm thickness.
- Asphalt roles, 5 mm thickness.
- Reinforced concrete 200 kg/cm^2, 25 cm thickness, from grid to grid 20 cm.
- Asphalt, 5 mm thickness.
- Polystyrene boards, 10 cm.
- Steel sheet, 2 mm thickness.

B) Partitions

The Partitions construction is as follows:

- Steel sheet, 2 mm thickness.
- Polystyrene boards, 10 cm.

- Brick, 10 cm.
- Polystyrene boards, 10 cm.
- Steel sheet, 2 mm thickness.

C) Floor Construction

The floor of the station is all composed of the same construction materials, which are:

- Reinforced concrete 200 kg/cm^2, 10 cm thickness.
- Base course, thickness 10 cm.
- Grade, 50 cm thickness.
- Base stone, 3 m thickness for calculations.

The ground is loaded by concrete bridges 60X50 cm.

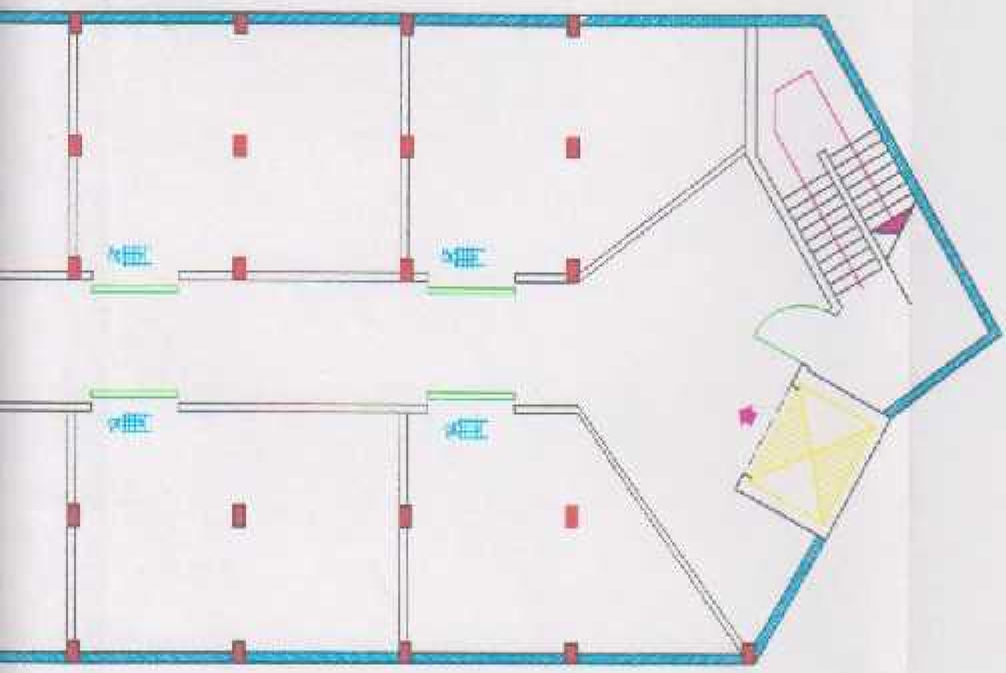
D) The Ceiling of the Station

The ceiling of the station is also the same for all station. Same construction materials:

- Brick, 20 cm thickness
- Reinforced concrete, 10 cm thickness.
- Asphalt, 5 mm thickness.
- Polystyrene boards, 10 cm.
- Steel sheet, 2 mm thickness.

E) The Columns

All are composed from the reinforced concrete 300 kg/cm^2. Dimensions are 30X50 cm, covered by polystyrene boards, 10 cm thickness. And steel sheets of 2 mm thickness.



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Project

Refrigeration Station -Halhul

Sheet Title

Archticture Plane

Senior Project

Date
2006-2007

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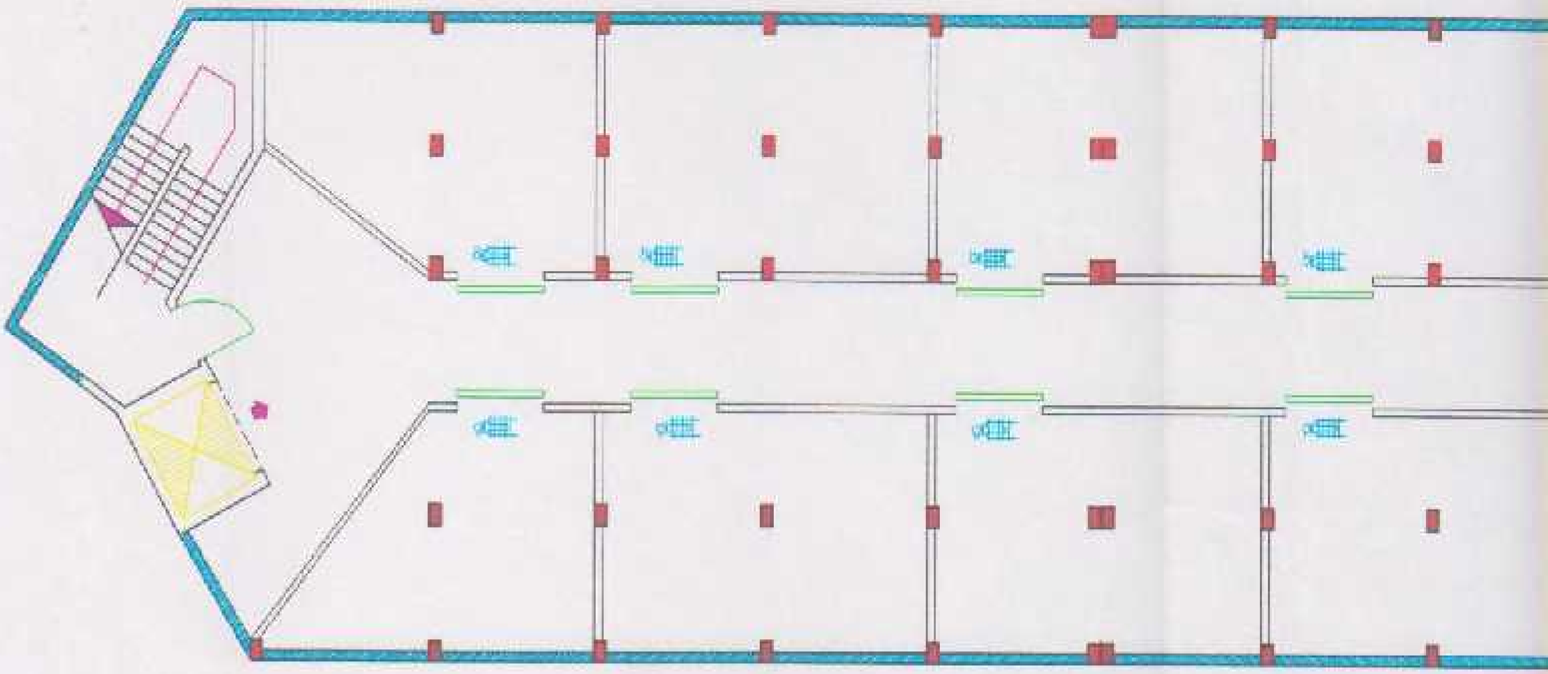

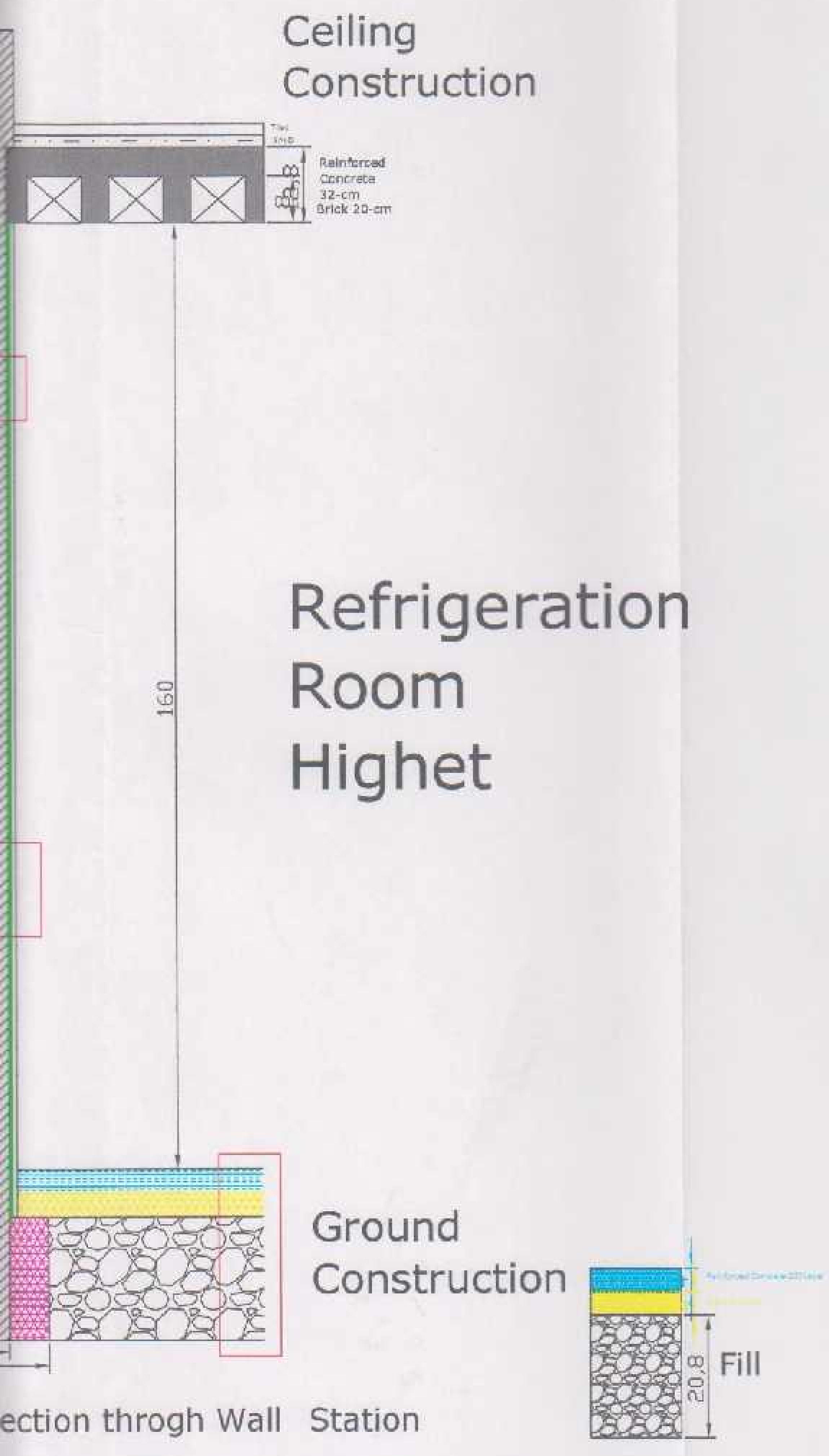

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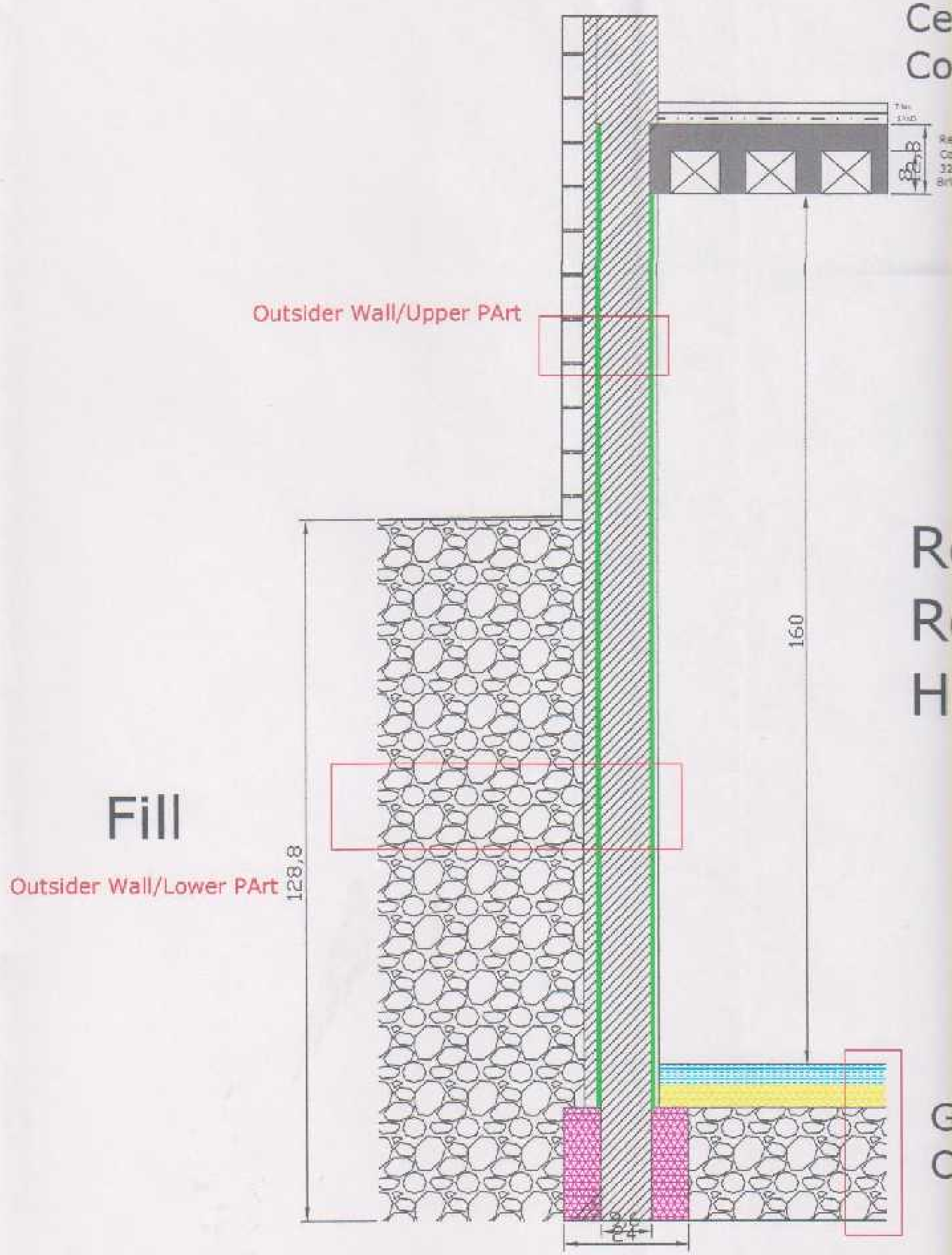

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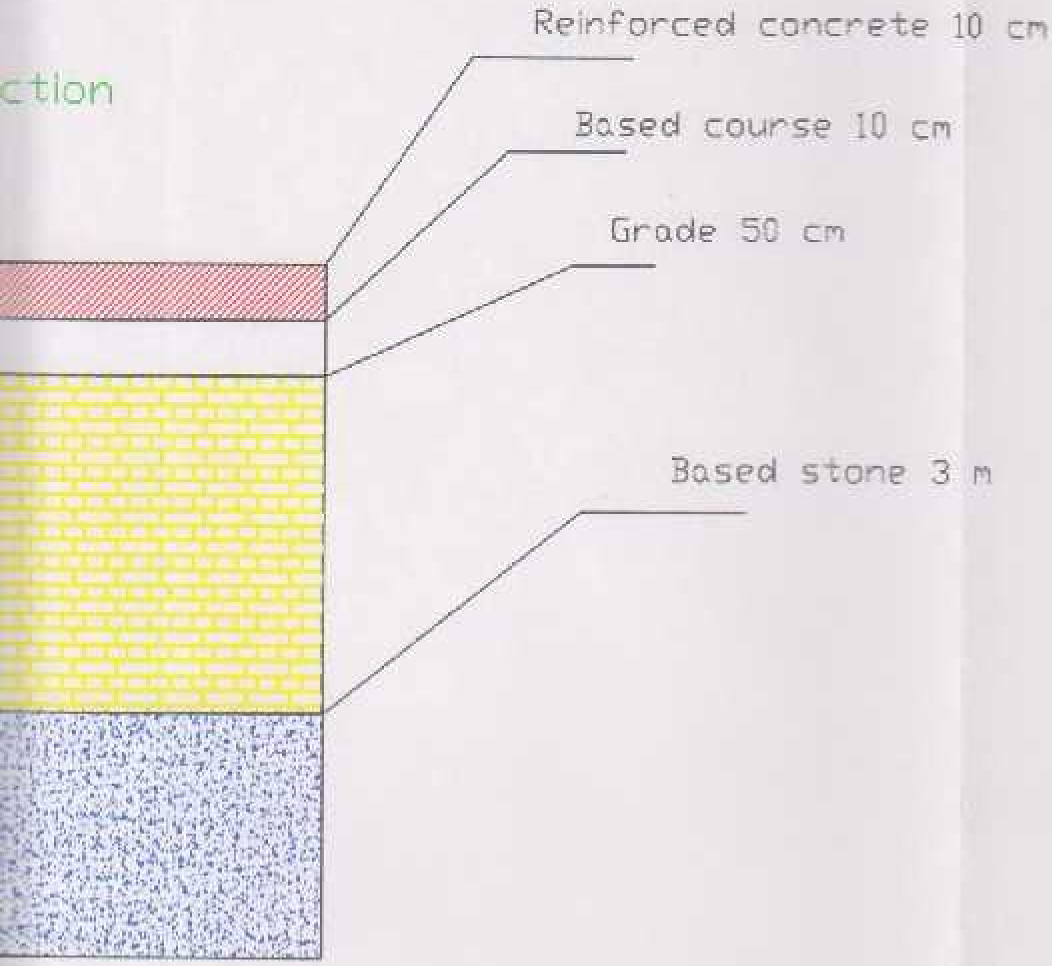

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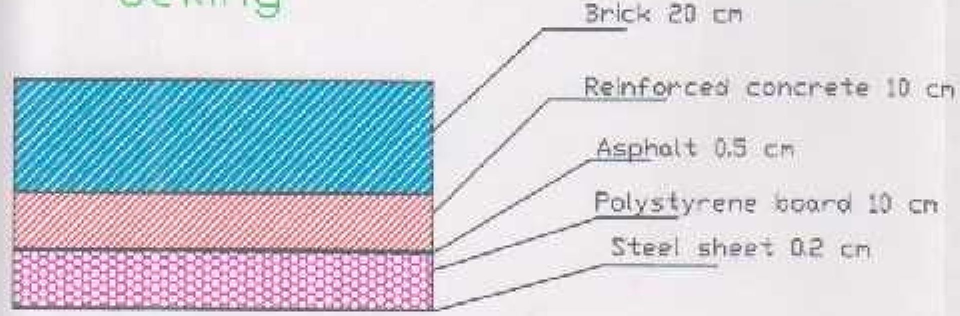

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Wall construction

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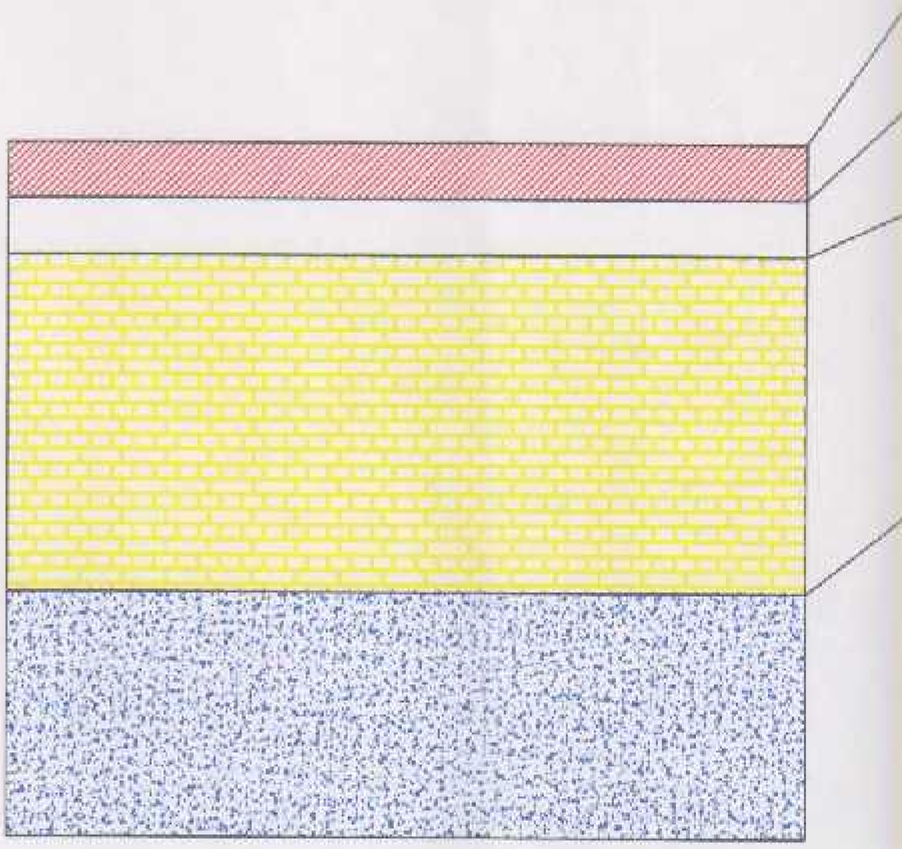
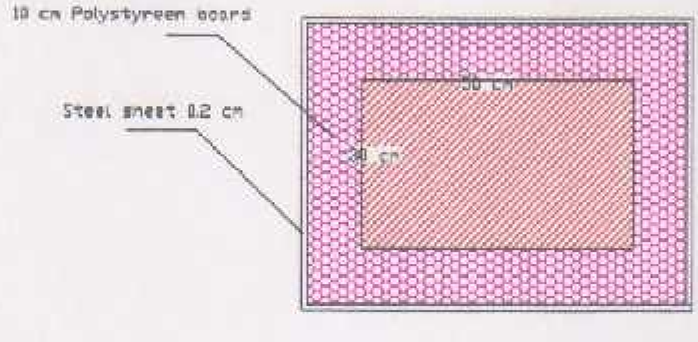
Floor constrction



Columns



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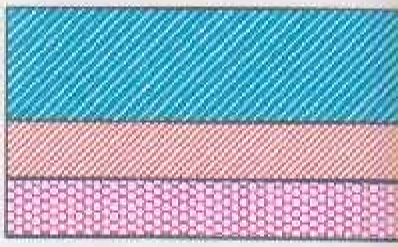
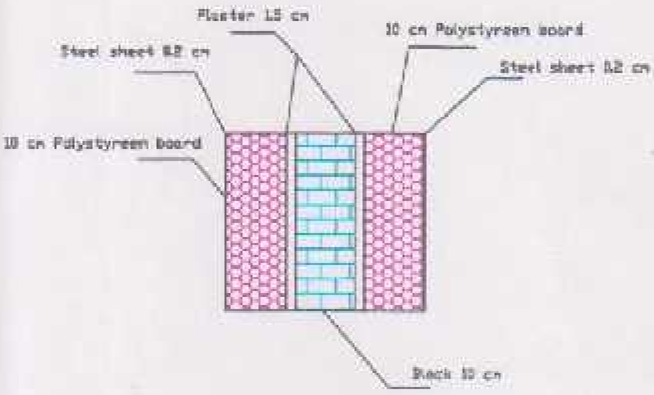
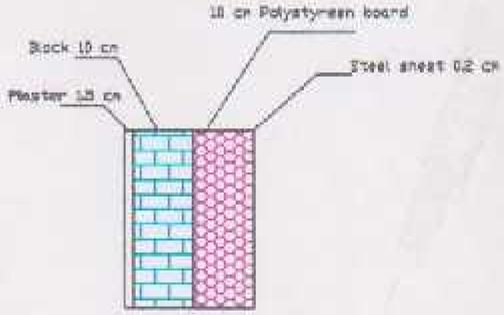


Walls between rooms



Wall between room and Caridors



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Project

Refrigeration Station -Halhul

Sheet Title

Wall construction

Senior Project

Date

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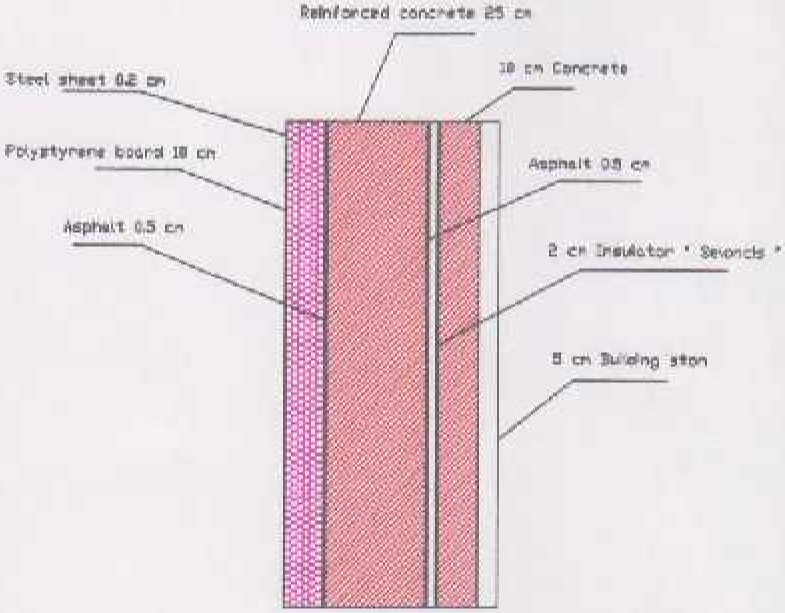
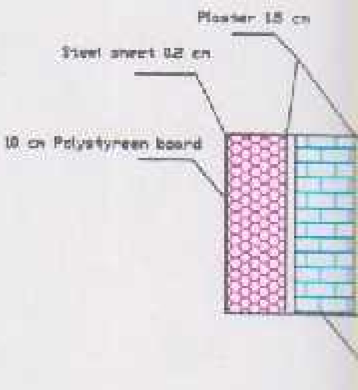
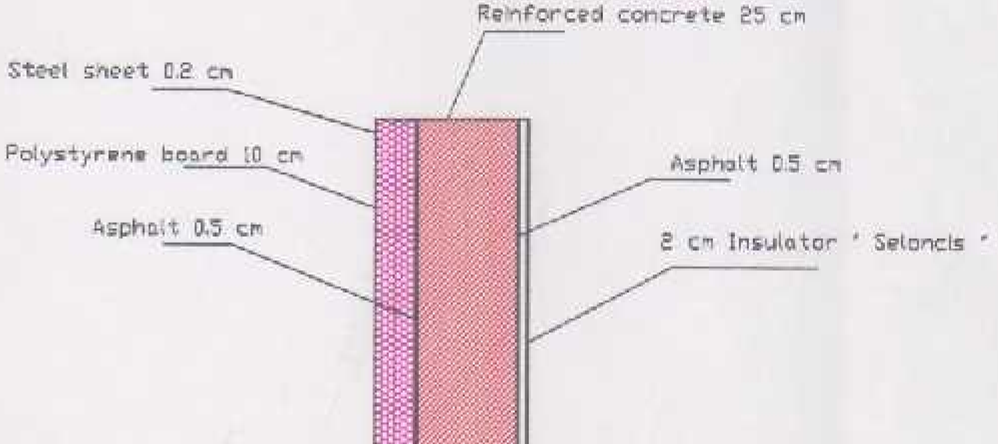
Heigher part of outside walls



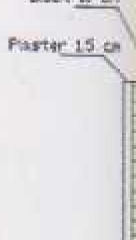
Walls between



Lower part of out side walls



Wall be



F) General Civil Plan Declaration:

As shown in the civil top plan, the refrigeration station has two entrances; from the south and the other from the north, with two elevators beside each entrance for the product lifting. And the doors width is 1.55 m and constructed of galvanized steel.

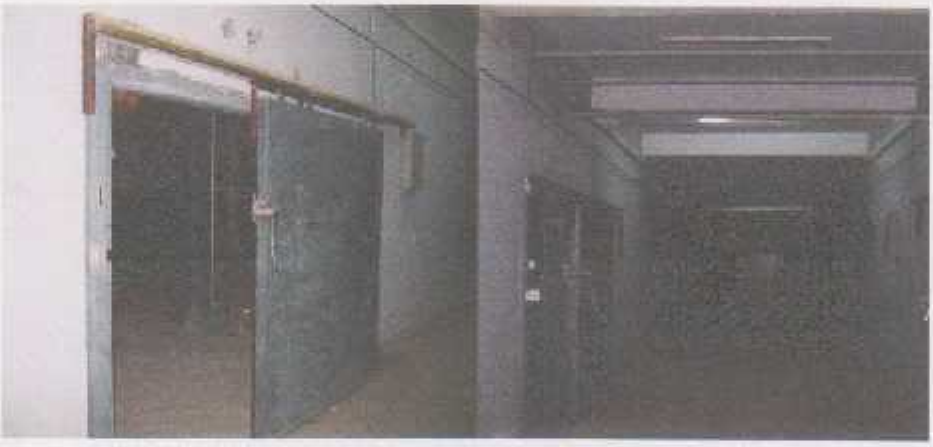


Figure (3-1).Photos Taken inside the Refrigeration Station

The refrigeration station is consisted of 10 refrigeration rooms, designed for variable cooling loads differs according to the type of fruit or vegetable that will be saved in. and one for the freezing low temperatures. Each room is 58.5 m^2, 8.3 m length and 5.85 m width, and with a 4 m height. The door of each room is 10 cm thickness, 2.2X1.9 m, composed of two reinforced galvanized steel sheets with internal polyurethane isolation foam. The corridor length is 45 m and width of 3 m. (See Figures 3-1, 3-2)

Figure (3-2) Photos Taken inside a Refrigeration Room

3.3 The Mechanical Design of the Station

3.3-1 The Refrigeration System

The refrigeration station is composed of the familiar refrigerating system. Where as each room is refrigerated by single unit. This unit components and accessories are declared in Figure 3-3 below:

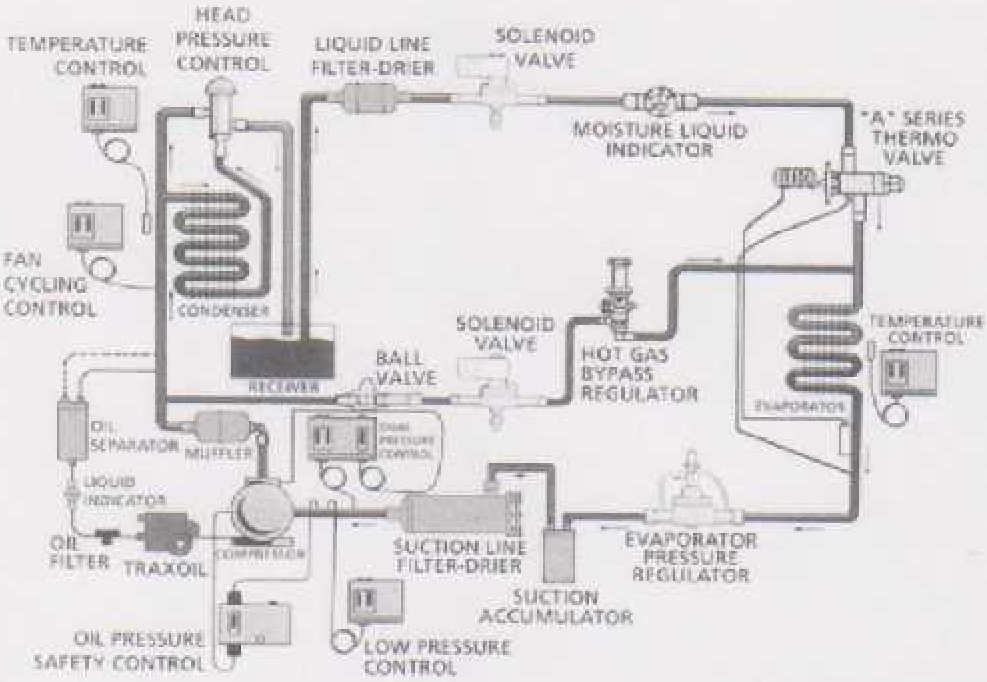


Figure (3-3) Total Refrigeration Circuit with Its Accessories

A) Compressor

There are two types of compressors used in the refrigeration system of the station. The first type of compressors used in the refrigeration units is reciprocating type compressors with the following name plate:

DWM COPELAND
TYP DMRH – 750 – EWL – 68XL
Falor. Nr 457434 Bajuahr 1988 R
Zul Betr. – Druck HD/ND 20.5-25 Bar

3~ n =1455 rpm $\dot{V}$ = 26.8 m^3/hr.

Schaltk IP64. CosΦ = 0.87

Hz	Blackierter Rotor Storm	A
50.... 220/240 Δ	123/134	31.3 A
50.... 380/420 Y	71/78	18.1 A

Figure (3-4) Compressor Used in the Station

And so the output power of this compressor is:

$W = I * V * \cos\Phi * \eta = 31.1 * 220 * 0.87 * 0.90 = 5.47$ KW

And the second type is:

DMRH – 750 –EWL – 000

With same characteristics mentioned above. Except their power is less because of the less efficiency and size. The Figure 3-4 shows the compressor used in the station.

B) Evaporator

Figure (3-5). Indoor Evaporator

The evaporator unit in the station is composed of a fan-coil unit. Two fans of mechanical power of 0.37 KW, 4500 rpm. And coil units of dimensions 1.30X65 cm and 45 cm depth. And the height of the evaporator from the ground is 2.8 m. and from the entrance of the room 50 cm. The evaporator output power is not available. The evaporator is shown in Figure (3-5).

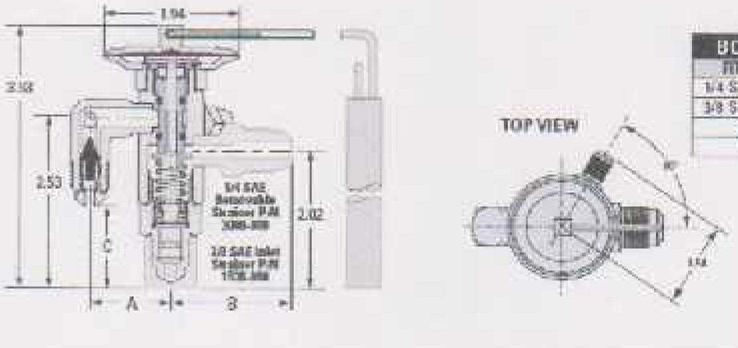
C) Condenser

The condensed unit that used in the station is composed of also fan-coil unit. With two fans of mechanical power of 0.37 KW, the condenser dimensions are 110X75 cm and 20 cm depth. And output power of the condenser is not available. (See Figure 3-6)

Figure (3-6). Outdoor Condenser

D) Expansion Valve

Ordinary thermostatic expansion valve with sensing bulb installed at the evaporator inlet. (See Figure 3-7).



BOL DIMENSIONS – Inches			
FITTING SIZE	A	B	C
1/4 SAE, 90° Angle	1.19	-	1.44
3/8 SAE, 90° Angle	1.36	-	1.09
5/8 SAE	-	1.81	-
1/2 SAE	-	1.90	-

Figure (3-7) Thermostatic Expansion Valve

E) The Accessories of the Refrigeration System

The Accessories of the refrigeration system is as follows:

E-1) Filter Dryer

Ceramic Filter core Model is CD – 164 – B

Design Pressure is 500 Psi. R12, R22. (Figure 3-8).

E-2) Solenoid Valve

The solenoid valve is sighted as shown in Figure (3-8). The model is:

SC EM
Coil ZB09
220V/~50 Hz, 9 W ±10%.

Figure (3-8) Filter Dryer and Solenoid Valve

E-3) Pressure and Temperature Controls for Compressor

The pressure control is adjusted on 21 bar discharge pressure, and 65 C cut off temperature limit. Figure (3-9).

Figure (3-9) Pressure and Temperature Controls and Refrigerant Accumulator

E-4) Refrigerant Accumulators

Refrigerant accumulator max capacity is 0.4 L. horizontal types, For providing pure gas suction. (See Figure 3-9).

E-5) Oil Separators

Oil separator capacity is 3 L. with feedback line to the bottom of the compressor. (See Figure 3-10)

Figure (3-10) Oil Separator

E-6) Thermostats

The thermostat is located up the door of the refrigerating room. And this is wrong since it should be placed at the inlet of the evaporator's fans to measure the room temperature precisely. (See Figure 3-11).

Figure (3-11) Thermostats

F) Refrigerant Piping:

All refrigerant piping is composed of copper. With the following diameters:

- The suction line of the compressor is 1.25 inch.
- The discharge line of the compressor is 1 inch.
- Condenser discharge line 0.5 inch.
- Pipe From receiver tank to filter drier 0.5 inch.

All refrigerant piping is not insulated absolutely. This causes heat leakage from the cycle.

GENERAL NOTES

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Refrigeration Station -Halhul

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Evaporator

3.3-2 The Humidifying system

The station is provided with humidifying system used to maintain the desired relative humidity for the storage circumstances. The system is composed of:

- The steam generator.
- The steam piping network.
- The solenoid valves used for steam injection control.

Now we will declare the characteristics of each component:

A) Steam Generator

The steam is generated using 3~ electrical heaters. The heating capacity is not available. Rated power is 10 kW, cylindrical shape. With total insider volume of 250 liter, and provided with level sensor that prevents steam generation if the reservoir is not full. And the steam generator is provided with control box and 3 fuses to control its working. (See Figure 3-12).

Figure (3-12) Steam Generator

B) Steam Piping

The piping of the steam is constructed of copper pipes. Diameter is 1/4". And the branches to the rooms are of 10 mm diameter. And the steam injection is coming from the up corner of the room. And a humidistat is put at the upper part of the room. Linked to a solenoid of the humidifying tunnel to the room.

C) Solenoid Valves and Pressure Relief Valve

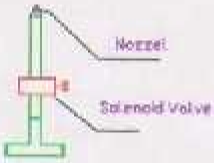
These solenoids are used to control the steam flow to the room, Affected by the humidistat in the refrigerated room space. (See Figure 3-13)

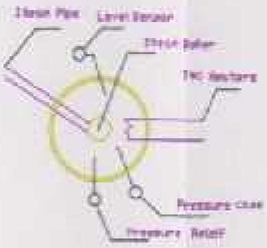
Figure (3-13) Solenoid and Pressure Relief Valves

D) Water Nozzle Sprayers

There are double sprays installed beyond the fans of the evaporator to humidify the rooms. But all were clogged and offline. (See Figure 3-14)

Figure (3-14) Steam Injection Tunnel





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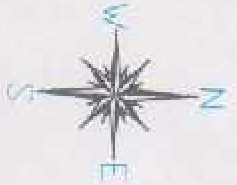

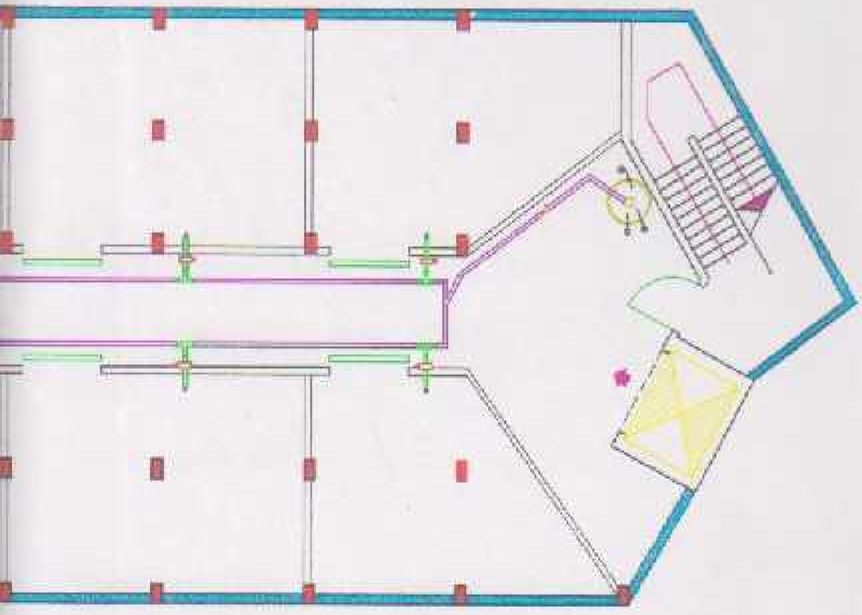

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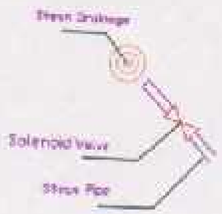
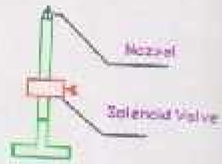
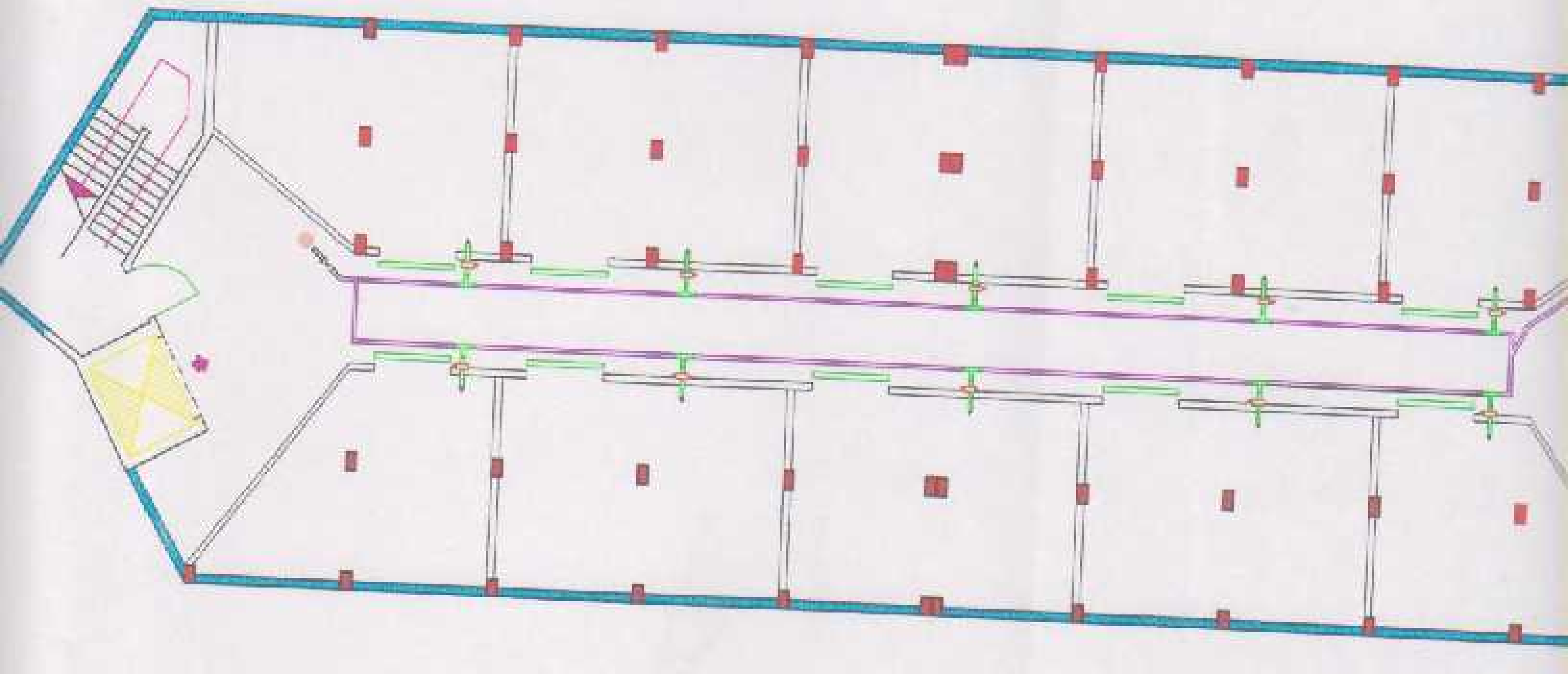
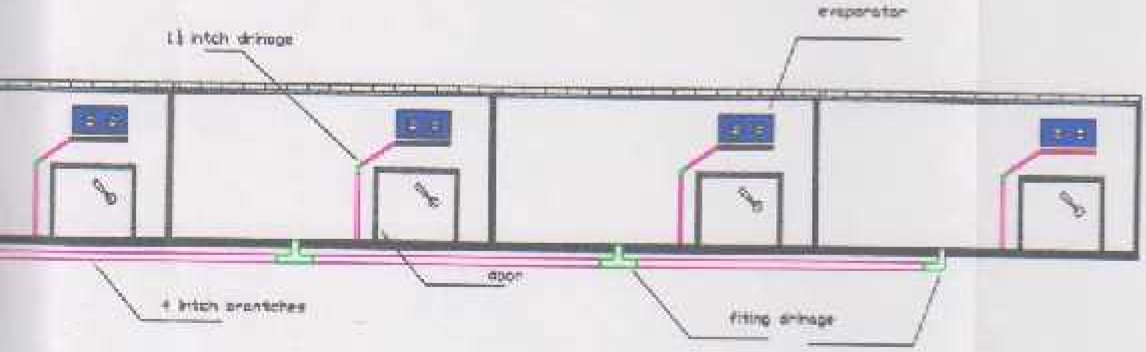

Solenoid Valve


Nozzel
Solenoid Valve





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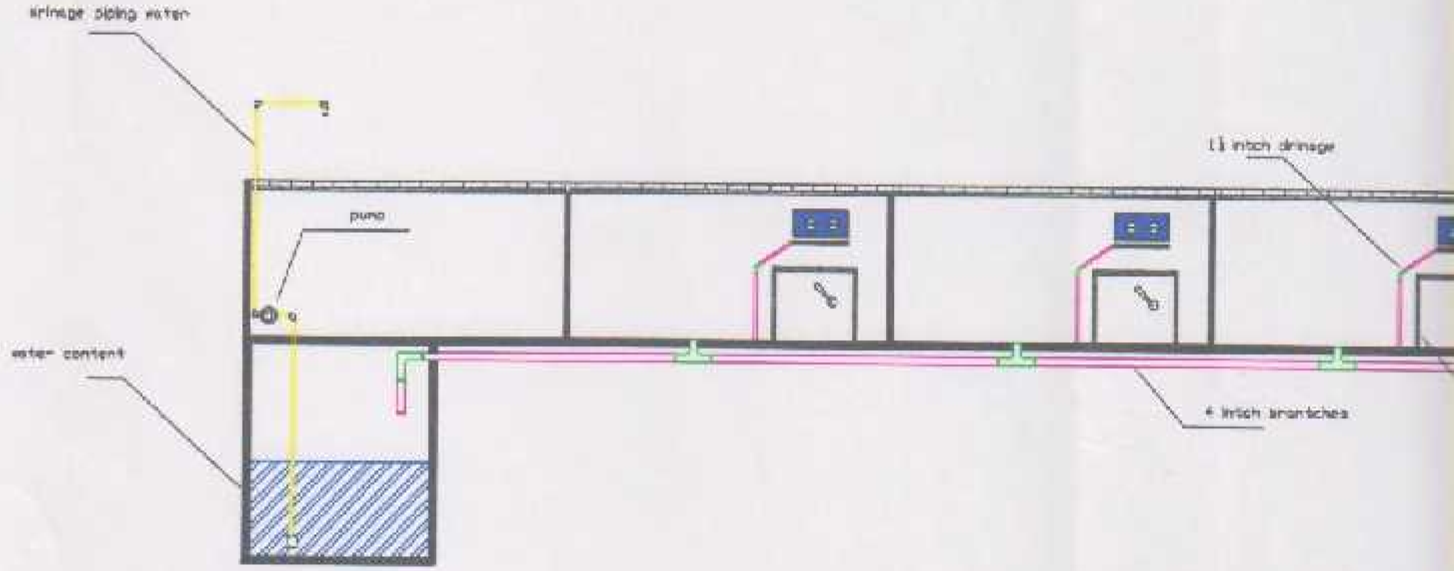


drinage piping water
pump
water content
1½ inch drinage
4 inch branches

3.3-3 The General Mechanical Plan Declaration

The mechanical refrigeration system is composed of 11 main refrigerating units, each unit refrigerate two split rooms. These units are put outside the building in the shadow. No ventilation system is installed for the station. And drainage piping is installed for the water produced from defrosting. (See Figure 3-15)

Figure (3-15) Outside View of Outdoor Refrigeration Units

3.4 Electrical System Design Station

The last layer of the station design is the electrical system design of the refrigerating station. In this part we will declare the electrical wiring design for the station.

General wiring network of system:

The wiring system is based on 3 ~ph wiring, because many components operates on 3~ power sources such as steam generator heaters, compressors…

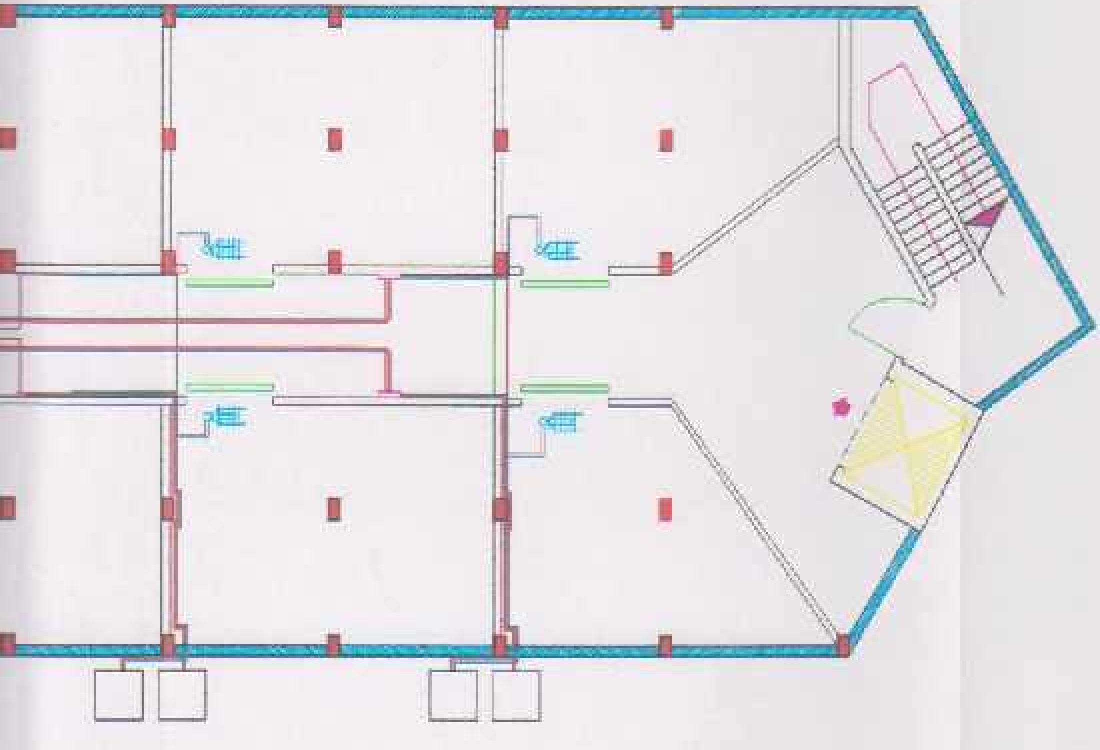
The net starts from the main network for Halhul's power network. Then the lines are connected to 12 group of fuses; 3 lines for each application. 10 groups is for the refrigeration unit compressors, 2 for the elevators. And the heaters fuses are placed in their own control box. Then from these fuses to the electrical power consumption counters. Then to the system circuit breakers, and then from them to the control boxes of the refrigerating units and elevator. (See Figure 3-16)

Figure (3-16) Electrical Cabinet of Station

These control boxes contains many electronic and electrical components such as timers, latches…., to maintain some control for these units.(See Figure 3-17)

Figure (3-17) Control Cabinet for each Room

Each refrigeration rooms are lighted by 4 bulb lights. With rated power of 100 W. and the corridor is lighted by 12 florescent tubes. Rated power is 40 W.



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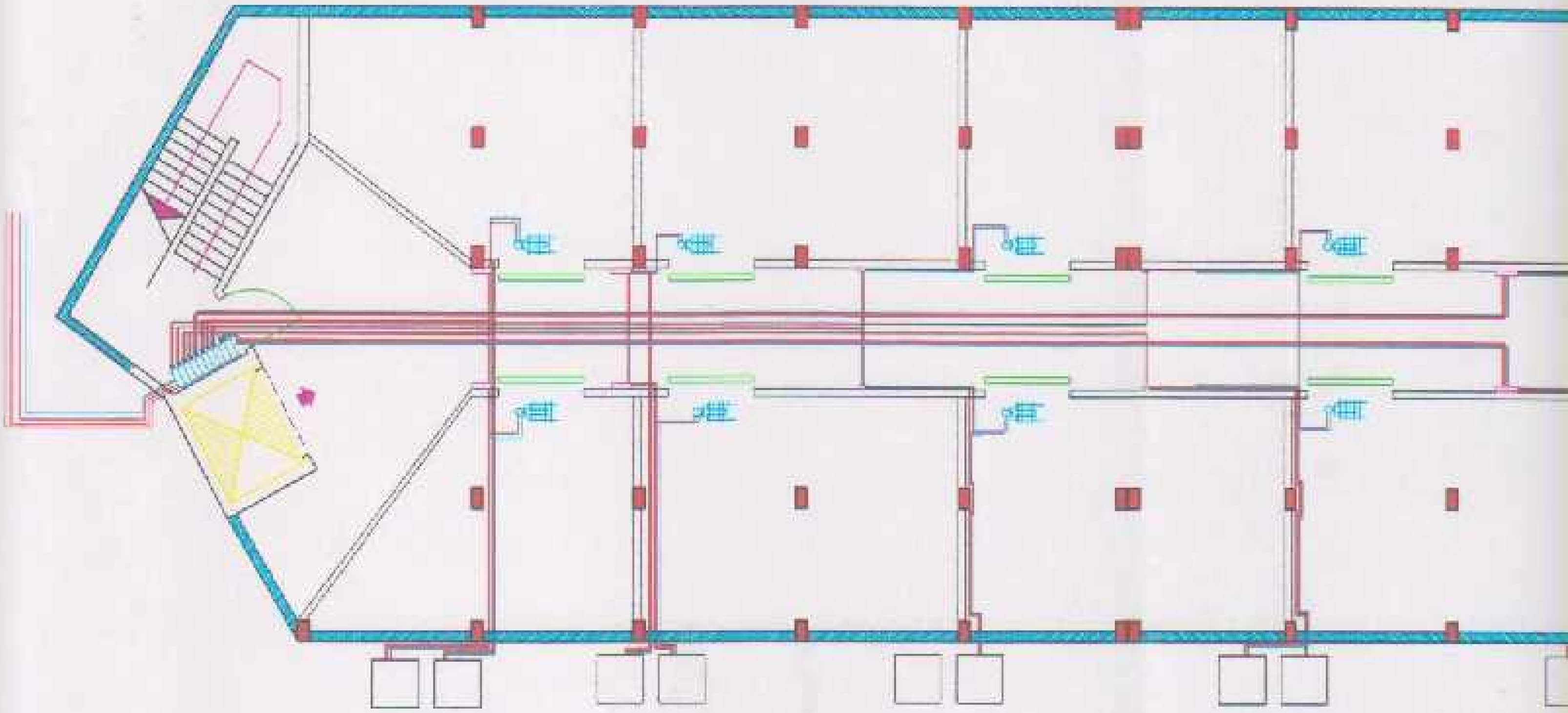


Chapter Four

GENERAL DESIGN PRAMETERS

4.1 Introduction

In the previous chapter, we declared all the designs of the refrigeration room; the civil construction design, mechanical refrigeration design and steam injection system, and the electrical wiring design. All operation characteristics have been analyzed part by part.

In this chapter we will determine the general design parameters necessary for the design process, these parameters are divided into two categories: the general outside condition, Such as the wind velocity, the average relative humidity, the outside temperature. And the ground's mean temperature.

And the other category is the storage conditions in each refrigeration room. This depends on the product type that will be stored and refrigerated, i.e. every product has its specific storage conditions. Such as the storing temperature, relative humidity, chilling time...

4.2 The General Outside Design Parameters

Halhul's Refrigeration station is located in Hebron, at longitude 35.1 degree east, latitude of 31.5 degree north. And the elevation is 1010 m from the sea level. (See Figure 4.1)

المعلومات الجغرافية للمحطة	
اسم المحطة	الخليل
خط الطول	35 شرقا
خط العرض	31 شمالا
الارتفاع عن سطح البحر	1005 متر

جداول ماهوتي لمحطة الخليل

Figure (4.1) Hebron Position

4.2-1 Temperature

The first outside design parameter is the temperature. For Halhul, its lactation is in the 3rd climate area of the West Bank. And the temperature records for the region at which the refrigeration station located is in Figure (4-2) below.

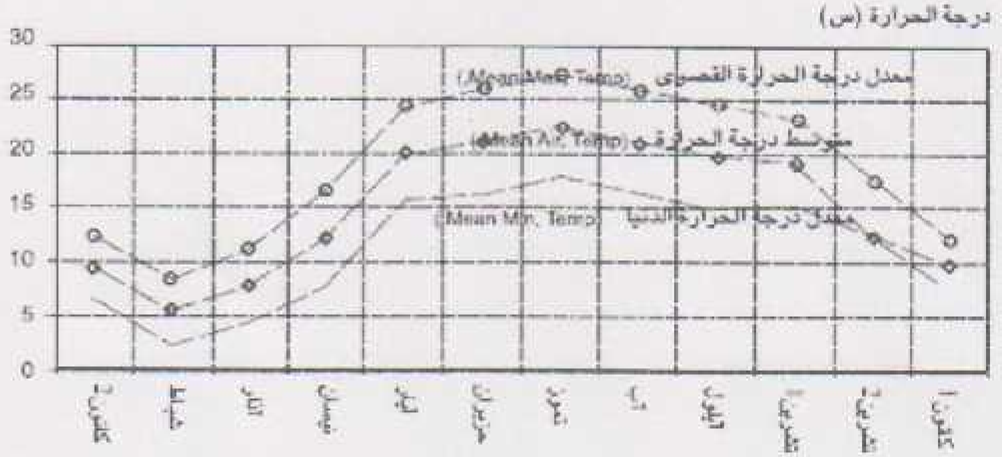


Figure (4-2) Temperature Records for Hebron

As shown in this record. The maximum temperature is about 28 C. and the minimum temperature 3 C. the maximum temperature that is gained last for about 4 months. By taking in consideration the high temperature waves that happens in this region which may reach 33 C, we will consider 35 C as an outside design temperature.

4.2-2 Relative Humidity

Second parameter is the outside relative humidity. The relative humidity records in Hebron are shown in Figure (4-3):

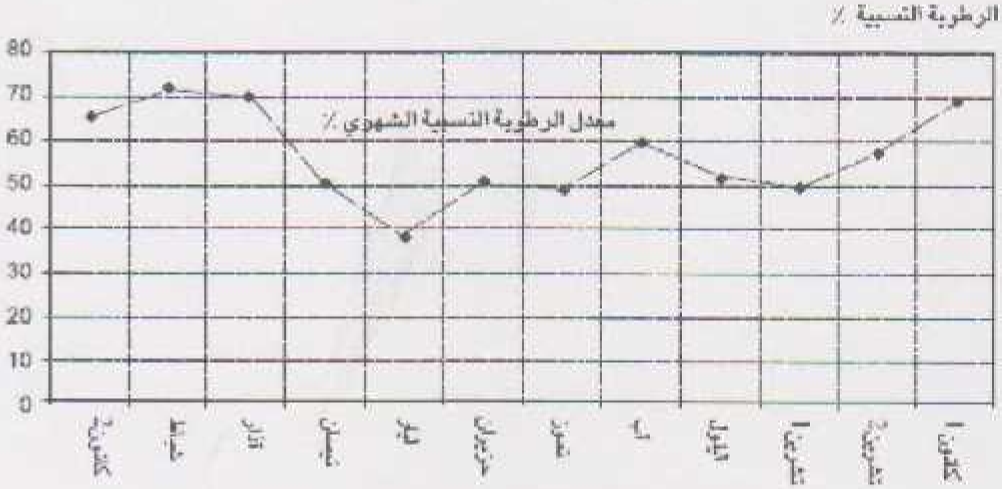


Figure (4-3) Humidity Records for Hebron

The maximum relative humidity is in February, and the minimum is about May. The relative humidity affects on two components; the ventilation system, and the condensation unit. So the design relative humidity is 50%.

For the ventilation system, the outside relative humidity, when the relative humidity of the entering air increases, this will cause an increase in the relative humidity in the storage room, which will affect on the product and its quality. In addition, the heat rejection from the ventilation air increases, this causes an increase in the cooling load. This occurs in February, This has the maximum relative humidity.

And for the condenser, the most critical relative humidity is the minimum relative humidity. Which occurs in May, i.e. specific volume increase, and so the enthalpy will increase, because the heat capacity of an air stream at these circumstances will have less heat capacity, Recording to the psychometric chart of air at the atmospheric pressure at that height.

4.2-3 Wind Velocity

The third outside parameter is the outside wind velocity. That's as shown in the figure below, Shows the wind velocity all around the year for Hebron area. (See Figure (4-5))

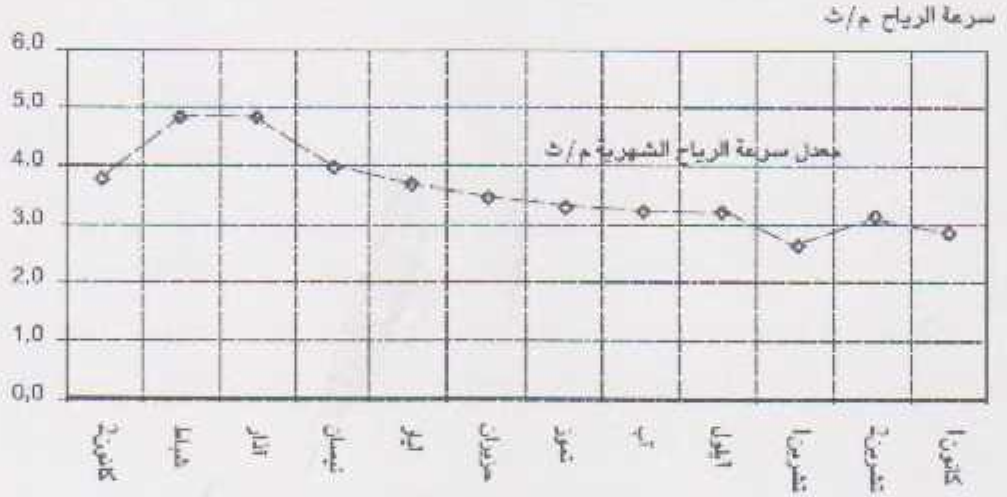


Figure (4-4) Wind Velocities for Hebron

The wind velocity ranges from 2.5 m/s to 4.8 m/sec. the wind velocity effects by two ways, increasing the value of the outsider convection heat transfer coefficient, which in turn increases the heat gain to the refrigeration room.

The design wind velocity will be taken at the mid-summer. That means to have wind velocity of 3.3 m/s. and for the safety of design we will consider 4 m/s as a design wind velocity.

4.3 The General Inside Design Parameters

The inside design parameters are set depending on the type of product that will be stored in the refrigeration room. Since each type of stored product require specific storage conditions.

The products that are stored in the refrigeration room are:

1- Lemons.
2- Oranges.
3- Potatoes.
4- Carrot.
5- Grapes.
6- Apples.
7- Apricot.
8- Cucumber.
9- Avocadoes.

The storage characteristics includes the storage temperature, relative humidity, chilling time, delay time, type of storage, maximum storage period. All declared in Table (4-1).

Product	Type of Storage	Storage Temperature "C"	Storage Period	Chilling Time "H"	Relative Humidity	Delay Time "H"	Cp "kJ/kg. K"	Water Content "%"	Air Velocity "m/s"	Respiration Rate "W/kg"
Lemons	Short Period Long Period Chill Start Chill Finish	7 10 20 7	2 month 3 month	20	85% To 95%	34	3.81	89%	1.25	0.040
Oranges	Short Period Long Period Chill Start Chill Finish	8 10 28 8	8 weeks 12 weeks	35	85% To 90%	18	3.81	81%	1.25	0.816
Potatoes	Short Period Long Period Chill Start Chill Finish	4 4 38 4	5 month 10 month	14	95%	24	3.6	78.5%	0.75	0.027
Carrot	Short Period Long Period Chill Start Chill Finish	0 0 32 0	7 month 9 month	24	98% To 100%	8	3.6	88%	0.3	0.047
Grapes	Short Period Long Period Chill Start Chill Finish	-2 -0.5 26 0	2 month 6 month	28	90% To 95%	6-8	3.56	75%	3.60	0.030
Apple	Short Period Long Period Chill Start Chill Finish	2 4 28 2	2 month 7 month	24	90% To 95%	24	3.72	84%	0.65	0.012
Apricot	Short Period Long Period Chill Start Chill Finish	-0.5 0 30 -0.5	1 week 2 week	16	90% To 95%	4	3.68	85%	0.45	0.150
Cucumber	Short Period Long Period Chill Start Chill Finish	10 13 30 10	10 days 14 days	16	90% To 95%	6	3.89	95.5%	0.45	0.070
Avocados	Short Period Long Period Chill Start Chill Finish	4 4 28 4	4 week 8 week	20	85% To 90%	6	3.81	94%	0.46	0.080

Table (4-1) Indoor Storage Requirements for Various Products

Chapter Five

COOLING LOAD CALCULATION

5.1 Introduction

A detailed study and analysis of the existing refrigeration station in Halhul's center market was performed previously. Also, all the required and necessary data and parameters regarding the design process of the refrigeration station was searched for and gathered for the redesign of the station.

As noticed, the existing station has no chilling room to handle the products before sending them to the storing rooms. In the redesign of this station, we will assign one of the rooms as a chilling room. For convenience, the room that is assigned to be designed as a chilling room is the room with star sign as shown on the station plan. And the room which was designed for cooling poultry will be redesigned to be used as a storing room for products.

The structural design parameters for walls, ceiling, floors, insulation thicknesses...etc, available form the civil project plans provided by Halhul's Municipality will be the base for our calculations.

The redesign of the station will proceed in two stages, the first stage will be the redesign of the chilling room, and the second stage will be the redesign of the storage rooms.

5.2 Product Distribution

The arrangement of products in the refrigeration rooms is very important to achieve uniform air distribution throughout the room and make it possible to reach all the stored products. So taking into account the product storing rules and recommendations, the products should be stored as specified in the product arrangement plan.

The recommended rules for the distribution of the products can be specified as follows:

- Put all packages of all kinds of products onto wood or plastic or metal cages allow the passage of air provided to rise above the surface of the earth by (10 cm) at least.

- Packages should be away from the walls by (15 cm) and from the ceiling a distance of 40-50 cm.

- Leaving corridors inside cooling rooms to allow inspection and sampling, that at least the main corridor width is 60-70 cm, leaving a distance of not less than 10 cm between Packages to facilitate the movement of air.

- Never store under refrigeration evaporator.

- Never stop operation coolers for any reason as long as rooms containing vegetables or fresh fruit.

- Non-perishable crops stored near the main door.

And so, as in the project plans; the distribution of the product obeys the product saving recommendations.

The rest of packaging information is:

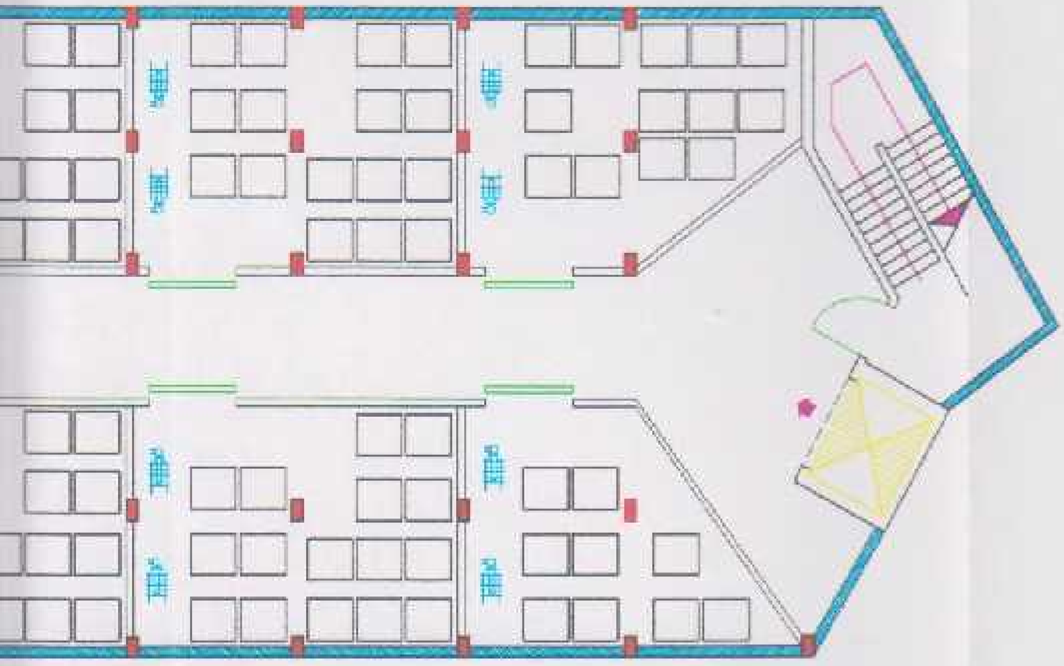
→ Box size is 50 X 35 X 18 cm, and each pack contains 9 layers each containing 6 boxes cartoon of product. Each box weight depends on its content. And so the dimension of the package is 1 X 1.15 X 1.72 m.

→ Steel stands are required to carry the couple of packages vertically, and these steel stands weigh 70 kg for each one.

→ The wooden boards used for carrying the product is waiting about 15 kg.

→ The cartoon box used weighs about 200 g.

→ The weights of the product boxes are as shown in Table (5-1).



GENERAL NOTES

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Refrigeration Station -Halhul

Product Ditribution Plane

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Date
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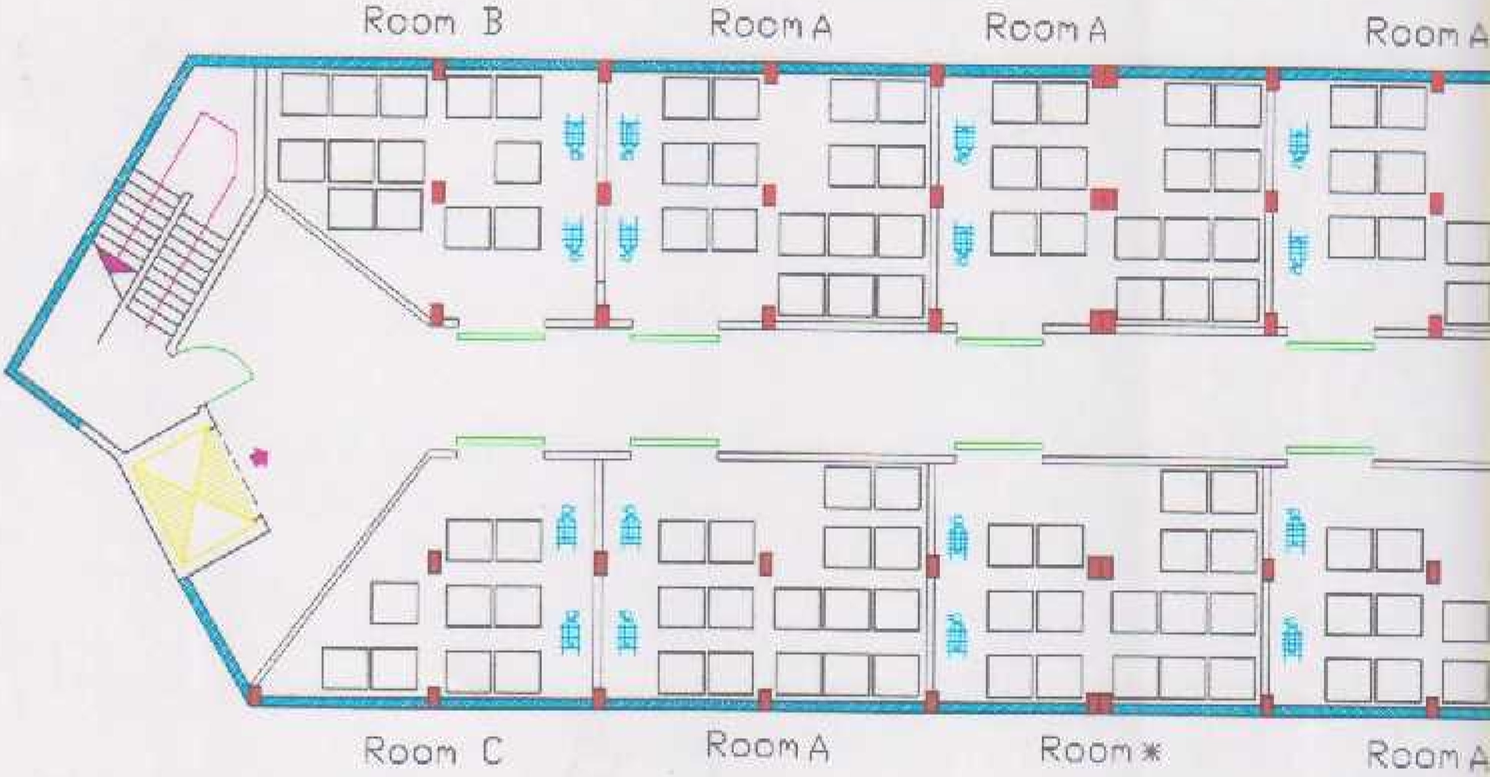

Room B
Room A
Room A
Room A
Room C
Room A
Room *
Room A

GENERAL NOTES

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Product Distribution

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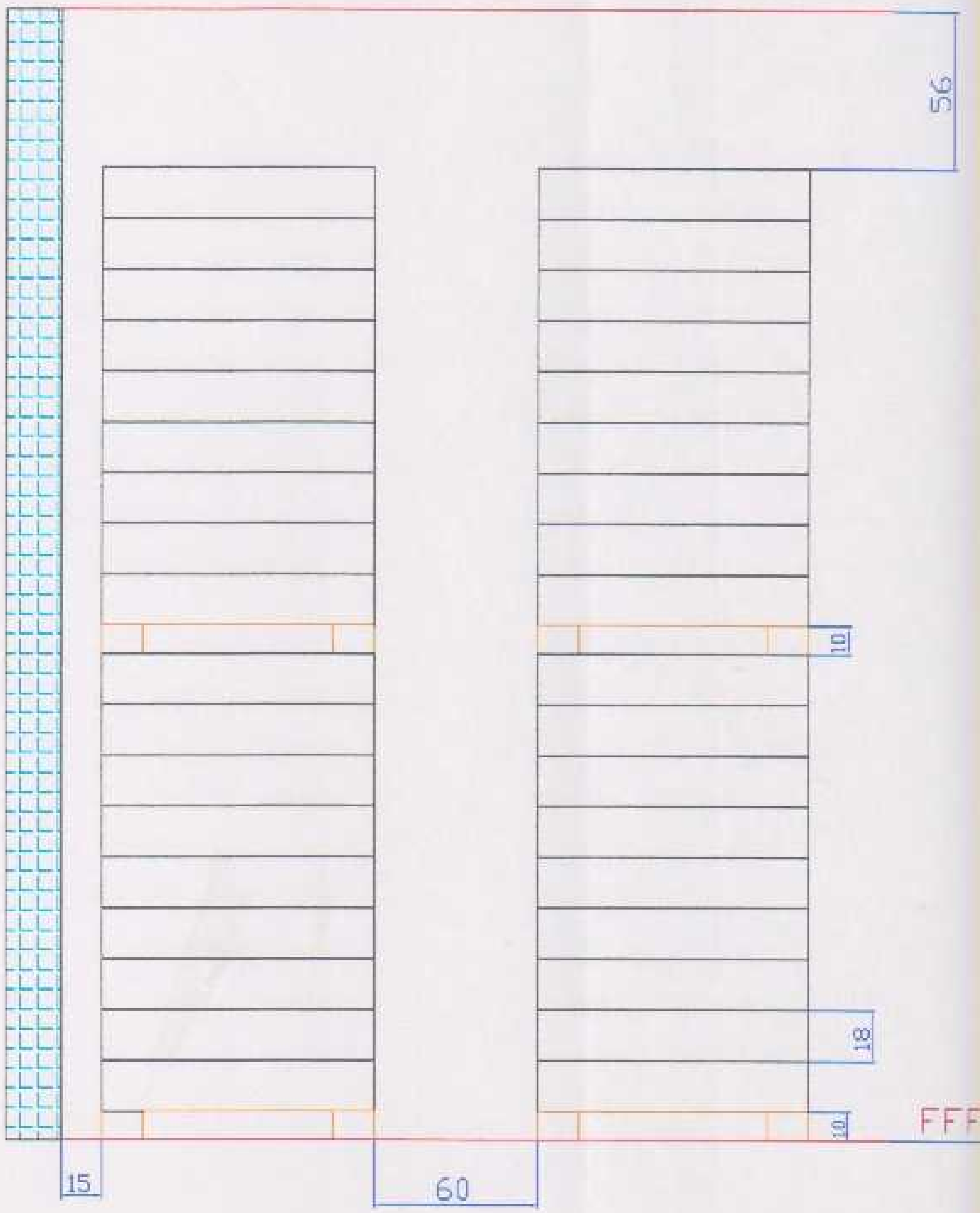


56
10
18
10
FFF
15
60

Product	Weight of Box
Lemon	15
Orange	15
Potatoes	20
Carrot	10
Grapes	10
Apple	12
Apricot	10
Cucumber	15
Avocadoes	10

Table (5-1) Product Box Average Weight
These values were obtained empirically

According to this information, (refer to the product distribution plan):

- The rooms assigned "A" can handle 32 package of product.
- The rooms assigned "B" can handle 26 package of product.
- The rooms assigned "C" can handle 18 package of product.

5.3 Cooling Load Calculations

In the cooling load calculations for the station rooms, we will consider the critical load (the maximum cooling loads) that may occur in the rooms of the station. Those rooms are divided into two categories; the chilling unit that will be designed at room with star sign (refer to the station plan), and the rest of the rooms as storing rooms of the refrigeration station. As the structural design parameters for all rooms are mostly identical, the predominant factor for the maximum cooling load comes from the type of the product to be stored, i.e. the cooling load will be calculated with respect to the product that needs much cooling for storage.

5.3-1 Cooling Load Components

The total cooling load for refrigeration rooms will consist of the following components:

- ☒ Heat gain through room surfaces.

- ☒ Product load.
- ☒ Respiration load.
- ☒ Air change and humidification load.
- ☒ Miscellaneous load, which consists of:

  - ➔ The lighting load.
  - ➔ Electric motor.
  - ➔ The people occupied the room.
  - ➔ Defrost Heat source.
  - ➔ The Packaging heat removal.

5.3-2 Cooling Load Demand

A) Storage Rooms

The cooling units for the storage rooms should be designed to handle the maximum cooling load demand. As all rooms are identical in size and construction, the product type; more precisely the respiration load, will play the main rule in the calculation of the cooling load demand for the storing room.

Table (5-2) shows the respiration rate for the various products, the storage mass, and the total heat generated by respiration. It's obvious that apricot has the highest respiration load.

Product	Average Storage "kg"	Respiration Rate "W/kg"	Heat Generation "W"
Lemons	25920	0.040	1037
Oranges	25920	0.016	415
Potatoes	34560	0.027	933
Carrot	17280	0.047	812
Grapes	17280	0.050	864
Apple	20736	0.012	250
Apricot	17280	0.150	2592
Cucumber	25920	0.070	1815
Avocados	17280	0.080	1383

Table (5-2) Respiration Load for Various Types
Of Products in Full Loading of Room

From this Table, we conclude that apricot is the critical product that will be chosen for the calculation of the critical cooling load; because it causes the maximum refrigeration storing demand under the same circumstances.

A-1) Wall gains:

This load is caused by the heat gain through the walls, ceiling, and floor when this wall at both sides has a different temperature.

Wall gain load sometimes called the wall gain load, which is a measure of heat flow rate by conduction through the walls of the refrigerated space from the outside to the inside.

The parameters that determine the gain heat are:

- The materials which the wall consists of.
- The temperature difference between two sides of the walls

The heat gain through walls equation is:

$$Q = U * A * (\Delta T) \tag{5-1}$$

Where:

Q: the rate of heat gain. [W]

A: the surface area which gain the heat. [m^2]

ΔT: the temperature difference across the walls. [C]

U: the overall heat transfer coefficient. [$W/m^2.C$]

Where U is identified in the following equation:

$$U = \frac{1}{\frac{1}{h_{in}} + \sum_{i=1}^{n} \frac{\Delta X_i}{K_i} + \frac{1}{h_{out}}} \tag{5-2}$$

Where:

→ h_{out}: Convection coefficient (surface Conductance) of inside wall, floor, or ceiling.

→ h_{in} : Convection coefficient (surface conductance) of outside wall, floor, or roof.

→ ΔX : Thickness of constructing material.

→ K : Thermal conductivity.

Design Data:

➔Temperature:

Temperature	Value "C"
T_{inside}	0
$T_{outside}$	35
T_{ground}	43
$T_{corridor}$	18

Table (5-3) Design Temperatures

➔Overall Heat Transfer Coefficient (OHTC): the overall heat transfer coefficient U was calculated by equation (5-2) for each wall and listed in Table (5-4) below:

OHTC	Value "$W/m^2.K$"
$U_{ceiling}$	0.2770
U_{floor}	0.4600
$U_{wall\ upper\ part}$	0.3022
$U_{wall\ lower\ part}$	0.3080
$U_{internal\ walls\ btn\ rooms}$	0.1730
$U_{btn\ room\ and\ corridor}$	0.3250
U_{door}	0.3200

Table (5-4) OHTC Values

➔Humidity:

Relative Humidity	Value "%"
$\Phi_{outside}$	50
Φ_{inside}	95

Table (5-5) Relative Humidity

➔Wind Velocity:

Wind velocity	Value "m/s"
V_{wind}	4

Table (5-6) Wind Velocity

Assuming nothing stored next to room from both sides. Firstly, we will take the rooms Assigned "A", the result of all wall-gain loads.

$$\mathbf{Q}_{walls} = \mathbf{Q}_{outsider\ wall} + \mathbf{Q}_{corridor} + \mathbf{Q}_{floor} + \mathbf{Q}_{ceiling} + 2*\mathbf{Q}_{adjacent\ room} + \mathbf{Q}_{door} \quad (5\text{-}3)$$

$$\rightarrow Q_{outsider\ wall} = Q_{upper} + Q_{lower} \quad (5\text{-}4)$$

$$= (U_{upper}*A_{upper}*\Delta T) + (U_{lower}*A_{lower}*\Delta T)$$

$$= (0.3022*10.8*35) + (0.308*22.4*43)$$

$$= 411\ W.$$

And by the same way we can obtain the wall gain for all sides. And so the wall gains are:

$\rightarrow Q_{corridor} = 177\ W$

$\rightarrow Q_{door} = 24\ W$

$\rightarrow Q_{Floor} = 961\ W$

$\rightarrow Q_{ceiling} = 471\ W$

$\rightarrow Q_{btn\ rooms} = 146\ W$

And So, The total heat gain (Q_{gain}) for the room will be 2190 W.

$Q_{gain} = 2190\ W.$

A-2) Product Load

The product enters the storing room at its storing temperature. And so there is no field heat to remove from the product; which means a product load goes to zero.

$Q_{product} = 0\ W$

A-3) Respiration Load

When a product experiences a respiration activity (metabolic activity); this produces heat as an output of this process. This heat forms a cooling load in the refrigeration room.

Respiration load is characterized by the respiration rate of the product, and calculated by this equation:

$$Q_{Respiration} = \text{Mass of Product} * \text{Respiration Rate} \qquad (5\text{-}5)$$

Where:

$Q_{Respiration}$: Respiration load [W].

Respiration Rate: factor [W/Kg]. (Refer to Table 4-2)

And for the apricot:

$Q_{Respiration} = 17280 * 0.15 = 2592$ W

A-4) Air Change and Humidification Load:

Most fruit and vegetables produce CO_2 and Aldehydes and other alcohols. These Organic gases cause the fast repining of the products and it distinct with its high density and molar mass. And so the room air needs to be refreshed either by the natural opening of doors or by forced ventilation system for the refrigeration room.

For the storage rooms in the station, each room needs to be ventilated since the room may subjected to the long term storage for long times without door opening. And it is depending on the volume of the refrigeration room. On the other hand the chilling rooms may have a little ventilation because it is opened after the end of the chilling process if it will not continue storing, In addition to the humidification process needed for maintaining the recommended storage humidity.

A-4.1) Ventilation Air Change:

The ventilation rates required for the refrigeration room applications are shown below in Table (5-7) below:

Refrigerated Space * "m^3"	No of Air Change per 24 h "n"
10	33
20	22
40	15
60	12
80	10
100	9
150	7
200	5.8
300	4.8
400	4.2
600	3.4
800	2.9
1000	2.5
1500	1.9
2000	1.7
3000	1.4
4000	1.2

Table (5-7) Required No. of Air Change for Room Ventilation

* Refrigerated space stands for the volume of the space that surrounds the product in the refrigeration room
→ *ASHREA REFRIGERATION HANDBOOK 2005*

And the ventilation air change load is calculated by the following equation:

$$Q = n*V_s*(h_o-h_i)*\rho*(1000/24*3600) \qquad (5\text{-}6)$$

Where:

- n = number of air change for refrigeration room.
- V_s = volume of air surrounding the refrigerated product. [m^3]
- $h_{i,o}$ = the outside and inside enthalpies. [KJ/kg]
- ρ = the outside air density. [Kg/m3]

And so, the refrigeration rooms Assigned "A" the desired air room change At 100 m^3 = 9.

Now by applying the equation above:

$Q_{Ventilation}$ = 833 W

A-4.2) Humidification

The humidification rate needed can be calculated with respect to the ventilation air change in the refrigeration room. Such that the amount of water removed by air change must be compensated by the humidification system, by the mass balance:

$$m^{o}_{steam} = m^{o}_{air} * (\omega_{out} - \omega_{in}) \tag{5-7}$$

Then, $m^{o}_{steam} = 0.00018$ kg/sec

And so the humidification load = $m^{o}_{steam} * h_{g @ 120 C} = 487$ W

A-4.3) Door Opening Infiltration:

The air infiltration due to door opening should be taken into consideration. Table (5-8) bellow shows the infiltration rates due to the door opening:

Refrigerated Space Volume "m^3"	Infiltration Rate V^{o} "l/s"	
	$T_{R/S} < 0$	$T_{R/S} > 0$
7	2.3	3.1
8.5	2.6	3.4
10	2.8	3.7
15	3.3	4.4
20	3.8	5.0
25	4.2	5.5
30	4.6	5.9
40	5.4	6.8
50	5.8	7.5
75	6.9	9.0
100	7.9	10.2
150	9.4	12.2
200	10.9	13.4
250	11.9	15.3
300	12.9	16.7
400	14.9	19.0
500	16.8	21.4
600	18.1	23.6
700	18.6	24.3
800	20.4	25.9
900	21.9	27.1
1000	23.1	28.9

Table (5-8) Infiltration Rates Due to Door Opening

Where $T_{R/S}$: Refrigerated Space Temperature.

This load is calculated by:

$$Q_{infiltration} = V^{o} * \Delta h * 10^{3} \qquad (5\text{-}8)$$

Where Δh is the enthalpy difference between infiltrated air and refrigeration room.

Now, the infiltration load is calculated depending on the volume of the refrigeration room:

$\rightarrow Q_{infiltration} = 1083$ W

Where the value of the heat gain due to air infiltration is obtained from Table (5-9) below:

Entering Air heat content to the refrigerated space

(kJ/l) $T_{R/S} > 0$ C

Entering Air temperature °C										$T_{R/S}$ °C
40		35		30			25			
Relative Humidity %										
60	50	60	50	70	60	50	70	60	50	
0.0795	0.0663	0.0563	0.0500	0.0441	0.0357	0.0281	0.0246	0.0186	0.0128	15
0.0992	0.0792	0.0694	0.0591	0.0574	0.0491	0.0319	0.0382	0.0323	0.0266	10
0.1036	0.0906	0.0810	0.0708	0.0693	0.0610	0.0536	0.0502	0.0445	0.0388	5
0.1141	0.1003	0.0910	0.0808	0.0794	0.0713	0.0639	0.0606	0.0550	0.0493	0

Entering Air heat content to the refrigerated space

(kJ/l) $T_{R/S} < 0$ C

Entering Air temperature °C										$T_{R/S}$ °C
35		30		25		10		5		
Relative Humidity %										
60	50	60	50	60	50	80	70	80	70	
0.0921	0.0820	0.0724	0.0650	0.0562	0.0505	0.0154	0.0142	0.0111	0.0092	0
0.1004	0.0903	0.0809	0.0736	0.0649	0.0592	0.0247	0.0235	0.0210	0.0193	-5
0.1071	0.0970	0.0877	0.0805	0.0719	0.0662	0.0321	0.0309	0.0288	0.0271	-10
0.1137	0.1037	0.0945	0.0873	0.0788	0.0732	0.0395	0.0383	0.0367	0.0350	-15
0.1203	0.1102	0.1013	0.0941	0.0857	0.0801	0.0468	0.0456	0.0444	0.0427	-20
0.1265	0.1165	0.1077	0.0998	0.0922	0.0866	0.0537	0.0525	0.0523	0.0501	-25
0.1325	0.1225	0.1138	0.1067	0.0985	0.0929	0.0604	0.0591	0.0588	0.0571	-30
0.1382	0.1283	0.1197	0.1126	0.1045	0.0989	0.0668	0.0656	0.0657	0.0640	-35
0.1440	0.1341	0.1256	0.1185	0.1106	0.1050	0.0732	0.0720	0.0725	0.0708	-40

Table (5-9) Energy loss Due to Door Opening

A-5) Miscellaneous load

This load represents the heat given off by the service offered in the refrigerated space, this service such as:

- Lighting.
- Electric motor and equipments which operate inside the refrigerated space.
- The people occupying the refrigeration room.
- Defrosting heat sources.

A-5.1) Lighting Load

The refrigeration room contains 4 lambs. Each one of 100 W rated.

Then ➔ $Q_{lighting}$ = (No of light * Power of light)/24 * time of use (5-9)
= (4*100)/24 * 2
= 35 W

A-5.2) Electric Motor

We have two electric motors with total power of 0.74 KW. The cooling load required for these motors is by the equation:

$$Q_{motors} = \frac{Z*n*P}{24}*T_{use} \quad (5\text{-}10)$$

Where:

Z : The correction factor (0.9)

n : The number of motors (same power)

P : Rated motor power. [W]

T_{use}: Time of use. [h]

And so ➔ Q_{motors} = 1000 W

And for the diesel medium forklift ...the output heat generation = 5 kWh:

Then ➔ $Q_{fork} = Q_{forklift}$ * 1/2 hour (5-11)
= 2500 W

A-5.3) People Occupying the Room

For heavy working conditions, 4 workers are needed. Each produces 250 W. as in Table (5-10) below:

Human Heat Generation

Q_t	$T_{R/S}$
450	-30
410	-25
390	-20
300	-10
275	-5
250	0
225	5
200	10

Table (5-10) Human Heat Generation Rate

→ $Q_{People} = (250*N_o)/24 * \text{Time of use}$ (5-12)

$= (280*4)/24 * 2$

$= 85\ W$

N_o: referred to number of persons.

A-5.4) Defrost Heat source

The defrost heat load is valued as 10% to 50% of the total capacity of the heating coil…

When the defrost heater is 3 KW capacity.

Then: $Q_{defrost} = 25\% * \text{Heater Capacity}$ (5-13)

$= 0.25 * 3000$

$= 750\ W$

And so, the total Miscellaneous cooling loads of this room:

$Q_{miscellaneous} = Q_{lighting} + Q_{people} + Q_{M\ \&\ Fork} + Q_{defrost}$

$= 35 + 85 + 3500 + 750$

$= 4350\ W$

And then the total cooling load demand:

$Q_{total} = Q_{miscellaneous} + Q_{Respiration} + Q_{gain} + Q_{product} + Q_{air\ change}$

$= 11.5\ kW$

And with taking 16 h operation and factor of safety of 10%:

$Q_{final} = 19.0\ kW.$

And for rooms assigned "B"; they have other geometric shape and capacity of product, by repeating the same above calculations; similarly for rooms B we get: $Q_{final} = 16.8$ kW.

And for rooms assigned "C"; they have other geometric shape and capacity of product. by repeating the same above calculations; similarly for rooms C we get:

$Q_{final} = 14.8$ kW.

These cooling load demand forms the maximum cooling loads that the refrigeration system should be designed to handle.

B) Chilling Room

Firstly, we will consider the stared room as a chilling room for the refrigeration station; after that, we will find out which product gives the maximum product refrigeration demand. The Table (5-11) below shows the product load for the various products in the refrigeration station.

Product	The Product Cooling Load "kW"
Lemons	28.80
Oranges	27.43
Potatoes	34.56
Carrot	20.16
Grapes	23.92
Apple	22.32
Apricot	32.92
Cucumber	36.00
Avocados	21.95

Table (5-11) Product Cooling Load Values under Full Loading

It's noticed clearly that the maximum cooling load is referred to the cucumber. Such that it gives the maximum available product load in the chilling room.

Now, the load calculations will be done based on the chilling requirements of the cucumber.

B-1) Wall Gain

Assuming nothing stored next to room from both sides.

$Q_{walls} = Q_{outsider\ wall} + Q_{corridor} + Q_{floor} + Q_{ceiling} + 2*Q_{adjacent\ room} + Q_{door}$

→ $Q_{outsider\ wall} = Q_{upper} + Q_{lower}$

$= (U_{upper}*A_{upper}*\Delta T) + (U_{lower}*A_{lower}*\Delta T)$

$= (0.3022*10.8*25) + (0.308*22.4*33)$

$= 303$ W.

And by the same way we can obtain the wall gain for all sides. And so the wall gains are:

→ $Q_{corridor}$ = 157 W

→ Q_{door} = 17 W

→ Q_{Floor} = 892 W

→ $Q_{ceiling}$ = 433 W

→ $Q_{btn\ rooms}$ = 129 W

Q_{gain} = 1831 W from wall gain.

B-2) Product Load

From Table (5-11) the product load for cucumber –which is referred to be the max–, is 36 kW.

$Q_{product}$ = 36 kW.

B-3) Respiration Load

In a same manner, using equation (5-5), the respiration load for the chilling room and for cucumber as the critical product will be:

$Q_{Respiration} = 25920 * 0.07 = 1815$ W.

B-4) Air Change Load

For the chilling rooms in the station, each room needs to be ventilated for removing the alcohols from refrigerated space. In addition to the humidification process needed for maintaining the storage recommended humidity.

B-4.1) Ventilation Air Change:

As calculated before; the ventilation load is calculated. Such that:

$Q_{Ventilation} = 833$ W

B-4.2) Humidification

Same way of calculations, we get:

$Q_{humidification} = 487$ W

B-5) Miscellaneous loads:

This load represents the heat given off by the service offered in the refrigerated space, this service such as:

- Lighting.
- Electric motor and equipments which operate inside the refrigerated space.
- The people occupying the refrigeration room.
- Defrosting heat sources.
- The Packaging heat removal

B-5.1) Lighting Load

The refrigeration room contains 4 lambs, each one of 100 W capacities.

Then ➔ $Q_{lighting}$ = (No of light * Power of light)/24 * time of use
= (4*100)/24 * 2
= 35 W

B-5.2) Electric motor

In a similar manner, the heat gain from the electric motors for chilling room will be obtained from equation (5-10) as follows:

➔ $Q_{motors} = 1000$ W

B-5.3) People occupying the room

As for storing rooms, the heat gain from people occupying the chilling room will be obtained from equation (5-12):

→ $Q_{People} = 85$ W

B-5.4) Defrost Heat source

The defrost heat load is valued as 10% of the total capacity of the heating coil...

When the defrost heater is 3 KW capacity.

$$\text{Then: } Q_{defrost} = 25\% * \text{Heater Capacity}$$
$$= 0.25 * 3000$$
$$= 750 \text{ W}$$

B-5.5) Packaging load

The heat removal stands is calculated as done with the product load. Each couple of packages is carried by one stand weighs 70 kg. And so, we need 16 stands. And the specific heat corresponding to the packaging materials are shown in Table (5-12).

$M_{stands} = 16 * 70 = 1120$ kg. And when C_p for steel = 0.5 kJ/kg.C

$$\rightarrow Q_{stands} = M * C_p * (\Delta T) / (16 * 3600) \quad (5\text{-}14)$$
$$= 175 \text{ W}$$

Package element	C_p "kJ\kg.K"
Steel	0.5
Wood	1.26
Cartoon	1.1

Table (5-12) Specific Heat for Packaging Materials

And by the same way for cartoon and wooden boards:

→ $Q_{cartoon} = 120$ W.
→ $Q_{wooden\ board} = 190$ W. → $Q_{packaging} = 485$ W

And so, the total Miscellaneous cooling loads of this room:

$Q_{miscellaneous} = 2350$ W

B-6) Chilling Rate Factor:

After all these calculations, the total chilling load demand should be divided by the chilling rate factor, which will make the chilling process more uniform along the chilling process. When the total chilling demand is divided by this factor the chilling load will increase to compensate the decay of the heat transfer at the last stages of chilling process. This factor depends on the type of chilled product. As shown in Table (5-13).

Product Type	Chilling Rate Factor
Lemons	1.00
Oranges	0.70
Potatoes	0.67
Carrot	0.80
Grapes	0.80
Apples	0.67
Apricots	0.67
Cucumber	0.71
Avocadoes	0.67

Table (5-13) Chilling Rate Factor

Then the cooling load = 61 kW

And with taking factor of safety by 10% and 16 h operation time:

$Q_{total} = 100$ kW.

5.4 Variable Cooling Load Consideration

After finishing the cooling load calculation required for the chilling and storage rooms of the station. Now we will consider the variable loading that the refrigeration system of the station should be designed to handle. The previous calculation determined the maximum cooling load demands required for both chilling and storing rooms. And the variable refrigeration demand required for both chilling and storing rooms now will be determined and explained.

There are some main factors to be taken into consideration while running or operating the refrigeration station, those factors are as followed:

- The refrigeration room either chilling or storing will run under 100%, 75%, or 50% of its total capacity. And it's forbidden to refrigerate a product quantity less than its 50% capacity.
- The chilling room runs in either chilling process or storing process, following the control signal given to its refrigeration system.

And now we will discuss and determine the variable cooling loads subjected to both storing and chilling rooms.

➔ Storing Rooms

The storing room variable loads depend on the type and quantity of the product under chilling process. And so, the cooling load for the storing room decrease from the maximum value obtained to some minimum value for this load. Table (5-14).

Product Type	Cooling Load Demand "kW"								
	100% Loading			75% Loading			50% Loading		
	A	B	C	A	B	C	A	B	C
Lemons	16.3	15.3	11.5	14.9	14.2	10.5	14.0	13.0	9.5
Oranges	15.3	14.4	11.0	14.3	13.6	10.0	13.3	12.7	9.3
Potatoes	16.5	15.5	11.7	15.2	14.3	10.6	14.0	13.5	9.7
Carrot	16.7	15.6	12.0	15.3	14.4	10.9	14.0	13.3	9.8
Grapes	16.7	15.7	12.0	15.4	14.4	10.9	14.0	13.3	9.8
Apples	15.4	14.6	11.0	14.4	13.6	10.2	13.3	12.7	9.3
Apricots	19.3	18.0	13.6	17.5	16.2	12.0	15.5	14.5	10.6
Cucumber	17.7	16.4	12.3	16.1	15.0	11.0	14.5	13.6	10.0
Avocadoes	17.3	16.0	12.2	15.8	14.8	11.0	14.3	13.5	10.0

Table (5-14) General Cooling Load Demand for Storing Rooms

→ **Chilling Room**

Also the chilling room variable loads depend on the type and quantity of the product under chilling process. And so, the cooling load for the chilling room decrease from the maximum value obtained to some minimum value for this load, the Table (5-15).

Product Type	Cooling Load Demand "kW"		
	100% Loading	75% Loading	50% Loading
Lemons	58.30	45.8	32.3
Oranges	78.60	62.0	45.3
Potatoes	101.0	79.0	57.0
Carrot	55.00	44.0	33.0
Grapes	63.40	50.4	37.5
Apples	70.00	55.8	41.4
Apricots	100.2	80.0	57.8
Cucumber	101.0	78.5	56.4
Avocadoes	74.6	57.0	42.3

Table (5-15) General Cooling Load Demand for Chilling Room

These are the refrigeration loads that form the base for the redesign of the refrigeration station system.

Chapter Six

STATION SYSTEM DESIGN

6.1 Introduction

The design of the refrigeration station should satisfy the cooling requirements of the products that the refrigeration station handle, in addition to keeping the marketing quality of these products such that it can be marketed at any time along the storage period.

In this chapter; the new designs of the refrigeration stations will be done, according to the product refrigeration characteristics. This design will include the total refrigeration system design, the humidification system design, the ventilation system design, and the control system design of the station.

6.2 Overview of Station Design

The design process will be performed according to the information and results gathered and calculated through the previous chapters. First, we will start with the refrigeration system design. Then, humidification system design, ventilation system design, control system, and finally the accessories of the refrigeration station designs.

6.3 Refrigeration System Design

As a useful approach, we will design the refrigeration system depending on using VSD (variable speed drive) compressors.

Traditionally, cooling capacity is matched to the load on a system by on-off control, pressure-regulation unloading valves, or hot gas bypass. In contrast, VSD compressor driven by variable frequency drive will constantly match the load, adjusting capacity at the source by varying the compressor speed in a speed range. Because of the speed increase a smaller compressor can be used than would be selected for fixed speed operation in the same system, resulting in a saving that offsets the cost of the variable frequency drive.

Energy savings occur after installation because varying the speed of a compressor on demand means that refrigeration capacity is closely matched to actual heat load whenever the system is in operation. In a carefully controlled laboratory setting, savings of up to 26% have been observed when cooling a cold storage room system with 68% average system load. Savings up to 62% were recorded in a process cooling application with 50% average system load.

The principle is that an AC Motor rotation speed is always proportional to power supply frequency. At 60 Hz, hermetic compressors typically run at 3500 RPM. An AKD frequency converter can deliver frequencies ranging from 30 to 90 Hz, resulting in a compressor speed from 1750 to 5250 RPM, with a corresponding variation in cooling capacity. Compressor cycling is also much reduced in a variable speed system.

With a compressor several sizes smaller than a fixed speed system would require, and speed reduction, costs are reduced. But during periods of partial load, the system also operates with an oversized condenser, which allows decreasing the condensing pressure, and consequently the power input. Thus there are three energy efficiency benefits derived from a variable speed system. This efficiency continues for the life of the installation.

6.3-1 Refrigerant Selection

When designing a refrigeration system, there are several refrigerants to choose from, such as CFCs, Ammonia, and other refrigerants.

Many important characteristics should be considered when refrigerant is selected. The most two important characteristics are the temperature of the refrigerated space, and the ambient temperature of the environment surrounding the refrigerated space. These two characteristic affects on the saturation pressures of the refrigeration cycle will differ from refrigerant to other, and in turn affects on the pressure ratio and the required sizes of equipments and its energy requirements for some refrigerant selection.

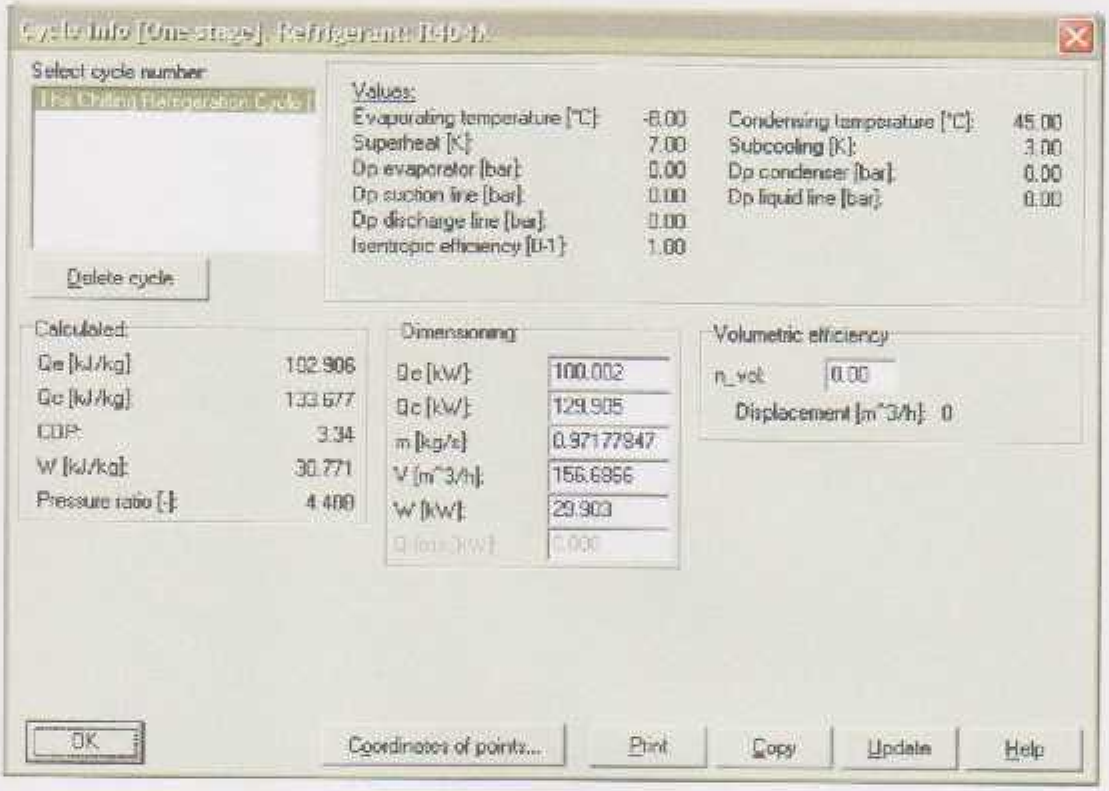


Figure (6-1).Thermodynamic Simulation of Refrigeration Effect Using R404A

The Figure (6-1) above shows the thermodynamic simulation for refrigeration cycle working depending on the refrigerant R404A. This refrigerant is the recommended refrigerant for the refrigeration room applications (ASHREA Refrigeration Handbook. Chapter 5, Refrigerants). The required pressure ratio is acceptable. And obtained COP at this temperature range is suitable and good. And this refrigerant is available at good price and environmentally less harmful or toxic. So, the most appropriate refrigerant is R404A.

6.3-2 Chilling Room

The chilling room must be designed to handle variable loads depending on the amount and type of product that needs to be chilled. In our design, the chilling room will be designed to handle 100%, 75%, and 50% of its total capacity of product.

The refrigeration system of the chilling room has specific refrigeration system used to handle the large cooling capacities. The refrigeration system of the chilling room consists of 4 main components:

1- The Compressor.
2- The Condensation Unit
3- The Room Cooler.
4- Expansion Valve.
5- The Piping of System.

1 → Compressor

The Chilling room works in two phases; the chilling phase, and the storing phase. And this will causes two main cooling demands; the chilling load demand, and the storing load demand. And so, two compressors will be attached to the condensation unit in parallel. Such that one of the two compressors will run depending on the input signal from the control system.

Each compressor will be selected alone, depending on the cooling capacity that it should handle. All compressors of the refrigeration station will be selected from DORIN Co. this company is specialized in the VSD refrigeration compressors. And all compressors of the refrigeration station are selected depending on the catalog of the company.

The compressor that works in the chilling phase is H8000CC VSD model. The working characteristics were taken from the VSD compressor catalog of the company and translated into English and listed below and in Table (6-1). (See Appendix 1)

Model H8OOOCC

380/420 V~AC
Gas R404A

Evaporating Temperature	-10 C
Condensation Temperature	45 C
Superheat	7 C
Subcooling	3 C

H8OOOCC		COMPRESSOR
Cooling Capacity	"kW"	93.60
Absorbed Power	"kW"	41.47
Mass Flow	"kg/h"	2.37
Absorbed Current	"A"	80.9
COP		2.26

Table (6-1) Operation Characteristics of Compressor Model

The other compressor is selected in the same way, and the compressor characteristic are listed below and shown in Table (6-2). (See Appendix 1)

Model C6015CC

380/420 V~AC
Gas R404A

Evaporating Temperature	-10 C
Condensation Temperature	45 C
Superheat	7 C
Subcooling	3 C

C6015CC		COMPRESSOR
Cooling Capacity	"kW"	18.70
Absorbed Power	"kW"	5.30
Mass Flow	"kg/h"	0.327
Absorbed Current	"A"	5.2
COP		2.35

Table (6-2) Operation Characteristics of Compressor Model

2 → Condensation Unit

The condensation unit in the refrigeration system of the chilling room will be selected from DWM Copland selection software. After taking manufacturer instruction, compressor can replace with other one with having same cooling capacity or ± 10%. The selection is taken at the desired cooling load at an ambient temperature of 35 C. and the model adopted was OLQ-48V-NLO-TWD. The parts and specification of this condensation unit is mentioned in appendix 1.

3 ➔ Evaporator Unit

The evaporator unit of the chilling room will be selected from the cooling units manufactured by FRIGA-BOHN Refrigeration Co. from the catalog of cooling units and following the manufacturer instructions, the model NKT 2X6D B4 L cooling unit will be selected. And the specifications of this cooling unit are shown in Table (6-3). Double cooling units are mounted in the room. (See Appendix 1)

NKT ... L T = large heat exchange surface

Models		NKT ... L	2x4D A2	2x4D A3	1x6D B2	3x4D A2	1x6D B3	3x4D A3	1x6D B4	1x8D C2	2x6Y B2	1x8D C3	2x6D B2	2x6D B3	2x6D B4
Capacity - [illegible]	DT1 = 8K - SC2 [illegible]	kW [illegible]	13.39	15.90	17.91	20.20	21.46	23.90	23.93	27.06	31.58	32.06	35.92	43.44	46.48
Surface		m2	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	3174
Circuit volume		dm3	[illegible]	[illegible]	274	[illegible]	368	[illegible]	456	[illegible]	[illegible]	514	[illegible]	730	913
Fans *	Num x Ø	mm	2x450	2x450	1x630	2x450	1x630	3x450	1x630	1x800	2x630	1x800	2x630	2x630	2x630
	Air flow	m3/h [illegible]	[illegible]	9500	[illegible]	[illegible]	13850	[illegible]	[illegible]	20450	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
	Air throw [illegible]	m [illegible]	[illegible]	[illegible]	30	22	34	[illegible]	33	40	[illegible]	38	37	36	[illegible]
Electric defrost EIU [illegible]	Ω	Num.	6	[illegible]	4	5	[illegible]	6	12	6	[illegible]	9	[illegible]	8	12
	400V/3	W	9000	9000	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	9000	[illegible]	13500	[illegible]	[illegible]	[illegible]
		A	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
Electric defrost ELU [illegible]	Ω	Num.	[illegible]	[illegible]	[illegible]	[illegible]	12	[illegible]	8	[illegible]	[illegible]	[illegible]	[illegible]	12	[illegible]
	400V/3	W	9000	13500	[illegible]	13200	[illegible]	[illegible]	[illegible]	12500	[illegible]	18000	[illegible]	[illegible]	[illegible]
		A	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
Electric defrost kit ECK [illegible]	Ω	Num.	-	[illegible]	3	-	[illegible]	[illegible]	[illegible]	3	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
	400V/3	W	-	[illegible]	[illegible]	-	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
		A	-	[illegible]	[illegible]	-	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
[illegible] ECK		Num.	0	1	1	0	1	1	1	1	1	1	1	1	1
Connections	Inlet	Ø	5/8"	5/8"	[illegible]	5/8"	7/8"	7/8"	1 1/8"	7/8"	1 1/8"	1 1/8"	1 1/8"	1 3/8"	1 3/8"
	Outlet	Ø	1 3/8"	1 3/8"	1 3/8"	1 3/8"	1 5/8"	1 5/8"	1 5/8"	1 5/8"	2 1/8"	2 1/8"	2 1/8"	2 1/8"	2 1/8"
Net weight		kg	180	190	[illegible]	200	200	230	[illegible]	270	[illegible]	300	[illegible]	350	390

Table (6-3) Indoor Cooling Unit Catalog

4 ➔ Expansion Valve

The expansion valve selection of refrigeration system in the chilling room will be based on multi evaporator refrigeration. As mentioned, the chilling room contains two indoor cooling units. This affects on the way of installing a proper expansion valve.

As a useful approach, the most suitable type to use is Thermostatic Expansion Valve with external equalizer; because of its high sensitivity to the temperature variations in the suction line, and their insensitivity for the pressure drop in the evaporator operation.

Before the expansion valve selection, the way of installation of the expansion valve within double evaporator system. As shown in Figure (6-2) below; the expansion valves are externally equalized from the main suction line, which starts after the two evaporator discharge lines are connected. The sensing bulb of the expansion valve should be followed by U configuration, this configuration enables the refrigerant gas to accumulate instantaneously just before it; this in turn encouraging more effective heat transfer between the evaporator discharge line and the sensing bulb of the expansion valve, and implies more accurate and sufficient performance.

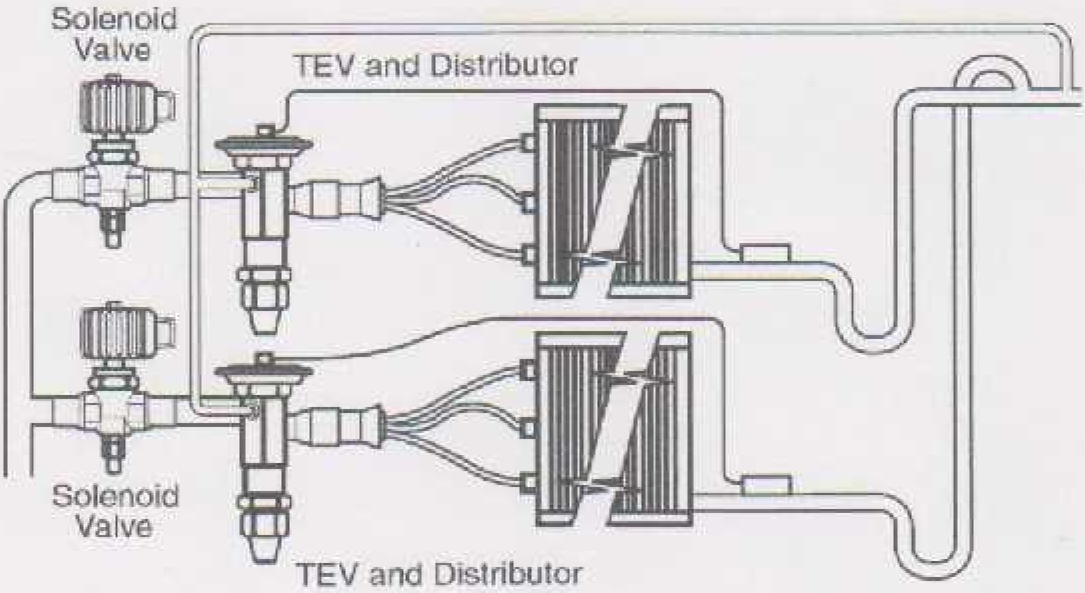


Figure (6-2) Thermostatic Expansion Valve Installation

The expansion valve is obtained from Parker Co. Referring to the company catalog the suitable expansion valve selected is the model (RE E 12 S), as shown in Table (6-4), the expansion valve nominal capacity is 12 tons (42 kW) for refrigerant R404A. For further information refer to the manufacturer catalog in Appendix 1.

Specifications

Refrigerant	Refrigerant Designation	Nominal Capacity	Body Style	Externally Equalized	Rainbow Changes™	Inlet Connection (Optional)	Outlet Connection
R-12 R-134a R-401A R-401B	J	6	1	RE 6 J	W, 280	5/8"	7/8"
		9	1	RE 9 J		7/8"	1-1/8"
		12	1	RE 12 J		7/8 (1-1/8")	1-3/8"
		16	1	RE 16 J		1-1/8"	1-3/8"
		23	2	RE 22 J		1-1/8"	1-3/8"
		40	2	RE 40 J		1-1/8"	1-5/8"
R-402B R-404A R-402A R-502 R-507	S	6	1	RE 6 S	W, 2, X1W, X35	5/8"	7/8"
		9	1	RE 9 S		7/8"	1-1/8"
		12	1	RE 12 S		7/8 (1-1/8")	1-3/8"
		21	1	RE 21 S		1-1/8"	1-3/8"
		30	2	RE 30 S		1-1/8"	1-3/8"
		45	2	RE 45 S		1-1/8"	1-5/8"
R-22 R-407C	V	10	1	RE 10 V	W, 2, X100, X35	5/8"	7/8"
		15	1	RE 15 V		7/8"	1-1/8"
		20	1	RE 20 V		7/8 (1-1/8")	1-3/8"
		30	1	RE 30 V		1-1/8"	1-3/8"
		40	2	RE 40 V		1-1/8"	1-3/8"
		70	2	RE 70 V		1-1/8"	1-5/8"

Table (6-4) Expansion Selection Table

5 ➔ Refrigerant Piping

The Refrigerant piping of the refrigeration system is one of the most important designs of the refrigeration system, i.e. the refrigerant piping should assure many purposes such as:

1- Ensure adequate velocity to return oil to compressor at all steps of unloading.

2- Avoid excessive noise.

3- Minimize system capacity and efficiency loss and pipe erosion.

4- Minimizing the refrigerant charge. As shown in Figure (6-3)

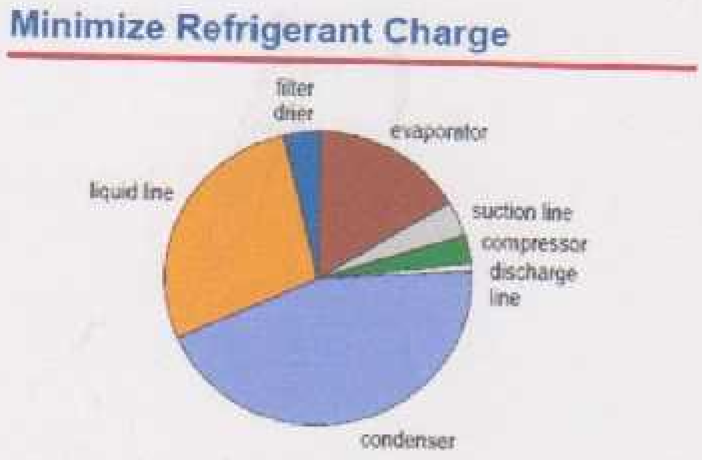


Figure (6-3).Refrigerant Charge Distribution in Refrigeration Circuit

The refrigeration design of the refrigeration rooms requires designing the liquid line that supplies the indoor cooling unit (evaporator & TEV) with liquid refrigerant, and the refrigerant suction line; which returns refrigerant from the evaporator to the compressor.

A) Liquid Line

This section of pipe conducts warm, high-pressure liquid refrigerant from the condenser to the expansion device and evaporator.

Routing Requirements and Sizing

The liquid line must be designed and installed to ensure that only liquid refrigerant (no vapor) enters the expansion device. The presence of refrigerant vapor upstream of a TXV can result in erratic valve operation and reduced system capacity. In order to meet this requirement, the condenser must provide adequate subcooling, and the pressure drop through the liquid line and accessories must not be high enough to cause flashing upstream of the expansion device. Subcooling allows the liquid refrigerant to experience some pressure drop as it flows through the liquid line, without the risk of flashing.

Oil and liquid refrigerant mix readily, so oil movement within the liquid line is not a concern. However, the design of the liquid line is the most critical when it comes to minimizing the system refrigerant charge. This is because, of the three lines, it has the greatest impact on the quantity of refrigerant required to charge the system. The diameter of the liquid line must be as small as possible to minimize the refrigerant charge, therefore improving reliability and minimizing installed cost. However, if the pipe is too small, the increased pressure drop may cause flashing upstream of the expansion device.

The refrigerant velocity in the liquid line shouldn't exceed 3 m/s to prevent noise and large pressure drops. And no under limit since the refrigerant will solute the oil and carry it by the way. As shown in Figure (6-4); the refrigerant velocity is about 2.8 m/s at the recommended liquid line diameter from the condensation unit manufacturer.

liquid line

Determine Refrigerant Velocity

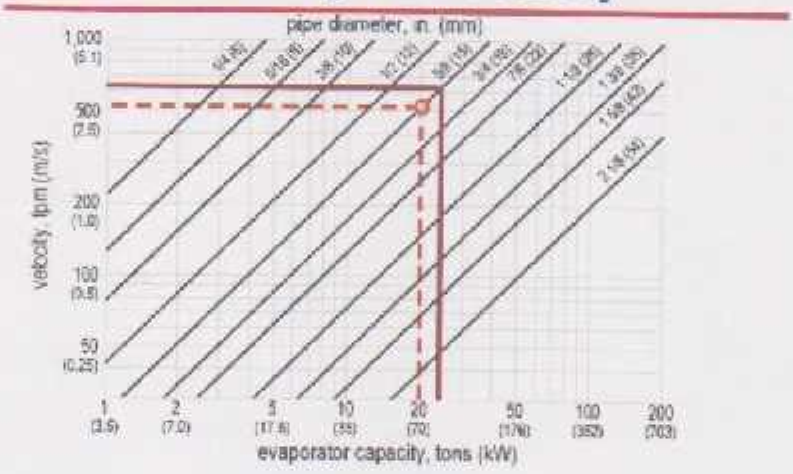


Figure (6-4) Evaporator Capacity and Velocity Diagram for Liquid Line

As mentioned, the refrigerant liquid line is recommended from DWM Copland for the liquid line to be 5/8 inch. This line will supply both evaporators with liquid refrigerant. (See appendix). And this diameter is around the recommended diameter for the liquid line obtained from the piping design process given in the TRANE piping design book. From Figure (6-3) below, the pressure drop corresponding to this diameter is 25 kPa/10 m. that means that there is a pressure drop in the liquid line about 8 kPa. And additional 1.8 kPa for the elbows and accessories. Totally 10 kPa pressure drop prevents the liquid refrigerant from flashing in the liquid line; because flashing happens at a pressure of about 18 Bar. And the saturation pressure in the condenser is about 20.5 Bar.

liquid line

Determine Pressure Drop

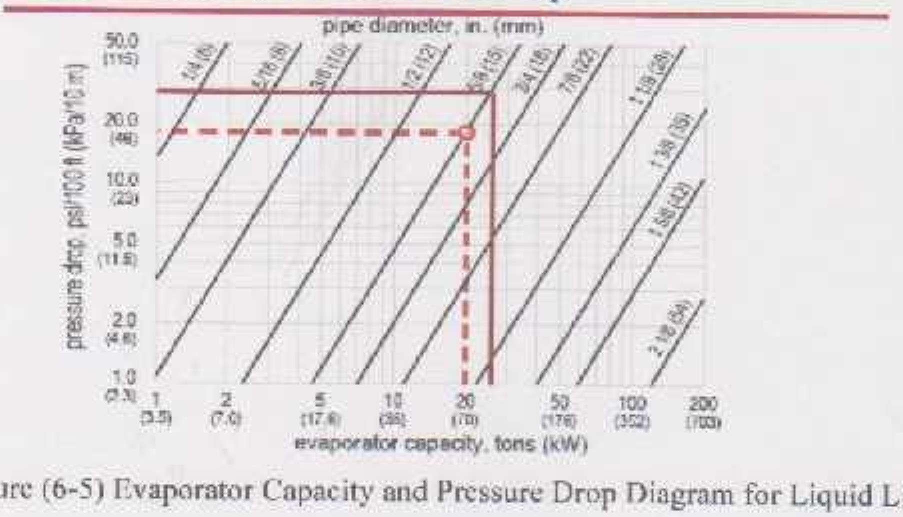


Figure (6-5) Evaporator Capacity and Pressure Drop Diagram for Liquid Line

B) Suction Line:

This pipe conducts low pressure refrigerant vapor from the evaporator to the compressor.

Routing Requirements and Sizing

The diameter of the suction line must be small enough that the resulting refrigerant velocity is sufficiently high to carry oil droplets, at all steps of compressor unloading. The refrigerant velocity inside a pipe depends on the mass flow rate and density of the refrigerant if the velocity in the pipe is too high, however, objectionable noise may result. Also, the pipe diameter should be as large as possible to minimize pressure drop and thereby maximize system capacity and efficiency.

The refrigerant velocity through the suction line has max limit of 20 m/s, more velocity causing vacuum and objectionable noise. And min limit depends weather the pipe is horizontal or upward or downward (see Table 6-5). Less velocity will cause the accumulation of lubricant oil resulting overheating of the compressor.

suction line

Minimum Allowable Velocities

pipe diameter, in. (mm)	minimum velocity, fpm (m/s)	
	riser	horiz/drop
3/8 (10)	370 (1.9)	275 (1.4)
1/2 (12)	460 (2.3)	350 (1.8)
5/8 (15)	520 (2.6)	390 (2.0)
3/4 (18)	560 (2.8)	420 (2.1)
7/8 (22)	600 (3.1)	450 (2.3)
1 1/8 (28)	700 (3.6)	525 (2.7)
1 3/8 (35)	780 (4.0)	585 (3.0)
1 5/8 (42)	840 (4.3)	630 (3.2)
2 1/8 (54)	960 (5.0)	735 (3.7)
2 5/8 (67)	1,080 (5.5)	810 (4.1)
3 1/8 (79)	1,180 (6.0)	885 (4.5)
3 5/8 (105)	1,270 (6.5)	950 (4.8)
4 1/8 (130)	1,360 (6.9)	1,020 (5.2)

Table (6-5) Min Allowable Velocities in Suction Line

The recommended suction line diameter is 1 3/8 inch from condensation unit manufacturer. From TRANE refrigerant piping book, the velocity of refrigerant in the suction line is more than 20 m/s (see Figure 6-6). So the refrigerant suction line will be designed as a double riser system.

suction line

Determine Refrigerant Velocity

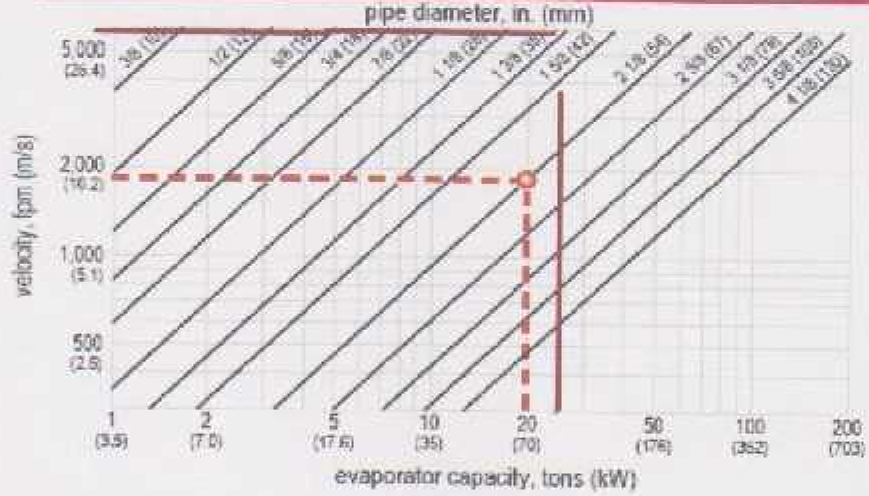


Figure (6-6) Evaporator Capacity and Velocity Diagram for Suction Line

The double riser system will be considered as in Figure (6-7); the main two pipes (to the condensation unit and from evaporator) will be selected as 1 5/8 inch. And the when the minimum load which is 32 kW; the first riser will be selected to be 1 1/8 inch, and the rest 68 kW load will be compensated from the second riser which its diameter is to be selected as 1 3/8 inch.

suction line

Double Suction Riser

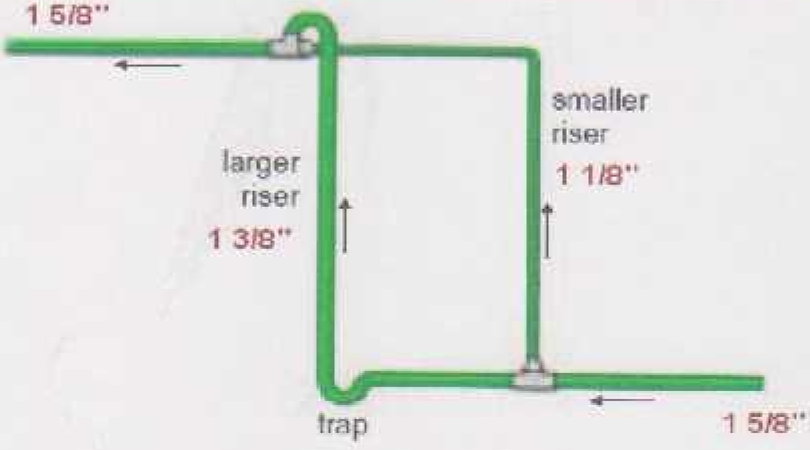
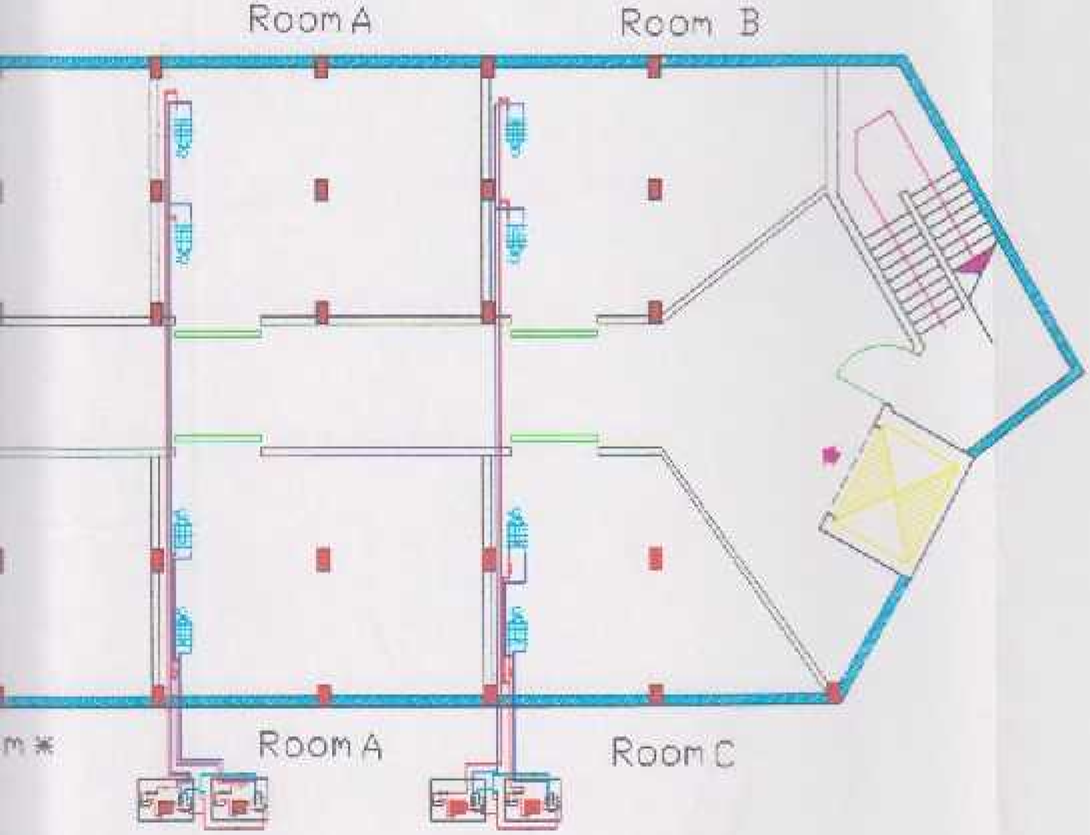


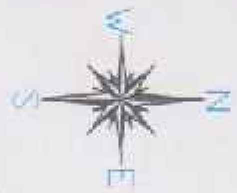
Figure (6-7) Double Riser System

At minimum capacity, the refrigerant velocity in the two risers becomes too low to carry the oil droplets. The oil from both risers therefore drains down, filling the trap at the base of the larger riser. When this trap becomes completely filled with oil, it prevents refrigerant vapor from flowing up the larger riser, and diverts the entire refrigerant up the smaller riser. This smaller riser is constructed of a pipe with a diameter that is small enough to maintain adequate velocity at minimum capacity. When system capacity is increased again, the higher refrigerant velocity clears the trap of oil, and refrigerant vapor again flows up both risers.

Notice the inverted trap at the top of the larger riser. When the trap at the base of the larger riser is filled with oil, and refrigerant flows up only the smaller riser, this inverted trap prevents oil from draining back into the larger riser. This configuration minimizes the amount of oil trapped in the double riser under this condition, therefore maximizing the amount of oil that is returned to the compressor.



GENERAL NOTES



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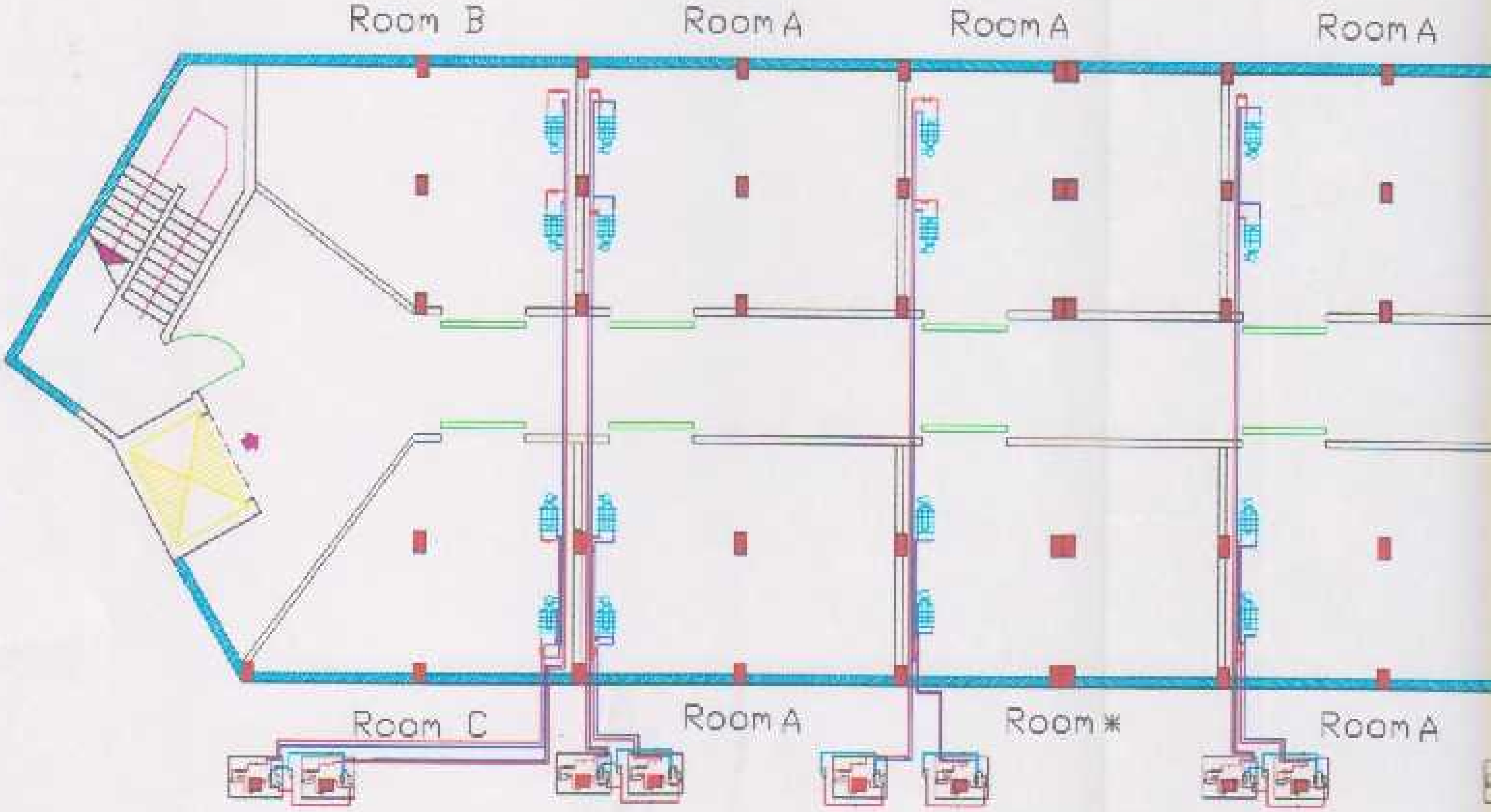
Project

Refrigeration Station -Halhul

Sheet Title

Refrigeration System Plane

Senior Project	Sheet
Date 2006-2007	A-3



Room B
Room A
Room A
Room A
Room C
Room A
Room *
Room A

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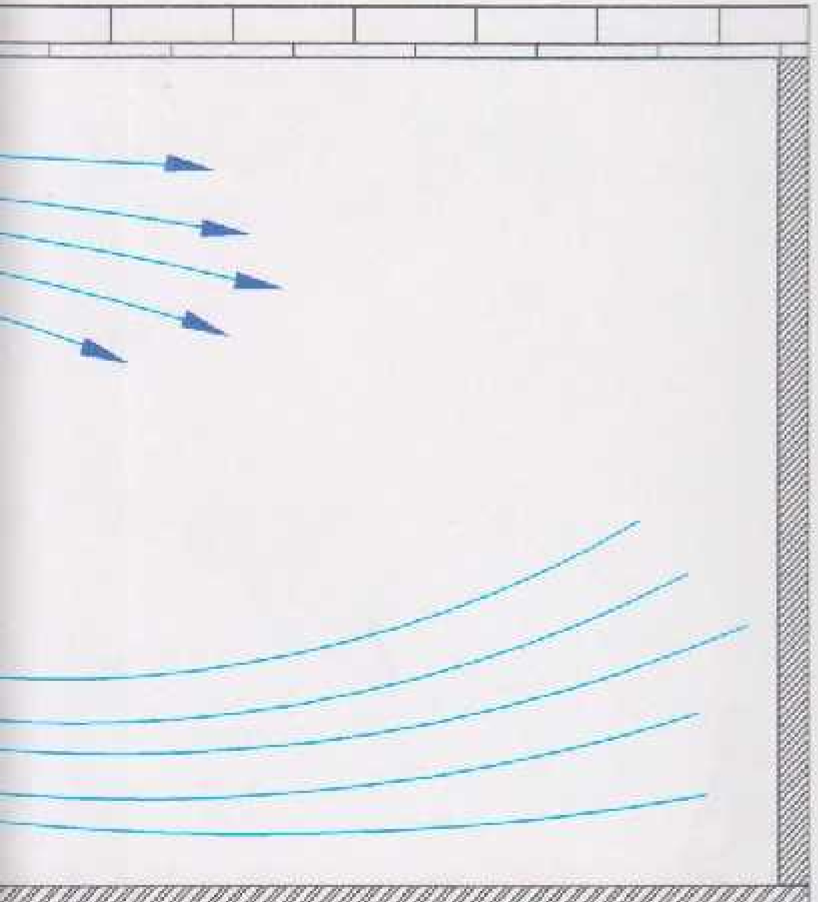

GENERAL NOTES

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Project

Refrigeration Station -Halhul

Sheet Title

Cold Air Throw

Senior Project

Date

2006-2007

Sheet

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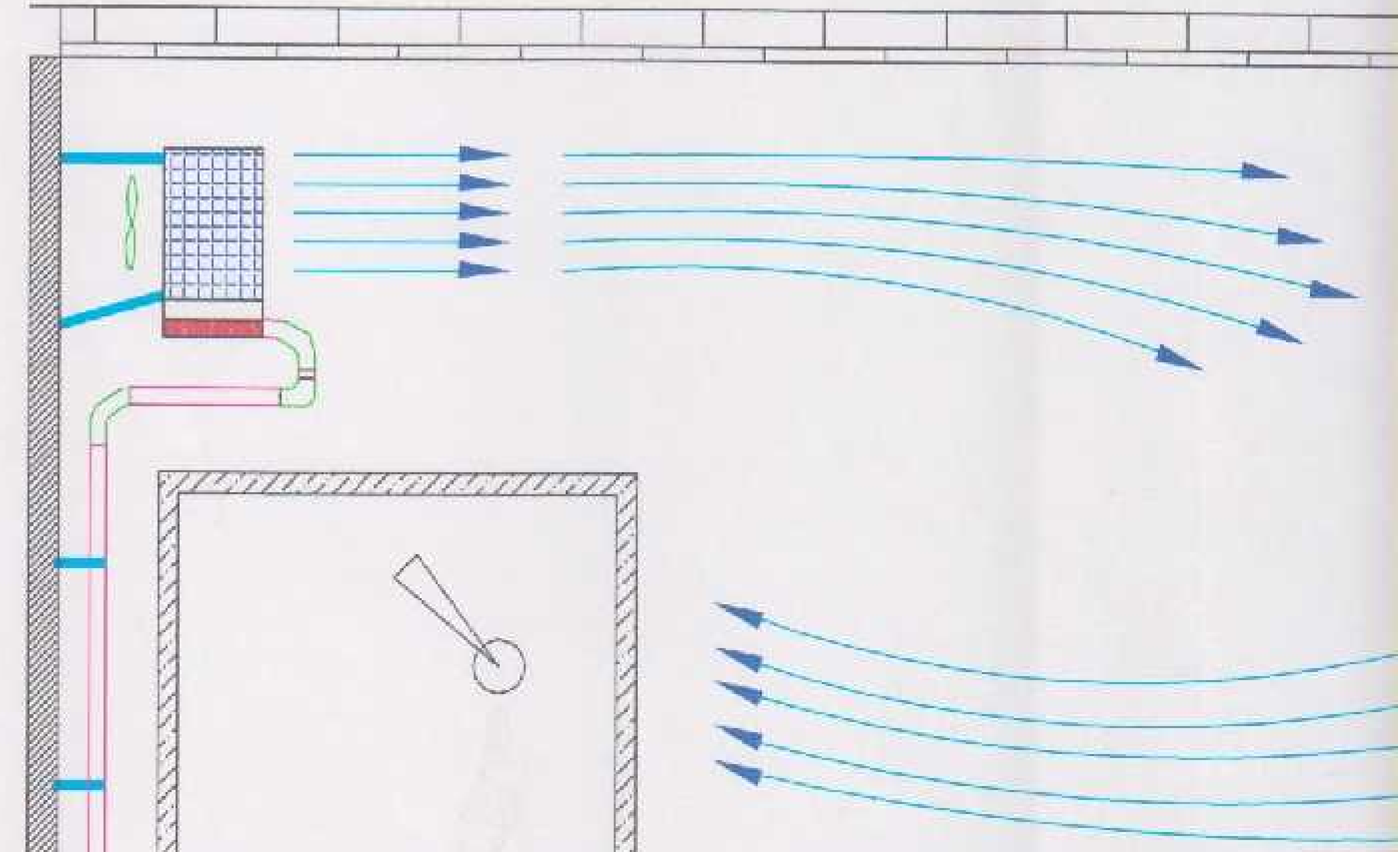
Cold Air Throw



6.3-3 Storing Rooms

The storing room must be designed to handle variable loads depending on the amount and type of product that needs to be stored. In our design for rooms assigned A, the storing rooms will also be designed to handle 100%, 75%, and 50% of its total capacity of product.

1 ➔ Compressor

The compressor will be selected depending on the cooling capacity that it should handle. Compressors of the refrigeration station are selected depending on the catalog of the DORIN Co.

The selected compressor is the same type that selected for the chilling room when working in the storing phase. The compressor characteristic declared below in Table (6-6).

Model C6015CC

380/420 V-AC
Gas R404A

Evaporating Temperature	-10 C
Condensation Temperature	45 C
Superheat	7 C
Subcooling	3 C

C6015CC		COMPRESSOR
Cooling Capacity	"kW"	18.70
Absorbed Power	"kW"	5.30
Mass Flow	"kg/h"	0.327
Absorbed Current	"A"	5.2
COP		2.35

Table (6-6) Operation Characteristics of Compressor Model

2 ➔ Condensation Unit

The condensation unit in the refrigeration system of the storing rooms also will be selected from DWM Copland selection software. After talking instruction form the manufacturer compressor can replace with compressors with same cooling capacity ±10%. The selection is taken at the desired cooling load at an ambient temperature of 35 C. and the model adopted was OLQ-33V-NLO-TWD. The parts and specification of this condensation unit is mentioned in Appendix 1.

3 ➔ Evaporator Unit

The evaporator unit of the storing room will be selected from the cooling units manufactured by FRIGA-BOHN Refrigeration Co. from the catalog of cooling units and following the manufacturer instructions, the model NKT 2X4D A3 T cooling unit will be selected. And the specifications of this cooling unit are shown in Table (6-7).

NKT _ T   T = large heat exchange surface

Models		NKT _ T	2x4D A2	2x4D A3	1x6D B2	3x4D A2	1x6D B3	3x4D A3	1x6D B4	1x8D C2	2x6Y B2	1x8D C3	2x6D B2	2x6D B3	2x6D B4
Capacity - R404A	DT1 = 7K - SC3	kW	8.00	9.91	10.63	12.10	13.15	14.91	15.26	16.12	19.03	19.99	21.64	26.75	31.00
	DT1 = 6K - SC4	kW	6.13	7.64	8.13	9.29	10.10	11.49	11.77	12.36	14.70	15.40	16.60	20.61	23.98
Surface		m2	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
Circuit volume		dm3	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
Fan	Num x Ø	mm	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
	Air flow	m3/h	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
	Air throw	m	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	42	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
Electric defrost		Num.	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
	400V	W	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
		A	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
Connections	Inlet	Ø	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
	Outlet	Ø	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
Net weight		kg	[illegible]	170	190	[illegible]	[illegible]	240	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]

Table (6-7) Indoor Cooling Unit Catalog

4 ➔ Expansion Valve

The expansion valve selection of refrigeration system in the storing rooms will be based on multi evaporator refrigeration. As mentioned, the storing rooms contain two indoor cooling units.

By the same way of selection and selection catalog, the selected expansion valve is (SE 4 S), as shown in Table (6-8). For further information refer to Appendix 1.

Specifications

Refrigerant	Refrigerant Designation	Capacity Range Tons	Internally Equalized Models	Externally Equalized Models	Rainbow Charges™	Inlet Connection ODF (Optional)	Outlet Connection ODF (Optional)	Equalizer Connection (Optional)
R-12	J	1/8 - 1/2	S 1/2J	SE 1/2J	W, X60	1/4 (3/8)	1/2 (3/8)	1/4" ODF (1/4" SAE)
R-134a		1/4 - 1	S 1J	SE 1J		3/8 (1/4)	1/2 (3/8)	
R-401A		1 - 2	S 2J	SE 2J		1/2	5/8 (7/8)	
R-401B		1 1/2 - 3	S 3J	SE 3J		1/2	7/8 (5/8)	
		3 1/2 - 5	S 5J	SE 5J		5/8	7/8	
R-402B	S	1/8 - 1/2	S 1/2S	SE 1/2S	W, Z, X110 X35	1/4 (3/8)	1/2 (3/8)	1/4" ODF (1/4" SAE)
R-404A		1/4 - 1	S 1S	SE 1S		3/8 (1/4)	3/8	
R-402A		1 - 2	S 2S	SE 2S		1/2	5/8 (7/8)	
R-502		1 1/2 - 4	S 4S	SE 4S		1/2	7/8 (5/8)	
R-507		4 1/2 - 6	S 6S	SE 6S		5/8	7/8	
R-22 R-407C	V	1/5 - 3/4	S 3/4V	S 3/4V	W, Z, X100 X35	1/4 (3/8)	1/2 (3/8)	1/4" ODF (1/4" SAE)
		1/2 - 1 1/2	S 1-1/2V	SE 1-1/2V		3/8 (1/4)	1/2 (3/8)	
		1 1/2 - 3	S 3V	SE 3V		1/2	5/8 (7/8)	
		3 1/2 - 5	S 5V	SE 5V		1/2	7/8 (5/8)	
		5 1/2 - 7 1/2	S 7 1/2	SE 7-1/2V		5/8	7/8	
		8 - 10	S 10	SE 10V		5/8	7/8	
R-410A	K	1/2 - 1 1/2	S 1 1/2K	SE 1 1/2K	KX200	3/8	1/2	1/4" ODF (1/4" SAE)
		1 1/2 - 3	S 3K	SE 3K		3/8	1/2	
		3 1/2 - 5	S 5K	SE 5K		1/2	5/8	
		5 1/2 - 7 1/2	S 7K	SE 7K		1/2	5/8	
		7 1/2 - 9	S 9K	SE 9K		5/8	7/8	

Table (6-8) Expansion Selection Table

4➔ Refrigerant Piping

By the same previous steps in the chilling room section:-

A) Liquid Line

The refrigerant velocity in the liquid line shouldn't exceed 3 m/s to prevent noise and large pressure drops. And no under limit since the refrigerant will solute the oil and carry it by the way. As shown in Figure (6-8); the refrigerant velocity is about 0.8 m/s at the recommended liquid line diameter from the condensation unit manufacturer.

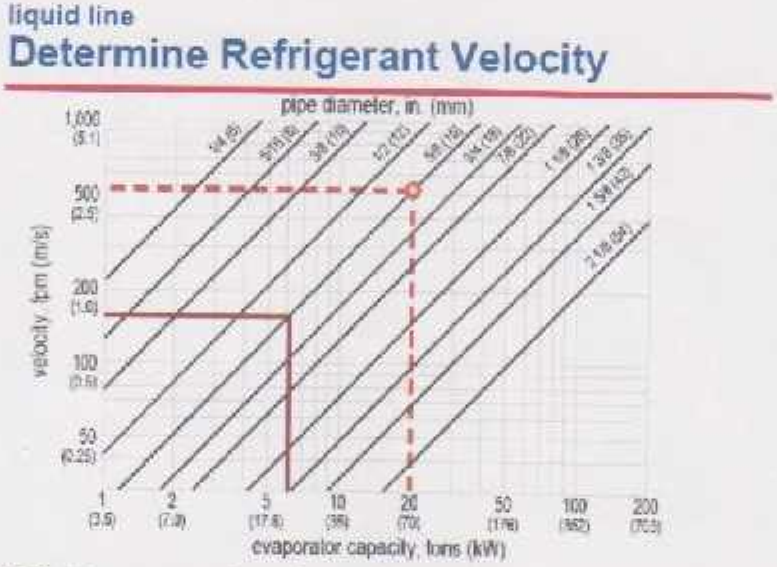


Figure (6-8) Evaporator Capacity and Velocity Diagram for Liquid Line

As mentioned, the refrigerant liquid line is recommended from DWM Copland for the liquid line to be 5/8 inch. This line will supply both evaporators with liquid refrigerant. (See appendix). And this diameter is around the recommended diameter for the liquid line obtained from the piping design process given in the TRANE piping design book. From figure (5-8) below, the pressure drop corresponding to this diameter is 4.6 kPa/10 m. that means that there is a pressure drop in the liquid line about 1.5 kPa. And additional 1.8 kPa for the elbows and accessories, Totally 3.3 kPa pressure drop prevents the liquid refrigerant from flashing in the liquid line; because flashing happens at a pressure of about 18 Bar. And the saturation pressure in the condenser is about 20.5 Bar.

Figure (6-9) Evaporator Capacity and Pressure Drop Diagram for Liquid Line

B) Suction Line:

The refrigerant velocity through the suction line has max limit of 20 m/s, more velocity causing vacuum and objectionable noise. And min limit depends weather the pipe is horizontal or upward or downward (see Table 6-9). Less velocity will cause the accumulation of lubricant oil resulting overheating of the compressor.

suction line

Minimum Allowable Velocities

pipe diameter, in. (mm)	minimum velocity, fpm (m/s)	
	riser	horiz/drop
3/8 (10)	370 (1.9)	275 (1.4)
1/2 (12)	460 (2.3)	350 (1.8)
5/8 (15)	520 (2.6)	390 (2.0)
3/4 (18)	560 (2.8)	420 (2.1)
7/8 (22)	600 (3.1)	450 (2.3)
1 1/8 (28)	700 (3.6)	525 (2.7)
1 3/8 (35)	780 (4.0)	586 (3.0)
1 5/8 (42)	840 (4.3)	630 (3.2)
2 1/8 (54)	980 (5.0)	735 (3.7)
2 5/8 (67)	1,080 (5.5)	810 (4.1)
3 1/8 (79)	1,180 (6.0)	885 (4.5)
3 5/8 (105)	1,270 (6.5)	950 (4.8)
4 1/8 (130)	1,360 (6.9)	1,020 (5.2)

Table (6-9). Min Allowable Velocities in Suction Line

The recommended suction line diameter is 1 3/8 inch from condensation unit manufacturer. From TRANE refrigerant piping book, the velocity of refrigerant in the suction line is more than 9.2 m/s (see Figure 5-9).for the vertical riser minimum allowable velocity is 4 m/s. So the refrigerant suction line will be designed to be 1 3/8 inch.

suction line

Determine Refrigerant Velocity

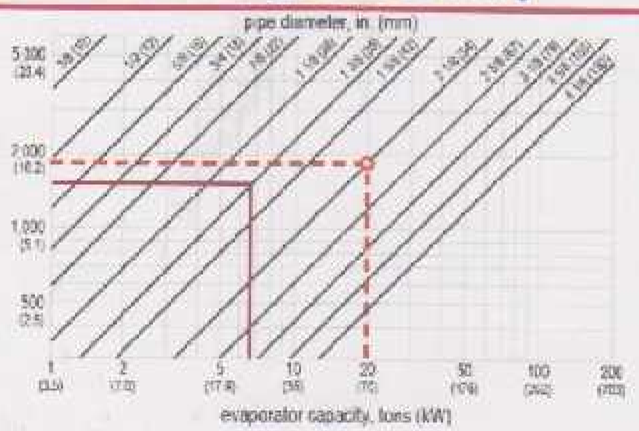


Figure (6-10). Evaporator Capacity and Velocity Diagram for Liquid Line

The amount of refrigerant charge depends on the instruction of the manufacturer. The total system design is clarified in the refrigeration civil plan. And same designs are for storing rooms B and C

6.4 Humidification System Design

The humidification system design will be designed to compensate the moisture loss caused by the cooling unit frosting, and the air change of the refrigeration room. This process needs calculation to determine the mass flow rate for the steam needed to compensate the moisture loss.

The moisture loss due to the air change by ventilation is calculated as follows:

$$\text{Moisture Demand Rate} = V^{o}\,(\omega_{in} - \omega_{out}) \qquad (5\text{-}1)$$

From psychometric chart and obtaining needed values:

= 0.65 kg steam/ h for rooms assigned A, and stared room.
= 0.60 kg steam/ h for rooms assigned B.
= 0.56 kg steam/ h for rooms assigned A.

By accumulating results:

Q^{o}_{steam} = 6.2 kg steam/h. ➔ So the required humidifier capacity = 6.2 kg steam/h.

The selection of the humidifier system will be done depending on the recommendations and products of NEPTRONIC Co. which is a company specialized in the steam and water spraying systems.

The selected steam generator is NEPTRONIC SKE-05M… (See Figure 6-11).

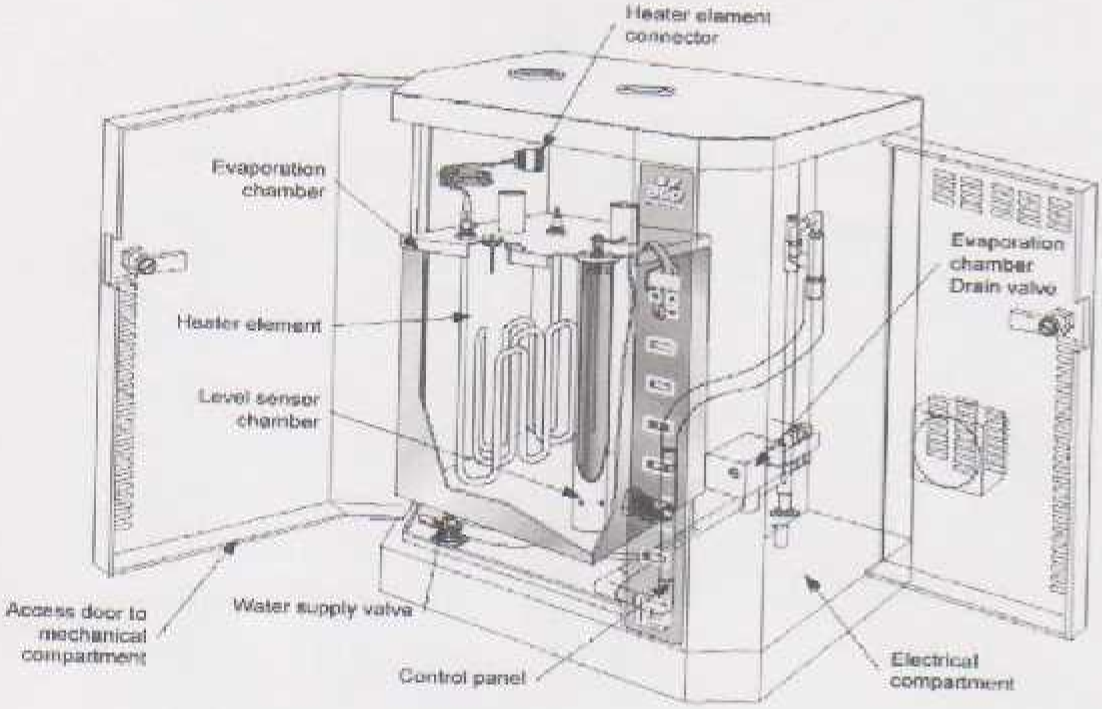


Figure (6-11) Humidifier Construction

Which its data as followed:

Model		NEPTRONIC SKE-05(M)
Capacity	"kg/h"	5
Voltage	"V"	400 V 3~
Power	"kW"	3.7
Current	"A"	5.5
Steam Outlets	"mm"	1 X 35 mm

Table (6-10) Some Performance Characteristics
See Appendix for future information

This humidifier also has the following characteristics:

1- Steam with pressure of 2.2 Bar.

2- Outlet steam temperature 120 C.

The steam pipes pressure should be from 1.5 – 3 Bars to enable high spraying performance, and the steam water hardness from 1 – 750 P.P.M. pressure relief valve is set at 3.5 bar. And water should be supplied at 1 – 4.18 bar.

The steam is ejected by steam pipe ejector (See Figure 6-12). The steam ejector for NEPTRONIC ejector must be at 8 mm. and operates in the pressure rang from 1 – 3 bar gauge.

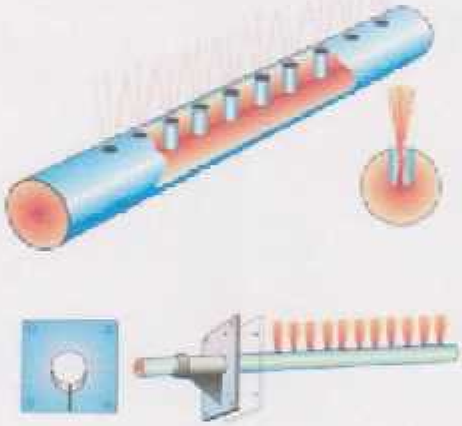


Figure (6-12) Steam Ejector

The steam ejectors are installed in front of the indoor Evaporators, i.e. the cold air is mixed with the superheated vapor and absorbs its moisture content.

The humidification system piping is shown in the Humidification plan. The corresponding diameters are:

1- The main steam supply 35 mm.
2- Each branch at 16 mm.
3- The ejector pipe is 8 mm.

The humidification stops when the refrigeration process stops or by control signal for saturation.

System Accessories

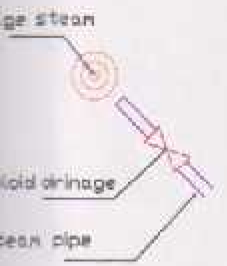
The system accessories are:

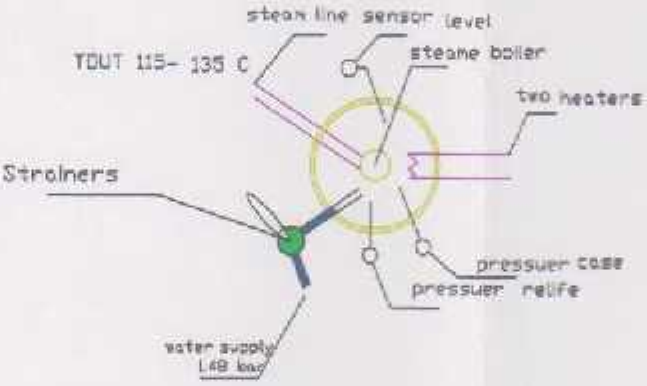
➔ <u>Solenoid Valves</u>: For humidification control.

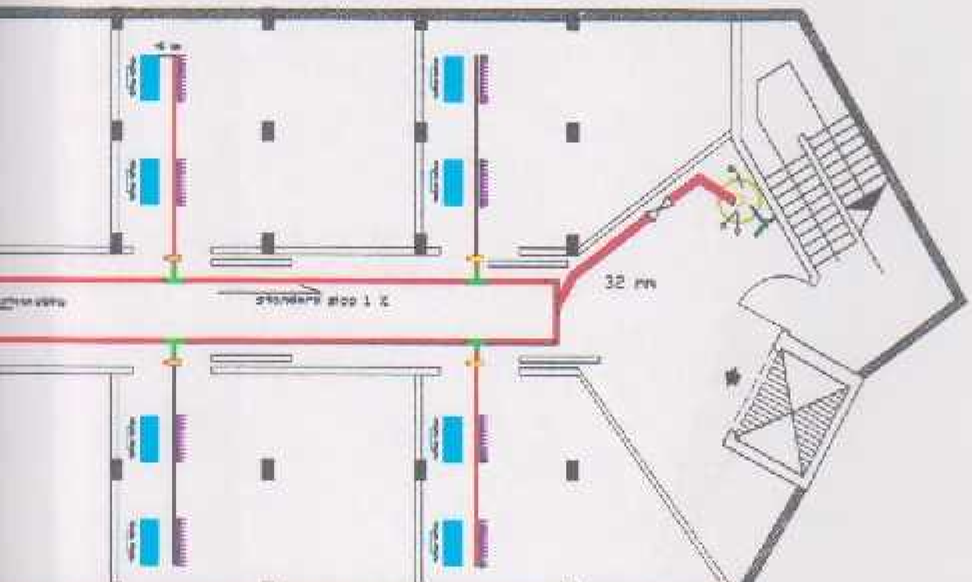
➔ <u>Strainer</u>: For collecting dirt and copper corrosion dust protecting the steam nozzles.

➔ <u>Pressure Relief Valve</u>: For protection from high pressures.

on System







GENERAL NOTES

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Project

Refrigeration Station -Halhul

Sheet Title

Humidification system Plane

Senior Project

Date

2006-2007

Sheet

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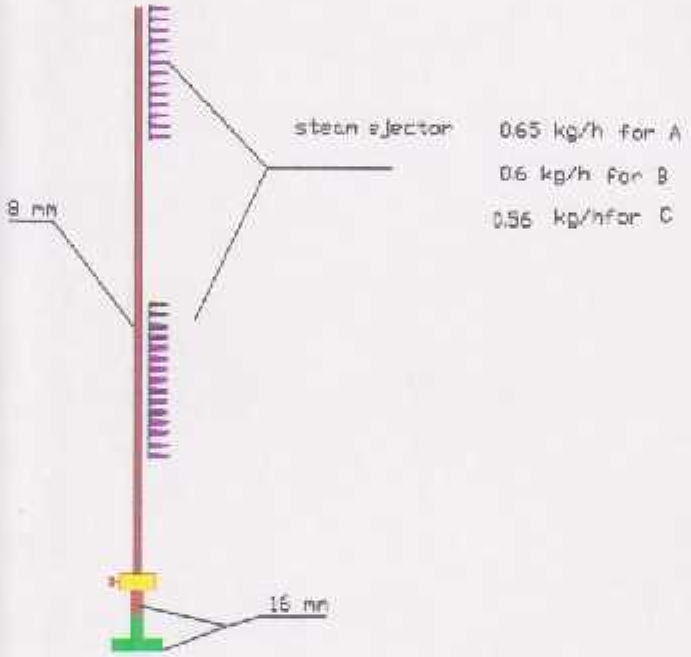
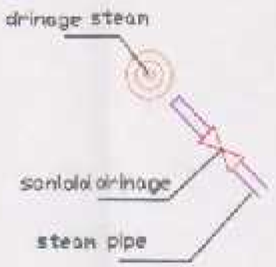
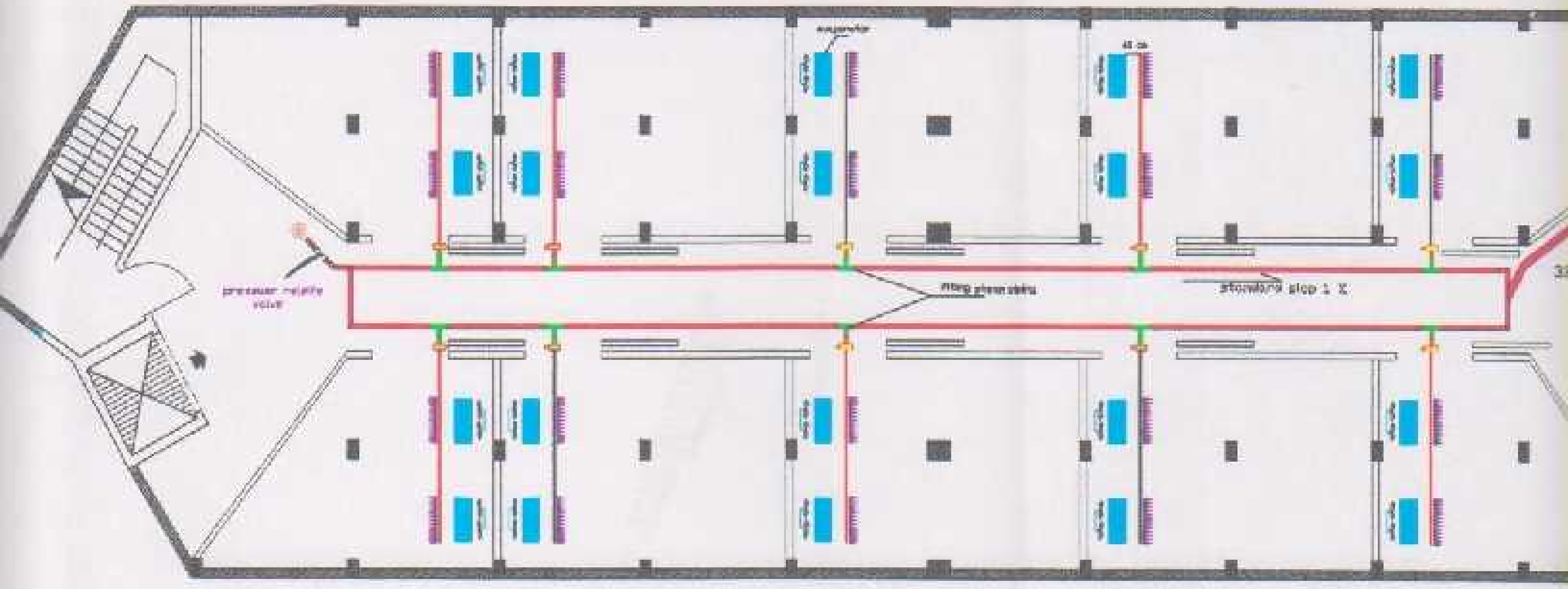
Humidification System







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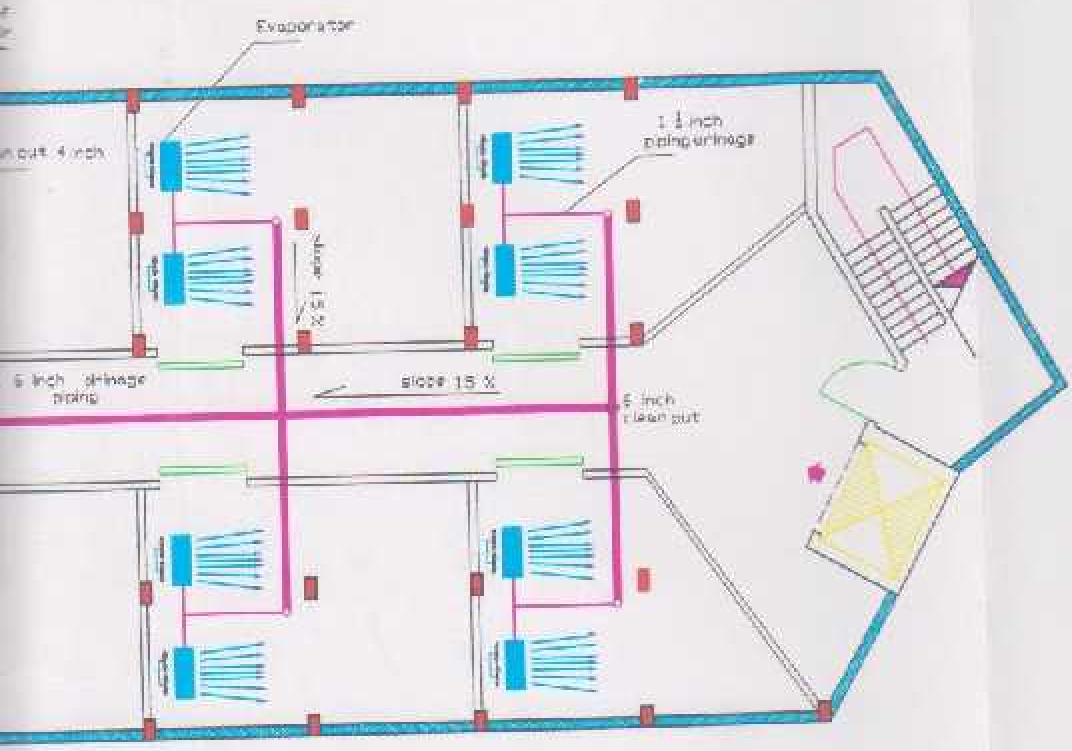

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Project

Refrigeration Station -Halhul

Sheet Title

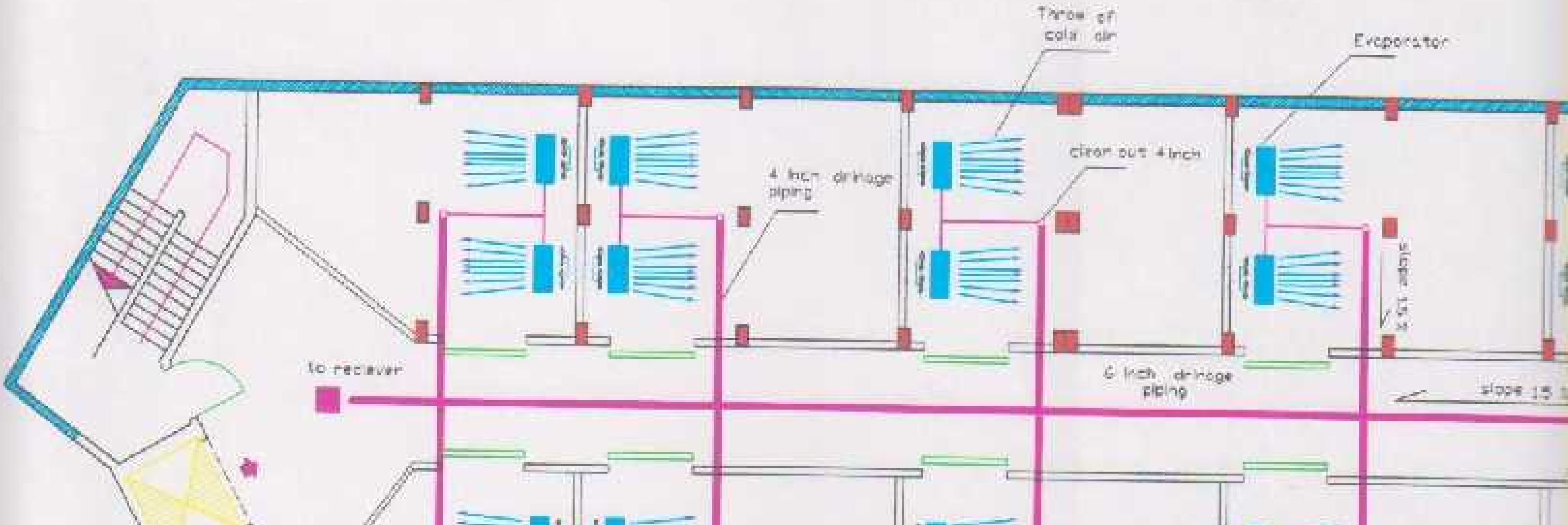
Top View for Drainage system

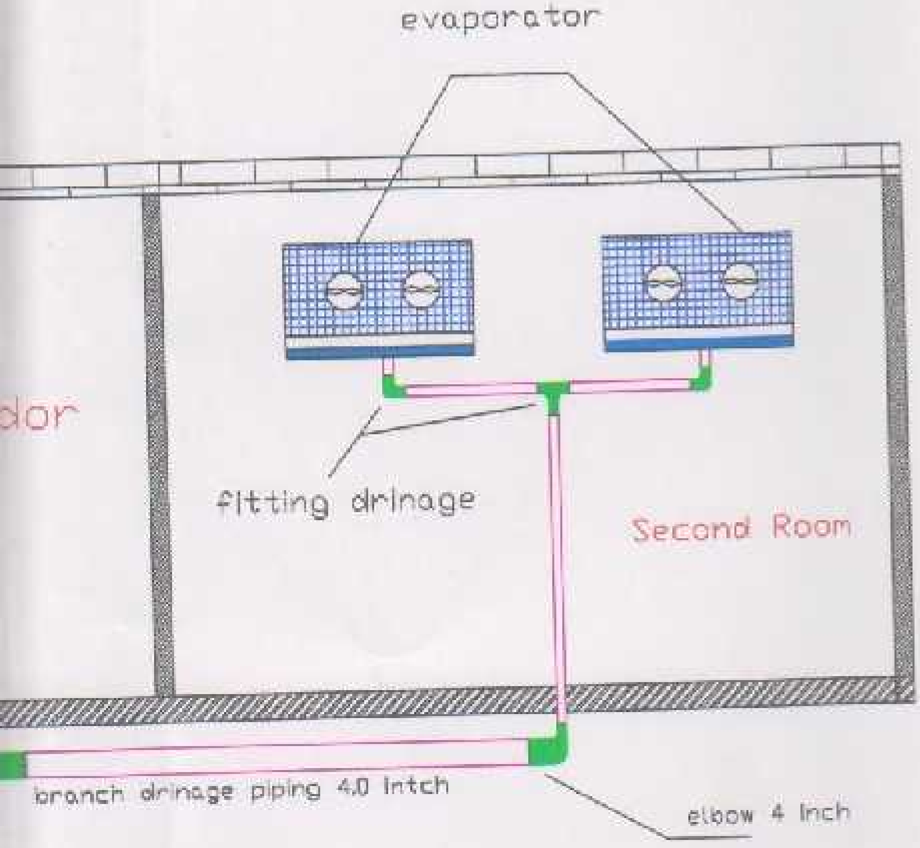
Senior Project

Date
2006-2007

Sheet
A-3

Drinage system piping





GENERAL NOTES

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Refrigeration Station -Halhul

Drinage Plane

Senior Project

Date 2006-2007

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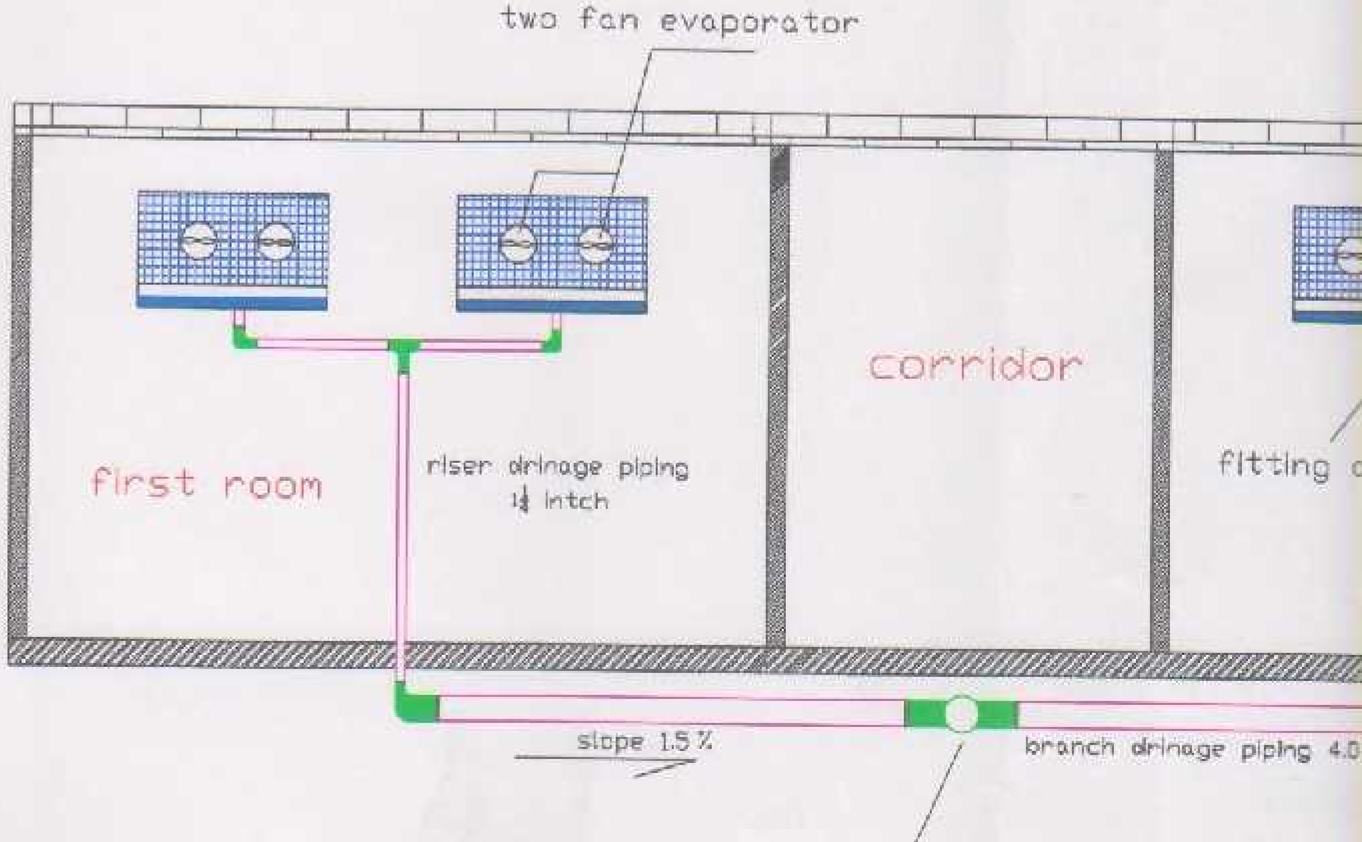

two fan evaporator
corridor
first room
riser drinage piping
1½ intch
fitting c
slope 1.5 %
branch drinage piping 4.0
T fitting 6 inch

6.5 Air Handling System Design

The Air Handling system of the refrigeration station is divided into two main parts; the ventilation system, and air curtains. Each system will be designed referring to the required characteristic of air handling.

6.5-1 Ventilation System

The ventilation system for the refrigeration station rooms is designed to get rid of the harmful gases generated by the products in the refrigeration room; which are:

1- CO_2, which causes the high reduction of the metabolic activity of product; which in turn causes the product dieing and changing of its color.

2- The alcoholic organic gases; such as ethylene and Aldehydes, which causing the high repining rate of the product and consumes its nutritional content.

3- O_2, which accelerates the metabolic activity of the products, which in turn consumes its nutritional content.

The ventilation process is designed and controlled such that when the refrigeration system starts the defrosting process, during defrosting; the gases mentioned above will go down under the effect of its high density and molar mass; and then, they will be drawn out by the ventilation system of the room from ground vents.

The ventilation system of the refrigeration station rooms consists of:

A) Drawing fan.

B) Drawing ducts.

C) Louvers to supply make up air, electrically controlled.

A) Fan Selection:

Two main parameters are necessary to select suitable fan:

a- The required cfm.

b- The Static pressure.

There are two main types of fans; centrifugal and propeller fans. Propeller fans provide an economical method to move large air volumes (5,000+ cfm) at low static pressures (0.50 in. or less). Motors are typically mounted in the air stream which limits applications to relatively clean air at maximum temperatures of 110°F.

Centrifugal fans are more efficient at higher static pressures and are quieter than propeller fans. Many centrifugal fan models are designed with motors mounted out of the air stream to ventilate contaminated and high temperature air. And for our application; a centrifugal fan is suitable.

➔CFM Calculation:

The required number of air change for rooms assigned A is 9/ 24 h.

$$V_{room} = \text{Width}*\text{Length}*\text{height}$$
$$= 8.3*5.85*4$$
$$= 194 \text{ m}^3$$

By subtracting the product volume ➔ $V_{room} = 112 \text{ m}^3$

This volume should be changed 9 times during the day (refer to chapter 5, ventilation load), but because the system just ventilates the room during defrosting process; the total ventilation time is 4 h.

So:

Required $V^{o} = \dfrac{N_o * V_{room}}{4} = 350 \text{ m}^3/\text{h}$ ➔ 321 cfm.

➔Static Pressure PS:

The pressures generated by fans in ductwork are very small. Yet, accurately estimating the static pressure is critical to proper fan selection. Fan static pressure is measured in inches of water gauge. One pound per square inch is equivalent to 27.7 in. of water gauge. Static pressures in fan systems are typically less than 2 in. of water gauge, or 0.072 psi. The drawing below illustrates how static pressures are measured in ductwork with a manometer (see Figure 6-13).

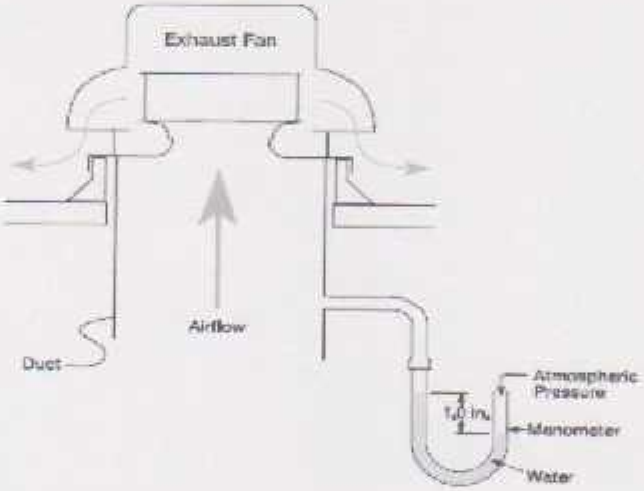


Figure (6-13) Pressure Measuring in Ducts

A pressure differential between the duct and the atmosphere will cause the water level in the manometer legs to rest at different levels. This difference is the static pressure measured in inches of water gauge. In the case of the exhaust fan at right, the air is being drawn upward through the ductwork because the fan is producing a low pressure region at the top of the duct. This is the same principle that enables beverages to be sipped through a straw. The amount of static pressure that the fan must overcome depends on the air velocity in the ductwork, the number of duct turns (and other resistive elements), and the duct length. For properly designed systems with sufficient make-up air, the guide lines in the Table (6-11) below can be used for estimating static pressure:

STATIC PRESSURE GUIDELINES	
Non-Ducted	0.05 in. to 0.20 in.
Ducted	0.2 in. to 0.40 in. per 100 feet of duct (assuming duct air velocity falls within 1000-1800 feet per minute)
Fittings	0.08 in. per fitting (elbow, register, grill, damper, etc.)
Kitchen Hood Exhaust	0.625 in. to 1.50 in.
Important: Static pressure requirements are significantly affected by the amount of make-up air supplied to an area. Insufficient make-up air will increase static pressure and reduce the amount of air that will be exhausted. Remember, for each cubic foot of air exhausted, one cubic foot of air must be supplied.	

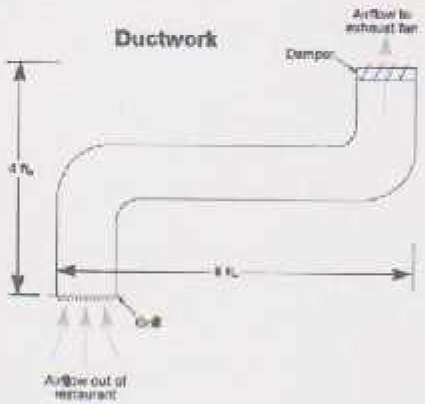


Table (6-11) Static Pressure Due to Fittings

Depending on Table (6-12) above, the duct static pressure 0.035 in, adding three elbows equals to 0.24 in, totally about 0.28 – 0.30 mm water.

The fan selection will be obtained from GREENHECK Ventilation Co. from the catalog of the centrifugal fans (see Table 6-13) the suitable fan is model SFD 6-6B at 0.25 in. direct drive. (See Figure 6-14).

PERFORMANCE DATA

Model SFD	RPM	HP	Static Pressure in Inches of W.G. .125		.250		.500		.750		1.00		1.25		1.50		2.00		2.50	
			CFM	BHP	CFM	BHP	CFM	BHP	CFM	BHP	CFM	BHP	CFM	BHP	CFM	BHP	CFM	BHP	CFM	BHP
6-6B	1140	1/6	407	.06	339	.05														
6-4A	1725	1/4	667	.24	639	.22	541	.17	424	.13										
7.5-6B	1140	1/6	672	.16	612	.14	467	.10												
7.5-5A	1725	1/2	1062	.59	1026	.56	940	.50	859	.46	775	.38	636	.30						
9-4C	860	1/4	830	.16	746	.15	487	.09												
9-6B	1140	1/2	1159	.48	1097	.42	957	.34	782	.26										
9-15A	1725	1 1/2	1806	1.64	1765	1.59	1663	1.46	1595	1.36	1502	1.24	1407	1.13	1298	1.01	969	.70		
10-3C	860	1/3	1259	.38	1176	.32	995	.25												
10-7B	1140	3/4	1713	.88	1653	.82	1526	.72	1378	.64	1163	.52								
10-30A	1725	3	2641	3.09	2609	2.99	2525	2.87	2444	2.73	2362	2.60	2275	2.44	2179	2.32	1942	2.01	1621	1.62

Performance certified is for installation Type B - Free inlet, Ducted outlet. Performance ratings do not include the effects of appurtenances (accessories).

Table (6-12) Fan Selection Chart Depending on Static Pressure and cfm

Figure (6-14) Selected Fan

B) Duct Sizing:

The duct sizing depends on the air flow and the recommended velocity. For the duct work in the industrial applications; the sound is not an important parameter for the design process. And suitable duct sizes are selected for application. The main parameters in our system design are the velocity of air and restricted space for the duct work.

For industrial ventilation systems; recommended velocities are about 10-15 m/s in open areas (non-ducted) and in ducted ventilation the velocity is recommended about 6-8 m/sec. 7 m/sec duct velocity will be considered as a design value, and when duct lengths are short; the effect of pressure drop on duct sizing diminishes.

Two air risers from both sides of the central column will be placed. Each riser draws air from its side of the room. And then draw all out. See the station air handling design plan.

1- Main Duct:

$V^o = 5.83\ m^3/sec$

By Continuity Equation $V^o = v*A$ ➔ $A_{main} = 0.63\ m^2$

And when $A = \Pi/4\ D^2$ ➔ $D_{main} = 0.7\ m$

And the Equivalent Rectangular duct is 60X35 cm

2- Risers

$V^o = 2.915\ m^3/sec$

By Continuity Equation $V^o = v*A_{main}$ ➔ $A_{main} = 0.32\ m^2$

And when $A_{main} = \Pi/4\ D_{main}^2$ ➔ $D_{main} = 0.35\ m$

And the Equivalent Rectangular duct is 42X35 cm

All is shown clearly on the air handling system plan.

6.5-2 Air Curtain System

Air Curtains are used for a wide variety of applications from thermal barriers at shopping mall entrances, to fly and insect prevention at restaurant service doors, to factory freight doors for wind resistance and temperature separation. The air curtain enables traffic to flow unobstructed through openings while maintaining distinct environments, thus resulting in energy savings.

Air curtains provide environmental separation for open doorways. This is achieved without the drawbacks of visibility-limiting solid barriers such as strip and impact doors. Air curtains reduce the risk of accidents by allowing vehicles and people to move through without obstructions to visibility or movement. While allowing this freedom of movement through doorways, air curtains minimize the natural convection flow of air, provide excellent resistance to the penetration of outside winds, and prevent the intrusion of flies and insects, fumes and dust.

Air curtains also reduce mechanical door maintenance costs, and are more aesthetically appealing than plastic strip doors or high- speed doors.

6.5-2.1 Importance of Air Curtains

When dividing spaces with temperature differences, the air curtain prevents the natural flow of air caused by differing temperatures. The difference between indoor and outdoor temperature creates an imbalance in density, and therefore a pressure variation which causes the infiltration of outside air.

Figure (6-15) illustrates the loss of conditioned air before and after the installation of an air curtain at the doorway. As an example, during the winter heating season, the outside colder denser air flows through the bottom half of the doorway into the nterior of the building; The warmer, lighter air flows to the outside through the top half of the open doorway. For large openings, this results in excessive energy losses.

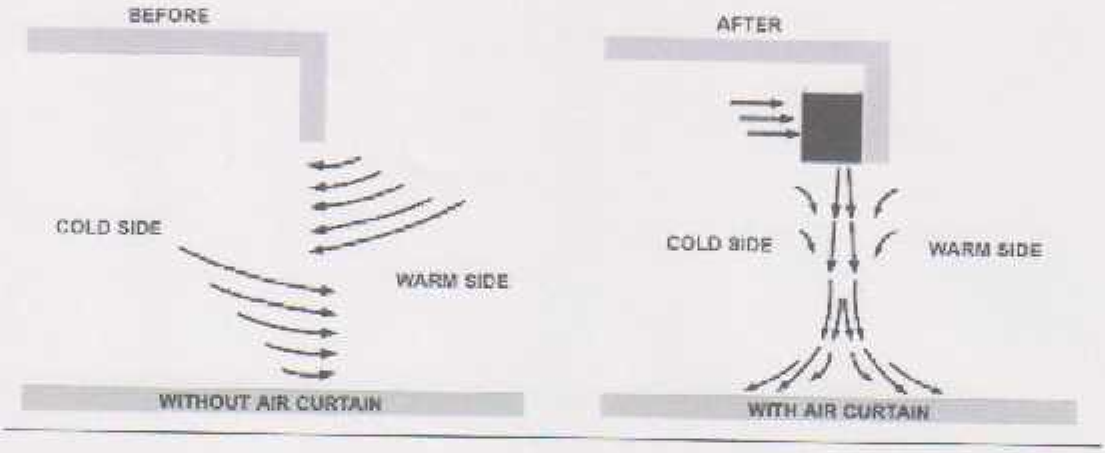


Figure (6-15) Air Curtain Working Principle

6.5-2.2 Air Curtain Selection

The air curtain is selected depending on type of application, and door dimension. The application here is to obtain a thermal barrier for the refrigeration rooms when opened. And the door dimensions are 190 cm height X 220 cm width (6' height X 86" width).

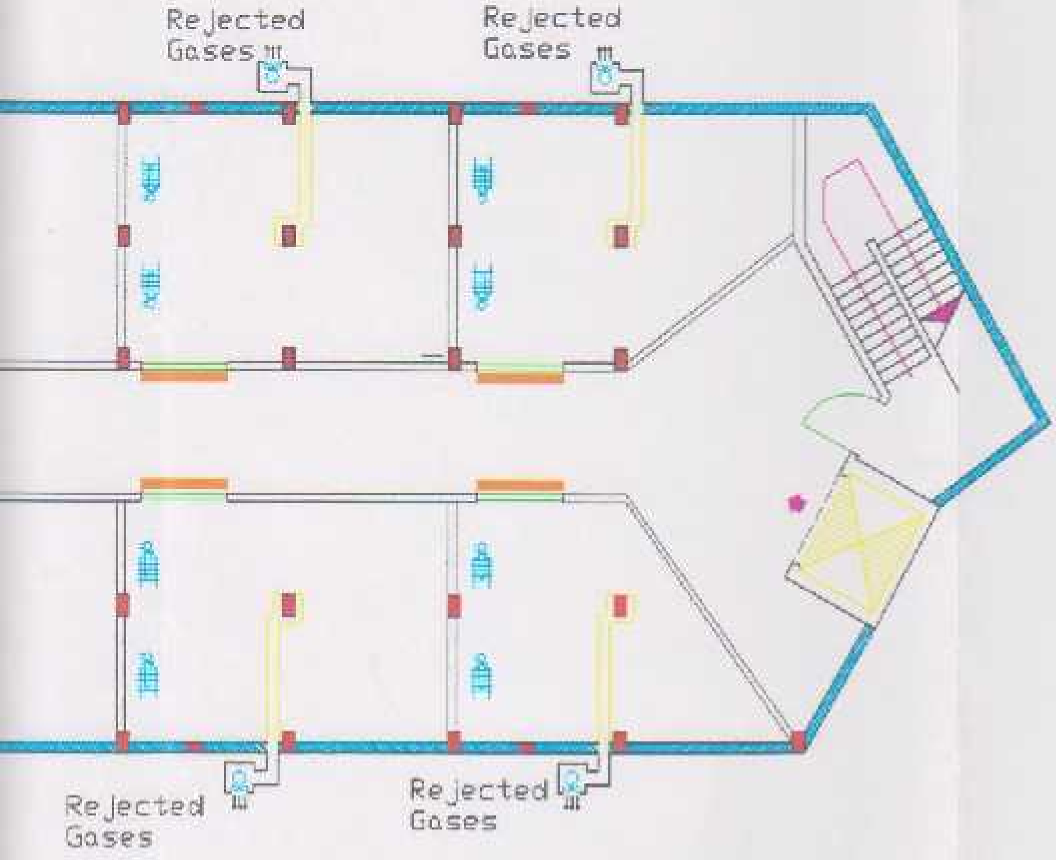
The air curtains are selected depending on Leading Edge Co catalog. From Table (6-13), the fan power can be selected, and the selected model is E9600-2125. For further information refer to the product catalog in appendix 1.

NOTE: Guide selection is based on refrigerated areas within facilities. Air curtains must be mounted on warm side. For exterior loading dock applications consult factory.
Selection guide does not include negative pressure or wind tunnel conditions.

AIR CURTAIN SELECTION GUIDE

Mtg. Ht. (Ft.)	Thermal Barrier Applications Type of Door	1/8 HP**	1/4 HP	1/2 HP	3/4 HP
7'	WALK-IN COOLER	✔			
8'	COOLER		✔		
10'	COOLER			✔	
12'	COOLER				✔
8'	FREEZER			✔	
10'	FREEZER				
12'	FREEZER				✔

Table (6-13) Motor Rating Selection for Air Curtain



GENERAL NOTES

Palestine Polytechnic University

Mechanical Eng. Department

Designed By:
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Supervisor :
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Project

Refrigeration Station -Halhul

Sheet Title

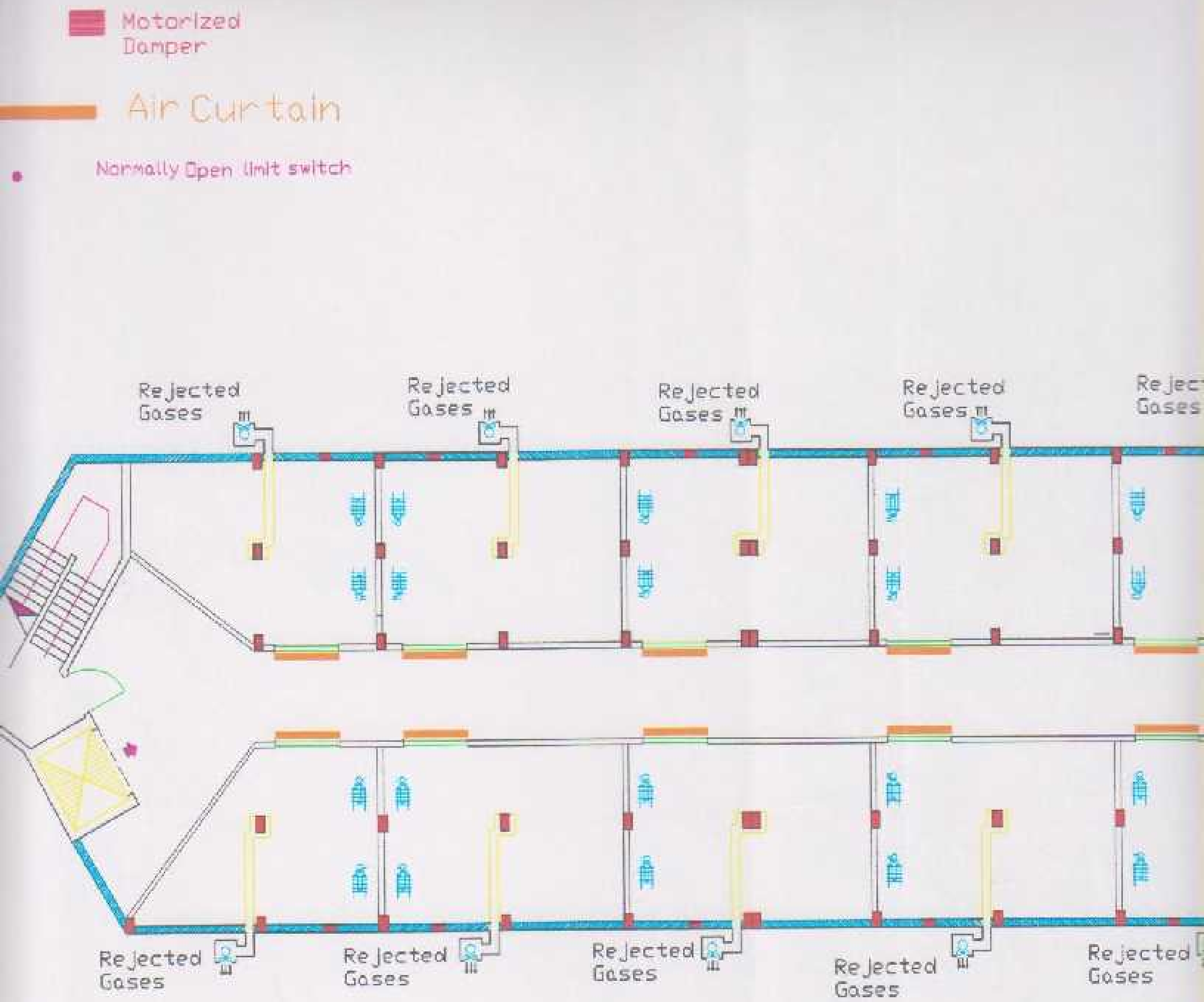
Air Handling System

Senior Project

Sheet

Date

2006-2007



Motorized
Damper
Air Curtain
Normally Open limit switch
Rejected
Gases
Rejected
Gases
Rejected
Gases
Rejected
Gases
Rejected
Gases
Rejected
Gases
Rejected
Gases
Rejected
Gases
Rejected
Gases

6.6 Control System Design

The control system main function is to synchronize and regulate the performance of the other systems of the refrigeration station. Such that it works on accomplish specific conditions and the way to accomplish these conditions. However, any control system what ever its type consists of three main units:

1- Inputs.
2- Data Processing.
3- Outputs.

The control systems in the refrigeration station are classified into two main categories: built-in control, this type of control is embedded in the systems itself. And the PLC control system. That implies the control requirements of the refrigeration station.

In this section of the system design of the refrigeration station; the control principles and parameters will be discussed and based on the PLC, and finally describes the control system of the refrigeration station.

6.6-1 Refrigeration System Control

The refrigeration system of the refrigeration station must work in specific way to achieve its best performance and obtaining the desired results of temperatures. The control system must achieve the following working characteristics:

- All system is working for 50 min in the cooling phase, and the following 10 min are for the ice defrosting. And this is accomplished by the built-in control of the refrigeration unit.

- The system stops working on high or low pressures at the compressor discharge and suction line. This is accomplished also by the built-in control of the condensation unit.

- Oil level and temperature are automatically causing the shut off the cycle.

- The system takes its temperature reading from the thermometer that is installed at the inlet of the indoor cooling unit. And depending on this temperature, the system continues refrigeration process until reaching temperature set point. The temperature range in ± 2 C.

Also the VSD compressors should be controlled and wired accurately, the working specification of the VSD compressors as follows:

- Firstly, when the product is placed in the chilling room, three Buttons should be pushed on, one for the product type, one for the quantity 50, 75, or 100%. And the last one for initiating the chilling room performance. Then the PLC sends the compressor operating frequency signal. Then the compressor will start at the suitable discharge rate, and then its velocity decreases linearly with the retaining Temperature-PID signal until the stop of the refrigeration system. The last frequency of the compressor then saved to be initiated from next cycle of operation.

- After the chilling process is accomplished, the chilling compressor valves are shut off and the storing compressor starts working.

- Any failure of any compressor starts the standby one automatically.

- All compressor works with the same working principle.

This control satisfies the safety of performance and reducing the costs of operation. The total system can be efficiently designed by automation engineers; who are specialized in control engineering.

6.6-2 Humidification System Control

The humidification system also should be controlled to obtain the set point of humidity required, by same way; the required specification of working is:

- The pressure in the steam pipes must be maintained at 2.8 bar (operating pressure from manufacturer), such that the humidity is modified by opening the room solenoid valve. Enabling the steam flow to the room increasing its relative humidity.

- The humidification runs only during the refrigeration process in the refrigeration station room. To prevent the condensation of the steam before be absorbed by the room envelope air.

- The humidity range of performance is +- 5%, the humidistat position is at the inlet of the indoor cooling unit. When the system reaches the desired set point. Automatically the humidification valve of the room is shut off. And the humidifier when reaching the operational pipe pressure; it stop working and stands by.

6.6-3 Air Handling System Control

This system consists of the air curtains, and the ventilation system of the rooms. The performance specifications are:

- The air curtains are only operating when the door of the room is opened. Such that when the door is opened, a limit switch runs the air curtain to prevent air leakage from refrigerated space.

- The ventilation process is initiated with the initiating of the defrosting process. And ends with its end, to pump out the alcoholic gases and CO_2 from refrigerated space.

The total system design should be designed and implemented by an automation engineer. With accompany with refrigeration engineer.

Chapter Seven

REDESIGN PROCESS
EVALUATION
&
STATION FAILURE
CONCLUSIONS

7.1 Overview

All refrigeration systems work depending on specific characteristics; such that the evaluation process is based on these operation characteristics. After finishing the redesign process of the refrigeration station; these new designs should be evaluated and compared with the existing designs of the refrigeration station, showing the new operation characteristics that the new designs distinct from the existing designs.

7.2 System Evaluation

The most suitable way to perform this evaluation, to take the station systems – the existing and new systems– and comparing them with the right engineering approach to design desired system, and showing the new accessories and systems installed in the new design and showing its necessity. Firstly, we will start with the refrigeration system evaluation, and continue the evaluation of the other systems.

7.2-1 Refrigeration System Evaluation

The refrigeration system design of the refrigeration station will be taken firstly with consideration. The new design of this system distinct with the following characteristics:

1- Multiple Cooling Capacities: The refrigeration design can handle a wide range of the cooling load demands, such that the system is designed to start its working at a starting cooling load taken from control signal to the VSD compressor. And then the system cooling capacity decreases along with the decreasing of the cooling load demand. In case of the chilling room, the end of chilling process shuts of the chilling compressor and starts the storing compressor to complete refrigeration in the storage phase.

This principle has concurrent with the system requirements. For example, energy savings since the system will absorb less energy, also the increase in compressor reliability; thanks to the significant decrease of compressors startups; infect a standard system stops when it has reached the desired temperature and then suddenly re-start when this temperature has increased. On the contrary VSD systems reduce their refrigeration duty while approaching the desired temperature; theoretically the compressor can even never stop working, continuing to give the minimum duty that is enough to keep the ambient constantly cold. All of these characteristics are missing in the existing refrigeration design, which only operates as on/off operation and one rated cooling capacity.

Also, this enables the product handling at various quantities and types. In this system the cooling capacity that the systems serve is around the cooling capacity for the desired product quantity with small difference. This in turn also serves more energy than the normal system installed in the existing design.

2- Uniformity of Cooling Process: The new system distinct with its uniform cooling performance of the cooled products. This is referred to the way of installation of the indoor cooling units. Their positioning in one direction along the wall near the door facilitates their refrigerant piping and enables the uniform circulation of the cold air inside the refrigeration room. That in turn, implies the uniform temperature distribution in the refrigeration room and ensures the cooling of all product quantity uniformly.

This was a problem in the existing design. The evaporator was installed in the corner of the room. This doesn't imply the uniform cooling process of the product. And then affects on the product quality.

3- More Precise Monitoring: The temperature sensor in the refrigeration room in the new design is positioned at the air inlet of the indoor cooling unit. This gives the right temperature measuring of the refrigeration room temperature. This adjusts the working of the refrigeration unit in the correct way of working.

In the existing design of the refrigeration system, the temperature sensor is positioned on the top of room door, this causes the temperature sensor to read a significant raise in the room temperature every time door in opened. And so, effects wrongly on the refrigeration system performance.

4- Chilling Availability: In the new refrigeration design, the design considered the building of a refrigeration room with special chilling characteristic. To be used for chilling the product just it arrives to the refrigeration room. This room is very important to save the product quality and prolong the product life.

The existing refrigeration design ignored the design of this room absolutely. It just designed all the rooms as a storing room.

5- The Standby Units: multiple standby compressors are included in the new refrigeration system design. Such that any failure in the compressor performance, causes its stop working and closing the solenoids supplying it and initiates the stand by compressor implemented within the system. This protects the products from the severe damage referred to the temperature raise after cooling of the products.

The existing refrigeration design doesn't consider the standby compressors in its design, which may cause high cost losses if compressor performance decays suddenly.

7.2-2 Humidification System Evaluation

Now the humidification system will be taken by evaluation, comparing it with the existing humidification system of the refrigeration station.

1- Sized Humidification Capacity: The rated humidifier capacity in the new system design is 5 kg/h and rated power of 3.7 kW. This capacity is suitable for the system humidification process.

Where as the rated capacity of the existing design is not known, and the rated power is 10 kW. This means high energy consumption without any engineering objectives.

2- Piping and Steam Injection: The new humidification design used insulated piping for the humidification system. Provided with solenoid control valves and strainers to control and purify for the steam before entering the steam ejectors. The ejecting process enables the maximum efficiency of steam absorption. Such that it prevents the steam condensation of the steam on the product.

In the existing humidification the no strainers are implemented in the steam piping, and this may cause the piping blockage due to piping corrosion. And the steam injection is done on wrong way that may cause the steam condensation on the products causing their rot.

3- Process Timing and Set Point: in the new design of system, the timing of the humidification process is synchronized to operate in the refrigeration operating phase. To ensure the uniform humidification at the desired set point and prevents the condensation of steam on the product.

In the existing design, the humidification is performed any time the humidity drops down the whole only set point of the system. This means that systems will achieve more or less humidity in insufficient and correct way.

7.2-3 Air Handling System Evaluation

Also the air handling system distinct with the following characteristics:

1- Air Leakage Control: the system includes the installation of air curtains. These operate during the door opening. To decrease cold air leakage to minimum. This decreases the air infiltration load. This in turn, reduces the required initial and running cooling costs.

The existing design of the station doesn't include any air curtains to prevent air leakage.

2- Ventilation Availability: a ventilation system is installed in order to get rid of alcoholic gases and reduces the high concentrations of CO_2. Which distinct the system with the ability of controlling the gaseous content.

The existing station design doesn't include this system absolutely, that means the deterioration of the products because of the repining gases.

3- Process Timing: the process timing is during the defrosting process of the refrigeration system. To pump high density alcoholic gases from the bottom of the refrigerated space.

7.3 System Failure Conclusions

After carrying out all the previous search and design process, total point of view can be formed describing the causes of the system failure in the desired objective.

The system failure is referred to two main types of errors; either operational error, which is referred to the station system design and working itself. And preservation rules errors, which referred to the errors happening during the agricultural preparation of the products to the cooling stage. Each type will be briefly discussed and analyzed.

7.3-1 Operational Errors

Many operational errors was detected in the refrigeration station design, they are as following with respect to the system:

A) Refrigeration System Design:

1- The evaporator positioning provides non-uniform cooling process. This affects on the biological response of the product, this in turn causing part to be cooled more rapid than others.

2- The temperature sensor positioning over the door gives erratic temperature sensing for the refrigerated space, which in turn affected on the refrigeration performance and caused chilling injury of the products.
3- The absence of chilling room design, all the rooms are designed to handle just in storing phase only. So, the products cooled were chilled at very long time exceeds the permitted time limit. And this caused the product damage due to product deterioration until reaching store temperature.

B) Humidification System Design:

The humidification piping wasn't insulated, this caused power loss. And the other operation characteristics are not clear.

C) Air Handling System Design:

No air handling system was implemented in the station design. Which causes the deterioration of products due to ethylene and

7.3-2 Product Preparation Errors

1- The refrigeration rooms were not sanitized and ventilated before the product cooling
2- The product wasn't classified and prepared for the cooling process
3- The product delay time wasn't taken in consideration while transferring to the cold storage.
4- The product was put irrationally and more than one type in the same room
5- The refrigeration system wasn't designed depending on specific requirements, and so not all the products cooled save their quality.

Appendix One

Selection Catalogs

1- Chilling Room Compressor

DORIN

Calcoli Su Modello

Gas R404A | Modello H8000CC

Tensione / fasi / frequenza | 380-420 V / 3 / 50Hz

Condizioni di funzionamento

Rugiada (dew temp.)

Temperatura evaporaz.	°C	-10.0
Temperatura condensaz.	°C	45.0
Surriscald. uscita evap.	K	7.0
Sottoraffredd. del liquido	K	2.0
Surrisc. in aspirazione	K	7.0

Risultati

H8000CC		Al compressore	All' evaporatore	Condizioni standard
Resa Frigorifera	Kw	93.60	93.60	93.96
Portata in massa	Kg/h	2,370		2,088
Potenza assorbita	Kw	41.47		41.47
Corrente assorbita	A	80.92		80.92
COP		2.26		2.27
Calore al condensatore	Kw	135.07		135.43

Officine Mario Dorin S.p.A. 26/04/2007

2- Storing Room Compressor

Calcoli Su Modello

Gas **R404A** Modello **C6015CC**

Tensione / fasi / frequenza 380-420 V / 3 / 50Hz

Condizioni di funzionamento

Rugiada (dew temp.)

Temperatura evaporaz.	°C	-10.0
Temperatura condensaz.	°C	45.0
Surriscald. uscita evap.	K	7.0
Sottoraffredd. del liquido	K	2.0
Surrisc. in aspirazione	K	7.0

Risultati

H8000CC		Al compressore	All' evaporatore	Condizioni standard
Resa Frigorifera	Kw	18.70	18.70	93.96
Portata in massa	Kg/h	0.327		0.327
Potenza assorbita	Kw	5.20		5.29
Corrente assorbita	A	5.34		5.34
COP		2.35		2.35
Calore al condensatore	Kw	23.57		23.57

Officine Mario Dorin S.p.A. 26/04/2007

3- Storing Refrigeration system Condensation Unit

Copeland Selection Software

REFRIGERANT	R404A
Operating Conditions:	
Evaporating Temperature:	-10.0°C
Ambient Temperature:	35.0°C
Suction Return Temperature:	20.0°C
Required Capacity:	**20.0 kW**
Condensing Unit Selected:	OLQ-33V-NLO-TWD

PERFORMANCE AT SPECIFIED OPERATING POINT
OLQ-33V-NLO-TWD Data at 50 Hz

Capacity kW	21.60
Total Power Input kW	11.20
Compressor Current 400V, A	19.5
Mass Flow g/s	133.0
Heat Rej. kW	32.00
Condensing Temp. °C	54.2
Liquid Temperature °C	17.80

CONDENSING UNIT MECHANICAL DATA

Number of Fans 2
Height, mm 950
Receiver Capacity, l 24.0
Air Flow, cu.m/sec 1.83
Max. Standstill Pressure 21.0
Suction Diameter, inch 1 3/8
Condenser/Fan Type V9.6P/145
Total Fan Power Input, Watts 310
Base mounting (hole dia), mm 1300 x 620 (12)

Depth/Width, mm 670/2100
Gross Weight, kg 284.0
Liquid Line, inch 5/8
Max. High Pressure 28.8
Suction Type Cu Tube
Oil reservoir capacity, l 8

CONDENSING UNIT ELECTRICAL DATA (380/420V - 3~ - 50Hz)

Condenser Fan Current for 1 fan (single phase) 0.67
Compressor Maximum Operating Current, A 22.30
Compressor Locked Rotor Current, A 127.0

ACCESSORIES INCLUDED

Crankcase Heater	70W External
Oil separator	ALCO OSH-407
Liquid sight glass	ALCO AMI-1SS5-5/8S
Liquid solenoid valve	ALCO 200RBT5 fitted
Compressor contactor	ABB A16-30-01
Control circuit fuse	S7-904-5355-0
Internal Suction Line	Insulated
Liquid Subcooler	Plate Heat Exchanger
Fan speed control	ALCO FSP150
LP Switch	ALCO PSAW3A (Automatic reset)
HP Switch	ALCO PS3 WDS (Automatic reset)
Electronic controler	ALCO EC2 551
Sound attenuation	Compressor blanket
Filter Drier	ALCO ADK Plus
Vapour Injection	Expansion Valve
Oil Control System	ALCO Trax-Oil

ACCESSORIES OPTIONAL

Liquid solenoid valve	ALCO 200RBT5 shipped loose
Main Isolator Switch	20A

4- Chilling Refrigeration system Condensation Unit

Version 6.21 / 38853 (10/05) www.eCopeland.com **11 May 2007**

Copeland Selection Software

REFRIGERANT	R404A
Operating Conditions:	
Evaporating Temperature:	-10.0°C
Ambient Temperature:	35.0°C
Suction Return Temperature:	20.0°C
Required Capacity:	**100.0 kW**
Condensing Unit Selected:	OLQ-48V-NLO-TWD

PERFORMANCE AT SPECIFIED OPERATING POINT
OLQ-48V-NLO-TWD Data at 50 Hz

Capacity kW	28.90
Total Power Input kW	19.80
Compressor Current 400V, A	32.0
Mass Flow g/s	180.0
Heat Rej. kW	47.00
Condensing Temp. °C	57.8
Liquid Temperature °C	19.30

CONDENSING UNIT MECHANICAL DATA

Number of Fans 2
Height, mm 950
Receiver Capacity, l 24.0
Air Flow, cu.m/sec 2.16
Max. Standstill Pressure 21.0
Suction Diameter, inch 1 3/8
Condenser/Fan Type V6 6P/301
Total Fan Power Input, Watts 580
Base mounting (hole dia), mm 1300 x 620 (12)
Depth/Width, mm 670/2100
Gross Weight, kg 303.0
Liquid Line, inch 5/8
Max. High Pressure 28.8
Suction Type Cu Tube
Oil reservoir capacity, l 8

CONDENSING UNIT ELECTRICAL DATA (380/420V - 3~ - 50Hz)

Compressor Maximum Operating Current, A 30.60
Compressor Locked Rotor Current, A 198.0

ACCESSORIES INCLUDED

Crankcase Heater	70W External
Oil separator	ALCO OSH-407
Liquid sight glass	ALCO AMI-1SS5-5/8S
Liquid solenoid valve	ALCO 200RBT5 fitted
Compressor contactor	ABB A26-30-01
Control circuit fuse	S7-904-5355-0
Internal Suction Line	Insulated
Liquid Subcooler	Plate Heat Exchanger
Fan speed control	ALCO FSP150
LP Switch	ALCO PSAW3A (Automatic reset)
HP Switch	ALCO PS3 WDS (Automatic reset)
Electronic controler	ALCO EC2 551
Sound attenuation	Compressor blanket
Filter Drier	ALCO ADK Plus
Vapour Injection	Expansion Valve
Oil Control System	ALCO Trax-Oil

ACCESSORIES OPTIONAL

Liquid solenoid valve	ALCO 200RBT5 shipped loose
Main Isolator Switch	32A

RE Series

The Parker RE series valve utilizes balanced port construction to provide optimum operation on medium to large tonnage air conditioning and refrigeration systems. Two brass body styles with copper ODF connections and a removable Rainbow Charged™ thermostatic power element provide the stability and control required in a variety of applications, especially where there are wide changes in load conditions. Body Style 1 has an R-22 nominal capacity up to 30 tons, while Body Style 2 extends the capacity range to 70 tons.

Applications

- Air Conditioning
- Process Chillers
- Industrial Refrigeration
- Transport Refrigeration

Features and Benefits

- Balanced port design
- Removable stainless steel power element
- Field adjustable superheat
- 1/4" sweat external equalizer
- Rainbow Charges™
- Weight: Body Style 1 - 1.7 lbs. / .77 kg
  Body Style 2 - 2.5 lbs. / 1.13 kg
- Forward or reverse flow
- 60" capillary tube (120" optional)

Specifications

Refrigerant	Refrigerant Designation	Nominal Capacity	Body Style	Externally Equalized	Rainbow Charges™	Inlet Connection (Optional)	Outlet Connection
R-12 R-134a R-401A R-401B	J	6	1	RE 6 J	W, X60	5/8"	7/8"
		9	1	RE 9 J		7/8"	1-1/8"
		12	1	RE 12 J		7/8 (1-1/8")	1-3/8"
		16	1	RE 16 J		1-1/8"	1-3/8"
		23	2	RE 23 J		1-1/8"	1-3/8"
		40	2	RE 40 J		1-1/8"	1-5/8"
R-402B R-404A R-402A R-502 R-507	S	6	1	RE 6 S	W, Z, X110, X30	5/8"	7/8"
		9	1	RE 9 S		7/8"	1-1/8"
		12	1	RE 12 S		7/8 (1-1/8")	1-3/8"
		21	1	RE 21 S		1-1/8"	1-3/8"
		30	2	RE 30 S		1-1/8"	1-3/8"
		45	2	RE 45 S		1-1/8"	1-5/8"
R-22 R-407C	V	10	1	RE 10 V	W, Z, X100, X35	5/8"	7/8"
		15	1	RE 15 V		7/8"	1-1/8"
		20	1	RE 20 V		7/8 (1-1/8")	1-3/8"
		30	1	RE 30 V		1-1/8"	1-3/8"
		40	2	RE 40 V		1-1/8"	1-3/8"
		70	2	RE 70 V		1-1/8"	1-5/8"

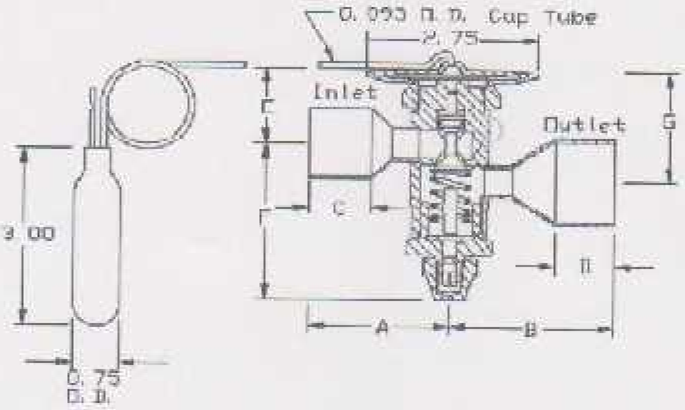
Notes:

1. Maximum operational pressure 500 psig (35 bars) high side and 275 psig (19 bars) low side.
2. Maximum storage temperature 130°F (55°C).
3. Consult Parker for pressure and temperature exceptions.
4. Do not use "W" or "Z" liquid charges in applications where bulb temperatures can exceed 130°F (55°C). For these applications use type "X" MOP gas charge only.

Parker Hannifin Corporation
Climate & Industrial Controls Group
Cleveland, OH

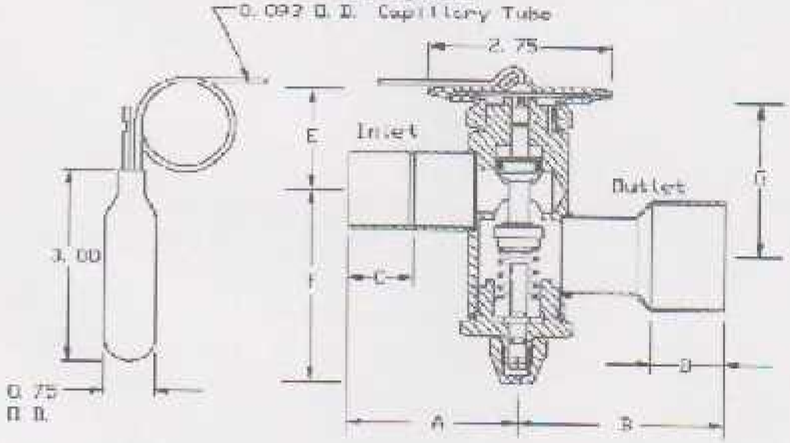
Dimensions

RE Body Size 1



Fitting Size	RE Body Size 1						
ODF	A	B	C	D	E	F	G
5/8	1.55	–	0.38 Min	–	1.45	2.56	2.06
7/8	2	2	.73 Min.	.73 Min.	1.45	2.56	2.06
1 1/8	2.3	2.3	.95 Min.	.93 Min.	1.45	2.56	2.06
1 3/8	–	2.56	–	.93 Min.	1.45	2.56	2.06

RE Body Size 2



Fitting Size	RE Body Size 2						
ODF	A	B	C	D	E	F	G
1 1/8	2.7	2.7	.91 Min.	.91 Min.	1.83	2.96	2.81
1 3/8	–	2.95	–	.97 Min.	1.83	2.96	2.81
1 5/8	–	3.19	–	1.09 Min.	1.83	2.96	2.81

S Series

Parker's S series is well suited for new or replacement installations on a variety of small to medium tonnage air-conditioning, heat pump, and refrigeration systems. A brass body with standard ODF solder connections and balanced port construction lends itself to installation on systems requiring stability and control under low load and other varying conditions.

Applications

- Air Conditioning
- Heat Pumps
- Commerical Refrigeration
- Transport Refrigeration
- Beverage Dispensors
- Dehumidifiers
- Ice Machines

Features and Benefits

- 60" capillary tube with shock loop
- Optional external equalizer
- Stainless steel power element
- Weight: .7 lbs. (.32 kg)
- Optional bleed
- Field adjustable superheat

Specifications

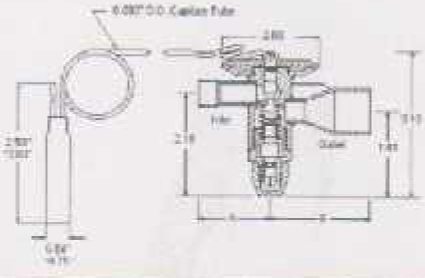
Refrigerant	Refrigerant Designation	Capacity Range Tons	Internally Equalized Models	Externally Equalized Models	Rainbow Charges™	Inlet Connection ODF (Optional)	Outlet Connection ODF (Optional)	Equalizer Connection (Optional)
R-12 R-134a R-401A R-401B	J	1/8 - 1/2	S 1/2J	SE 1/2J	W, X90	1/4 (3/8)	1/2 (3/8)	1/4" ODF (1/4" SAE)
		1/4 - 1	S 1J	SE 1J		3/8 (1/4)	1/2 (3/8)	
		1 - 2	S 2J	SE 2J		1/2	5/8 (7/8)	
		1 1/2 - 3	S 3J	SE 3J		1/2	7/8 (5/8)	
		3 1/2 - 5	S 5J	SE 5J		5/8	7/8	
R-402B R-404A R-407A R-502 R-507	S	1/8 - 1/2	S 1/2S	SE 1/2S	W, Z, X110 X35	1/4 (3/8)	1/2 (3/8)	1/4" ODF (1/4" SAE)
		1/4 - 1	S 1S	SE 1S		3/8 (1/4)	3/8	
		1 - 2	S 2S	SE 2S		1/2	5/8 (7/8)	
		1 1/2 - 4	S 4S	SE 4S		1/2	7/8 (5/8)	
		4 1/2 - 6	S 6S	SE 6S		5/8	7/8	
R-22 R-407C	V	1/5 - 3/4	S 3/4V	SE 3/4V	W, Z, X100 X35	1/4 (3/8)	1/2 (3/8)	1/4" ODF (1/4" SAE)
		1/2 - 1 1/2	S 1-1/2V	SE 1-1/2V		3/8 (1/4)	1/2 (3/8)	
		1 1/2 - 3	S 3V	SE 3V		1/2	5/8 (7/8)	
		3 1/2 - 5	S 5V	SE 5V		1/2	7/8 (5/8)	
		5 1/2 - 7 1/2	S 7 1/2	SE 7-1/2V		5/8	7/8	
		8 - 10	S 10	SE 10V		5/8	7/8	
R-410A	K	1/2 - 1 1/2	S 1 1/2K	SE 1 1/2K	KX200	3/8	1/2	1/4" ODF (1/4" SAE)
		1 1/2 - 3	S 3K	SE 3K		3/8	1/2	
		3 1/2 - 5	S 5K	SE 5K		1/2	5/8	
		5 1/2 - 7 1/2	S 7K	SE 7K		1/2	5/8	
		7 1/2 - 9	S 9K	SE 9K		5/8	7/8	

Notes:
1. U.L. recognized for maximum operational pressure of 600 psig (45 bars).
2. Maximum storage temperature 130°F (55°C).
3. Consult Parker for pressure and temperature exceptions.
4. Do not use "W" or "Z" liquid charges in applications where bulb temperatures can exceed 130°F (55°C). For these applications use type "X" MOP gas charge only.

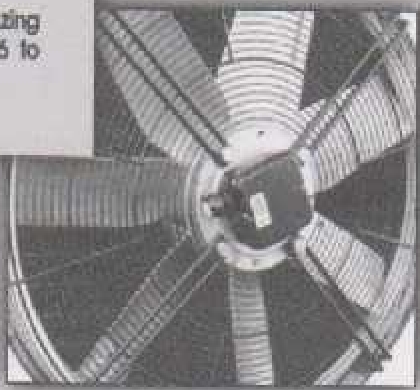
Dimensions

Fitting	Inlet A	Outlet B
1/4 ODF	1.48	-
3/8 ODF	1.69	1.65
1/2 ODF	1.48	1.46
5/8 ODF	1.87	1.87
7/8 ODF	-	2.07

* VX100, KX200 only



NK range unit coolers are suitable for chilling, storage or freezing applications. 187 basic models with capacities ranging from 5 to 130 kW.



NK

NOMENCLATURE ...

NKH 3X6D B2 R

Unit cooler

Fin types:
I : Large heat exchange surface
H : High efficiency coil

Number of fans x Ø
A - Ø 450 mm - 6 - Ø 630 mm - 8 - Ø 800 mm

Fin spacings:
R - 4 mm
L - 5 mm
C - 6 mm
S - 9 mm
T - 12 mm

Module

Motor connection
D : Delta - Y : Star

DESCRIPTION ...

CASING

- The casing is made of pre-painted galvanised steel which offers a high resistance to corrosion and impact damage.
- The main painted galvanised steel drain pan is hinge-mounted for easy maintenance.
- An internal aluminium drain pan limits the effects of condensation under the main drain pan during defrost.
- The side panels offer easy access to electrical and refrigerant circuit connections.
- The NK unit coolers are delivered in mounting position in reinforced crates (ECB option) to make lifting easier.

PURPOSE-BUILT HEAT EXCHANGER

- The finned coils of the NK line are composed of aluminium fins spaced at 4 - 6 - 9 or 12 mm, crimped to copper tubes.
- Two types of fins are available depending on the application:

-Hohweld H type fins offer an economic solution. This type of fin is well adapted to the storage of pre-packed products or products for which the cold storage room hygrometry is of little importance. The reduced size of the heat exchanger also enables fast defrosting.
-I type fins for a greater heat exchange surface. Thanks to the high surface area, this type of fin limits product dehydration and also saves energy by reducing the number of defrost cycles per day.

- The coils are equipped with RADIA optimised "Nozzle" distributors.
- For all other refrigerants, please contact us and note the details on your order form.

VENTILATION

- The external rotor ZIEHL-ABEGG fans are equipped with a fan guard conforming to standard NF E51 190.
- The external rotor enables very easy access for any work required.
- 3 fan types for the NK line:

-Ø 450 mm 4/6 Poles (1500/1000 rpm).
-Ø 630 mm 4/6 Poles (1500/1000 rpm).
-Ø 800 mm 6/8 Poles (1000/750 rpm).

- The motors are of the 3-phase type, 400V, 50 Hz, IP54, class F.
- Various fan number/diameter combinations are possible to adapt the unit cooler dimensions and air flow to the exact cold room requirements.

DEFROSTING

NKH ... C, NKH ... S, NKT ... C, NKT ... S, and NKT ... T

- The screened electrical defrost heaters are placed in tubes located in the finned coil. Two or three heaters are placed under the intermediate drain pan.
- This method enables homogeneous heat dispersion for a fast and efficient defrost cycle.
- The heaters are factory wired in the terminal box for a 400V 3-ph supply.
- A totally (HGT) or mixed (HG1) hot gas defrost is optionally available.

NKH ... R, NKH ... L, and NKT ... L

- The light electrical defrost (ETU) and electrical defrost of "low temperature" models (ELU) are optionally available.
- The light electrical defrost (ETK) is also available in kit form.
- A water defrost (DAE) is optionally available for room temperatures equal to or higher than +4°C.
- Maximum water flow-rate: NK 1 fan: 5 m3/h, NK 2 fans: 10 m3/h, NK 3 fans: 15 m3/h.

OPTIONS ...

KIT

ETK	Light electric defrosting NKH ... R, NKT ... L and NKH ... L
ECK	Additional electric defrosting
RVK	Fan ring electric heaters
VGT	Flange for textile duct
VPA	Air pressure flange
MSD	Air pressure flange with defrost shut up

DEFROST

ETU	Light electrical defrost
ELU	Electrical
RVU	Fan panel electrical heaters
DAE	Water defrost
HG1	Hot gas (coil: hot gas, drain pan: electrical heaters)
HGT	Hot gas (coil and drain pan)

MOTORFANS

MMI	230-400V/3/60Hz fan assembly
CMU	Motors factory wired

COIL

BAE	Coating of the fins
BYP	Polual Blygold coating of the fins
WCO	Glycol water and brine
EGU	Glycol water extension (please consult us)

MISCELLANEOUS

EIS	Insulated drain pans
ECB	Full crate

OTHER OPTIONS

Please consult us.

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A wide choice for all power ratings:
- T-type fins for a greater heat exchange surface or high-efficiency H type fins offering an economic solution,
- 4 fin-spacing distances 4 - 6 - 9 or 12 mm,
- 3 fan diameters for adaptation of air throw to the application,
- Various height x width combinations for perfect adaptation to the cold storage dimensions, enables selection ;
of the unit cooler best adapted to specific requirements.

NKT PRESELECTION

Fin spacing	Positive applications SC2 tAI = 0 °C ΔT 8 K	Negative applications SC3 tAI = -18 °C ΔT 7 K	 SC4 tAI = -25 °C ΔT 6 K
6 mm	NKT...L*	NKT...C	NKT...C
9 mm	-	NKT...S	NKT...S
12 mm		NKT...T	NKT...T
Defrost	ERP* / ELU*	Integrated	Integrated

* Add defrosting
ERP for a room temperature between +4 °C and +2 °C
ELU for a room temperature between +2 °C and -6 °C

NKH PRESELECTION

Fin spacing	Positive applications SC2 tAI = 0 °C ΔT 8 K	Negative applications SC3 tAI = -18 °C ΔT 7 K	 SC4 tAI = -25 °C ΔT 6 K
4 mm	NKH...R*	-	-
6 mm	NKH...L*	NKH...C	NKH...C
9 mm	-	NKH...S	NKH...S
Defrost	ETP* / ELU*	Integrated	Integrated

* Add defrosting
ETP for a room temperature between +4 °C and +2 °C
ELU for a room temperature between +2 °C and -6 °C

CERTIFICATIONS

EUROVENT : The performance published of our products are certified in conformity with european standards EN328
ISO 9001 : Our company is certified by LRQA to comply with quality standards ISO 9001 : 2000
RoHS - WEEE : Our products are compliant with regards to european guideline 2002/95/CE and 2002/96/CE concerning electric and electronic components.
CE : Our products are in conformity with european guidelines
GOST : Products in conformity with "GOST" agreement

AVERAGE CORRECTION FACTORS FOR STANDARD MOTORS CONNECTIONS IN Y INSTEAD OF D*

NKT	Fin spacing 6 mm Air flow	 Capaci.	 Air throw	Fin spacing 9 mm Air flow	 Capaci.	 Air throw	Fin spacing 12 mm Air flow	 Capaci.	 Air throw
A2	0,76	0,87	0,76	0,76	0,87	0,76	0,75	0,87	0,76
A3	0,75	0,84	0,75	0,76	0,87	0,76	0,76	0,87	0,75
B2	0,76	0,87	0,76	0,76	0,88	0,76	0,76	0,89	0,76
B3	0,76	0,86	0,76	0,76	0,88	0,76	0,77	0,88	0,77
B4	0,75	0,85	0,76	0,76	0,86	0,76	0,75	0,87	0,76
C2	0,72	0,86	0,72	0,73	0,86	0,73	0,73	0,86	0,73
C3	0,72	0,85	0,72	0,73	0,85	0,73	0,73	0,85	0,73

* If unit will be used permanently with motors connected in star please specify when ordering for optimisation of circuiting arrangement and distribution.

AVERAGE CORRECTION FACTORS FOR STANDARD MOTORS CONNECTIONS IN Y INSTEAD OF D*

NKH	Fin spacing 4 mm Air flow	 Capaci.	 Air throw	Fin spacing 6 mm Air flow	 Capaci.	 Air throw	Fin spacing 9 mm Air flow	 Capaci.	 Air throw
B1	0,76	0,87	0,76	0,76	0,87	0,76	0,76	0,89	0,76
B2	0,76	0,86	0,76	0,76	0,86	0,75	0,76	0,87	0,76
B3	0,76	0,85	0,76	0,76	0,84	0,76	0,75	0,86	0,76
C1	0,73	0,85	0,73	0,74	0,84	0,74	0,74	0,86	0,74
C2	0,72	0,82	0,72	0,72	0,82	0,72	0,73	0,85	0,73

* If unit will be used permanently with motors connected in star please specify when ordering for optimisation of circuiting arrangement and distribution.

SOUND POWER LEVEL Lw

Ø 450 mm	1 fan D	1 fan Y	2 fans D	2 fans Y	3 fans D	3 fans Y	4 fans D	4 fans Y
dB(A)	77	72	80	75	82	77	83	78

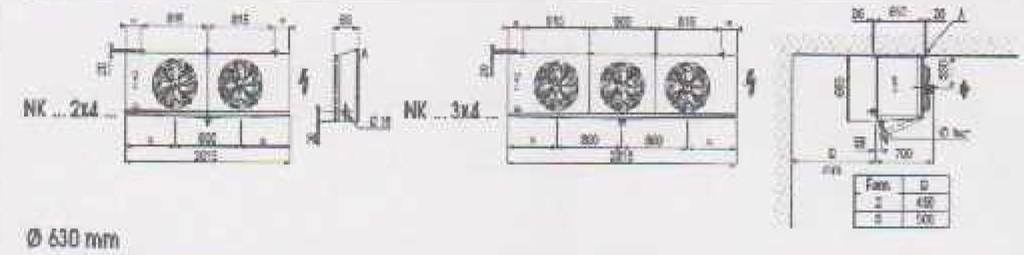
Ø 630 mm	1 fan D	1 fan Y	2 fans D	2 fans Y	3 fans D	3 fans Y	4 fans D	4 fans Y
dB(A)	90	82	93	86	95	87	96	88

Ø 800 mm	1 fan D	1 fan Y	2 fans D	2 fans Y	3 fans D	3 fans Y	4 fans D	4 fans Y
dB(A)	84	77	87	80	89	82	90	83

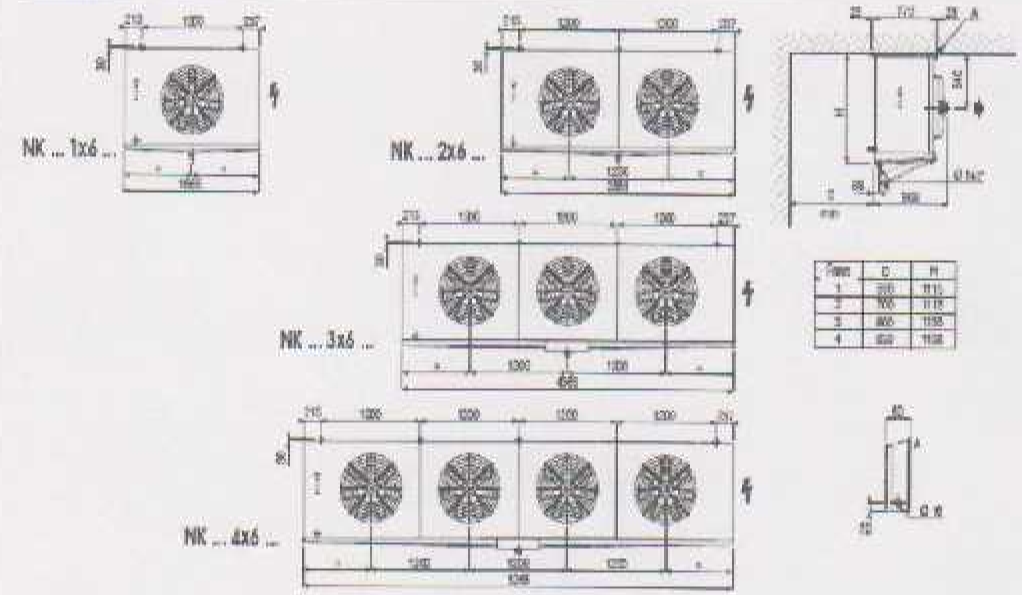
Motor connection: D: Delta - Y: Star

DIMENSIONS

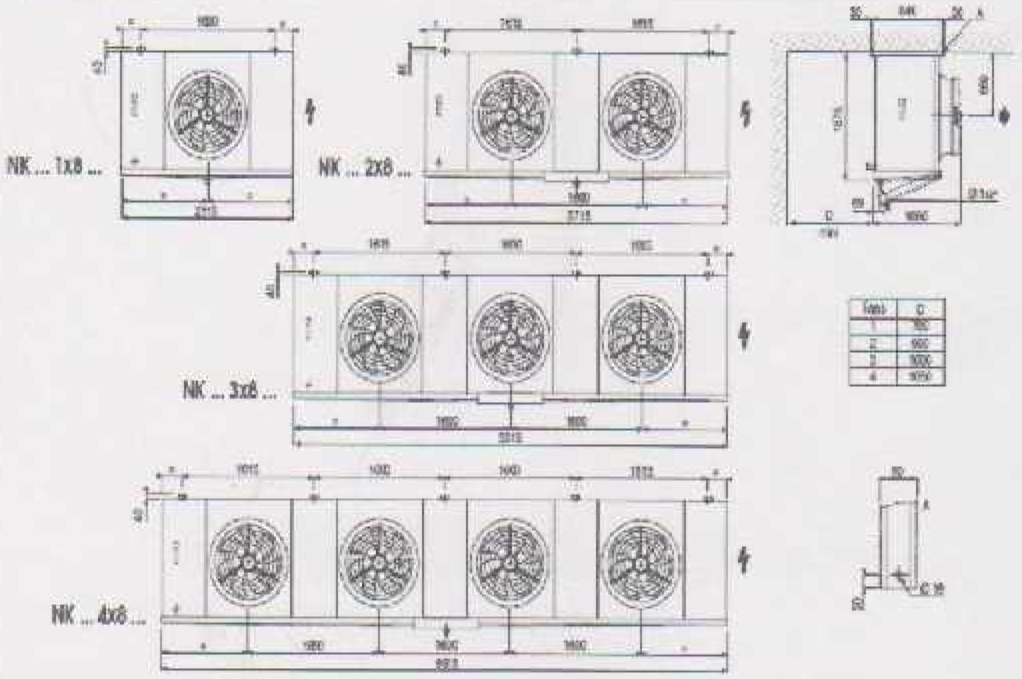
Ø 450 mm



Ø 630 mm



Ø 800 mm



TECHNICAL DATA...

NKT ... L T = large heat exchange surface 6 mm

Models		NKT ... L	2x4D A2	2x4D A3	1x6D B2	3x4D A2	1x6D B3	3x4D A3	1x6D B4	1x8D C2	2x6Y B2	1x8D C3	2x6D B2	2x6D B3	2x6D B4
Capacity - TNDAA	DT1 = 8K - SC2 (1)	kW (5)	13.39	15.90	17.91	20.23	21.46	23.90	23.93	27.06	31.58	32.66	35.92	43.44	48.43
Surface		m2	810	1060	1012	1215	1360	1590	1687	1420	2025	2160	2025	2700	3374
Circuit volume		dm3	219	292	274	329	365	503	451	458	548	584	548	730	913
Fan *	Num x Ø	mm	2 x 450	2 x 450	1 x 630	3 x 450	1 x 630	3 x 450	1 x 630	1 x 800	2 x 630	1 x 800	2 x 630	2 x 630	2 x 630
	Air flow	m3/h (5)	10000	9600	14650	15000	13850	14250	12850	20450	29300	19800	28700	27700	26800
	Air throw (2)	m (5)	19	18	36	20	34	20	33	40	28	38	37	36	34
Electric defrost ETU (3)	Ω	Num.	6	6	6	6	4	6	12	6	8	9	8	9	12
	400V/3	W	9000	9000	6900	13500	11380	13500	13800	9000	12200	12900	12200	19800	25400
		A	130	130	100	191	165	191	199	130	191	195	191	286	361
Electric defrost ELU (3)	Ω	Num.	6	9	9	8	12	9	15	9	9	12	9	12	15
	400V/3	W	9000	13500	10350	13200	13800	19800	17250	13500	19800	18000	19800	26400	33000
		A	130	195	149	191	199	286	249	195	286	260	286	381	476
Electric defrost kit Kit ECK (4)	Ω	Num.	-	3	3	-	3	3	3	3	3	3	3	3	3
	400V/3	W	-	4500	3470		3450	6600	3450	4500	6600	4500	6600	6600	6600
		A	-	65	50	-	50	95	50	65	95	65	95	95	95
Kit ECK		Num.	0	1	1	0	1	1	1	1	1	1	1	1	1
Connections	Inlet	Ø	5/8"	5/8"	5/8"	5/8"	7/8"	7/8"	1 1/8"	7/8"	1 1/8"	1 1/8"	1 1/8"	1 3/8"	1 3/8"
	Outlet	Ø	1 3/8"	1 3/8"	1 3/8"	1 3/8"	1 5/8"	1 5/8"	1 5/8"	1 5/8"	2 1/8"	2 1/8"	2 1/8"	2 1/8"	2 1/8"
Net weight		kg	180	180	185	200	220	230	220	270	300	320	310	380	400

Models		NKT ... L	3x6D B2	2x8D C2	2x8D C3	3x6D B3	3x6D B4	4x6D B2	3x8D C2	4x6D B3	4x6D B4	3x8D C3	4x8D C2	4x8D C3
Capacity - R404A	DT1 = 8K - SC2 (1)	kW (5)	54.21	55.13	64.69	65.02	71.04	72.40	82.40	86.65	94.89	96.98	109.95	129.20
Surface		m2	3037	3239	4319	4049	5062	4049	4859	6099	6749	6479	6479	8638
Circuit volume		dm3	822	876	1168	1095	1369	1095	1314	1460	1826	1753	1752	2337
Fan *	Num x Ø	mm	3 x 630	2 x 800	2 x 800	3 x 630	3 x 630	4 x 630	3 x 800	4 x 630	4 x 630	3 x 800	4 x 800	4 x 800
	Air flow	m3/h (5)	43050	40300	38600	45500	39600	57400	61050	56600	52800	57900	69800	77200
	Air throw (2)	m (5)	40	41	39	42	40	40	45	47	45	45	54	51
Electric defrost ETU (3)	Ω	Num.	6	6	9	9	12	6	6	9	12	9	8	9
	400V/3	W	19800	17400	26700	29250	39000	25300	25800	28700	51600	36700	34200	51300
		A	281	251	387	422	562	372	372	359	745	531	494	740
Electric defrost ELU (3)	Ω	Num.	9	9	12	12	18	9	9	12	15	12	9	12
	400V/3	W	29250	26100	34800	39000	48750	38600	38700	51600	54500	51600	50000	68800
		A	422	377	502	563	704	558	559	745	811	745	720	997
Electric defrost kit Kit ECK (4)	Ω	Num.	3	3	3	3	3	3	3	3	3	3	3	3
	400V/3	W	9750	8700	8700	9750	9750	12900	12900	12900	12900	12900	17100	17100
		A	141	126	126	141	141	186	186	186	186	186	247	247
Kit ECK		Num.	1	1	1	1	1	1	1	1	1	1	1	1
Connections	Inlet	Ø	1 3/8"	1 3/8"	1 5/8"	1 5/8"	1 5/8"	1 5/8"	1 5/8"	1 5/8"	1 5/8"	1 5/8"	2x1 3/8"	2x1 5/8"
	Outlet	Ø	2 1/8"	2 1/8"	2 5/8"	2 5/8"	2 5/8"	2 5/8"	2 5/8"	2 5/8"	2 5/8"	2 5/8"	2x2 1/8"	2x2 5/8"
Net weight		kg	440	480	510	500	550	560	610	640	720	770	870	980

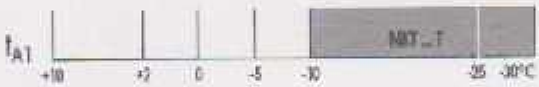
* Ø 450 mm: 400 V/3/50/60 Hz Δ = 610 W max - 1.3 A max - Y = 410 W max - 0.7 A max (5)
* Ø 630 mm: 400 V/3/50 Hz Δ = 1100 W max - 2.0 A max - Y = 810 W max - 1.16 A max (5)
* Ø 800 mm: 400 V/3/50 Hz Δ = 2000 W max - 4.0 A max - Y = 1260 W max - 2.3 A max (5)

(1) See pages "APPENDIX"
(2) Residual air velocity 0.5 m/s
(3) Electric defrost options
(4) Electric defrost kit
(5) [illegible]
(6) [illegible]
(7) [illegible]

OPTIONS...

	EIK	ECK*	RVK	VGT	VPA	MSD	ETU	ELU	RVU	DAE	HG1	HGT	M60	CMU	BAE	BYP	WCO	EGU	EIS	ECB
NKT...L	○	○	○	○	○	○	○	○	○	○	○	○	○	○	○	○	○	○	○	○

* Except NKT 2x4D A2 L - NKT 3x4D A2 L



TECHNICAL DATA...

NKT ... T   T = large heat exchange surface   12 mm

Models		NKT ... T	2x4D A2	2x4D A3	1x6D B2	3x4D A2	1x6D B3	3x4D A3	1x6D B4	1x8D C2	2x6Y B2	1x8D C3	2x6D B2	2x6D B3	2x6D B4
Capacity (ENV)	DT1 = 7K - SC3 (1)	kW (5)	8.00	9.91	10.63	12.10	13.16	14.91	15.26	16.12	19.05	19.99	21.64	26.75	31.00
	DT1 = 6K - SC4 (1)	kW (5)	6.13	7.64	8.13	9.29	10.10	11.49	11.77	12.36	14.70	15.49	16.60	20.61	23.99
Surface		m2	425	579	544	652	724	565	906	869	1086	1159	1066	1443	1811
Circuit volume		dm3	216	310	274	323	365	459	456	433	543	584	545	733	910
Fan *	Num. x Ø	mm	2x450	2x450	1x630	3x450	1x630	3x450	1x630	1x800	2x560	1x800	2x630	2x630	2x630
	Air flow	m3/h (6)	10650	10500	14960	15900	14600	15150	14850	21500	22600	20650	29900	29000	28100
	Air throw (2)	m (6)	20	19	37	23	36	22	35	42	39	40	39	38	38
Electric defrost	Ω	Num.	6	9	9	8	12	4	5	8	6	12	9	12	15
	400V/3	W	9000	10800	10800	12000	13500	10800	17250	12600	16800	18000	10800	18400	23000
		A	13.0	19.5	16.9	19.1	19.3	28.6	24.9	18.5	28.6	26.0	26.6	36.1	40.6
Connections	Inlet	Ø	5/8"	5/8"	5/8"	7/8"	7/8"	7/8"	1 1/8"	1 1/8"	1 1/8"	1 1/8"	1 1/8"	1 3/8"	1 5/8"
	Outlet	Ø	1 3/8"	1 3/8"	1 3/8"	1 5/8"	1 5/8"	1 5/8"	2 1/8"	2 1/8"	2 1/8"	2 1/8"	2 1/8"	2 1/8"	2 5/8"
Net weight		kg	160	170	190	220	215	240	250	260	300	310	320	350	420

Models		NKT ... T	3x6D B2	2x8D C2	3x6D B3	2x8D C3	4x6D B2	3x6D B4	3x8D C2	4x6D B3	3x8D C3	4x6D B4	4x8D C2	4x8D C3
Capacity (ENV)	DT1 = 7K - SC3 (1)	kW (5)	32.56	33.10	39.24	40.47	43.16	46.54	49.50	53.06	60.66	60.77	66.30	81.14
	DT1 = 6K - SC4 (1)	kW (5)	25.00	25.44	30.64	31.27	33.08	35.91	37.99	40.95	46.81	46.84	50.99	62.70
Surface		m2	1630	1738	2173	2274	2173	2716	2607	2897	3476	3621	3476	4635
Circuit volume		dm3	822	874	1085	1148	1096	1364	1304	1463	1753	1826	1742	2337
Fan *	Num. x Ø	mm	3x630	2x800	3x630	2x800	4x630	3x630	3x800	4x630	3x800	4x630	4x800	4x800
	Air flow	m3/h (6)	42650	43000	43800	41000	59800	42150	64800	58000	61650	56200	86700	82200
	Air throw (2)	m (6)	45	43	44	41	46	42	50	45	49	45	56	54
Electric defrost	Ω	Num.	9	9	12	12	8	15	9	12	12	15	9	12
	400V/3	W	29050	28800	39000	34580	35500	48750	39700	51400	51600	64500	52000	56800
		A	42.2	47.7	56.3	50.2	46.9	70.4	60.0	74.5	74.8	93.1	76.0	99.7
Connections	Inlet	Ø	1 5/8"	1 3/8"	1 5/8"	1 5/8"	1 5/8"	1 5/8"	1 5/8"	2x1 5/8"	1 5/8"	1 5/8"	2x1 3/8"	2x1 5/8"
	Outlet	Ø	2 5/8"	2 5/8"	2 5/8"	2 5/8"	2 5/8"	2 5/8"	2 5/8"	2x2 1/8"	3 1/8"	3 1/8"	2x2 5/8"	2x2 5/8"
Net weight		kg	460	450	520	550	600	640	700	670	700	750	690	920

* Ø 450 mm - 400 V/3/50/60 Hz - Δ = 620 W max - 1.15 A max - Y = 450 W max - 0.72 A max (5)
* Ø 630 mm - 400 V/3/50 Hz - Δ = 1900 W max - 3.20 A max - Y = 1220 W max - 1.95 A max (5)
* Ø 800 mm - 400 V/3/50 Hz - Δ = 2000 W max - 4.00 A max - Y = 1250 W max - 2.30 A max (5)

(1) See page "APPENDIX"
(2) Residual air velocity 0.5 m/s
(3) Electric defrost options
(4) Electric defrost kit
(5) [illegible]
[illegible]
(6) [illegible]

OPTIONS...

	ETK	ECK	RVK	VGT	VPA	MSD	ETU	ELU	RVU	DAE	HGI	HGT	M60	CMU	BAE	BYF	WCO	EGU	EIS	ECB
NKT...T	-	-	O	O	O	O	-	E	O	-	O	O	O	O	O	O	-	-	O	O

S: Standard

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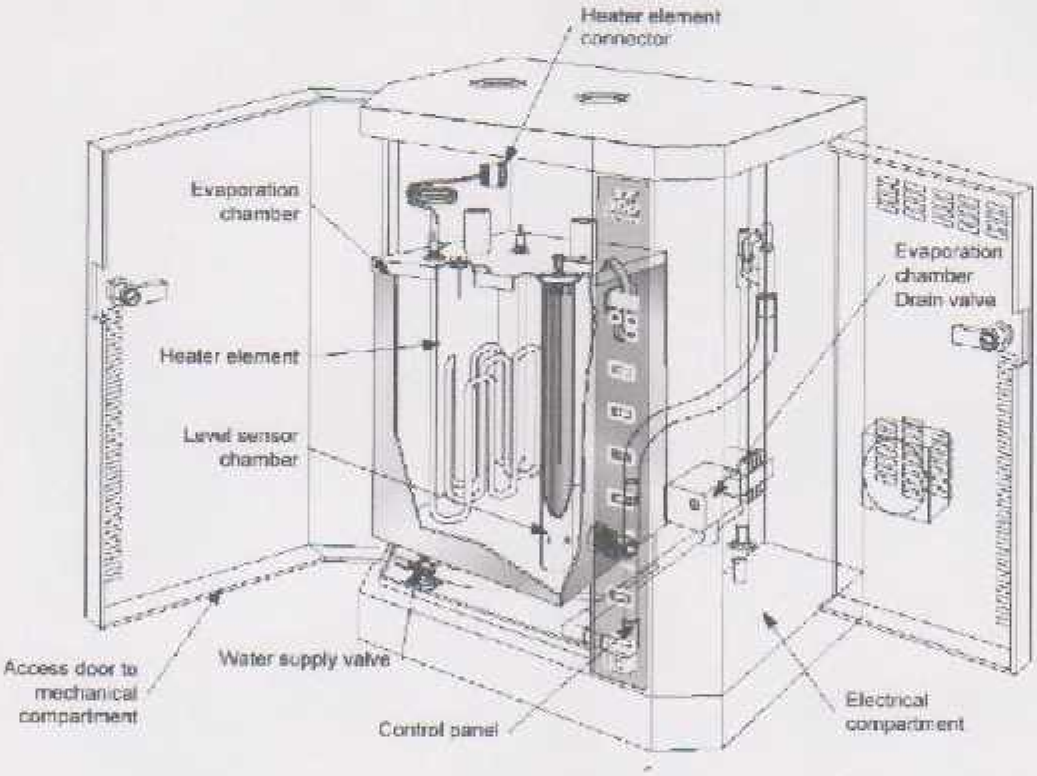
8- Humidifier

SKE Steam Humidifier

Introduction

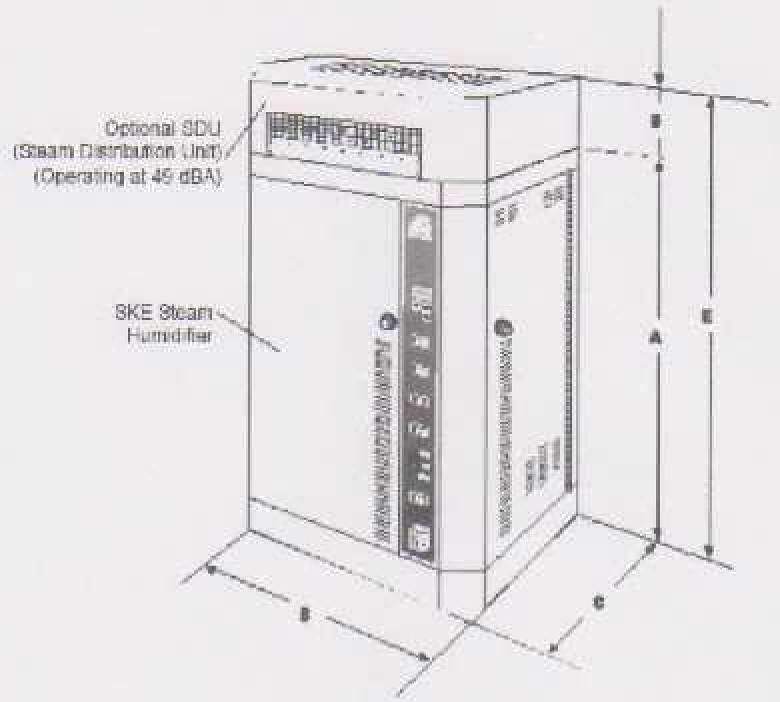
Technical Specification



Model	Capacity (kg/h)	Voltage	Power (kW)	Current (Amps)	Steam Outlets	Weight (kg)	Duct Pressure
SKE 05(M)	5	400V 3ph or (230V)	3.7	5.5 (16)	1 x 35mm	20	
SKE 10(M)	10	400V 3ph or (230V)	7.5	11 (33)	1 x 35mm	30	
SKE 20(M)	20	400V 3ph	15	22	1 x 35mm	30	
SKE 30(M)	30	400V 3ph	22	33	2 x 35mm	30	+/-1250Pa
SKE 40(M)	40	400V 3ph	30	44	2 x 35mm	30	
SKE 60(M)	50	400V 3ph	36	53	2 x 51mm	50	
SKE 60(M)	60	400V 3ph	44	64	2 x 51mm	50	
SKE 80(M)	80	400V 3ph	60	87	3 x 51mm	50	

For static duct pressures higher then +/-1250 Pa please consult JS Humidifiers

Dimensions



Model	Dimension A mm	Dimension B mm	Dimension C mm	Dimension D mm	Dimension E mm
SKE 05(M)	597	470	292	140	737
SKE 10(M)	724	533	318	165	890
SKE 20(M)	724	533	318	165	890
SKE 30(M)	724	533	318	324	1048
SKE 40(M)	724	533	318	324	1048
SKE 50(M)	794	813	318	N/A	N/A
SKE 60(M)	794	813	318	N/A	N/A
SKE 80(M)	794	813	318	N/A	N/A

All Dimensions in mm

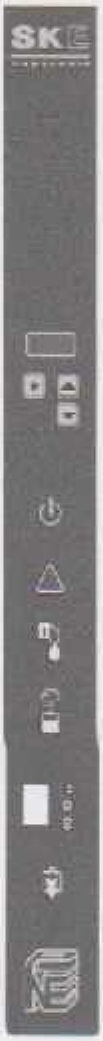
SKE Steam Humidifier
Introduction
Principle of Operation

Introduction

The Neptronic SKE steam humidifier range is designed to produce dry sterile steam for the purpose of humidification. The range consists of modulating humidifiers giving a duty of between 5-80kg/h available as a standard model or an Ultra version for close control applications with a feed water supply of 1ppm or less total dissolved solids (TDS) concentration.
Combining the latest microprocessor based electronic design with carefully matched metallurgical and mechanical technology, the design has been perfected over two decades. The SKE system is based on permanent rather than costly disposable containers.

Principle of Operation

The SKE steam humidifiers consist of an automatically water fed 316 stainless steel evaporation chamber with an easily removable lid, containing the self cleaning heating elements and patented AFEC technology water level sensor. The sheathed resistive elements are manufactured with Incoloy 825 alloy and have an output capacity of 12.8 w/sq cm, a low intensity of heat per sq cm.



The unique patented AFEC (Anti-Foaming Energy Conservation System) technology offers a unique safety and energy conservation management of expensive boiling water and steam. The patented AFEC system is made of a PTFE coated stainless steel mass measuring water sensor, anti foam sensor, electronic high temperature sensor, interactive LCD display and microprocessor controller.

The patented AFEC system is unique because the water level sensor cannot be falsely triggered by foam formation on the top of the boiling water, which eliminates the risk of elements burning out in free air, due to low water level. The foam sensing probe will automatically initiate a drain cycle if foam is sensed, unlike some other humidifiers, where foaming is controlled by the continuous skimming to drain of expensive boiling water.

The SKE Humidifier is also unique in having a fast acting electronic safety temperature sensor inside the evaporation chamber in very close proximity to the heating elements as well as a standard bimetallic external electro-mechanical temperature switch. Therefore the patented AFEC system provides an additional layer of protection.

The alphanumeric display continuously displays the status of the unit. It is also used to program variable parameters into the memory of the control microprocessor. The intelligence built into the control system ensures simple interaction between the service personnel and the humidifier.

The microprocessor control incorporates frequency and duration parameters of water dilution cycles for scale management of the SKE steam humidifier. This allows for compensation of all types of feed water qualities.

The humidifier is able to modulate its output from 0-10Vdc, 2-10Vdc or a 4-20mA control signal (also available for on/off control to special order). The maximum output can be minimised by using the "Lock On" feature. Modulation of all elements is achieved using silent Solid state relays that have zero voltage crossing. These are backed up by an electro-mechanical contactor and wiring is to the TEW 105 deg C, 600V CSA thermoplastic wire standard.

Drain Connections

The SKE Humidifier has two separate drains.

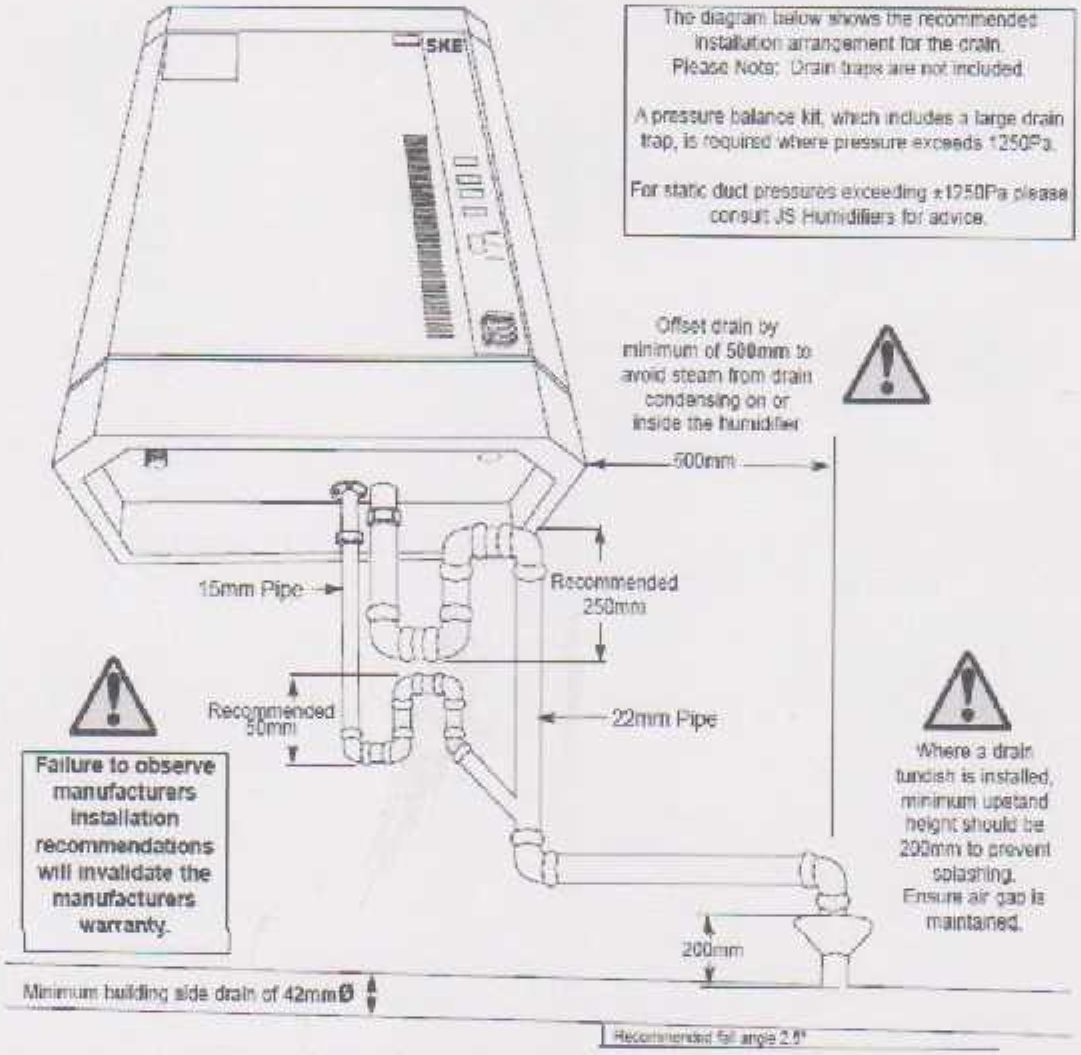
The main drain outlet located on the right side is the main hot water drain from the evaporation chamber. This drain must be trapped as shown to ensure the positive or negative pressures from the duct do not either draw air from or blow air into the main drain. This could occur during a humidifier drain cycle if this drain trap is not installed. The recommended drain trap size is 250mm. This will accommodate any start up pressure surges where an SKE Humidifier is installed on the limits of its maximum operating duct pressures specification.

The second drain is for the humidifier cabinet. This to facilitate drainage of any spillage from the evaporation chamber during a service. This drain must also be trapped to stop steam from the main drain condensing inside the humidifier cabinet.

The diagram below shows the recommended installation arrangement for the drain. Please Note: Drain traps are not included.

A pressure balance kit, which includes a large drain trap, is required where pressure exceeds 1250Pa.

For static duct pressures exceeding ±1250Pa please consult JS Humidifiers for advice.

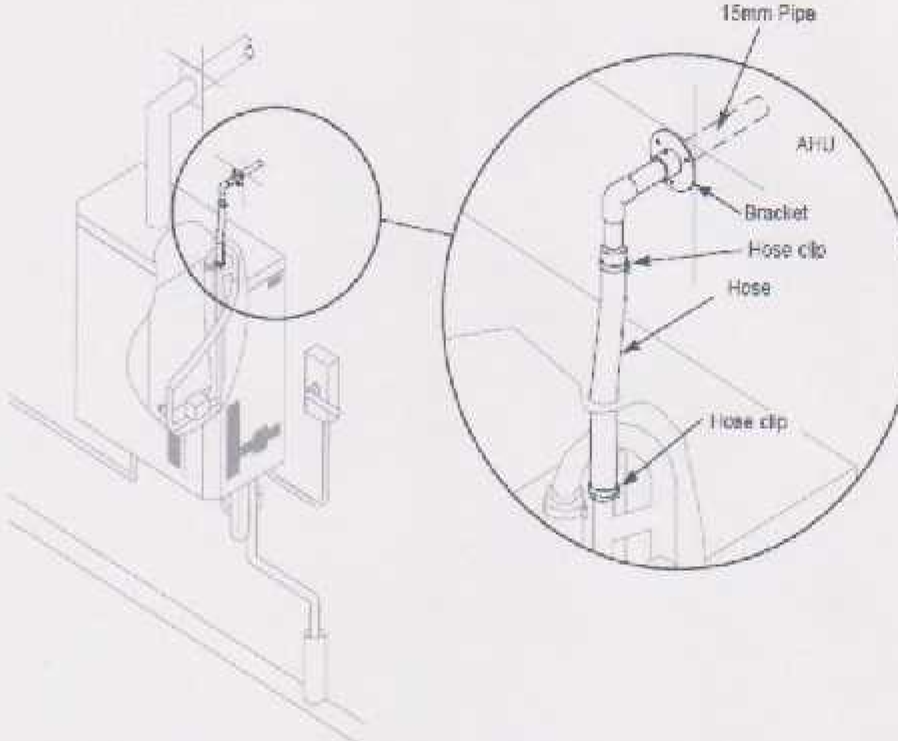


Failure to observe manufacturers installation recommendations will invalidate the manufacturers warranty.

SKE Steam Humidifier
Installation
Pressure balancing pipework installation

Pressure Balancing Pipework Installation
(Only required on high pressure installations)

Where duct pressure exceeds the normal operational limit of 1250 Pascals it will be necessary to fit a pressure balancing system to compensate. 2500 Pa peaks are experienced in ducts on normal operating pressure of 1250 Pa. As shown in the diagram below, a 15mm pipe is to be installed into the upstream duct using the bracket provided. This 15mm pipework is then connected to the top of the water filling circuit inside the humidifier.



Drain pipework for higher duct pressure

Wherever duct pressures exceed the normal operational limit of 1250 Pascals it will also be necessary to adjust the minimum drain trap depth of the main drain and any condensate drains.

When sizing the required drain trap size the normal running duct pressures should be doubled to take into consideration any pressure surge upon start up of the AHU.

Running Duct Pressure (pascal's)	Maximum Duct Pressure (Pascal's)	Drain trap size (mm)
1250	2500	250
2000	4000	400
2500	5000	500
3000	6000	600

etc..............

Failure to observe manufacturers installation recommendations will invalidate the manufacturers warranty.

SKE Steam Humidifier
Installation
Stage 6 - Electrical Supply and Installation

Electrical Connection

All work concerned with electrical installation MUST only be performed by skilled and qualified technical personnel (e.g. electrician or technicians with appropriate training). The customer MUST be responsible for ensuring their suitability.

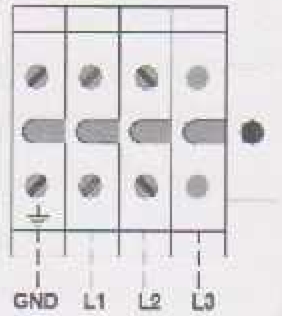
Please observe the local regulations concerning the provision of electrical installations

WARNING - Danger of Electric Shock. Ensure the electrical supply is isolated before commencing any installation.

The installation engineer must ensure the following:

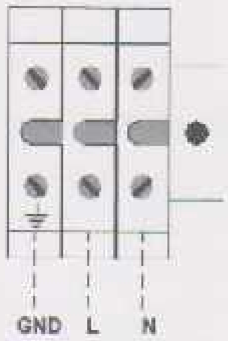
- The size of the power conductors are suitable for the maximum current supplied.
- The incoming power cable are secured via a suitably sized cable gland.
- That each terminal connection is secured firmly with a cable ferrule.

Electrical Power Connection for 400V 3ph SKE Humidifier



400V 3 phase power supply

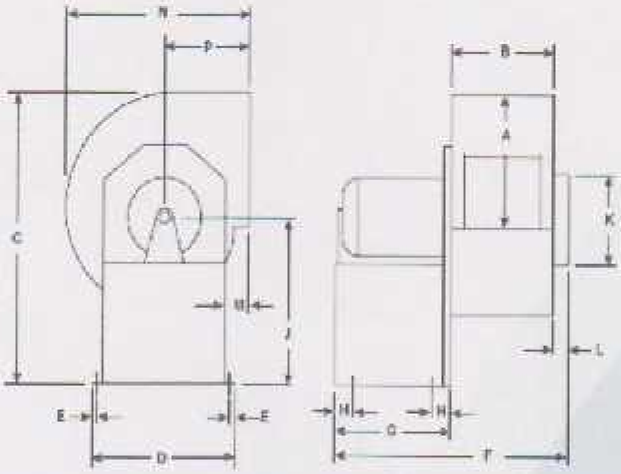
Electrical Power Connection for 230V single phase SKE Humidifier



230V single phase power supply

9- Fan Selection:

DIMENSIONAL DATA



Model SFD	A	B	C	D	E	F	G	H	J	K	L	M	N	P	Max. Motor Frame Size	*Unit Weight lbs. (kg)
6	7¾ (197)	5¼ (133)	16 (406)	11 (279)	5/8 (16)	14¾ (375)	8¼ (208)	1 1/16 (27)	9⅜ (238)	6 (152)	1¼ (32)	7/8 (22)	11⅝ (295)	5¼ (132)	56	30 (14)
7.5	9 (229)	5¼ (133)	18 (459)	12¼ (311)	5/8 (16)	16⅞ (454)	8¼ (209)	1 1/16 (27)	9⅝ (244)	8 (203)	1¼ (32)	1⅛ (29)	13¾ (348)	6⅞ (162)	56	40 (18)
9	10⅛ (257)	6¼ (159)	20¾ (576)	13¾ (349)	5/8 (16)	17⅞ (454)	10¼ (260)	1 1/16 (27)	10¾ (273)	10 (254)	1¼ (32)	1⅜ (35)	15⅞ (403)	7⅛ (181)	145T	75 (34)
10	12⅛ (308)	6⅞ (175)	24⅜ (619)	16¼ (413)	9/16 (14)	19⅝ (498)	11⅛ (283)	1 1/16 (27)	12⅛ (308)	12 (305)	1¼ (32)	1⅜ (35)	18⅝ (473)	8½ (216)	184T	113 (51)

All dimensions in inches (millimeters). *Approximate unit weight with largest frame size motor (60 Hz).

For additional discharge positions and dimensions, see page 19.

PERFORMANCE DATA

Model SFD	RPM	HP	Static Pressure in Inches of W.G.																	
			.125		.250		.500		.750		1.00		1.25		1.50		2.00		2.50	
			CFM	BHP	CFM	BHP	CFM	BHP	CFM	BHP	CFM	BHP	CFM	BHP	CFM	BHP	CFM	BHP	CFM	BHP
6-6B	1140	1/6	407	.06	338	.05														
6-4A	1725	1/4	667	.24	638	.22	541	.17	424	.13										
7.5-6B	1140	1/6	672	.16	612	.14	467	.10												
7.5-5A	1725	1/2	1062	.59	1028	.58	949	.50	869	.46	775	.38	636	.30						
9-4C	860	1/4	939	.19	749	.15	467	.09												
9-5B	1140	1/2	1159	.46	1007	.42	857	.34	762	.28										
9-15A	1725	1 1/2	1806	1.64	1765	1.59	1683	1.49	1595	1.36	1502	1.24	1407	1.13	1298	1.01	939	.70		
10-3C	860	1/3	1259	.36	1178	.32	965	.25												
10-7B	1140	3/4	1713	.86	1653	.82	1528	.72	1378	.64	1163	.52								
10-30A	1725	3	2641	3.03	2602	2.98	2525	2.97	2444	2.73	2362	2.60	2275	2.44	2179	2.32	1942	2.01	1621	1.62

Performance certified is for installation Type B - Free inlet, Ducted outlet. Performance ratings do not include the effects of appurtenances (accessories).

7

REFRIGERATION Applications

Model shown E7200-2125RF

ENVIRONMENTAL SERIES

For use as a Thermal Barrier to prevent the loss of refrigerated air, controlling condensation and icing.

APPLICATIONS:

- For use in freezers, coolers, cold storage facilities, processing plants and other refrigerated applications.

SPECIAL FEATURES:

- Galvanized, corrosion-resistant cabinet and intake grill.
- Corrosion-resistant dual-intake scrolls, precision-balanced twin blower wheels.
- Heavy duty, totally-enclosed permanent split capacitor type motors.
- Rubber-sealed and permanently-lubricated bearings.
- For use with optional instant automatic door switch.
- Energy-efficient 2-speed motors *(For single phase only)*.

HOW AIR CURTAINS WORK

AIR CURTAINS create a thermal barrier keeping cold areas at maximum efficiency, minimizing freezer condensation while conserving energy.

NOTE: Guide selection is based on refrigerated areas within facilities. Air curtains must be mounted on warm side. For exterior loading dock applications consult factory.
Selection guide does not include negative pressure or wind tunnel conditions.

AIR CURTAIN SELECTION GUIDE

Mtg. Ht. (Ft.)	Thermal Barrier Applications Type of Door	1/8 HP**	1/4 HP	1/2 HP	3/4 HP
7'	WALK-IN COOLER	✔			
8'	COOLER		✔		
10'	COOLER			✔	
12'	COOLER				✔
8'	FREEZER			✔	
10'	FREEZER				
12'	FREEZER				✔

***Walk-in cooler (Low-Profile Series) model information found on page 13.*

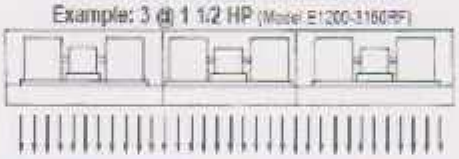
MODEL NUMBER	WIDTH	AIR DELIVERY @ NOZZLE OUTLET MAX VELOCITY (FPM) HI/LOW	AVG VELOCITY (FPM) HI/LOW	AVG. AIR FLOW RATE (CFM) HI/LOW	#MOTORS @ H.P.	*VOLTS	MAX AMPS	APPROX. SHIPPING WEIGHT (LBS)
1/4 HP MODELS								
E3600-1125	36"	2500/1850	2000/1480	1620/1200	1@1/4	120	3.4	66
E4800-1125	48"	2500/1850	1850/1370	2000/1480	1@1/4	120	3.4	77
E6000-2125	60"	2500/1850	2125/1575	2880/2135	2@1/4	120	6.8	128
E7200-2125	72"	2500/1850	2000/1480	3240/2400	2@1/4	120	6.8	130
E9600-2125	96"	2500/1850	1850/1370	4000/2960	2@1/4	120	6.8	154
E1200-3125	120"	2500/1850	1940/1440	5240/3880	3@1/4	120	10.2	207
E1440-3125	144"	2500/1850	1850/1370	6000/4440	3@1/4	120	10.2	231
1/2 HP MODELS								
E3600-1150	36"	3800/2810	3040/2250	2460/1820	1@1/2	120	5.2	74
E4800-1150	48"	3800/2810	2815/2085	3040/2250	1@1/2	120	5.2	86
E6000-2150	60"	3800/2810	3230/2390	4375/3240	2@1/2	120	10.4	148
E7200-2150	72"	3800/2810	3040/2250	4920/3640	2@1/2	120	10.4	148
E9600-2150	96"	3800/2810	2815/2080	6080/4500	2@1/2	120	10.4	172
E1200-3150	120"	3800/2810	2950/2180	7980/5890	3@1/2	120	15.6	234
E1440-3150	144"	3800/2810	2815/2080	9120/6750	3@1/2	120	15.6	258
3/4 HP MODELS								
E3600-1175	36"	4850/3590	3880/2870	3140/2325	1@3/4	120	7.8	83
E4800-1175	48"	4850/3590	3590/2660	3880/2870	1@3/4	120	7.8	96
E6400-2175	64"	4850/3590	4125/3055	5570/4125	2@3/4	120	15.6	164
E7200-2175	72"	4850/3590	3880/2870	6280/4650	2@3/4	120	15.6	166
E9600-2175	96"	4850/3590	3590/2660	7780/5740	2@3/4	120	15.6	192
E1200-3175	120"	4850/3590	3780/2795	10160/7520	3@3/4	120	23.4	261
E1440-3175	144"	4850/3590	3590/2660	11640/8610	3@3/4	120	23.4	280

* Optional voltages available - 208 - 230/ 460 volt 3-phase and 50 Hz voltages.
• For accessories, see pages 20-21

All specifications subject to changes and improvements

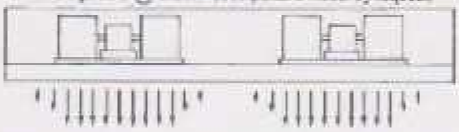
LEADING EDGE 120" AIR CURTAIN

Example: 3 @ 1 1/2 HP (Model E1200-3160RF)



BRAND X 120" AIR CURTAIN

Example: 2 @ 1/2HP (Competitors model by request)



ANATOMY OF LEADING EDGE AIR CURTAINS

Low-Profile Series*

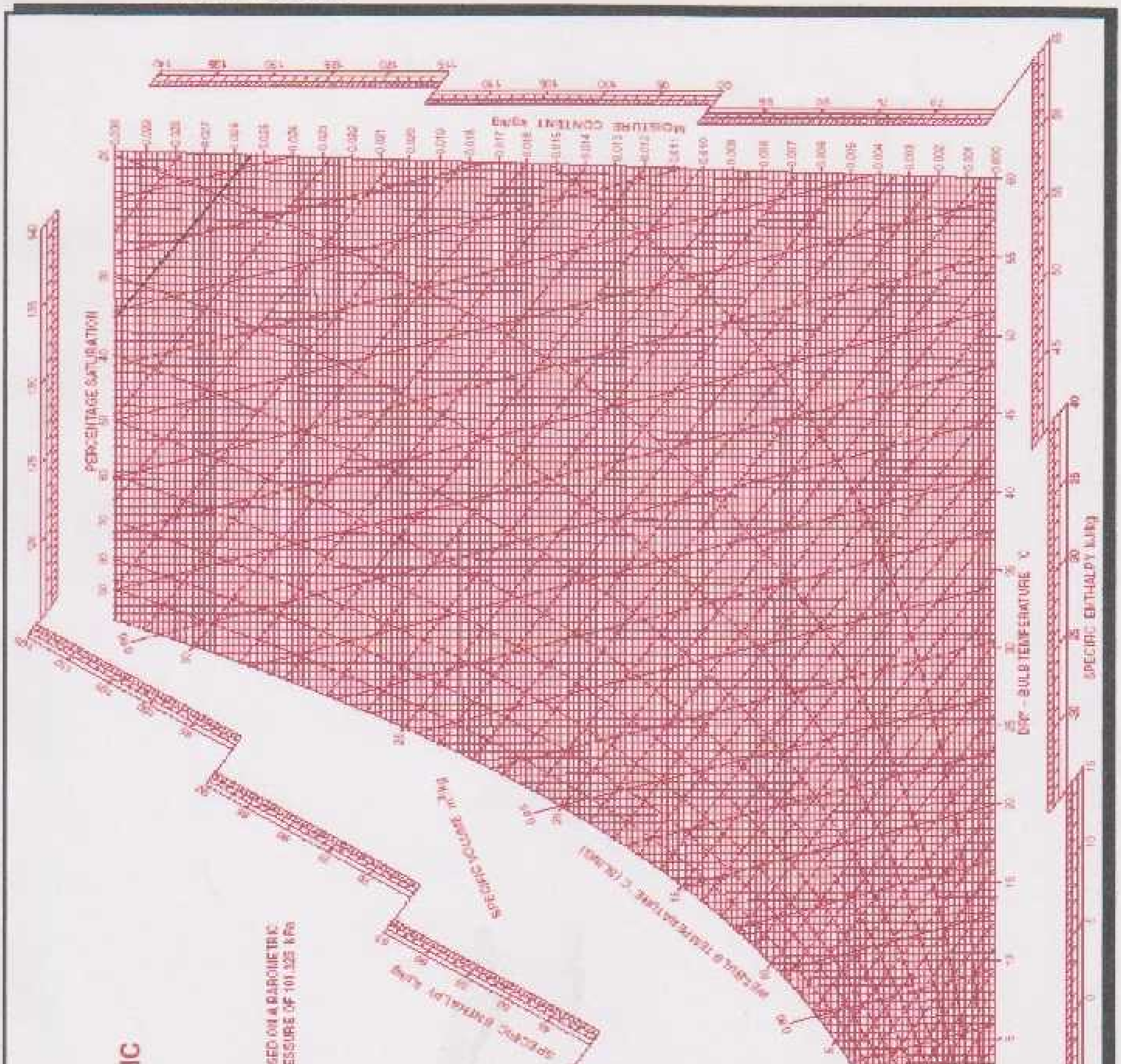
Unit Shown CLP3600 - 1110RE, 1/10HP

*All specifications subject to changes and improvements.

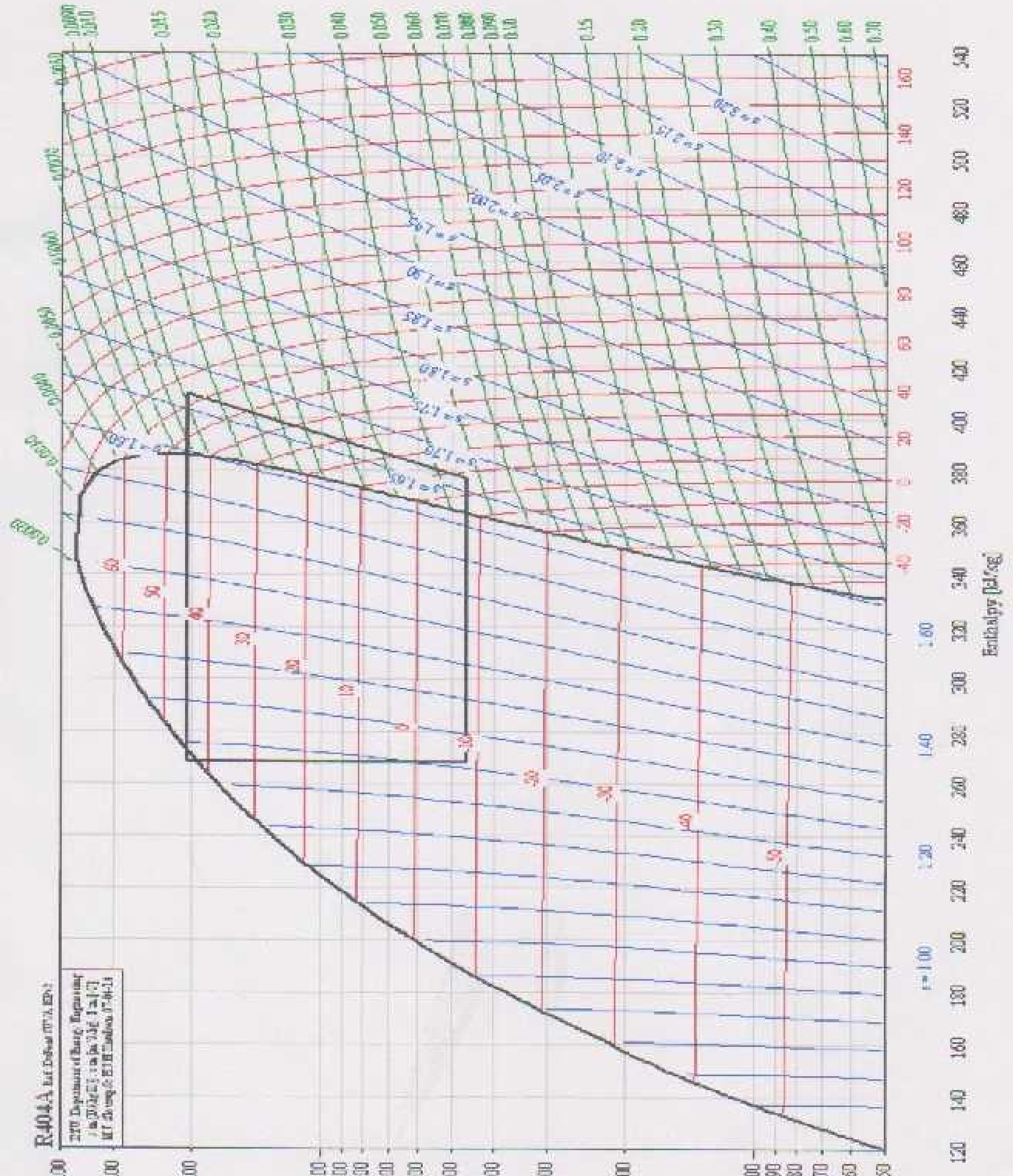
Appendix Two

Tables and Charts

1- Psychometric Chart



2- Thermodynamic Cycle of R404A



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