Addis Ababa University

Addis Ababa Institute of Technology

School of Mechanical and Industrial Engineering

Design of Refrigerated Railcar for Addis Ababa – Djibouti Railway

by

Mekonnen Seid

Advisor ;Demiss Alemu (Dr.Eng.)

May, 2015

Addis Ababa University

Addis Ababa Institute of Technology

School of Mechanical and Industrial Engineering

Design of Refrigerated Railcar for Addis Ababa – Djibouti Railway

by

Mekonnen Seid

May, 2015

Approved by Board of Examining:

Demiss Alemu (Dr.Eng.)	________________	________________
Advisor	Signature	Date
Fasil Gesessew (MSc.)	________________	________________
Internal Evaluator	Signature	Date
Tesfaye Damma (Dr.)	________________	________________
External Evaluator	Signature	Date
Birhanu Besha (Dr.)	________________	________________
Railway Center Head	Signature	Date

DECLARATION

I hereby declare that the work which is being presented in this thesis entitled "Design of Refrigerated Railcar for Addis Ababa – Djibouti Railway" is original work of my own, has not been presented for a degree of any other university and all the resource of materials used for this thesis have been duly acknowledged.

______________________________ __________________________

Mekonnen Seid Date

This is to certify that the above declaration made by the candidate is correct to the best of my knowledge.

______________________________ __________________________

Dr. Eng. DEMIS ALEMU Date

(Advisor)

ACKNOWLEDGEMENT

I would like to express my gratitude towards my advisor **Dr. Eng. DEMISS ALEMU**, for his invaluable guidance, encouragements and inspiration during the path of this work. I would like to extend my special thanks to all my families and friends for their kind help and cooperation in various ways. They have all been a tremendous source of support and encouragement throughout all of this work.

ABSTRACT

*Refrigerated railroad cars, known as reefers, are railroad rolling stock designed to carry perishable freight at specific temperatures. They are insulated boxcars that keep the cargo at a regulated temperature. As soon as railway bulk freight transportation has emerged and railroad cars had the capability to keep a load cold, the potential growth of the meat, vegetables, fruits and brewery industries became nearly unlimited. In this thesis, a current technology and achievement of a cold-room system on transportation system especially in a railroad car was investigated. A refrigerated railroad car with electric powered integrated R134A vapor compression refrigeration system needs to be designed for local manufacturing. The refrigerated railroad car with internal dimensions (17.72 * 2.66 * 2.52) has to be able to preserve and transport a maximum charge of 60000kg of chilled/frozen meat or 42000 kg of chilled vegetables in 24 hours (1 day) to the next mode of transportation, sea transport to Middle East Market. The meat to be frozen and transported enters the refrigerated rail car already frozen to a temperature of about -10°C. Also the meat and vegetables to be chilled and transported enters the car at a temperature of about 2°C and 3°C respectively. The frozen meat leaves the refrigerated railcar and transferred to the refrigerated ship or reefer container at -18°C.*

A pre-fabricated outer facing with an injected polyurethane is used as an insulation system for the car envelop design and the thermal heat resistance of walls, ceiling and floor is calculated with standard wall thickness. In this design of car envelop the refrigerating space and air circulation is determined for obtaining the optimum air velocity inside the refrigerated railcar. In these design work we use condensing temperature 48°C. Average summertime temperature of AYSHA is 38°C, so there is an average margin of 10°C. For our design purpose we will assume an ambient temperature of 40°C. With these parameters we made cooling load calculations analytically and using computer program and the total cooling load is found to be approximately 19 kW. We use this amount refrigeration capacity for the design process like system component selection and sizing. And finally using the refrigerated railcar envelope and selected refrigerating equipment the most appropriate system configuration has been made.

Keywords: Thermodynamic cycles, heat transfer, transport refrigeration

Table of Contents

ACKNOWLEDGEMENT iii

ABSTRACT iv

NOMENCLATURE viii

LIST OF FIGURES x

LIST OF TABLES xi

LIST OF APPENDIX xii

CHAPTER ONE 1

INTRODUCTION 1

1.1 Background 1

1.2 Significance of the Study 3

1.3 Statement of the Problem 4

1.4 Objective 5

1.4.1 Main Objective 5

1.4.2 Specific Objective 5

1.5 Scope and Limitation of the Study 6

1.5.1 Scope of the Study 6

1.5.2 Limitation 6

1.6 Organization of the Thesis 6

1.7 General Descriptions of the Refrigerated Railcar 7

CHAPTER TWO 9

LITERATURE REVIEW 9

2.1 Introduction to the History of Refrigeration 9

2.2 Vapor compression refrigeration system 10

2.3 Overview of Today's Transport Refrigeration 12

CHAPTER THREE 17

REFRIGERATED RAILCAR ENVELOPE DESIGN 17

3.1 Introduction 17

3.2 Types of Insulation System 18

3.3 Heat Resistance of Insulation System 18
3.4 Refrigerating Space Calculation 21
3.4.1 Introduction 21
3.4.2 Calculations 21
3.5 Air Circulation in the Refrigerated Railcar 23
3.5.1 Introduction 23
3.5.2 Air Flow Rate Calculation 25
CHAPTER FOUR 27
REFRIGERATING LOAD CALCULATION 27
4.1 Introduction 27
4.2 Refrigerating Load Calculation 28
4.2.1 Product Load 28
4.2.2 Transmission Load 30
4.2.3 Infiltration through the Vehicle Body and Closed Doors 35
4.2.4 Internal Load 36
4.2.5 Equipment Related Load 37
4.2.6 Total Refrigerating Load 37
4.3 Cooling Load Calculation Using Computer Program 38
CHAPTER FIVE 41
EQUIPMENT SELECTION 42
5.1 Introduction 42
5.2 Evaporator 42
5.2.1 Introduction 42
5.2.2 Selection 44
5.3 Evaporator Defrosting 46
5.4 Air Cooled Condensing Unit 47
5.4.1 Introduction 48
5.4.2 Air Cooled Condenser 48
5.4.3 Compressor 50
5.4.4 Condenser Selection 52
5.4.5 Compressor Selection 53
5.4.6 Oil Management 54

5.5 Piping 54
5.5.1 Introduction 54
5.5.2 Pipe Material 55
5.5.3 Pipe Sizing 55
CHAPTER SIX 56
REFRIGERATION SYSTEM CONFIGURATION 56
6.1 Thermodynamic Cycle 56
6.1.1 Introduction 56
6.2 System Layout 59
CHAPTER SEVEN 64
RESULTS AND DISCUSSION 64
CHAPTER EIGHT 66
CONCLUSION AND RECOMMENDATION 66
8.1 Conclusion 66
8.2 Recommendation 66
8.3 Future Work 67
REFERENCE 68
APPENDIX 72

NOMENCLATURE

ASHRAE = American Society of Heating, Refrigerating and Air-Conditioning Engineers

HFC = Hydro fluorocarbon

EES = engineering equation solver

PAG = Polyalkylene Glycols

POE = Polyol Esters

METEC: Metal and Engineering Corporation

GTP1: Growth and Transformation Plan One

GTP2: Growth and Transformation Plan Two

U: Overall heat transfer coefficient

R: Heat resistance

$\dot{V}$: Air flow rate

$\dot{Q}$: Refrigeration load

v_{air}: air velocity

T_f: film temperature

T_w: wall temperature

T_{ext}: ambient temperature

T_{int}: railcar inside temperature

ρ: Air density

μ: Air dynamic viscosity

k: Air thermal conductivity

k_{ss}:Thermal conductivity of Galvanized Steel

k_{pu}: Thermal conductivity of Polyurethane

k_{al}: Thermal conductivity of Aluminum

C_p: Air specific heat

L: Significant length

β: Air coefficient of thermal expansion

g: Gravitational acceleration

ΔT: Temperature difference T_{ext} - T_f

LIST OF FIGURES

Figure 1.1 Schematic of the vapor compression refrigeration cycle 2

Figure 1.2 An overview of the refrigerated railcar envelop 8

Figure 2.1 Schematic of the refrigeration cycle 11

Figure 2.2 Simple vapor compression refrigeration system 13

Figure 5.1 Air – cooling evaporating unit 43

Figure 5.2 Air - cooled condensing unit 48

Figure 5.3 Bitzer Octagon Semi – hermetic compressor 51

Figure 6.1 Cycle analysis of single stage refrigeration system 57

Figure 6.2 Components of the refrigeration plant 58

Figure 6.3 System layout of the refrigeration equipment, nose mounted condensing unit 59

Figure 6.4 System layout of the refrigeration equipment, evaporator inside the car 60

Figure 6.5 2D drawing of car body of the refrigerated railcar 61

Figure 6.6 Exploded view of the refrigerated railcar 62

Figure 6.7 A refrigerated container equipped with a top-air delivery system. 63

Figure 6.8 A refrigerated container equipped with a bottom-air delivery system. 63

LIST OF TABLES

Table 2.1 Ambient Temperatures for Equipment Design in Several Geographical Regions 16

Table 3.1 Thermal resistances recommended by ASHRAE 20

Table 3.2 Calculation results of the heat transfer through the ceiling, sidewalls, front & back walls and floor with standard insulation size. 20

Table 3.3 Beef meat composition 21

Table 3.4 Component densities at -10°C and –18°C 22

Table 3.5 Beef meat density 22

Table 3.6 Beef meat volume 23

Table 3.7 Percentage of the freezing space volume filled with meat 23

Table 3.8 Free section per 2.664 m wide of the refrigerated railcar 26

Table 3.9 Approximate air velocity inside the refrigerated railcar 26

Table 4.1 Conventional temperature differential increase to compensate for sun effect 32

Table 4.2 Surface temperature of vertical walls, ceiling and floor due to sun effect 34

Table 4.3 Calculation results of vehicle body transmission load 35

Table 4.4 LU – VE Software result of total cooling load for frozen meat 39

Table 4.5 LU – VE Software result of total cooling load for chilled meat 40

Table 4.6 LU – VE Software result of total cooling load for chilled vegetables (General) 41

Table 5.6 Refrigerant pipe materials and size selection output 55

Table 7.1 Calculated Product Loads 64

LIST OF APPENDIX

[A] Thermal Conductivity and Thermal Resistance of Insulation Materials	72
[B] Heat Load from Air Leakage through a –18°C vehicle resulting from infiltration of ambient air at various conditions.	73
[C] Heat equivalent of electric motor	74
[D] Selected evaporator characteristics	75
[E] Selected condenser Characteristics	76
[F] Selected compressor performance data with arbitrary operating points	77
[G] Selected compressor output characterstics	79
[H] Selected compressor technical data	80
[I] Selected compressor dimension	81
[J] Refrigerant Receiver output characterstics	82
[K] Technical Data for the Horizontal Receiver	83
[L] Horizontal Refrigerant Receiver Dimension	83
[M] Assembly of the Body, Sealing and Bonding in Interior and Exterior of the car envelope	84
[N] Technical Specification of Refrigerated Car	84
[O] Average summer time weather conditions in AIYSHA	85

CHAPTER ONE

INTRODUCTION

1.1 Background

The need for perishable products transportation is one of the most common applications where mechanical refrigeration is used. The design of the mechanical transport refrigeration storage and required capacity of the unit is critical for the technical requirements of transport refrigeration units are more severe than for many other applications of refrigeration. The equipment has to operate in a wide range of ambient temperatures and under extremely variable weather conditions (sun radiation, rain…). It also has to be able to carry any one of a wide range of cargoes with differing temperature requirements, and it must be robust and reliable in the often severe transport environment. [4]

Refrigeration is the process of removing heat from a lower-temperature zone and discarding it to a higher-temperature zone. Heat naturally flows from hot to cold. Refrigeration is therefore the opposite of the natural flow of heat. It has many applications in everyday life including chilling, freezing, and air-conditioning. The availability and application of refrigeration technology is critical to a society's standard of living. Preservation throughout the food chain and medical applications are examples of key contributors to quality of life. The refrigeration method during transportation is using an independent refrigeration unit. An advantage is that it can keep a container at low temperature for a relatively long period. It is generally used for long distance transportation of bulk foodstuffs. The working principle (see figure 1.1) adopted is a compressor provides refrigeration for the containers; therefore, unavoidably extra energy will be consumed for the system. [1]

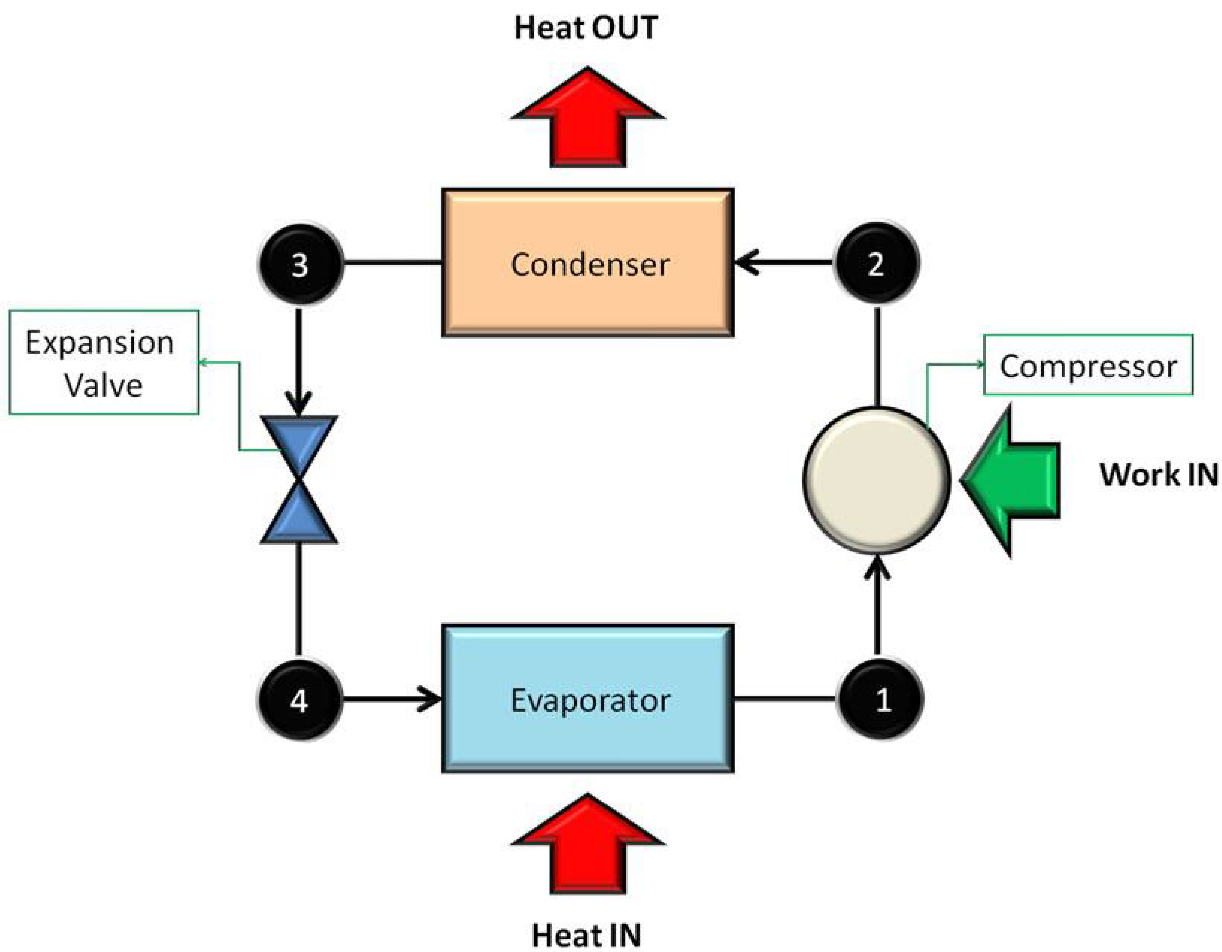


Figure 1.1 Schematic of the vapor compression refrigeration cycle

Refrigeration is a remarkable tool for preserving perishable food. [1] In addition to preventing deterioration, refrigeration is of real economic importance, since it promotes national and international trade and maintains the quality of agricultural and produced products. The low temperature environment can be achieved and maintained through cold room of transport refrigeration in which the temperature is consistently at required levels. Railway is one of the transport industries which are utilizing this refrigeration system for long distance transportation of bulk foodstuffs and non-food items. Currently in Ethiopia many commodities which are perishable travel to distant markets on highway using refrigerated trucks and vans with inefficient temperature controls.

Railway transportation is one of the mega projects that the Government of the Federal Democratic Republic of Ethiopia has planned to implement during the Growth and Transformation Plan (GTP) between 2010/11-2014/15. Construction of about 2000km of standard gauge railway infrastructure is planned to be undertaken. Among this the Addis Ababa – Djibouti Railway line, which is now under construction, is regarded as a major import and export corridor of the country.

1.2 Significance of the Study

Ethiopia has a big potential to grow more fruits and vegetables as well as a potential of exporting beef, lamb and poultry meat. Ethiopia's current meat, fruit and vegetable export is very limited. Scope for expanding and diversifying its export markets is large. Demand for Ethiopian different meat products, fruits and vegetables are growing in nearby international markets, like the Middle East. Although Ethiopia is land locked, it is geographically close to the Middle East. To reach the Middle East, and other international markets, produce has to be transported to Djibouti where it can be shipped. Roads to Djibouti are generally good, but transportation takes up to 2 and 3 days including long waiting times at the border, sometimes up to ten hours.[23] The currently emerging mode of transportation in the country, railroad transportation to Djibouti is the perfect and better option of the currently used road transportation. It has an outstanding advantage of economically visible for its nature of mass and electric driven transportation. Fresh produce is sensitive and needs to be refrigerated to remain fresh and of good quality during such transportation time. So to support this exportation of perishables to the Middle East it is of paramount importance to design a reliable refrigerated railcar or reefers which are used to deliver the meat, fruit and vegetables in good condition. This refrigerated railcar or reefer container transport offers possibilities to make use of the potential to export more meat products, fruits and vegetables which arrives the market in good quality.

1.3 Statement of the Problem

Transport refrigeration is essential in today's society to preserve and protect perishable foodstuffs or non-food goods such as flowers, plants, pharmaceuticals, chemical products, etc. Frozen goods such as meat are transported at a temperature of –18°C or lower, chilled goods, such as vegetables and fruits, at a temperature above the freezing point. Some goods are also transported at controlled temperatures above 15°C (e.g. cocoa, coffee, flavors, certain fruit and vegetables, certain pharmaceuticals). [3]

Food transport refrigeration is a critical link in the food chain, not only in terms of safety by properly maintaining the temperature of the transported goods, but also through its impact on energy consumption or CO_2emissions. One of the greatest issues in refrigerated transport is the high dependency on oil for both refrigeration and motion purposes. Refrigerated transport equipment has been progressively improved over the past 60 years in line with the development of food trade worldwide. [3]

While the use of transport refrigeration such as refrigerated railcars is fairly mature in developed countries, it remains at the early stage of use in many developing countries like our country, Ethiopia, and is a necessary part of solution to the problem of high food spoilage rate. With increased central urbanization and hence greater distances between rural production areas and highly populated areas, refrigerated railcar mass transport will play a major role in the addressing of these requirements.

Basically the government of the Federal Democratic Republic of Ethiopia has planned in the new Growth and Transformation Plan (GTP2) that Bishoftu Automotive and Locomotive Industry under Metal &Engineering Corporation will locally manufacture different types of railway passenger and freight locomotives including refrigerated railcars. These refrigerated railcars greatly support the growth of cold chain of the country especially for the potential export markets of vegetables, fruits and meat products to international market.

The idea of thought for any study is instigated on the curiosity to satisfy a need at hand. So this study seeks to solve the problem of designing and developing a refrigerated railcar for transporting different perishable commodities like vegetables, fruit and meat products on Addis Ababa – Djibouti Railway line.

1.4 Objective

1.4.1 Main Objective

The main objective of this thesis is to design a refrigerated rail car for transporting chilled or frozen perishable products like vegetables, fruits and meat products on A.A. – Djibouti railway line for locally manufacturing purpose so that preliminary transfer of technology is achieved.

1.4.2 Specific Objective

- Ascertain the current achievements of today's transport refrigeration specially refrigerated railcars.
- Determine the optimum thermal insulation system and thickness of the refrigerated railcar envelop. And calculate the required refrigeration space for product load to be transported and air circulation inside the envelope so that the air velocity is optimum.
- Determine the total cooling load due to the structural heat leakage from heat transferred through the walls and due to the product loads which have to be cooled and transported by the refrigerated railcar car.
- Using the calculated cooling load, select the required cooling system units or equipments for maximum design load condition of the refrigerated railcar on Addis Ababa – Djibouti railway line.
- Forward the system layout and configuration of the designed cooling system with the proposed refrigeration system units and refrigerated railcar envelop.

1.5 Scope and Limitation of the Study

1.5.1 Scope of the Study

The scope of the study shall address the conceptual design work of a single temperature compartment refrigerated railcar which is supposed to be manufactured locally. The refrigerated rail car which is intended to be designed supposed to be worked on Addis Ababa – Djibouti Railway line which is one of the harshest weather conditions of the national railway networks of Ethiopia.

1.5.2 Limitation

In this design work while calculating the total cooling load it is found a bit difficult to include the infiltration load due to door opening during loading and unloading of the products due to the lack of appropriate information about the location and types of refrigerated rail cross dock facilities used at starting and ending terminals. However it can be assumed that this infiltration load due to door openings is insignificant provided that we have a well-organized and conditioned refrigerated road – rail distribution link with refrigerated rail cross dock facilities.

1.6 Organization of the Thesis

This thesis work is grouped into eight chapters. In this first chapter the background of transport refrigeration and the principles of mechanical vapor compression system utilized on cold chain have been discussed in brief. The main significance of the study, statement of the problem which initiates us for doing this study, general and specific objectives that we are going to attain, and scope and limitations of the study have also been outlined in this chapter.

On the literature review, second chapter, history of refrigeration and overview of transport mechanical refrigeration system, current technology of transport refrigeration and its achievement have been discussed. The working Principle of vapor compression system has also reviewed in brief.

The third chapter explains the design of refrigerated railcar envelope. The appropriate insulation system of the car envelop has been selected and heat resistance of the insulation system has been calculated. The required frozen meat holding space inside the refrigerated railcar is calculated to make sure that an optimum air velocity is maintained.

In the fourth chapter the total cooling load for three types of commodities (chilled vegetables, frozen and chilled meat) is calculated analytically and using computer program. The maximum total refrigerating load which includes product load, transmission load, infiltration load, internal load plus20% safety factor of the three products is taken as design cooling load.

On the fifth chapter, using the calculated cooling load, the refrigeration system components are selected from manufacturer's catalogue and also the necessary pipe material recommendation and sizing for the required refrigeration plant are made.

The sixth chapter describes the thermodynamic cycle and overall system configuration of the refrigeration plant of the refrigerated railcar.

On the seventh chapter the result and discussion of the design work has been discussed,

Finally on the last chapter conclusion and recommendation together with future work have been addressed

1.7 General Descriptions of the Refrigerated Railcar

The refrigerated railcar which is intended to be designed is essentially thermally-insulated envelop container with an electrically powered cooling unit. The railcar cooling unit performs two basic functions:

- Pre-cooling the mass of the railcar (along with the air inside the car) to the required operating temperature prior to loading;
- Maintaining a protective air envelope around the load inside the railcar by removing conduction heating coming into the railcar through the floor, ceiling and walls.

The refrigerated railcar has a capacity of loading about 60,000kg with an overall dimension of: -
Length = 18.00 m.

Width = 2.874 m.

Height = 2.920 m.

The refrigerated railcar has an insulated hinged door on the center of the side wall. The dimension of the door opening is: -
Width = 2.40 m

Height = 2.52 m

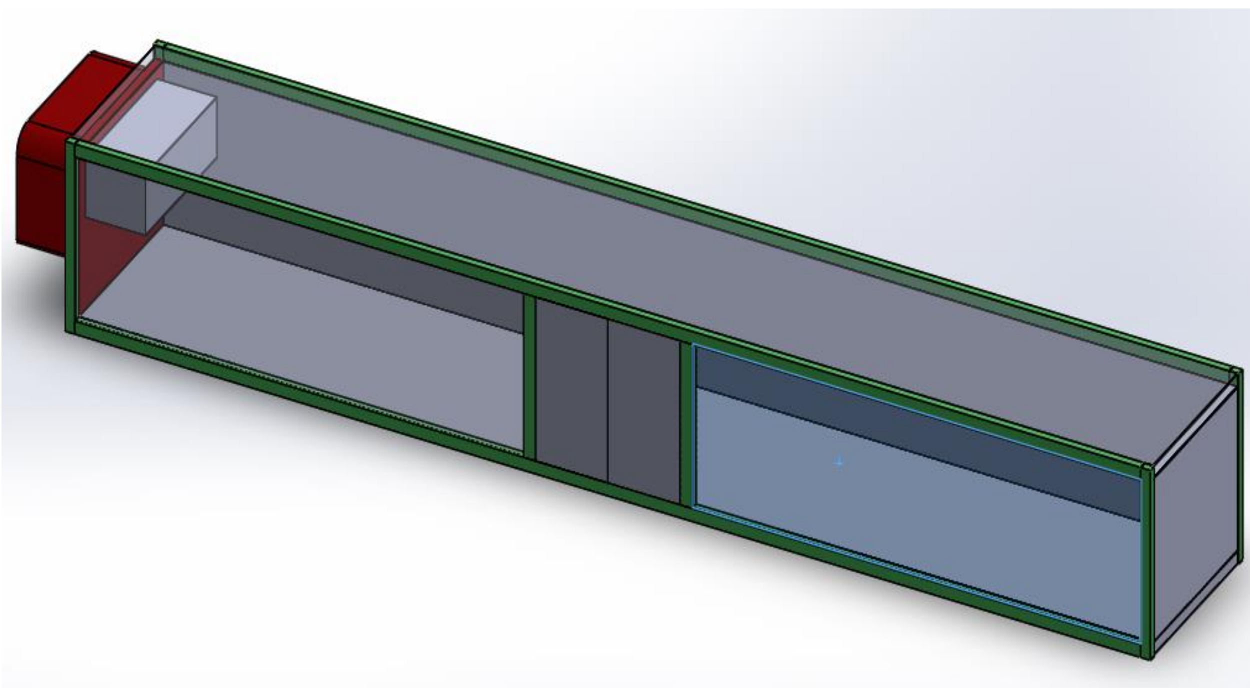


Fig. 1.2 An overview of the refrigerated railcar envelop

CHAPTER TWO

LITERATURE REVIEW

2.1 Introduction to the History of Refrigeration

Refrigeration is a process widely used to lower temperature in a closed environment and is nowadays found in anything from an air-conditioned car to a warehouse with frozen foods.

Refrigeration is a process in which heat is moved from one environment to another by means o f a transportation medium, commonly called a refrigerant. An American working in Britain named Jacob Perkins discovered the cooling effect of certain liquids when they evaporate and invented the refrigeration cycle in 1834. Not knowing about the invention of Jacob Perkins, John Harrison, a Scottish printer living in Australia, discovered the cooling effects also. He used ether to clean metal printing type and discovered the cooling effect of the substance while using it.

In 1862 Perkins started his first refrigeration company in a Brewery in Bendingo, Victoria in 1879; refrigeration did not come into homes on a large scale until refrigeration became electric in 1920. It was, however, 1938 before the first mobile refrigeration unit was built [32]. There had been attempts to coo l trucks in earlier times, but the unit usually fell apart due to the jarring and jolting of the truck as the first refrigeration units were very fragile and difficult to operate. The refrigeration units mainly ran on electrical power, which was not available on trucks. In 1938 however, Frederick McKinley Jones developed the first working mobile refrigeration unit. His refrigeration unit became a great success, and it was patented. Joe Numero founded the ’United States Thermo-ControlCompany1, which soon became ‘Thermo King ’, which changed into the Thermo King Cooperation of today.

Mobile refrigeration opened the market for the transportation of perishable foods such as fruit, vegetables and meat. In the early days, most of these products were canned or alternatively, fresh produce was transported over short distances with the use of a lot of dry ice. A huge risk, however, was that the truck would break down during transport, the ice would melt and that the cargo would be spoiled. Other techniques for food preservation were drying, smoking, pickling or salting, but these techniques had the disadvantage of severely changing the appearance and taste of the food and only worked with specific types of food for a limited time period.

The invention of mobile refrigeration changed all of this. Grocers quickly realized the advantage of this invention where a fresher and greater variety of perishable foods became available everywhere. The market for mobile refrigeration was born. [33]

2.2 Vapor compression refrigeration system

Compression refrigeration cycles take advantage of the fact that compressed fluids at a certain temperature have a tendency of getting colder when allowed to expand. If the pressure change is sufficiently high, then the compressed gas will become hotter than the source of cooling (ambient air) and the expanded gas will become cooler than the desired cold temperature. In this case, fluid is used to cool a low temperature environment and reject heat to a high temperature environment. Vapor compression refrigeration cycles have two advantages. First; a large amount of thermal energy is required to change state of refrigerant from liquid to vapor hence extracting a lot of heat from the conditioned space. Secondly, the isothermal nature of the vaporization allows extraction of heat without raising the temperature of the working fluid to that of the substance to be cooled. This means that the heat transfer rate remains high, because the closer the working fluid temperature approaches that of the surroundings, the lower the heat transfer rate. The refrigeration cycle is shown in figure 2.1 & 2.2 below and consists of the following stages;

1-2: Low pressure liquid refrigerant in the evaporator absorbs heat from its surroundings, usually air, water or some other process liquid. During this process, it changes its state from liquid to gas and at the evaporator exit it is slightly superheated.

2-3: The superheated vapor enters the compressor where its pressure is increased. The temperature will also increase due to the fact that some of the energy put into the compression process is transferred to the refrigerant.

3-4: The high pressure superheated gas passes from the compressor into the condenser. The initial part of the cooling process (3-3a) de-superheats the gas before it is turned back into liquid (3a-3b). The cooling for this process is usually achieved by using air or water. Further reduction in temperature happens in the pipe work and liquid receiver (3b-4), so that the refrigerant liquid is sub cooled as it enters the expansion device.

4-1: The high pressure sub cooled liquid passes through the expansion device, which both reduces its pressure and controls the flow into the evaporator.

The condenser has to reject the combined heat inputs of the evaporator and the compressor. It should be noted that there is no heat loss or gain in the expansion device. [27]

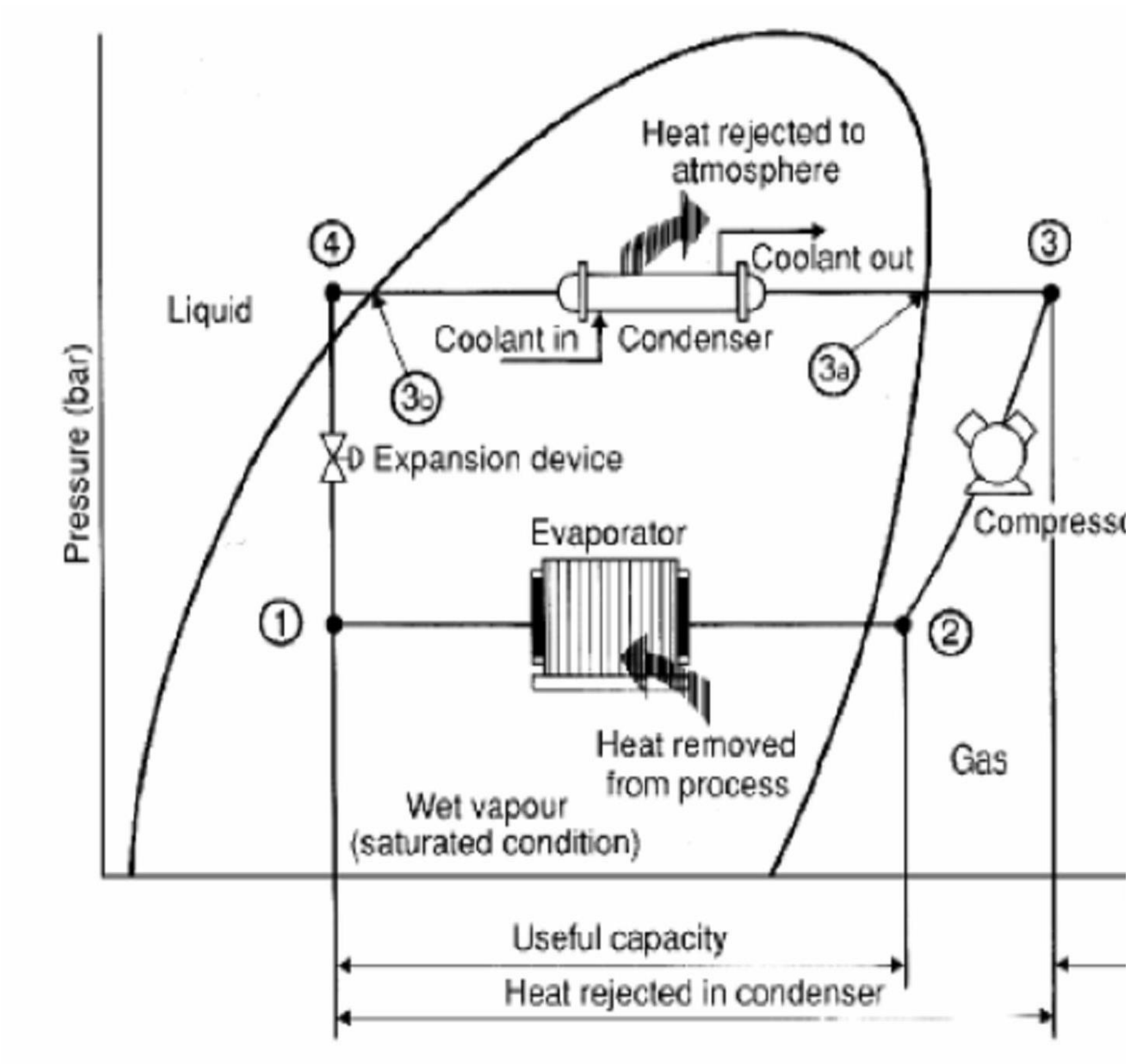


Fig. 2.1 Schematic of the refrigeration cycle (Source: Bureau of Energy Efficiency, 2004)

2.3 Overview of Today's Transport Refrigeration

Transport refrigeration consists of refrigeration systems for transporting chilled or frozen goods. Transport takes place by road, rail, air and sea; further, containers as refrigerated systems are used with moving carriers. All transport refrigeration systems must be sturdily built to withstand movements, vibrations and accelerations during transportation, and be able to operate in a wide range of ambient temperatures and weather conditions. [14]

Transport refrigeration is essential in today's society, to preserve and protect food, perishable goods, flowers, drugs and medical supplies in the cold chain. This sector includes transport of refrigerated products with reefer ships, intermodal refrigerated containers, refrigerated railcars, refrigerated marine and air applications, and road transport including trailers, diesel trucks and small trucks and vans. [3]

As a general rule, transport refrigeration equipment is designed to maintain temperature, not to cool down goods, so goods should be loaded already pre-cooled to carriage temperature. Exceptions are shipments of bananas and citrus fruit, for which appropriate cooling regimes are established. It is essential to establish rules governing hygiene, particularly with respect to cleaning of the body following transport. [4] The majority of new transport refrigeration units use vapor-compression refrigeration with HFC refrigerants. Some equipment uses total-loss refrigerants (carbon dioxide or liquid nitrogen) for limited journey times. Road transport units may operate either from the vehicle engine or from an independent diesel engine. Rail units may operate on electricity supplied from a generator wagon. Marine refrigeration is electrically driven from ships' supplies. Intermodal containers are also electrically driven, but can have portable diesel generator sets attached to them. Refrigerated equipment in air transport is relatively rare, and commonly uses "dry ice" (solid carbon dioxide), though some battery-operated systems are available. [4]

All contemporary space cooling and process cooling equipment exploits the fact that a liquid absorbs heat when it evaporates. (There are a few exceptions, which are limited to unusual applications.) A liquid used for cooling is called a "refrigerant." The energy absorbed by the refrigerant in changing from a liquid to a vapor is called its "latent heat." When this heat is drawn from a body, the body is cooled (see figure 2.2). For a good demonstration of this, swab some alcohol on your arm. The wet spot will chill your skin until it all evaporates.

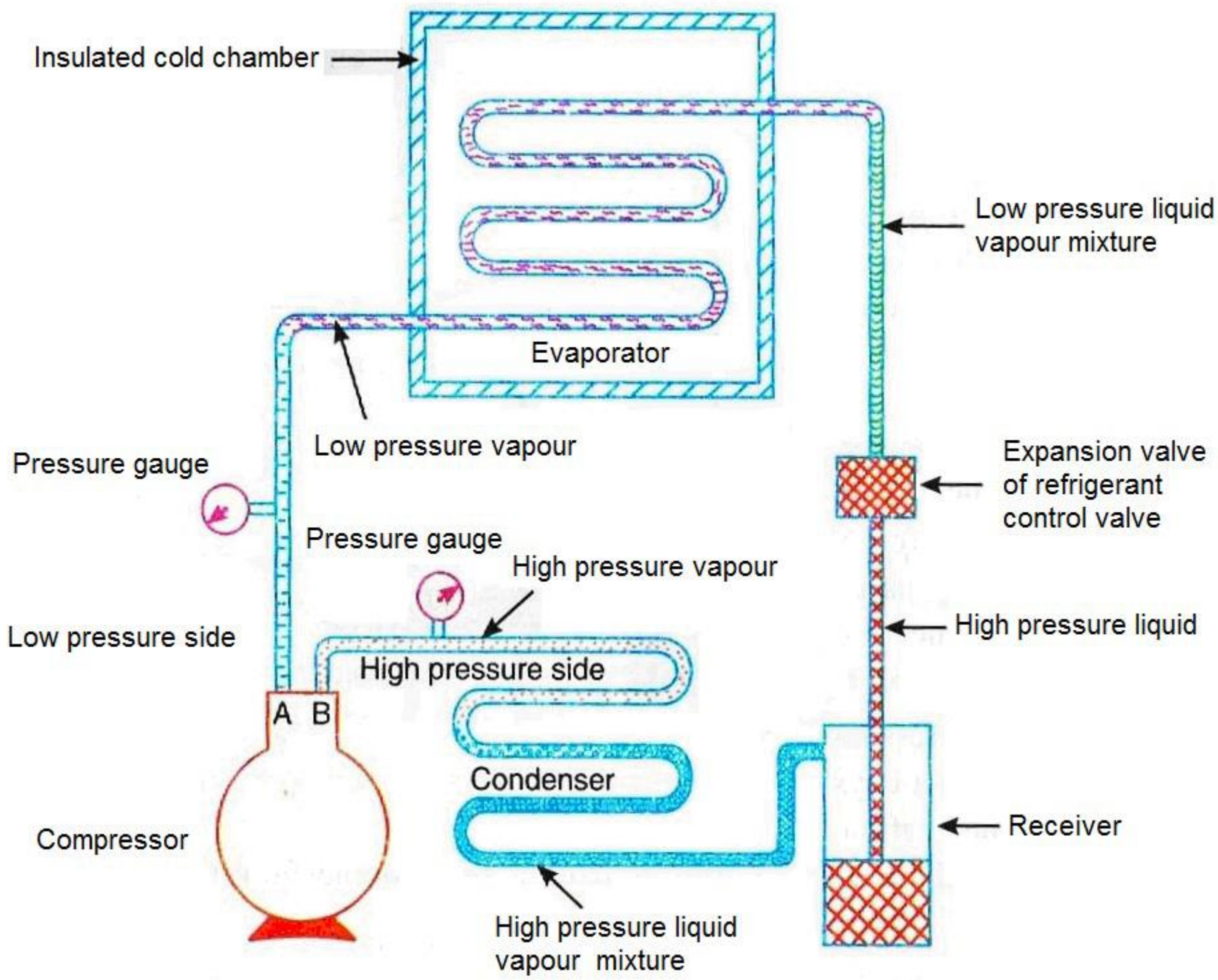


Figure 2.2 Simple vapor compression refrigeration system (Source: Bureau of Energy Efficiency, 2004)

You can create cooling without any machinery if you have enough liquid to evaporate. Water has been used for cooling since before the dawn of humanity. However, it is not very satisfactory as a coolant under ambient conditions because it evaporates too slowly. There are many liquids that provide a large evaporative cooling effect, but they are not cheap enough to be used on a once-through basis. Mechanical cooling can use more effective refrigerants and recycle them indefinitely. The process of doing this is called a “cooling cycle.” The outer shell or casing of the cooling equipment serves as a pressure vessel, isolating the refrigerant from air and atmospheric pressure. This allows the cooling cycle to operate under conditions that provide the greatest cooling capacity and efficiency from the refrigerant.

The most common kind of cooling equipment uses a compression cooling cycle. After the refrigerant has been evaporated by heat from the cooling load, the vapor is compressed. This raises the temperature of the gas well above ambient temperature, so that the heat in the gas can be removed by cooling it with air or water at ambient temperature. Removing the heat causes the compressed gas to condense back to a warm liquid.

The warm refrigerant liquid from the condenser is metered into the cooling area (evaporator) by a flow control device of some kind. The pressure in the evaporator is determined by the suction of the compressor and by the rate of evaporation. Because the evaporator pressure is lower than the condenser pressure, a small portion of the liquid refrigerant “flashes” into vapor when it passes through the control device. This flashing cools the remaining liquid to the temperature of the evaporator. The liquid refrigerant is now ready to absorb heat from the cooling load, repeating the cycle.

Nowadays, selecting the refrigerant may be the first choice you make in selecting or designing a cooling system. The choice of refrigerant limits the system’s efficiency and affects the design of all the system hardware. Safety concerns with some refrigerants dictate where the equipment can be located, and how the surrounding space must be designed. Refrigerant selection has recently become more important and complex because of environmental concerns. [17]

Auxiliary electric-power system is used for the power source for controlling each train system (i.e. air-conditioning or refrigeration system units, heater units, lighting, and battery chargers), and for the power sources for the train-monitoring system, on-board signaling equipment, and communication equipment. [38]

ASHREA (2010) discuss that shock and vibration, on vehicles used for temperature-controlled transport, is primary design concerns because equipment travels with the vehicle it serves, often includes an internal combustion engine and compressor, and may be subjected to occasional rough handling. It is good design practice to avoid equipment natural frequencies of 10 Hz or less (these are typical vehicle wheel rotation and suspension inputs). Also, equipment should not have natural frequencies close to the equipment engine's firing frequency, compressor's pumping frequency, or rpm of any of its rotating components. Vibration criteria are difficult to establish because of the many variables involved. For example, vehicle speed and road characteristics have an input that varies widely. Equipment manufacturers often have criteria based on their testing and experience. [2]

Xie R., et al., 2009 stated that "Refrigerated transportation is a key of Cold-Chain. Condign transport condition is the basis for ensuring quality of perishable goods such as meat, fish, fruit, vegetable, egg, milk, etc. It also plays an important role for saving food resource, keeping food nutrition value, increasing sale income and decreasing energy consumption." [34]

CKP Hui, et al., 2008 stated that "Solar radiation increases significantly the refrigeration load of the refrigerated railcar. The cooling requirements of stationary vehicles have been found to increase by 20% when exposed to sunlight for several hours. That is why aluminum plates and reflective paints are usually used on the exterior of refrigerated vehicles to reduce the heat gain through radiation." [21]

ASHREA (2010) discuss the infiltration load from air entering through cracks in the front of a moving vehicle. The ram air pressure is 300 Pa at 80 km/h, and an exposed 645 mm2 opening can allow 32.6 m3/h of air to enter. At ambient conditions of 38°Cand 50% RH and a vehicle temperature of –18°C, the extra infiltration load is approximately 1115 W. [2]

ASHREA (2010) presents very general ambient temperatures for equipment design for vehicle equipment operation in several geographical regions (see table below). Caution is required when using climatic design information because (1)more extreme temperatures may be encountered at locations in the region that are some distance from the station where data are taken, and (2) higher transient values of 55°C or more may occur because of recirculation in sheltered areas or heat rejected from other nearby sources. [2]

Table 2.1 Ambient Temperatures for Equipment Design in Several Geographical Regions [2]

	Asia	Europe	Mideast	North America	Tropics	Global
Maximum, °C	50	45	50	50	45	50
Minimum, °C	-40	-35	-35	-40	0	-45

Source: ASHRAE

Tassou S.A., et al (2008) stated that "The most common refrigeration system in use for refrigerated food transport applications today is the vapor compression system. The performance and power requirements of these systems are normally assessed at full load. In reality however, transport refrigeration systems operate over a wide range of loads. Depending on the system design envelope, the expected Coefficient of Performance (COP) is generally in the range between 0.5 and 1.75."

CHAPTER THREE

REFRIGERATED RAILCAR ENVELOPE DESIGN

3.1 Introduction

Railway refrigerator cars are insulated boxcars, usually 15 to 20 m long. As illustrated in figure 1.2, they may have a machinery compartment at one end. Vehicles used for temperature-controlled transport are similar in construction and outward appearance to those in general freight service, but have three fundamental differences: they have (1) insulation that is usually foamed in place, (2) provisions for conditioned air circulation through and around the cargo, and (3) machinery for cooling and/or heating.

Envelope design factors to be considered are similar to those for stationary refrigerated facilities, and include the following:

- Extremes of exterior conditions: temperature, relative humidity, wind, and solar effect
- Desired interior conditions: temperature and relative humidity
- Insulation properties: thermal conductivity, moisture permeability and retention, chemical and physical stability, adhesion, uniformity of application, fire resistance, cost of material and application, and presence of structural members
- Infiltration of air and moisture
- Physical deterioration from the shock and vibration of travel and cargo shifting

Closed-cell foamed-in-place insulation, such as polyurethane, is generally recommended to achieve an approximate thermal conductivity k of 0.0230 W/(m·K). It also helps limit air and water vapor infiltration. Rail cars usually have polyurethane insulation of 75 to 150 mm in walls, and 125 to 200 mm in floors and ceilings. Car front walls are sometimes thicker to resist cargo shifting and support equipment.

Infiltration of moisture and air is affected by the integrity of a car’s exterior surfaces (usually sheet metal with riveted joints). The molded glass-fiber-reinforced plastic sometimes used for truck and trailer exteriors is quite effective. There is some experimentation with composite materials for cargo container bodies. Inside, it is common to use a vapor barrier, such as aluminum foil coated with plastic binder and sealed at joints. Integrity of foamed-in-place insulation (the absence of voids and breaks) is also important. Other physical contributors include the effectiveness of door gaskets and sealing around all exterior-to-interior penetrations. Operational factors are vehicle travel speed and the frequency and duration of door openings. [2]

3.2 Types of Insulation System

Pre-fabricated outer facings with an injected polyurethane core are used as insulation system for ceiling, floor and vertical walls of the car. Polyurethane foam is the preferred insulates used in the prefabricated panels. A vapor seal stainless steel sheets of thickness 2 mm are used for external facing materials of the pre-fabricated panels. A vapor seal aluminum foils coated with plastic binder of thickness 2 mm are used as internal facing materials of the pre-fabricated panels of the refrigerated railcar envelop.

Door, at the center of side wall, is constructed of panels with the same insulating material and thickness of the side walls to ensure the chamber space can maintain inside temperatures against a worse case outside temperature scenario. The door panels seal against the chamber wall with inflatable or special gasket material held in aluminum foil profiles.

3.3 Heat Resistance of Insulation System

The insulation sandwich panels are composed of two facing materials covering a polyurethane board at both sides. The stainless steel sheets having a thickness of t_{ss} = 2 mm is used as external facing material and the aluminum foils coated with plastic binder having a thickness of t_{al} = 2 mm is used as internal facing material. Nevertheless this will not affect the heat resistance of the panel because of the high thermal conductivity of galvanized steel and aluminum foil coated with plastic binder.

Thermal conductivity of Stainless Steel: k_{ss} = 16.3 W/m•K

Thermal conductivity of Polyurethane: k_{pu} = 0.023 W/m•K

Thermal conductivity of Aluminum Foil: k_{al} = 204.3 W/m•K

The external heat resistance $R_{ss,ext}$(K/W) for a stainless steel sheet is defined as follows:

$R_{ss,ext}=\frac{t_{ss}}{K_{ss}S_{ext}}$, where S_{ext} : external surface area, in contact with ambient air.

Analogous we can define the internal heat resistance $R_{al,int}$ (K/W) for aluminum foil coated with plastic binder:

$R_{al,int}=\frac{t_{al}}{K_{al}S_{int}}$, where S_{int} : internal surface area, in contact with cooled air.

The heat resistance for a plane surface due to convection R_{conv}(K/W) is defined as follows:

External convection

$$R_{conv,ext} = \frac{1}{h_{ext}S_{ext}}$$

Internal convection

$$R_{conv,int}= \frac{1}{h_{int}S_{int}}$$

For the polyurethane panel the heat resistance R_{pu} (K/W) is defined as follows using the geometric mean surface:

$R_{pu}= \frac{t_{pu}}{K_{int}\sqrt{S_{int}S_{ext}}}$, where t_{pu}: thickness of polyurethane panel

The total heat resistance R_{total} of the insulation panel is the sum of the heat resistance due to internal convection, conduction in the external stainless steel sheet and internal aluminum foil, conduction in the polyurethane panel and external convection. Thus:

$$R_{total} = R_{conv,int} + R_{al,int} + R_{pu} + R_{ss,ext} + R_{conv,ext}$$

$$= \frac{1}{h_{int}S_{int}} + \frac{t_{al}}{K_{al}S_{int}} + \frac{t_{pu}}{K_{int}\sqrt{S_{int}S_{ext}}} + \frac{t_{ss}}{K_{ss}S_{ext}} + \frac{1}{h_{ext}S_{ext}}$$

Consider following recommended (by AHSRAE) insulation R-values for holding freezer built within an unconditioned space. The values are given for a temperature range of -18°C to -25°C.

Table: 3.1 Thermal resistances (R – value) recommended by ASHRAE [2]

Construction	R – value (m^2K/W)
Ground Floors	4.8 - 5.6
Walls, Vertical	6.2 - 7.0
Roofs/ Suspended Ceiling	7.9 - 8.8

Source: ASHRAE

For the calculation of transmission load we took the standard thickness of the polyurethane panel so that the total heat resistance of the walls accomplishes the above recommendations we find in ASHRAE. The internal and external heat transfer coefficients air calculated using EES (engineering equation solver) software.

Table 3.2 Calculation results of the heat transfer through the ceiling, sidewalls, front & back walls and floor with standard insulation size.

Car Wall	t, m	$S_{int/ext}$, m^2	k, $\frac{W}{m \cdot K}$	h_{int}, $\frac{W}{m^2 \cdot K}$	h_{ext}, $\frac{W}{m^2 \cdot K}$	U, $\frac{W}{m^2 \cdot K}$	R, $\frac{K}{W}$
Ceiling	0.200	47.21/51.73	0.023	12.42	5.22	0.112	0.181
Floor	0.200	47.21/51.73	0.023	12.42	1.57	0.106	0.190
Right Side	0.105	44.65/52.56	0.023	12.56	4.44	0.205	0.100
Left Side	0.105	44.65/52.56	0.023	12.56	4.44	0.205	0.100
Front	0.140	6.71/8.39	0.023	12.42	4.60	0.157	0.849
Back	0.140	6.71/8.39	0.023	12.42	4.60	0.157	0.849

k=thermal conductivity,
h_{int}=internal heat transfer coefficient,
h_{ext}=external heat transfer coefficient,
U=overall heat transfer coefficient,
R=heat resistance,
$S_{int/ext}$ =internal & external surface

3.4 Refrigerating Space Calculation

3.4.1 Introduction

In this section the space required to preserve and transport the frozen meat by the refrigerated railcar freezing chamber is calculated. The data here used is the maximum batch mass which amounts 60000 kg of frozen meat of beef carcasses. And also the internal volume of (17.72 * 2.664 * 2.520 = 118.96 m^3) is taken.

3.4.2 Calculations

To find the volume of that mass of frozen meat of beef we need the density. ASHRAE provides us the unfrozen composition data of meat of beef.

Table 3.3 Beef meat composition [2]

Components	Beef, meat
Moisture content x_{wo}(%)	57.26
Protein x_p (%)	17.32
Fat x_f (%)	24.05
Carbohydrate x_c(%)	0
Fiber x_{fb} (%)	0
Ash x_a (%)	0.81
Other components x_o (%)	0.56

Source: ASHRAE

ASHRAE offers thermal property models for food components between –40°C and 150°C.

Density (kg/m³)

Protein: $\rho = 1.3299 \times 10^3 - 5.1840 \times 10^{-1} T$

Fat: $\rho = 9.2559 \times 10^2 - 4.1757 \times 10^{-1} T$

Carbohydrate: $\rho = 1.5991 \times 10^3 - 3.1046 \times 10^{-1} T$

Fiber: $\rho = 1.3115 \times 10^3 - 3.6589 \times 10^{-1} T$

Ash: $\rho = 2.4238 \times 10^3 - 2.8063 \times 10^{-1} T$

Water: $\rho = 9.9718 \times 10^2 + 3.1439 \times 10^{-3} t - 3.7574 \times 10^{-3} T^2$

Ice: $\rho = 9.1689 \times 10^2 - 1.3071 \times 10^{-1} T$

Where, T is the food temperature.

The meat enters the refrigerated railcar freezing compartment at a temperature of -10°C and leaves at –18°C.

As usual we take the most extreme case: t = -10°C. In next table we list the densities of all components of the meat.

Table 3.4 Component densities at -10°C and –18°C

Density (kg/m³)	-10°C	-18°C
Water	-	-
Ice	918.2	919.24
Protein	1335.08	1339.23
Fat	929.77	933.11
Carbohydrate	1602.2	1604.69
Fiber	1315.16	1318.09
Ash	2426.61	2428.85

Now the density of the meat is calculated with following formulas:

Frozen meat at -10 °C and -18 °C

$$\boldsymbol{\rho_{meat}} = \frac{(\boldsymbol{\rho_{ice}} * \mathbf{x_{wo}}) + (\boldsymbol{\rho_{protein}} * \mathbf{x_p}) + (\boldsymbol{\rho_{fat}} * \mathbf{x_f}) + (\boldsymbol{\rho_{carb}} * \mathbf{x_c}) + (\boldsymbol{\rho_{fiber}} * \mathbf{x_{fb}}) + (\boldsymbol{\rho_{ash}} * \mathbf{x_a})}{\mathbf{x_{wo}} + \mathbf{x_p} + \mathbf{x_f} + \mathbf{x_c} + \mathbf{x_{fb}} + \mathbf{x_a}}$$

The results are collected in the table below.

Table 3.5 Beef meat density

Density (kg/m³)	-10°C	-18°C
Beef, meat	1005.895	1008.043

Table 3.6 shows the volume taken by the meat. The volume is found by dividing the total mass of the meat by the density.

Table 3.6 Beef meat volume

Volume (m^3)	-10°C	-18°C
Beef, meat	59.65	59.52

Previously we have decided that the available volume is 118.96 m^3. If we now take a closer look to percentage of the freezing space volume filled with meat (table 3.6) we can conclude that there is still enough free space left to have sufficient air circulation.

Table 3.7 Percentage of the freezing space volume filled with meat

Filled volume (%)	-10°C	-18°C
Beef, meat	50.14	50.03

3.5 Air Circulation in the Refrigerated Railcar

3.5.1 Introduction

Air circulation is one of the most important factors in protecting refrigerated loads of perishable foods. Refrigeration capabilities are meaningless if the refrigerated air is not properly circulated to maintain product temperature. Air is the secondary refrigerant that removes heat from the produce and its surroundings and carries it to the evaporator where it is cooled and discharged again into the refrigerated space. Heat, both sensible and latent due to water vapor condensation, is absorbed by the evaporator surface. [13]

Air movement inside the freezing chamber serves two main functions: first, the atmospheric temperature and relative humidity are homogenized to keep them reasonably uniform; second, evaporator efficiency is improved, in that the heat transfer coefficient is increased as air speed rises above it.

Air movement may be enhanced by differences in air density, known as natural convection, or may be promoted by mechanical action, which is called forced convection. Fans or blowers are used to control the air speed in the chamber and also over the evaporator. An air circulation pattern is established inside the freezing room, the evaporator being the focal point, while the rest is governed by smaller forces that direct the air as it moves through the stacking and recirculation spaces.

When using forced convection systems, uniform atmospheric conditions depend on the air being dispersed with reasonable consistency, and if the circulation pattern involves much horizontal travel the product stacking design and the air flow pattern should be compatible.
Stacked rows of produce sometimes serve to regulate the airflow by creating a uniform resistance all across the room without interfering with circulation.

The air circulation pattern is dependent on several factors: fan air flow; devices for distribution and suction; quantity of stored produce and stacking patterns; insulation efficiency; dimensions and shape of the chamber; external ambient temperature; pressure differences due to air flow; operating condition of the evaporator and frost accumulation. From this it can be seen that the circulation pattern changes according to the moment of the operating cycle and the state of the evaporator surface.

Air distribution in the cold store is achieved by using long-throw fans. The throw of the air delivery must be sufficiently far-reaching to ventilate adequately all points of the cold room.

It is clear that the most important factor in air distribution, no matter which system is used, is the stacking pattern. The air circulation in the freezing room must be correct no matter the state of loading. If the stacking pattern requirements are all met and the temperature still remains too high in certain positions, the reason is most probably an insufficient supply of cold air. This can be overcome with higher air velocity flows.

For freezing chambers the mean air speed in the zone where the product is placed is the characteristic factor, which defines air distribution. It is reported that in most freezers the air speed must be about 0.5 m/s or higher, measured in the empty section. The air velocity through the evaporator should be about 3m/s to achieve high overall heat exchange coefficients.

The air circulation must be permanently maintained, as a sufficient supply of cold air to the product surface is necessary to sustain adequate freezing rates, independently of air or evaporator temperatures.

3.5.2 Air Flow Rate Calculation

With following formula it is possible to estimate the airflow rate that has to be accomplished in the freezing chamber, with a certain refrigerating load given.

$$\dot{V} = \frac{\dot{Q}}{\Delta T * \rho_{air} * C_p}$$

Where

$\dot{V}$ = Flow rate (m^3/ h)

$\dot{Q}$ = Refrigeration load (kW)

ΔT = temperature difference incoming air - leaving air (K)

ρ_{air} = Air density (kg /m^3)

C_p = specific heat of air (kJ/kg K)

Before, we found the refrigeration load: Q = 19 kW. We assume the air heats 3.3°C when refrigerating the meat. This means ΔT = 3.3 K. At -18°C the air density ρ air = 1.3834 kg/m^3 and the specific heat Cp = 1.003 kJ/kg K.

Substituting these values in the above equation we get a total airflow rate:

$\dot{V}$ = 14,938m^3/h.

This is the airflow rate blown through a section of 2.664 meters wide and 2.52 meters high. Because we calculated the percentage of the freezing space volume filled with meat we are able to estimate the air velocity.

So, the section surface S = 2.664 x 2.52 = 6.71 m2. We define the free section S' as the free surface that will let pass the air. S'= S x (1 - percentage of volume filled with meat). In the table below we collect the results.

Table 3.8 Free section per 2.664 m wide of the refrigerated railcar

Free section S' (m^2)	**-10°C**	**-18°C**
Beef, carcass	3.346	3.353

The approximate air velocity v_{air} is calculated as follows:

$$\mathbf{v_{air}} = \frac{\dot{\mathbf{V}}}{\mathbf{S' * 3600}}\,\mathbf{m/s}$$

Table 3.9 Approximate air velocity inside the refrigerated railcar

v_{air} (m/s)	**-10°C**	**-18°C**
Beef, carcass	1.240	1.237

CHAPTER FOUR

REFRIGERATING LOAD CALCULATION

4.1 Introduction

The cold storage of perishable products maybe is the most important link in the cold chain and the cold rooms are the most important link through which pass the food products. One cold storage consists of usually several cold rooms in which food products are kept at lower temperatures in order to preserve them in short or longer term. The storage temperature of food products in one cold room is very important for many aspects such as the product quality and durability, but also for the energy consumption. The choice of proper temperature can be made from the recommendations published in. The temperature in the cold rooms should be from 0 to 4 ^{0}C for chilled and -18 to -25 ^{0}C for frozen products.

Due to the higher temperature of the surrounding air, the heat passes through the walls of cold room to its interior. In order to prevent or reduce heat gains, the enclosure surfaces are performed with thermal insulation. Besides, there are additional heat gains (of products, equipment, etc.) in the cold room which should be taken into account in order to obtain the total refrigeration load required for the selection of equipment.

Cold rooms may vary depending on the insulation material, the structure (layers) of walls, type of product, the temperature mode of use, etc. All these facts affect the larger or smaller amounts of the heat gains, so that the two cold rooms with equal size can have a large difference in cooling capacity. Therefore, it is incorrect that the cooling capacity, and choice of equipment to be done according to the volume or floor area of cold rooms.

It is very important to accurately determine the capacity of cold rooms. If less than required, the refrigerating system will run non-stop and may not achieve the required temperature regime. If the calculated capacity is greater than required, the refrigeration equipment will be bigger and unnecessary more expensive, and the system will work with a lot of stops because it will achieve the temperature regime very quickly. In this way, and the energy efficiency of the system will be lower. [8]

4.2 Refrigerating Load Calculation

4.2.1 Product Load

The primary refrigeration loads from products brought into and kept in the refrigerated space are (1) heat that must be removed to bring products to storage temperature and (2) heat generated by products (mainly fruits and vegetables) in storage. [20]To calculate independently the heat loads produced while transporting of the required mass of chilled beef meat, frozen beef meat and chilled vegetables we need the following data:

Maximum mass: m

Vehicle travel time: t

Maximum beef meat/vegetables temperature when entering the freezing room: T_{in}

Beef meat/vegetables temperature when leaving the freezing room: T_{out}

Initial freezing point: T_f

Specific heat above freezing: C_{p1}

Specific heat below freezing: C_{p2}

Latent heat: L_{h1}

ASHRAE provides us the initial freezing point, the specific heats and the latent heat of fresh beef and also vegetable.

Following formula calculates the heat load:

$$\dot{Q} = \frac{mCp1(T_{in} - T_f) + mL_h + mCp2(T_f - T_{out})}{t}$$

Beef Meat (Freezing)

Maximum mass: m = 60,000 Kg.

Vehicle travel time: = 24 Hour (1 day) = 86400 Sec

Maximum beef meat temperature when entering the freezing room, T_{in} = -10°C

Maximum beef meat temperature when leaving the freezing room, T_{out} = -18°C

Initial freezing point: T_f= -2.8°C

Specific heat above freezing: C_{p1} = 3.14kJ/kg K

Specific heat below freezing: C_{p2}= 1.67kJ/kg K

$$\dot{Q}_{beef/f} = \frac{mCp2(T_{in} - T_{out})}{t}$$

$$= \frac{60{,}000\ Kg * 1.67\ ^{kJ}/_{Kg.\ K}(-10 - (-18))K}{24\ Hr.}$$

= **9.28 kW**

Beef Meat (Chilled)

Maximum mass: m = 60,000 kg.

Vehicle travel time: = 24 hrs (1 day) = 86400 sec.

Maximum beef temperature when entering the car chilling room, T_{in}= 2°C

Maximum beef temperature when leaving the car chilling room, T_{out}= -1°C

Specific heat above freezing: C_{p1}= 3.520kJ/kg K

$$\dot{Q}_{beef/c} = \frac{mCp1(T_{in} - T_{out})}{t}$$

$$= \frac{60{,}000\ Kg.*3.52\ {}^{kJ}/_{Kg.\ K}\,(2\ -\ (-1))\ K}{24\ hrs.}$$

= **7.33 kW**

Vegetables (Mixed/General)

Maximum mass: m = 42,000 kg.

Vehicle travel time: = 24 hrs (1 day) = 86400 sec.

Maximum vegetable (mixed) temperature when entering the car chilling room, $T_{in} = 3^oC$

Maximum vegetable (mixed) temperature when leaving the car chilling room, $T_{out} = 0^oC$

Specific heat above freezing: C_{p1}= 3.768kJ/kgK

Heat of respiration of vegetable (mixed) at 0^oC= 0.1397 kJ/kg Hr

Product pull-down load will be

$$\dot{Q}_{beef/c} = \frac{mCp1(T_{in} - T_{out})}{t}$$

$$= \frac{42{,}000\ Kg.*3.768\ {}^{kJ}/_{kg.\ K}\,(3\ -\ 0)\ K}{24\ hrs.}$$

= 5.495 kW

Product heat of respiration of the fresh vegetable (mixed) at 0^oC,

$\dot{Q}_{resp.}$ = 42,000 kg. * 0.1397${}^{kJ}/_{kghr}$

= 1.630 kW

4.2.2 Transmission Load

First it will be given a theoretical expansion with the formulas and correlations that will need to calculate the heat transfer through a wall. It has to be taken into account all mechanisms of heat-transfer: convection, conduction and solar radiation.

Convection

Internal Convection

Flow parallel to plane surfaces:

Reynolds number:

$$Re = \frac{\rho V_{air} L}{\mu}$$

Prandtl number:

$$Pr = \frac{\mu C_p}{k}$$

Nusselt number:

The boundary layer flow regimes are laminar for $Re_x < 2 \times 10^5$ and turbulent for $Re_x > 3 \times 10^6$

For the laminar range

$Nu_x = 0.332\, Re_x^{\frac{1}{2}} Pr^{\frac{1}{3}}$ and

$Nu_L = 0.664\, Re_L^{\frac{1}{2}} Pr^{\frac{1}{3}}$

For the turbulent range

$Nu_x = 0.0288\, Re_x^{\frac{4}{5}} Pr^{\frac{1}{3}}$ and

$Nu_L = 0.036\, Re_L^{\frac{4}{5}} Pr^{\frac{1}{3}}$

Fluid properties should be evaluated at the film temperature. The film temperature T_f is defined as the arithmetic mean between the wall and the bulk temperature:

$T_f = \frac{T_w + T_{ext}}{2}$

Thus, Internal convection coefficient

$h_{int} = \frac{kNu_L}{L}$

We assume the Nusselt number to be constant and evaluated at x = L

External Convection

According to ASHRAE, the temperature difference To-Ti can be adjusted to compensate for the solar effect on heat load. Table 4.1 represents the degrees centigrade which must be added to the normal temperature difference to compensate for the sun effect.

Table 4.1 Conventional temperature differential increase to compensate for sun effect [2]

Exposition Color	**East**	**South**	**west**	**Flat roof**
Dark color	4.5	2.6	4.5	11
Medium color	3.3	2.2	3.3	8.6
Light color	2.2	2.1	2.2	5

Source: ASHRAE

For Horizontal Wall Plates

Grashof's number:

$Gr = \frac{\beta g \rho^2 \Delta T L^3}{\mu^2}$

Rayleigh number:

Ra = Gr Pr

The correlations suggested by McAdams are well accepted for this geometry. A distinction is made regarding whether the surface faces up or down. It is clear that the induced buoyancy will be much different for a hot surface facing up than down. McAdams's correlations are, for a hot surface facing up or cold surface facing down

$10^5 < Ra_L < 2 * 10^7$ $\quad$ $Nu_L = 0.54\, Re_L^{\frac{1}{4}}$

$2 * 10^7 < Ra_L < 3 * 10^{10}$ $\quad$ $Nu_L = 0.14 Re_L^{\frac{1}{3}}$

and for a hot surface facing down or a cold facing up

$3 * 10^{10} < Ra_L < 10^{13}$ $\quad$ $Nu_L = 0.27\, Re_L^{\frac{1}{4}}$

In each of these correlating equations the film temperature T_f, should be used for the air property evaluation. The length scale, L, is the ratio of the plate-surface area to the perimeter.

$$h_{ext} = \frac{kNu_L}{L}$$

For Vertical Wall Plates

Transition from laminar to turbulent flow in natural convection boundary layers adjacent to vertical plane surfaces has been determined to occur at, or near

$Gr_t\, Pr = Ra_t = 10^9$, where t, indicates transition.

Churchill and Chu have correlated a large amount of experimental data for natural convection adjacent to vertical planes over 13 orders of magnitude of Ra. They propose a single equation for Nu_L that applies to all fluids. This powerful equation is

$$Nu_L = \left\{ 0.825 + \frac{0.387\, Ra_L^{\frac{1}{6}}}{\left[1 + \left(\frac{0.492}{Pr}\right)^{\frac{9}{16}}\right]^{\frac{8}{27}}} \right\}^2$$

Churchill and Chu show this expression to provide accurate results for both laminar and turbulent flows. Some improvement was found for the laminar range ($Ra_t < 10^9$) by using the following equation:

$$Nu_L = 0.68 + \frac{0.670\, Ra_L^{\frac{1}{4}}}{\left[1 + \left(\frac{0.492}{Pr}\right)^{\frac{9}{16}}\right]^{\frac{4}{9}}}$$

Table 4.2 Surface temperature of vertical walls, ceiling and floor due to sun effect

Envelop Wall	Sun Effect, K	External Surface Temperature,
Ceiling	5	45
Floor	0	40
Wall, Front	2.1	42.1
Wall, Back	2.1	42.1
Wall, Right Side	2.2	42.2
Wall, Left Side	2.2	42.2

The heat transmission $\dot{Q}$ through the walls can be calculated using the following formula

$$\dot{Q} = \frac{T_{ext} - T_{int}}{R_{total}}$$

And with this result of heat transmission $\dot{Q}$, we can calculate the temperatures of the interior and exterior wall surface as follows:

$$T_{w,ext} = T_{ext} - \frac{\dot{Q}}{h_{ext} * S_{ext}}$$

$$T_{w,int} = T_{int} - \frac{\dot{Q}}{h_{int} * S_{int}}$$

Where

Tw, ext : Exterior wall surface temperature

Tw, int : Interior wall surface temperature

Table 4.3 Calculation results of vehicle body transmission load

Variables	Ceiling wall	Floor wall	Front/Back wall	R& L Sidewall
t_{pu}	0.200	0.200	0.140	0.105

h_{int}	12.42	12.42	12.42	12.56
h_{ext}	5.22	1.57	4.60	4.44
R_{tot}	0.181	0.190	0.849	0.100
$\dot{Q}$ (W)	**348**	**305**	**71**	**602**
T_{int}/T_{ext}	-18/45	-18/40	-18/42.1	-18/42.2
$T_{w,int}$	43.66	35.76	40.11	39.12
$T_{w,ext}$	-17.38	-17.41	-17.12	-16.8

4.2.3 Infiltration through the Vehicle Body and Closed Doors

Infiltration loads (the ingress of warmer air into a cold space) are significant, particularly for cold stores. The freezing of moisture entering with the air will be a latent load in the store and subsequently the frost removal will be an additional defrost load. [15]

Sensible Heat Gain due to Infiltration through the Vehicle Body and Closed Doors

Infiltration into insulated vehicles occurs even when they are stationary, probably because of stack effect caused by the inside-to-outside temperature difference. The infiltration driving force for a vehicle 2.4 m high with a 55 K temperature difference is about 7.5 Pa. Openings with an aggregate area of 645 mm^2each at the top and bottom allow infiltration of about 3.4 m^3/h, if assumed to be thin-plate orifices. [2]

Thus the sensible infiltration load can be calculated using the following formula.

$\dot{Q}$ = V * AirΔ_{rate} * ρ C_p (T_{ext} - T_{int}) where,

Volume of the refrigerator railcar compartment, V = 118.96 m^3

Infiltration air change factor, AirΔ_{rate} = 6.3 per 24 hr.

Air density and specific heat at 40°C outside condition, ρ C_p = 1.135 KJ/m^3 K,

Outside temperature, T_{ext} = 40°C,

Inside temperature, T_{int} = -18°C,

And substitution of the data in above formula gives:

$\dot{Q}$ = **0.596 KW**

Latent Heat Gain due to Infiltration through the Vehicle Body and Closed Doors

The humidity ratio of air is 0.04913 kg water vapor/kg dry air at 40°C and 95 percent relative humidity, and 0.00073 kg water vapor/kg dry air at -18°C and 85 percent relative humidity. The latent heat of vaporization of water at 0°C is 2501 kJ/kg. The density of air at the ambient temperature of 40°C and 1 atm. is 1.127 kg/m3, and its specific heat is cp = 1.007 kJ/kg°C.

According to ASHRAE the ambient air enters the cargo space at a rate of 32.6 m^3/hr, which, corresponds to amass flow rate of

$\dot{m} = \rho_{\text{air at amb temp.}} * 32.6 m^3/hr$

$= 1.127\ Kg/m^3 * 32.6 m^3/hr$

= 36.74 Kg/hr

Ambient air enters with a humidity ratio of 0.04913 kg water vapor/kg dry air and leaves at humidity ratio of 0.00073 kg water vapor/kg dry air. The difference condenses in the railcar and is drained out as a liquid, releasing h_{fg} = 2501 kJ/kg of latent heat. Then the latent heat gain of the railcar due to infiltration becomes

$\dot{Q}_{\text{inf.}} = (0.04913 - 0.00073) * 36.74 * 2501$

= **1.235 KW**

Thus total infiltration load becomes

$\dot{Q}$ = 0.679 kW+ 1.235 kW

$\dot{Q}$ = **1.831 kW**

4.2.4 Internal Load

All electrical energy dissipated in the refrigerated space (from lights, motors, heaters, and other equipment if any) must be included in the internal heat load. Heat equivalents of electric motors are listed in the table (see appendix B).

The air in the inside compartment of the refrigerated railcar is circulated by the evaporator fans of 1.460 kW. Remind that the meat is hung up on an internal metallic construction. The motor and the driven equipment are located inside the refrigerated railcar. If we look in the table of heat equivalents of electric motors, extracted from ASHRAE, we can make an estimation of the heat load due to evaporator fans ventilation. Because electrical motors in refrigerated room can work in overload the heat gain is bigger than motor power. Corresponding to a 1.820 kW electrical motor (taken from unit cooler data) the heat gain inside the refrigerated railcar due to the evaporator fans is about 1.928 W. As we assume the lights to be switched off when the freezer is in operation, there is no extra heat load due to lighting. So the total internal heat load will be:

$\dot{Q}$int. = **1.928kW**

4.2.5 Equipment Related Load

Heat gain due to evaporator defrosting is difficult to estimate analytically. We will choosehot gas defrosting which will give us the smallest heat gain possible regarding to defrosting. As you will see in the next section we take a 20 percent safety factor for the total heat load including defrosting. We assume that this will be sufficient to cover the heat gain due to defrosting.

4.2.6 Total Refrigerating Load

We recapitulate what the maximum total refrigerating load of the frozen meat consists of:

- Maximum Product load = 9.280 kW
- Transmission load = 1.999 kW
- Total Infiltration load = 1.831 kW
- Internal load = 1.928 kW
- Defrost load = included in the safety factor
- Safety factor (20%) = 3.008 kW

$$\dot{Q}_{\text{total}} = \mathbf{18.046\ kW}$$

4.3 Cooling Load Calculation Using Computer Program

To calculate the total refrigerating load using computer program, at first, the dimensions of cold room are input, and then start calculation of heat gains through enclosure surfaces (vertical walls, ceiling and floor). After calculating of each wall it appears an inter-result: the amount of heat gains for that wall, the coefficient of heat transfer and the specific heat flux by which the effectiveness of wall is assessed. The program contains a database for thermal conductivity of insulation and building materials that can be used if it is needed. Also, the program includes a table of equivalent temperature differences due to solar radiation.

In the calculation of heat gains from the product cooling (entering), the input data are the mass (amount) of products, their input temperature and cooling time. Also, there is a database for the specific heat of products above and below the freezing temperature, and for various packaging materials as well. For the calculation of respiration heat a database from ASHRAE is added for fruits and vegetables. A respiration occurs only at the storage temperature above 0 ^{0}C.

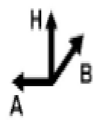
Infiltration heat gains from the surrounding air are calculated by the thermodynamic properties of external and internal air. Therefore, a separate subprogram is incorporated for calculation of properties for moist air.

At the end, a recapitulation of all heat gains are displayed, and the total load of system as well. There is an option to the user wish for printing a complete report which contains many of detailed calculations. [8] For our design process we use a LU – VE Group Cooling load calculation software and the result of the calculation is given below in the following tables. And it is almost in conformance with analytical calculation result.

Table 4.4 LU – VE Software result of total cooling load for frozen meat

Product: **Beef (lean)**			Process: **Frozen produce preservation**		
Freezing Temperature	[°C]	-1,7	Load density	[kg/m3]	504,4
Cp fresh	[kJ/kgK]	3,226	Initial product temperature	[°C]	-10,0
Cp frozen	[kJ/kgK]	1,676	Process time	[hh:mm]	24:00
Freezing heat	[kJ/kg]	234,640	Room relative humidity	[%]	85,0
Respiration heat	[kJ/kg]	0,000	Room temperature	[°C]	-18,0

COLD ROOM CHARACTERISTICS



A:	17,72	[m]
B:	2,66	[m]
H:	2,52	[m]

Internal volume	[m3]	118,96	Daily turnover	[%]	100,0
Outdoor T.	[°C]	40,0	Persons	[n]	0
Relative humidity	[%]	95,0	Movement		Medium

Wall	Surface [m2]	Insulation	Thermal conductivity coeff. [W/(mK)]	Thickness [m]	Outdoor T. [°C]	Loss [kW]
1)	47,2	Polyurethane	0,02303	0,20	45,0	0,335
2)	47,2	Polyurethane	0,02303	0,20	45,0	0,335
3)	44,7	Polyurethane	0,02303	0,10	42,2	0,565
4)	44,7	Polyurethane	0,02303	0,10	42,2	0,565
5)	6,7	Polyurethane	0,02303	0,14	42,1	0,064
6)	6,7	Polyurethane	0,02303	0,14	42,1	0,064

ROOM THERMAL LOADS		[kW]	[kW*24h]
Wall heat losses		1,927	46,259
Outdoor air inlet		2,723	65,356
Product cooling/freezing		9,302	223,256
Product respiration		0,000	0,000
Lighting		0,472	11,329
Persons		0,000	0,000
Addit. thermal load		1,500	36,000
Ventilation		2,800	67,200
Total	(= CT)	18,725	449,400

Table 4.5 LU – VE Software result of total cooling load for chilled meat

Product: **Beef (lean)**			Process: **Fresh produce preservation**		
Freezing Temperature	[°C]	-1,7	Load density	[kg/m3]	504,4
Cp fresh	[kJ/kgK]	3,226	Initial product temperature	[°C]	2,0
Cp frozen	[kJ/kgK]	1,676	Process time	[hh:mm]	24:00
Freezing heat	[kJ/kg]	234,640	Room relative humidity	[%]	85,0
Respiration heat	[kJ/kg]	0,000	Room temperature	[°C]	-1,0

COLD ROOM CHARACTERISTICS

A:	17,72	[m]
B:	2,66	[m]
H:	2,52	[m]

Internal volume	[m3]	118,96	Daily turnover	[%]	100,0
Outdoor T.	[°C]	40,0	Persons	[n]	0
Relative humidity	[%]	95,0	Movement		Medium

Wall	Surface [m2]	Insulation	Thermal conductivity coeff. [W/(mK)]	Thickness [m]	Outdoor T. [°C]	Loss [kW]
1)	47,2	Polyurethane	0,02303	0,20	45,0	0,244
2)	47,2	Polyurethane	0,02303	0,20	45,0	0,244
3)	44,7	Polyurethane	0,02303	0,10	42,2	0,405
4)	44,7	Polyurethane	0,02303	0,10	42,2	0,405
5)	6,7	Polyurethane	0,02303	0,14	42,1	0,046
6)	6,7	Polyurethane	0,02303	0,14	42,1	0,046

ROOM THERMAL LOADS		[kW]	[kW*24h]
Wall heat losses		1,392	33,396
Outdoor air inlet		2,219	53,253
Product cooling/freezing		6,715	161,163
Product respiration		0,000	0,000
Lighting		0,472	11,329
Persons		0,000	0,000
Addit. thermal load		1,500	36,000
Ventilation		1,400	33,600
Total	(= CT)	13,698	328,741

Table 4.6 LU – VE Software result of total cooling load for chilled vegetables (General)

Product: **General**			Process: **Fresh produce preservation**		
Freezing Temperature	[°C]	-1,0	Load density	[kg/m3]	353,1
Cp fresh	[kJ/kgK]	3,771	Initial product temperature	[°C]	3,0
Cp frozen	[kJ/kgK]	1,886	Process time	[hh:mm]	24:00
Freezing heat	[kJ/kg]	293,300	Room relative humidity	[%]	90,0
Respiration heat	[kJ/kg]	3,352	Room temperature	[°C]	0,0

COLD ROOM CHARACTERISTICS

A:	17,72	[m]
B:	2,66	[m]
H:	2,52	[m]

Internal volume	[m3]	118,96	Daily turnover	[%]	100,0
Outdoor T.	[°C]	40,0	Persons	[n]	0
Relative humidity	[%]	95,0	Movement		Medium

Wall	Surface [m2]	Insulation	Thermal conductivity coeff. [W/(mK)]	Thickness [m]	Outdoor T. [°C]	Loss [kW]
1)	47,2	Polyurethane	0,02303	0,20	45,0	0,239
2)	47,2	Polyurethane	0,02303	0,20	45,0	0,239
3)	44,7	Polyurethane	0,02303	0,10	42,2	0,396
4)	44,7	Polyurethane	0,02303	0,10	42,2	0,396
5)	6,7	Polyurethane	0,02303	0,14	42,1	0,045
6)	6,7	Polyurethane	0,02303	0,14	42,1	0,045

ROOM THERMAL LOADS		[kW]	[kW*24h]
Wall heat losses		1,360	32,639
Outdoor air inlet		2,180	52,319
Product cooling/freezing		5,495	131,875
Product respiration		1,628	39,074
Lighting		0,472	11,329
Persons		0,000	0,000
Addit. thermal load		1,500	36,000
Ventilation		1,400	33,600
Total	(= CT)	14,035	336,837

CHAPTER FIVE

EQUIPMENT SELECTION

5.1 Introduction

Refrigeration capacity up to 88 kW of a cold storage with a small plant refrigeration system is usually a single package unit containing evaporator, compressor, air-cooled condenser, receiver, halocarbon refrigerant, and control devices if suitable outside walls are available for the through-the-wall condenser section. Most of the times on mobile refrigeration system, semi-hermetic reciprocating or scroll compressors are most commonly used with integrated air-cooled condensers. One condenser should be used with each compressor. Refrigerant system is the direct expansion halocarbon type refrigerants R-134a or R404A depending on the application types. Evaporator fans are the direct mounted propeller type blowing through a coil bank. Control will be off and on from a thermostat either starting and stopping the compressor or operating a liquid line solenoid valve which will allow the compressor to pump down or shut off on a pressure control. Along with the evaporator an electrical or gas type of automatic defrosts is used. [25]

Now that the refrigeration load for this particular job (preserving and transporting frozen meat) is calculated, it is time to select the most suitable equipment for the application. In this section using our design criteria, the temperature difference or TD, between the room and the saturated suction temperature and the relative humidity in the cold room space together with product data and specifications from manufacturers' literature, we will result in an informed component selection.

5.2 Evaporator

5.2.1 Introduction

Evaporator is the name given to any heat exchanger where the refrigerant is evaporated at low temperature and therefore at low pressure, but usually above atmospheric pressure to prevent gas and/or water vapor leakages into the low-pressure circuit. The evaporator is the element of the refrigerating circuit through which heat is absorbed from the environment that is being cooled. It can absorb the heat necessary for refrigerant vaporization from the air being circulated through the air cooler inside the refrigerated room.

Air-cooling evaporator (see figure 5.1) can operate under natural convection, where air movement is governed by differences in air density, or under forced or mechanical convection when fans or blowers are employed to expedite air movement over the cooler and to facilitate air distribution inside the frozen meat preserving and transporting refrigerated railcar.

Fig. 5.1 Air – cooling evaporating unit *(Dean and Wood Product Catalogue Issue one)*

Natural convection systems have two main advantages: no energy is needed for air circulation and the desiccation of the produce is much less because air velocities are much lower and relative humidity is high. There are obviously some drawbacks, first, in their defrosting and, second, and more important, in their low overall heat exchange coefficient owing to low air velocities over the evaporator that lead to large exchange surface areas, making them bulky and very expensive. Their cost, including installation in the blast-freezing chamber, is from three to four times that of a forced convection evaporator because the exchange surface is larger, the tube manufacturing cost is high and erection in the chamber is difficult and time consuming.

Forced convection or forced draught evaporators have been developed because the higher overall heat transfer coefficient achieved by rapid air circulation over the coils permits a drastic reduction in evaporator surface areas. Further, they have great refrigerating capacity. Air velocities through the evaporator must be in the range of 1.5 to 3 m/s to achieve a high heat-transfer coefficient, but it should be borne in mind that above the upper limit (3 m/s) the heat created by the electric fan motor will surpass the increase in cooling capacity. Also, velocities above 3 m/s will tend to carry moisture deposited on the coils toward the produce. The throw draught must be sufficiently far-reaching to achieve uniform air circulation and an even temperature distribution in the cold store.

Evaporator coils are finned tubes built to produce a large heat exchange area in a compact element. Good thermal contact between the tube and the fins must be assured (soldering, biting into the tube surface by tube expansion, securing by straightening fin flares). Fin size and spacing depend on the operation and fundamentally on the operating temperature. Fin height should usually be equal to the diameter of the tube and the fin should be thick enough to provide mechanical resistance (0.7-1 mm).Fin spacing is between 6 and 15 mm or more, to minimize the risk of ice block. Sometimes fins are spaced wider at the coil air inlet to act as a frost catcher to avoid obstruction of the airflow.

Evaporators fall into two categories, depending on the way they are fed with liquid refrigerant: dry or direct expansion and flooded. In dry expansion operation the evaporator receives the fluid from thermostatic expansion as a mixture of liquid and a small proportion of vapor. The refrigerant leaves the evaporator totally evaporated and slightly superheated, about 2–4 °C above the evaporation temperature; this avoids any risk of liquid reaching the compressor. Liquid refrigerant expansion takes place in the thermostatic expansion valve, a fluid-metering device fitted with a thermometric sensor that is able to maintain the superheating of a moderate level of refrigerant vapor.

Flooded evaporators operate by a continuous pure liquid flow circulating through the evaporator, fed by gravity or by pump. The fluid leaves the evaporator as a mixture of liquid and vapor as only a part of the refrigerant volume delivered to it is evaporated. The mixture is directed to a liquid separator or surge drum from where the un-evaporated liquid is fed back to the evaporator inlet. The high-pressure refrigerant is released into the liquid separator at low pressure in such a way that the level is kept fairly constant. The expansion of the fluid can be controlled by a low-pressure float-valve, a hand-operated valve, a constant flow rate or thermostatically, and today by an electronic expansion valve allowing easy adjustment of the liquid level. [26]

5.2.2 Selection

Once the space and product loads have been established, we can select the evaporator type most suitable for our application. The total cooling capacity of the evaporator in the refrigerated railcar room has to be approximately 19 kW.

It is supposed earlier that the air is 3.3 K heated while flowing over the frozen meat. That means the air-on temperature amounts to about -18°C. The air-on temperature is the air temperature at the intake side of the coil block. The evaporating temperature is -25°C. Thus, the temperature difference between evaporation and entering air is about 7 K. For the relevant evaporator (R-134A DX system) the calculated maximum cooling room load is 19 kW. The internal width of the refrigerated railcar is 2.664 meters. That means the evaporator length has to be smaller than this 2.664 m. The selected model fabricated by "LU-VE Group" is**LS50H 5810 G 7**

Selected evaporator characteristics

Type: ***LS50H 5810 G 7***

Capacity: - R134a at evaporating temperature = -25°C and DT_1 = 7 K

- Frosted condition = 20.000 kW,
- Dry condition = 19.000 kW

Fin spacing: 7.5 mm

Coil surface: 238.8 m^2

Internal volume: 37.30 dm^3

Air flow rate: 13,400 m^3/ h

Fan motor consumption: 1.460 W

Dimensions:

- Length: 2,250 mm
- Width: 1,245 mm
- Height: 930 mm
- Weight: 327 kg

Defrost system: 1.570 kW electric defrost

5.3 Evaporator Defrosting

5.3.1 Introduction

Whenever the evaporator operates at temperatures below 0°C a layer of frost is deposited on the exchanger surface, increasing its thickness with time. Frost, a solid phase, is the result of moisture condensation and solidification which is deposited by the air circulating in the freezing space.

Frosting-up of the evaporator is inevitable despite all the normal precautions (adequate air circulation rate to minimize desiccation of produce; efficient thermal insulation and water vapor barrier; good quality gaskets on the door). Frost deposit on the evaporator first reduces the overall heat exchange coefficient because of the thermal resistance of ice and secondly hinders air circulation. Both cause deterioration in the performance of the evaporator and of the refrigerating installation. The decrease in the heat exchange makes the compressor work for longer periods, and also lowers the refrigerant boiling temperature. Both factors increase energy consumption. Further, the temperature of the frosted surfaces diminishes, increasing the relative humidity in the chamber.

Regular defrosting of the evaporator is necessary because equipment performance deteriorates with increasing thickness of ice, primarily affecting the free passage of air through the evaporator coils. First the method to supply heat to the evaporator to melt the ice has to be decided, and then the optimum defrosting frequency must be found.[26]

There are several ways to defrost an evaporator. Occasionally more than one can be used simultaneously.

- Electric defrost which is the most common method in use today. Equipment cost is about the same as with hot gas but installed cost can be lower. Operating cost is about

15% higher with electric defrost than with hot gas and a fair amount of heat and moisture is released in the room during defrost.

- Hot gases defrost is still the most efficient methods of defrosting regardless of storage temperature but, unfortunately, most contractors are reluctant to use it. Defrost is very quick with minimum room temperature rise. Hot gas defrost, however, requires care to ensure that the compressor is protected against liquid slugging.
- Water Defrost while not very common, it can be used on both medium and low temperature storages. Water must be at least 10°C and is sprayed on the coil at a rate of about 3g/m^2 foot of coil for five to 15 minutes, depending on severity of frosting. Water defrost is fast and efficient but some moisture is re-released into the room. These systems also require more maintenance than electric or hot gas systems.[16]

5.3.2 Defrost system selection

Besides electric defrost system of the selected evaporator, an independent hot gas defrosting is selected for the efficient defrosting system of evaporator coils. It is the fastest and most efficient method because an adequate supply of hot gas is available. Besides performing the defrost function, the hot refrigerant discharge gas internally clears the coil and drain pan tube assembly of accumulated compressor oil. This aids in returning the oil to the compressor. We choose the hot gas defrosting because it will give us the smallest internal heat load gain possible than other defrosting methods.

5.4 Air Cooled Condensing Unit

5.4.1 Introduction

An air-cooled condenser may be an integral part of the compressor unit (air-cooled condensing unit) or it may be remotely located (on the roof, for example). On most mobile refrigeration applications due to space limitation, compressors are most commonly used with integrated air-cooled condensers (see figure 5.2). In this design of refrigerated railcar an integrated one and factory assembled air-cooled condensing unit is considered. Condensing units are generally classified based on their approximate evaporating temperatures as high (-1°C to +10°C), medium (-23°C to -1°C) or low (-40°C to -23°C) temperature. [16]

Fig. 5.2 Air - cooled condensing unit *(Dean and Wood Product Catalogue Issue one)*

5.4.2 Air Cooled Condenser

Condensers are basically heat exchangers in which the refrigerant vapor is cooled and liquefied after compression. The evaporator heat load plus the heat of compression are released into the atmosphere via the condenser by means of a fluid normally air or by water: condensers can be cooled by water or air.

The air-cooled condensers operate under forced convection, using axial flow fans, which are noisy, to move air at high speeds over the finned tube. The usual air speed is between 2.5 and 5 m/s. However, as power consumption increases with the square of air velocity, 3 m/s is considered a reasonable speed.

Good air circulation is essential for satisfactory operation; therefore confined spaces must be avoided. For this reason the general trend is to mount condensers at a high level and in a cool location that is clean, dry and well ventilated, protected from the heat of the sun to prevent air recirculation and sited to take advantage of prevailing winds, particularly during periods of maximum load. They should be close to the evaporator, and whenever possible slightly above rather than below it.

Condensation pressure is the parameter to control for efficient and safe operation in the high pressure circuit. This pressure should not be excessively high, first for safety considerations and second because thermodynamically the refrigerating cycle operates less efficiently as the pressure rises, influencing energy consumption and raising operating costs.

It may also be necessary to regulate the condensation pressure to avoid it becoming too low, as this does not maintain a high enough pressure differential across the refrigerant expansion valve and the evaporator is fed with an insufficient refrigerant flow. To maintain a high condensation pressure, and corresponding high temperature, it is necessary to control the condenser capacity when the ambient temperature is low. This control is achieved either by reducing the flow of air through the condenser or by diminishing the effective heat exchange surface area or condensing area.

The condensing surface is modulated by retaining liquid refrigerant in the lower part of the condenser, using a valve known as a pressure regulator.

For air-cooled condensing units the air flow control system may cycle-off some of the fans, when there are more than one, or the air flow can be reduced with pressostatic shutters activated by the condenser pressure. When the pressure is low the shutters will close in relation to the decrease, reducing the flow of air circulating through the condenser. Thus it is possible to cut air circulation and so reduce the heat exchange and consequently raise the condensation pressure, but it is not recommended because of surface scaling, unless it is done for a long period in cold weather before condenser cleaning.

For vapor de-superheating and liquid sub-cooling (i.e., inlet and outlet regions of the coil) a cross-counter flow arrangement of refrigerant and air produces the best performance. In the condensing portion (i.e., at the center of the coil) the latent heat dissipates as a result of the refrigerant pressure drop. That is, the refrigerant enters the condenser as superheated vapor and cools by singe-phase convection to saturation temperature, after which condensation starts. If the coil's surface temperature is much lower than the saturation temperature, condensation occurs close to the gas entry area of the condenser coil. Eventually, in the last part of the condenser, the entire refrigerant vapor condenses to liquid. Because of the change of state, the velocity and volume of the refrigerant, as a gas, drops substantially as it becomes a liquid. In general, most designs provide an adequate heat transfer surface area and complementary pressure drop to cause the refrigerant to sub-cool an average of 1 to 3 K before it leaves the condenser coil. [26]

5.4.3 Compressor

In a refrigeration cycle, the compressor has two main functions within the refrigeration cycle. One function is to pump the refrigerant vapor from the evaporator so that the desired temperature and pressure can be maintained in the evaporator. The second function is to increase the pressure of the refrigerant vapor through the process of compression, and simultaneously increase the temperature of the refrigerant vapor. By this change in pressure the superheated refrigerant flows through the system.

Refrigerant compressors, which are known as the heart of the vapor-compression refrigeration systems, can be divided into two main categories which are displacement compressors and dynamic compressors. Note that both displacement and dynamic compressors can be hermetic, semi-hermetic, or open types.

Hermetic Compressors are preferable on reliability grounds to units primarily designed for the smaller range of temperatures required in air conditioning or cooling applications. In small equipment where cost is a major factor and on-site installation is preferably kept to a minimum, such as hermetically sealed motor/compressor combinations, there are no rotating seals separating motor and compressor, and the internal components are not accessible for maintenance, the casing being factory welded.

In larger sizes, refrigeration compressors are often semi-hermetic (see figure 5.3), that is, although motor and compressor are within one casing, this casing may be unbolted, and the refrigerant does not flow over the motor windings. Access for maintenance is straightforward, but the need for external motor cooling which aids efficiency in cooling applications is no advantage in refrigeration operations, and the cost is substantially higher than for hermetic units. As large motors are more efficient than small ones, overall efficiencies of up to 70% or more are theoretically possible, and in multi-cylinder compressors, capacity may be controlled by making one or more cylinders ineffective (e.g., by holding the inlet valve open). Cylinder unloading at start-up is also a convenient way of reducing starting torque. [14]

Fig. 5.3 Bitzer Octagon Semi – hermetic compressor *(Dean and Wood Product Catalogue Issue one)*

5.4.4 Condenser Selection

Knowing the room load, room temperature and desired suction temperature (room temperature - TD = suction temperature), an appropriate unit cooler made by "LU – VE" is selected as shown in the previous section. On the selection of the evaporator using LU-VE software it is found that required condenser capacity as 33.100 KW. So using the following data the required condenser can be selected form the same software.

- Required Capacity is 34.80 KW
- Ambient temperature is 40°C
- Sub cooling is 3 K
- Condensing Temperature is 48°C
- Refrigerant R134a

Selected evaporator characteristics

Type: **EAV9X 1111 V 1VENT (1X1) - SPECIAL EC FANS**

Capacity: - 35.71 kW (At condensing temperature = 47°C, Hot gas temperature= 82°Cand Sub-cooling = 3 K)

Power Consumption: - 1.450 kW

Fan diameter: - 1 * 910 mm

Fan speed: - 800 rpm

Fin spacing: - 2.1 mm

Coil surface: - 131.3 m^2

Coil operating maximal pressure: - 30 bar

Internal volume: - 23.00 dm^3

Air flow rate: - 19,500 m^3/h

Dimensions: -

- Length: - 2,623 mm
- Width: - 905 mm
- Height: - 1210 mm
- Weight: - 238 kg

5.4.5 Compressor Selection

Compressor selection of compressor totally depend on cooling load calculation of the system, Evaporating temperature and suction temperature because evaporating temperature of compressor affects the volumetric efficiency of compressor. Evaporating temperature decreases the volumetric efficiency of compressor decreases. It is also depend on type of thermodynamic cycle and refrigerant used in the system. In the selection of a proper refrigerant compressor, the following criteria have to be also considered:

- volumetric flow rate,
- compression ratio and,
- thermal and physical properties of the refrigerant.

As we remember the selected thermodynamic cycle is a single-stage cycle with suction gas heat exchanger. The evaporating saturated suction temperature of -25ºC, the suction gas temperature of 15ºCand the condensing saturated discharge temperature of 48ºC are taken as input data for selecting the semi-hermetic reciprocating compressor using BITZER Software. The result is shown in the tables below.

Type& Model: - Semi-hermetic compressor, **6FE-40Y**

Technical Data:

- Cooling capacity: - 19.48 kW
- Power input: - 11.76 kW
- Swept volume (Displacement)
  - 151,6 m^3/h at 1450 RPM 50Hz
  - 183,0 m^3/h at 1750 RPM 60Hz
- Dimensions: 795 x 452 x 445 mm

- Weight: 239 kg (including electric motor)
- Sound pressure level @ 1m (-10°C / 45°C) 73,0 dB(A) @ 50Hz
- No. of cylinder x bore x stroke 6 x 82 mm x 55 mm
- Horizontal receiver: - F302H (Volume 30.0 dm^3)

5.4.6 Oil Management

Compressor lubricants should be those recommended by the compressor manufacturer. They should also be compatible with the refrigerants and have a sufficiently low pour point to permit recovery from the low side of low pour point to permit recovery from the low side of the system at all expected temperature levels. HFC refrigerants will generally use a polyester synthetic lubricant. HFC refrigerants are chemically very stable and show very little tendency to degrade under conditions found in refrigeration and air-conditioning systems. HFC refrigerants are therefore not a factor in degradation of lubricants that might be used with them. Hygroscopic synthetic POE and PAG lubricants are less chemically stable with chlorinated refrigerants than mineral oil because of the interaction of moisture with the refrigerant at high temperatures.[2] Most oil recovery is accomplished directly by the refrigerant flow through the evaporator coil and suction line. The refrigerant tends to sweep the oil along. The heavier oil drains into the oil pots and may be recovered either manually or automatically.

5.5 Piping

5.5.1 Introduction

The design and operation of refrigerant piping systems should (1) ensure proper refrigerant feed to evaporators; (2) provide practical refrigerant line sizes without excessive pressure drop; (3) prevent excessive amounts of lubricating oil from being trapped in any part of the system; (4) protect the compressor at all times from loss of lubricating oil; (5) prevent liquid refrigerant or oil slugs from entering the compressor during operating and idle time; and (6) maintain a clean and dry system. [2] In this section we will choose the recommended material and size for all the pipes in our system. We follow the recommendations given by ASHRAE and change sizes properly when pressure drops are too significant. [2]

5.5.2 Pipe Material

For this design work a copper EN 12735-1 pipe material is selected for the HFC refrigerant flow.

5.5.3 Pipe Sizing

The pipe sizing for the single stage dry expansion R134a refrigeration system with suction gas heat exchanger is made by using SOLKANA REFRIGERANT SOFTWARE VERSION 8.0.0 and the result is given below.

Input: -

Refrigerating capacity = 19 kW

Evaporating temperature = -24.8 °C

Mean suction gas temperature = 14.75 °C

Condensing temperature = 47 °C

Liquid sub-cooling = 3 K

Table 5.1 Refrigerant pipe materials and size selection output

	Suction line	Discharge line	Liquid line	Suction riser	Hot gas riser
Inner diameter [mm]	46.76	20.68	13.04	64.90	42.32
Velocity [m/s]	16.21	9.41	0.87	8.42	2.25
Equivalent length [K/m]	0.04	0.04	0.02	-	-
Pressure drop [Pa/m]	194	1252	626	-	-
Total pressure drop [K]	0.4/@ L=10m	0.4/@ L=10m	0.2	-	-

CHAPTER SIX

REFRIGERATION SYSTEM CONFIGURATION

6.1 Thermodynamic Cycle

6.1.1 Introduction

Single-stage vapor-compression refrigerators are used by cold storage facilities with a range of +10 to −30 ◦C storage temperature. In this system, the evaporator installed within the refrigeration system and the ice-making unit, as the source of low temperature, absorbs heat. Heat is released by the condenser at the high-pressure side. (See fig. 6.2)

The refrigeration plant consists of the refrigerated railcar for preserving and transporting frozen meat with R134a as refrigerant. The air of the refrigerated railcar room is kept at -18°C during operation. This means that the refrigerant has to be cooled a few degree centigrade colder than -18°C. Let's say -25°C and hence the superheat is 7.0 K. The condensing temperature is 48.0 °C, the sub-cooling is 3.0 K, the refrigeration capacity is 19.00 kW, the mass flow rate of the compression is 0.1345 kg/sec, and the compressor suction temperature is 15.0 °C. The thermal efficiency of the suction gas heat exchanger is taken as 0.45. The resulting cycle analysis using CoolPack Software Program is shown in the fig below.

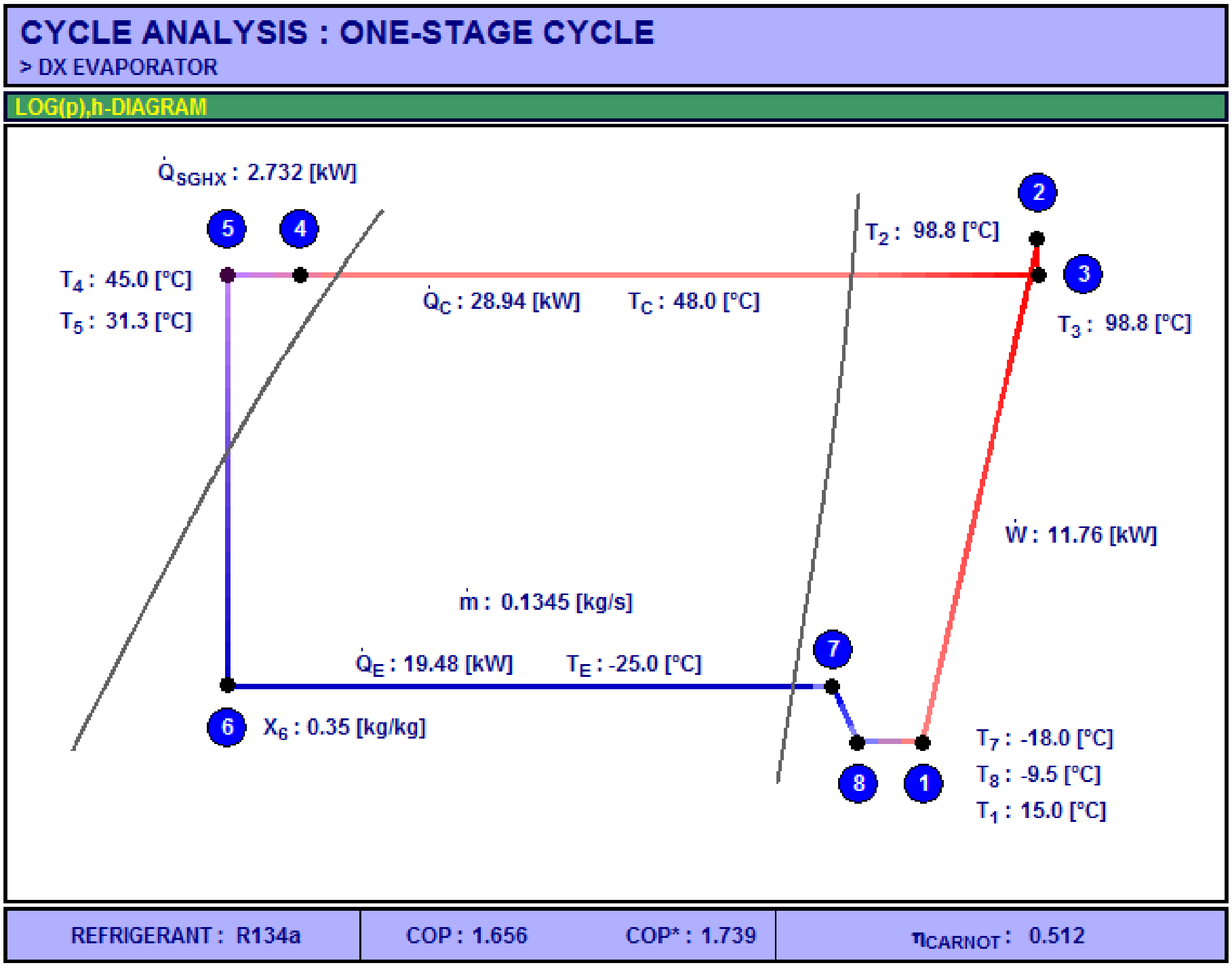


Fig. 6.1 Cycle analysis of single stage refrigeration system

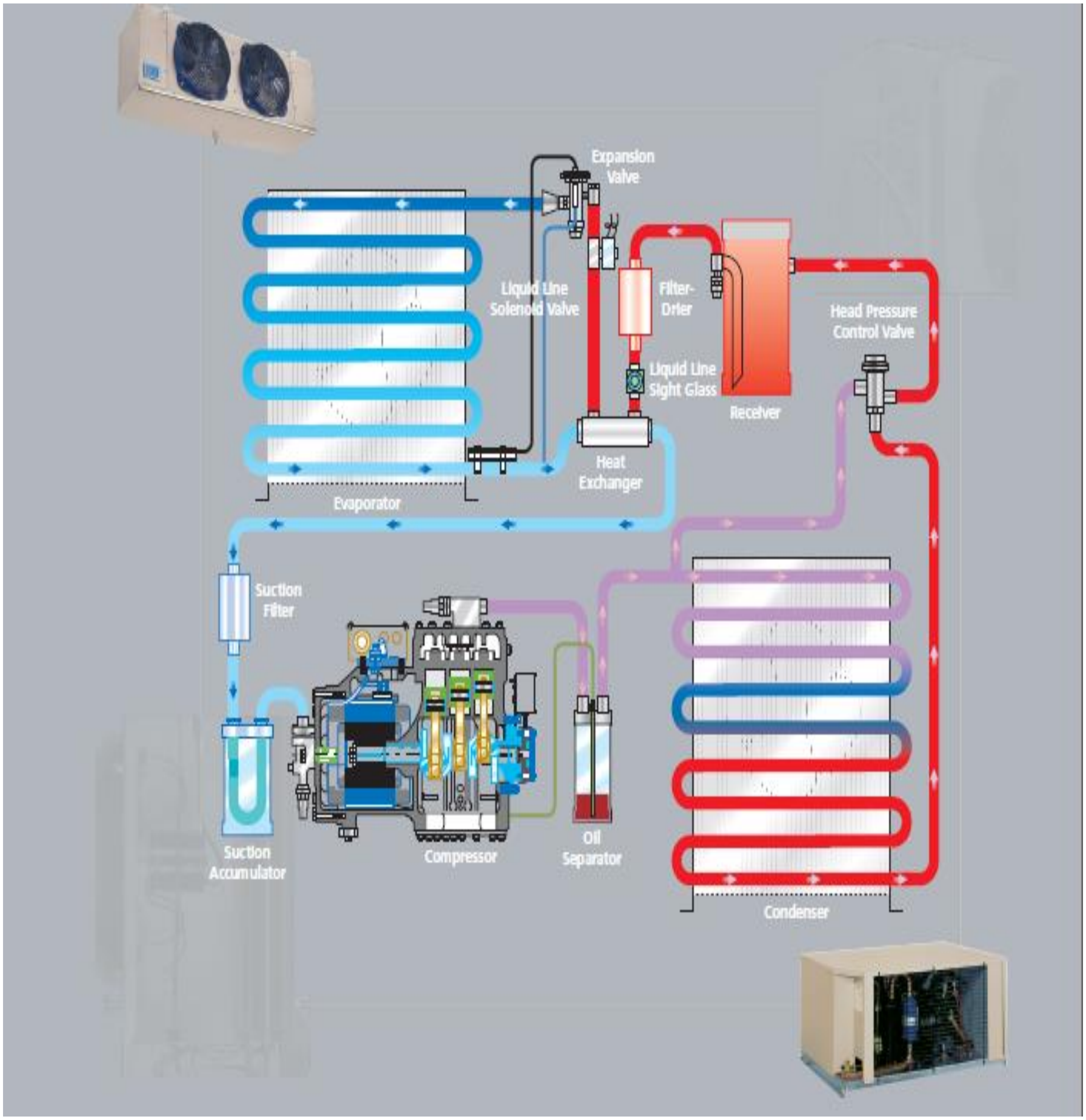


Fig. 6.2 Components of the refrigeration plant *(Heatcraft Refrigeration Engineering Manual)*

6.2 System Layout

The system layout or configuration of our refrigerated railcar design consists of a single temperature compartment configuration of a condensing unit incorporating a semi-hermitic compressor/motor assembly, and a receiver of refrigerant (R134a) and electrical operating controls which are all located at one end of the car (see figure 6.3& 6.5). The evaporator fan-coil package with selected defrosting system is located adjacent to the condensing unit but inside the insulated space (see figure 6.4& 6.5).

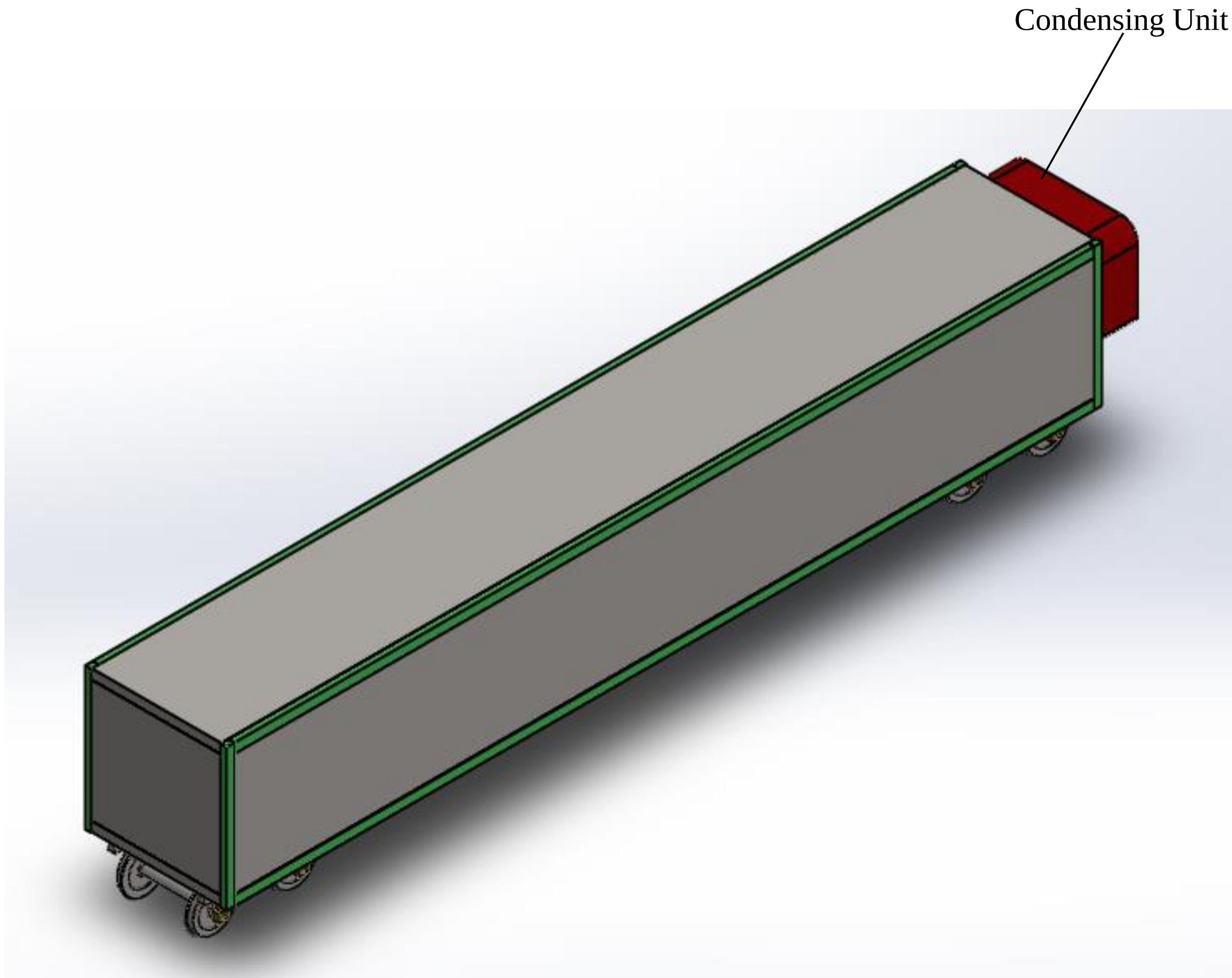


Fig. 6.3 System layout of the refrigeration equipment, nose mounted condensing unit

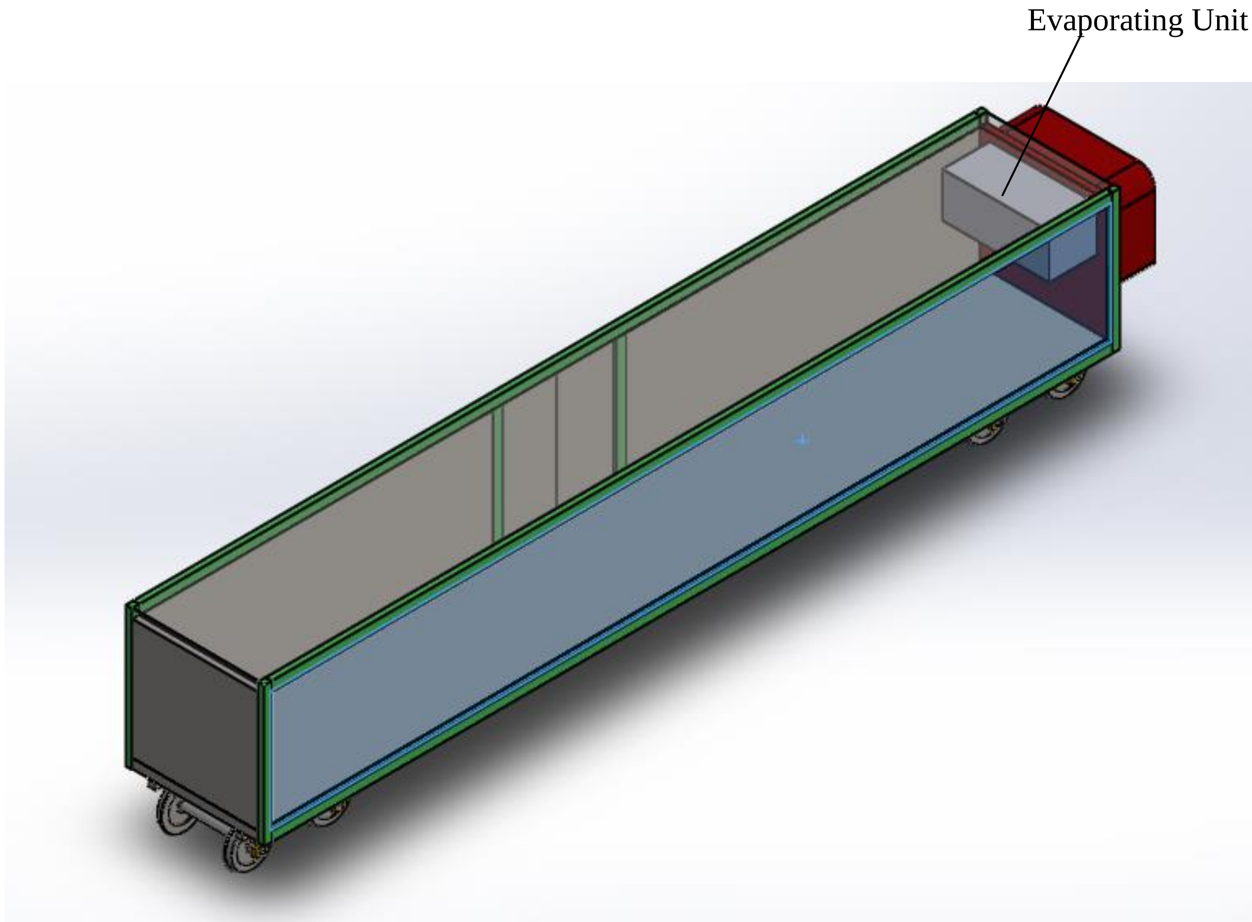


Fig. 6.4 System layout of the refrigeration equipment, evaporator inside the car

All components must be securely fastened to the vehicle to resist shock, vibration, and vandalism. Vehicle-to-equipment interfaces must have structure capable of secure support under all conditions of dynamic loading caused by vehicle travel and equipment operation (e.g., engine and compressor vibration). Suitable fastening provisions (mounting holes, studs, or captive nuts) are needed. Wall openings for equipment, which may be large (e.g., the entire car front wall of the machinery compartment), must have provisions to limit infiltration, using gaskets or other sealing methods (see figure 6.5).

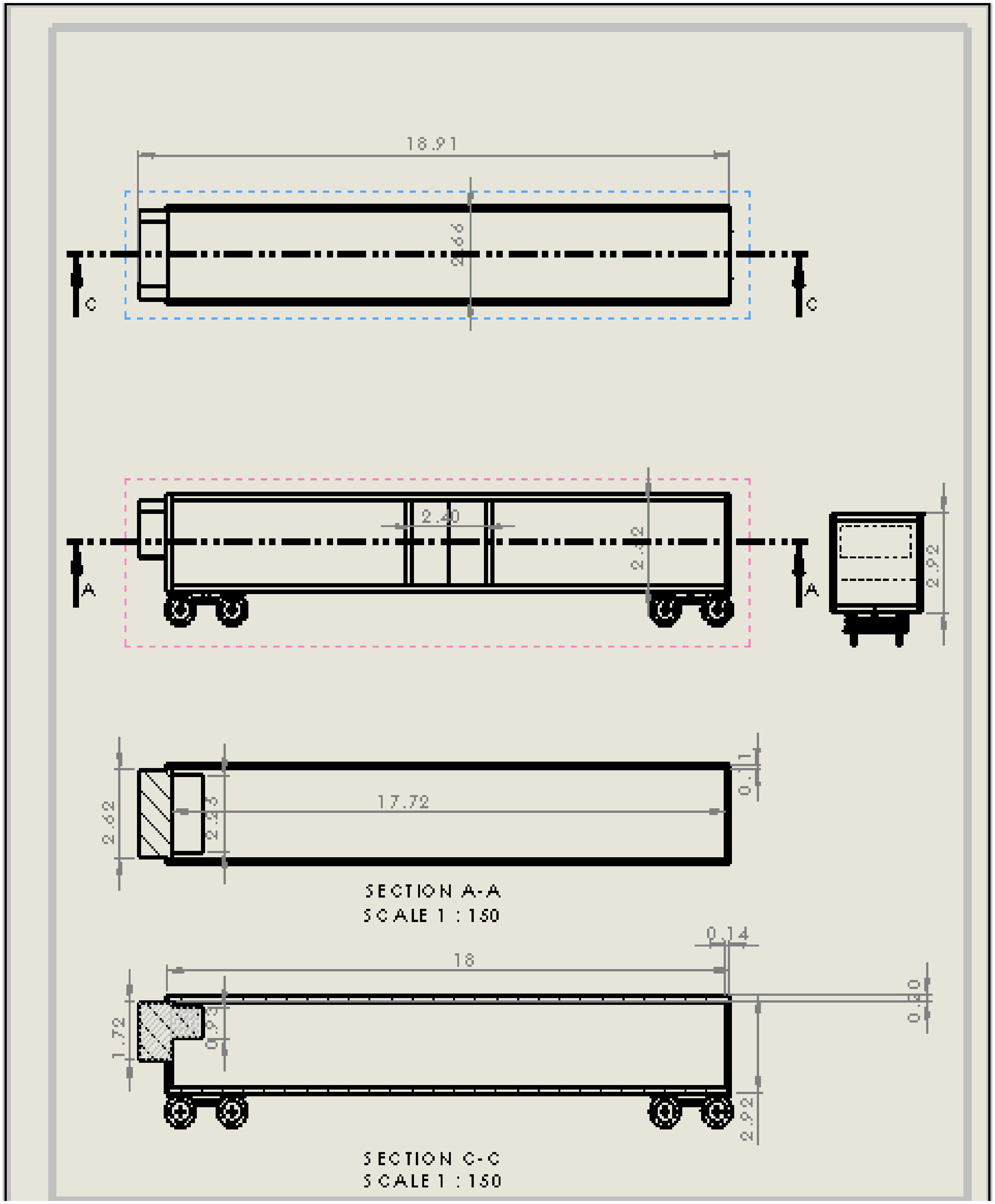


Fig. 6.5 2D drawing of car body of the refrigerated railcar

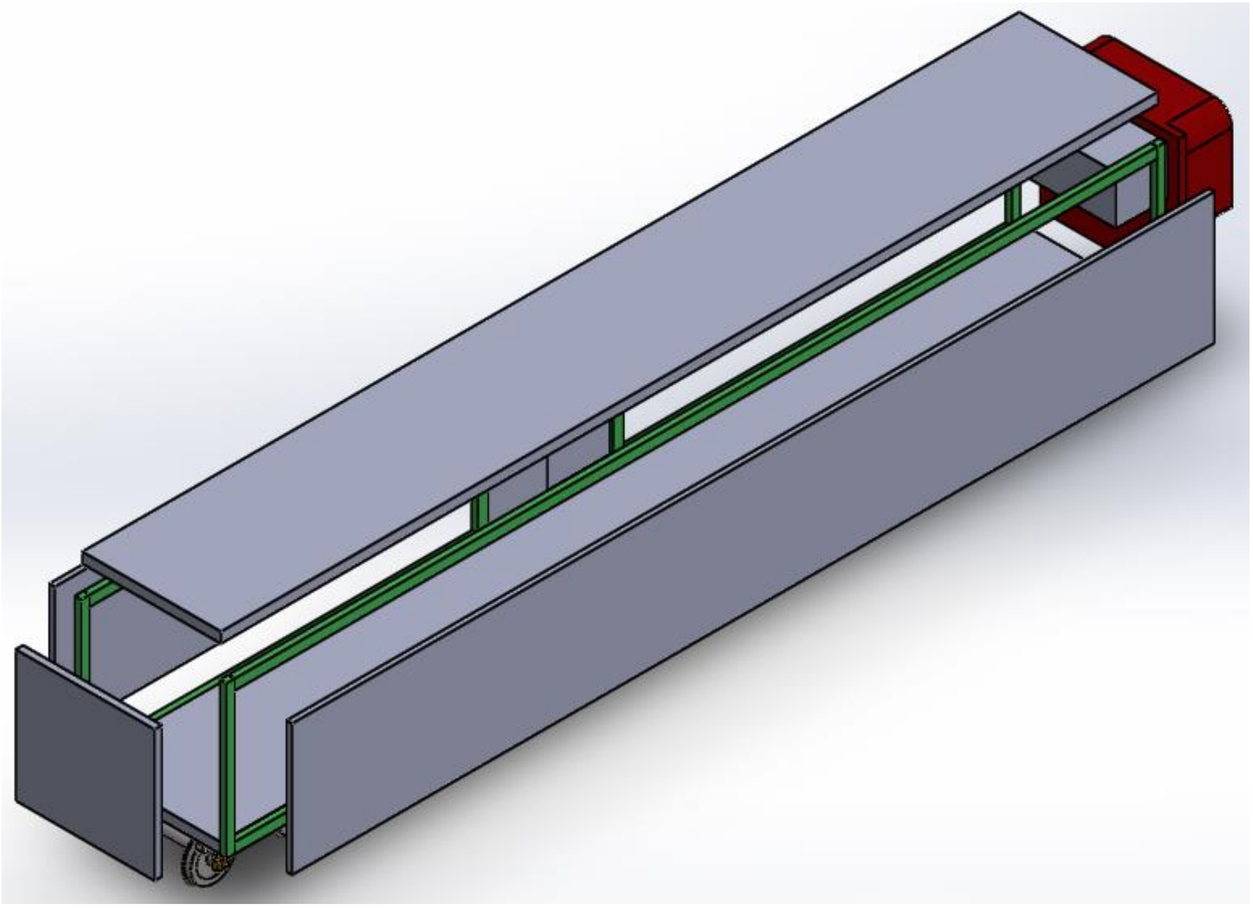


Fig. 6.6 Exploded view of the refrigerated railcar

Air circulation system configuration must be selected or designed to allow rapid heat removal and uniform air distribution. Rapid heat removal also helps to reduce temperature gradients across the evaporator and prevents frequent defrosting of the refrigeration unit. If air is unevenly distributed throughout the load, parts of the load may over-heat and other parts may be over-cooled [36], which accelerates produce spoilage.

There are two types of air circulation system configuration mostly used on transport refrigeration. The top-air delivery system (Fig. 6.7) issued for air circulation in refrigerated semi-trailers and railcars. The refrigeration unit blows cold, high velocity air along the ceiling above the load from the front to the rear of the trailer.

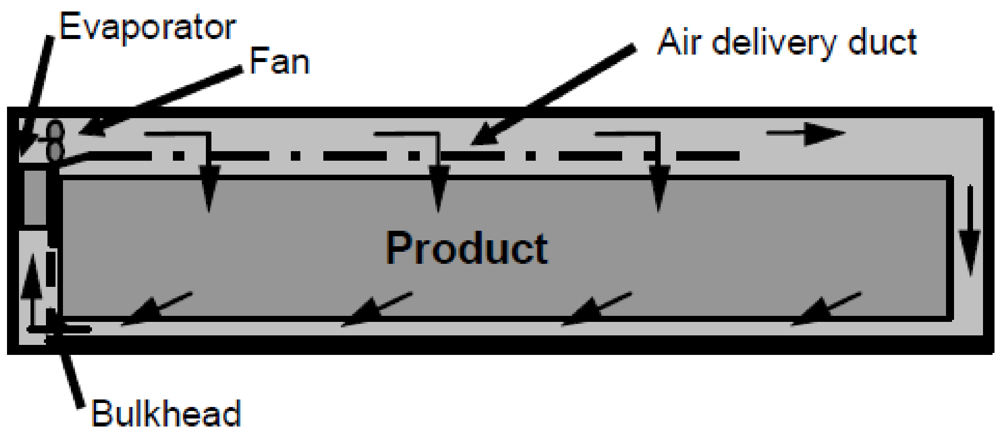


Fig 6.7A refrigerated container equipped with a top-air delivery system *(Postharvest Technologies for Horticultural Crops, 2009, Vol. 2: 13-14)*

The bottom-air delivery system (Fig. 6.8) is commonly used in intermodal and marine containers. In the bottom-air delivery system, air is blown to the bottom along T-beam floors and through pallets, rather than along the ceiling. As the air flows from the front to the rear, it is forced upwards through the cargo.

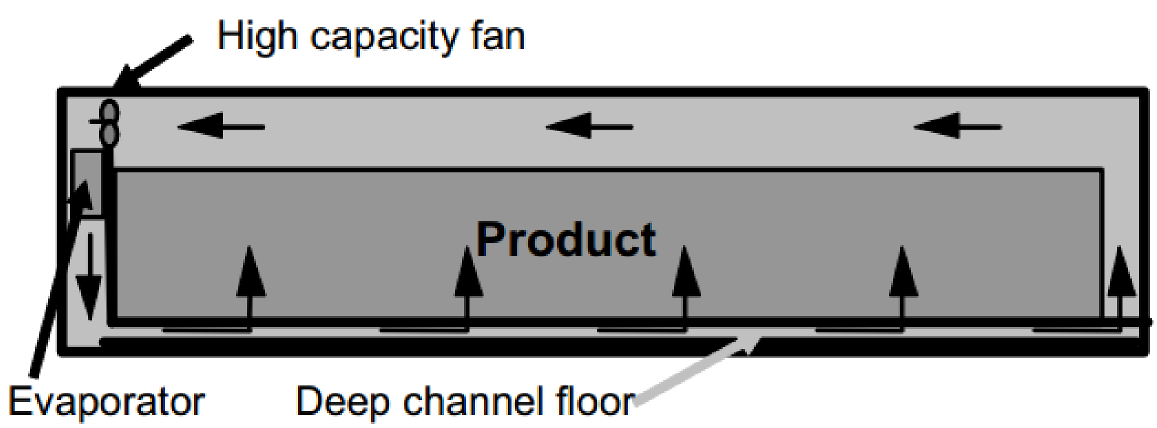


Fig 6.8A refrigerated container equipped with a bottom-air delivery system *(Postharvest Technologies for Horticultural Crops, 2009, Vol. 2: 13-14)*

Thus for this design work of refrigerated railcar a top-air delivery system configuration where the air from the system is delivered to the top of the container, with return air through the bottom of the evaporator coil section is used.

CHAPTER SEVEN

RESULTS AND DISCUSSION

For the designed refrigerated railcar envelop a standard wall thickness are selected for front & back walls = 140 mm, for two side walls = 105 mm and for ceiling and floor walls = 200 mm so that the volume of the car envelop is calculated to be 118.96 m3. The walls are constructed with closed-cell foam-in place polyurethane and outer facing a vapor seal galvanized steel and aluminum foils coated with plastic binder (see appendix I). The total heat resistance of the insulation panel to the heat transfer due to convection, conduction and solar radiation effect is calculated.

Taking the maximum mass of 60000 kg frozen meat to be transported and maximum volume of the car box 118.96 m3, the effective refrigerating space is calculated to be 59.6 % of the overall volume. Then the air circulation inside the refrigerated railcar is calculated to maintain an optimum air velocity of 1.240 m/sec which is acceptable for holding freezer (0.5 m/s and above).

While calculating the total cooling load, we took three different types of loads to be transported independently for traveling time of 24 hours (1 day).The following table show the maximum rate at which the product heat must be removed from the refrigerated railcar while transporting in order to maintain the required temperatures.

Table 7.1 Calculated Product Loads

Type of Load	Type of Commodities		
	Vegetable (mixed)	**Meat Chilled**	**Meat Frozen**
Pull down Load (KW)	5.495	7.333	9.280
Heat of Respiration (KW)	1.630	0	0
Total Product Load (KW)	7.125	7.333	9.280

Once the refrigeration load is calculated we made a selection of required fan-coil evaporator with gas defrosting system, air cooled condenser and a semi-hermetic reciprocating compressor from the manufacturer selection software catalogues considering the space limitation of system configuration (see appendix D – L). The single stage process, dx R134a system with suction gas heat exchanger is used for the thermodynamic cycle. Also the required refrigerant pipe material with specific equivalent dimensioning is made using the SOLKANE Refrigerant Software.

The four major elements of a refrigeration vapor compression cooling system together with the piping system can be configured or grouped in a variety of ways. As per our application, refrigerated railcar, and space optimization we select an appropriate system layout for our design. A nose mounted condensing unit inside a machinery compartment incorporating a semi-hermitic compressor/motor assembly and electrical operating controls is used. The evaporator fan-coil package is located inside the insulated car box such that the air from the system is delivered to the top of the container, with return air through the bottom of the evaporator coil section (i.e. top air delivery see).

CHAPTER EIGHT

CONCLUSION AND RECOMMENDATION

8.1 Conclusion

A refrigerated railcar of single temperature compartment for transporting 60000 kg mass of frozen/chilled meat products and/or 42,000 kg mass of vegetables from Addis Ababa to Djibouti has been designed. A standard prefabricated outer facing with an injected polyurethane insulation wall thickness is selected for car envelope design.

It is found that the effective refrigeration capacity of the designed refrigerated railcar for preserving and transporting a maximum mass of 60,000 kg frozen meat to Djibouti is 20.00 kW at condensing temperature 48 ^{o}C and evaporative temperature -25 ^{o}C. The maximum operating ambient temperature is taken as 40 ^{o} C. For this maximum ambient temperature the rated condenser capacity is found to be 35.71 kW. With the selected semi-hermetic reciprocating compressor working at full capacity the energy efficient ratio, EER/COP of the plant is found to be 1.656.

It is important to consider this design work of refrigerated railcar which is intended to realize the preliminary technology transfer for local manufacturing of different types of rail vehicles. By achieving this, the current inefficient road vehicle refrigerated transportation of exportable and perishable commodities like meat products, fruits and vegetables can be perfectly substituted. It is because of the refrigerated railcars can provide economical and environmental advantages over the refrigerated road vehicles.

8.2 Recommendation

During loading of the refrigerated railcar try to avoid blockage of air passages and hence maximize air circulation inside the compartment provided that maximize vehicle utilization - avoid partial loads. To minimize the unit refrigeration capacity the goods to be transported have to be fully pre-cooled to required set-point or below it.

For currently establishing national railway freight transportation system it is recommended that a central refrigerated road – rail distribution link is vital for sustainable cold transportation chain in the country. So it is important to implement a project to construct a well-organized refrigerated road – rail distribution link with refrigerated rail cross dock facilities on selected terminals.

Finally, I recommend that if the ERC uses such design work for local manufacturing of the refrigerated railcar through A.A. – Djibouti the potential export trade of meat, vegetables, and fruits hopefully will be successful.

8.3 Future Work

A multi-temperature and compartment refrigerated railcar for transporting perishables on Addis Ababa – Djibouti Railway line will be designed. This will make possible preserving and transporting two or more types of products at different storage temperature on a single car.

REFERENCE

[1] Huan, Z. “Energy Saving Opportunities In Food ‘Cold Chain’”, School of Mechanical Engineering, Vaal University of Technology, International Journal of Refrigeration Vanderbijl Park, 2008.

[2] ASHRAE Handbook, Refrigeration Systems and Applications, American Society of Heating, Refrigerating and Air conditioning Engineers, Inc., Atlanta, Georgia, 2010.

[3] Sustainable Refrigerated Road Transport, 21st Informatory Note on Refrigerating Technologies, Intergovernmental Organization for the Development of Refrigeration, December 2011. www.iifiir.org.

[4] Refrigerated transport: progress achieved and challenges to be met, 16 st Informatory Note on Refrigerating Technologies, Intergovernmental Organization for the Development of Refrigeration, August 2003. www.iifiir.org.

[5] TassouS. A.,De-Lille G., Ge Y.T., “Food transport refrigeration – Approaches to reduce energy consumption and environmental impacts of road transport”, Journal of Applied Thermal Engineering (2008), 2 July 2008.

[6] Tassou S.A., Lewis J.S., Ge Y.T., Hadawey A., Chaer I. “A Review of Emerging Technologies for Food Refrigeration Applications”, Journal of Applied Thermal Engineering (2009), 4 September 2009.

[7] Recommendations for Chilled Storage of Perishable Produce, IIR, 2000.

[8] RistoCiconkov,“Computer Program for Load Calculation of Cold rooms with Incorporated Databases and Recommendations”, Journal of Hygienic Engineering and Design.

[9] Refrigerated Transport in the UK, Sector review paper, ETSU for DETR, 2000.

[10] Stack A. and Finn D. P., “Modeling and Validation of a Multi-Evaporator Vapor Compression Cycle Subject to Non-Uniform Loading in Transport Refrigeration

Applications.” International Refrigeration and Air Conditioning Conference, Paper 617, 2002.

[11] Radhey Agarwal (India), Jason Anderson (USA), Donald Bivens (USA)…, PetterNekså (Norway), IPCC/TEAP Special Report: - “Safeguarding the Ozone Layer and the Global Climate System”. Page 225-268. (2010).

[12] Welty J.R., Wicks C.E., Wilson R.E., Rorrer G., “Fundamentals of Momentum, Heat, and Mass Transfer”, 4th edition, United Sates of America, (2001).

[13] Stegmann R., “Practical guide to low temperature refrigeration”, ASHRAE Journal 42 (1) Page 42–50, (2000).

[14] Ibrahim Dincer, Mehmet Kanoglu, “Refrigeration Systems and Applications”, Second Edition, A. John Wiley and Sons, Ltd., Publication, 2010.

[15] British Refrigeration Association, Institute of Refrigeration, System Design and Component Selection, Issue 1 September 2009.

[16] Carrier Corporation 10/19. Rev A 06/03, Refrigeration Selection Guide, For Condensing Units And Unit Coolers, 1991.

[17] WulfinghoffD. R., Reference Note 32 Compression Cooling, 1999

[18] Department of Defense, USA. Design: Refrigeration Systems for Cold Storage, Unified Facilities Criteria (UFC), UFC 4-826-10, 10 July 2002

[19] United State Department of Agriculture, Transport & Marketing Division, Hand Book Number 669, “Protecting Perishable Foods during Transport by Truck”, July 2008.

[20] Hadawey A. and TassouS. A., Reducing the Environmental Impact of Refrigerated Transport Vehicles Using Vacuum Insulation Panels (VIPS), School of Engineering and Design Brunel University May 2009.

[21] HuiCKP, Raghavan GSV, Vigneault C., Cooling requirements for refrigerated semi-trailers transporting fresh fruits and vegetables.

[22] Wiersinga R.C. (LEI), Snels J. (AFSG), Admiraal L. (LEI), Prospects and Challenges for Refrigerated Container Transport of Fruits and Vegetables from Ethiopia to the Middle East, January 2009.

[23] Ministry of Agriculture and Rural Development, Animal and Plant Health Regulatory Department, “Meat Cold Chain Guideline”, August 2009.

[24] Report on Refrigeration Sector Achievements and Challenges to serve as input to the World Summit on Sustainable Development, IIR, 2002.

[25] Naval Facilities Engineering Command, Refrigeration Systems for Cold STORAGE, Approved for Public Release, 200 Stovall Street, Alexandria, Virginia 22332-2300, 1986. Downloadable from http://www.everyspec.com.

[26] Wouter De Muynck& Davy Poelman, “Refrigerating Plant Design”, Thesis presented to obtain the Diploma of the Second Cycle of Electromechanical Engineering, 2006.

[27] Thomas Makumbi, “Investigating the Application of Environmentally Friendly Solutions in Refrigeration Applications of Uganda”, October 2013.

[28] Totem Ocean Trailer Express, Inc., Refrigeration Unit Manual, 2002.

[29] ASHRAE, Handbook - Fundamentals, American Society of Heating, Refrigerating and Air Conditioning Engineers, Atlanta, USA, 2005

[30] ASHRAE, Handbook - Refrigeration, American Society of Heating, Refrigerating and Air Conditioning Engineers, Atlanta, USA, 2006

[31] Heat Craft Engineering Manual, Commercial Refrigeration Cooling and Freezing Load Calculation and Reference Guideline, H-ENGM0408, 2008.

[32] Swanson G.M., Ott M.V., The story o f Frederick McKinley Jones; Runestone press, 1994.

[33] Lugien A. Jansen, “Evaluation and Optimization of Airflow Distribution within A Refrigerated Trailer”, A thesis submitted for the degree of Master of Engineering, 2006.

[34] Xie R., Alan F., Liu G., Zou Y., Huagn X. and Cheng-Min F., "Research on the Control Mechanism and Simulation of Safety Conditions of Perishable Foods Transportation and Logistics" ICRA Final Report (2007-2009).

[35] BING, Thermal insulation materials made of rigid polyurethane foam (PUR/PIR), Federation of European Rigid Polyurethane Foam Associations, volume number 1, 2006.

[36] Hui K.P.C., Vigneault, C., de Castro, L.R., and Raghavan, G.S.V., Effect of different accessories on airflow pattern inside refrigerated semi-trailers transporting fresh produce. Appl. Eng. Agric., (in press). 2006.

[37] Bureau of Energy Efficiency, "HVAC and Refrigeration System." In: Energy Efficiency in Electrical Utilities, Chapter 4: Ministry of Power. New Delhi, India, 2004.

[38] Fang S., Zhu B., Yoshio O., Toyoki A., Hirohisa K., and Katsuhiko F., Electrical Components and Air-conditioning Units for Low environmental- impact Trains for Subway Systems in China. 2008.

[39] WEB_1, http://www.et.dtu.dk/coolpack.

[40] WEB_2, http://www.bitzer.de-bitzer@bitzer

[41] WEB_3, http://www.luvegroup.com

[42] WEB_4, http://www.fchart.com

APPENDIX

APPENDIX A

Thermal Conductivity and Thermal Resistance of Insulation Materials

Thermal conductivity (k) is a specific material property. It represents the heat flow in watts (W) through a 1 m² surface and 1 m thick flat layer of a material when the temperature difference between the two surfaces in the direction of heat flow amounts to 1 Kelvin (K). The unit of measurement for thermal conductivity (k) is W/(m·K).

The thermal resistance (R) describes the thermal insulation effect of a constructional layer. It is obtained by dividing the thickness (t) by the design thermal conductivity value of a building component: $R = d/\lambda$. The unit of thermal resistance (R) is (m²·K)/W. In building components comprising several layers, the thermal resistances of the individual layers are added together.

The thermal transmittance (U) is the heat flow in watts (W) through 1 m² of a building component when the temperature difference between the surfaces in the direction of heat flow is 1K. U-value can be calculated from $U = 1/R$ for a given construction, and is generally represented in W/(m^2.K).

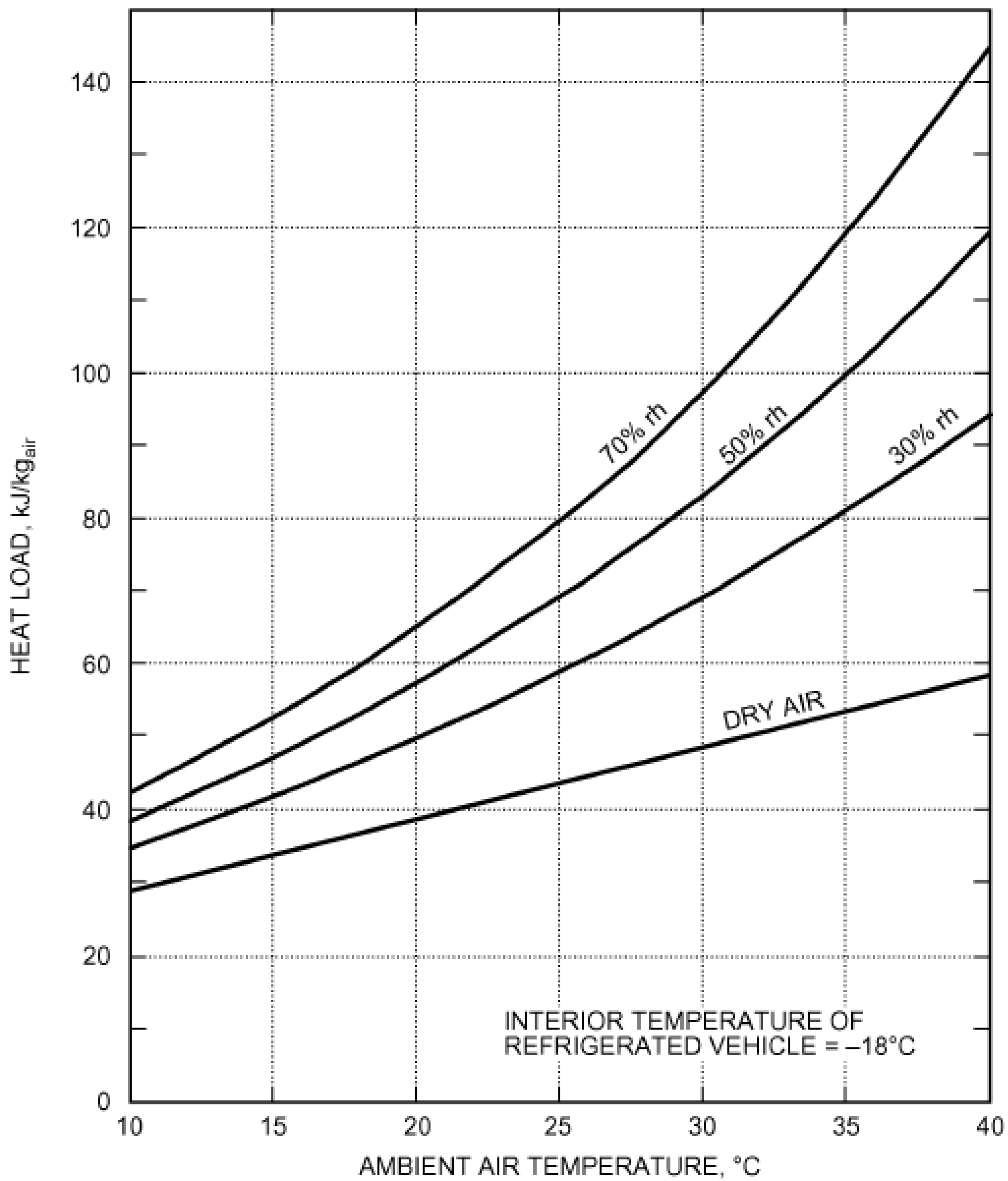
The specific heat capacity C_p states how much heat energy is required to increase the temperature of 1 kg mass of a material by 1 K. Specific heat capacity C_p is measured in J/(kg K). More heat energy is required to raise the temperature by 1 K of a material with a greater heat capacity. And inversely, less energy is required to produce a 1 K increase in temperature in materials with lower heat capacities.

Specific heat capacity C_p of various materials

Material	Specific heat capacity c_p = J/(kg·K)
Rigid polyurethane foam (PUR/PIR)	1400 – 1500
Wood-fibre insulation boards	1400
Mineral wool	1030
Wood and wood-based materials	1600
Plasterboard	1000
Aluminium	880
Other metals	380 – 460
Air (ρ=1.25 kg/m³)	1000
Water	4190

APPENDIX B

Heat Load from Air Leakage through a –18°C vehicle resulting from infiltration of ambient air at various conditions.



APPENDIX C

All electrical energy dissipated in the refrigerated space due to electric motors and driven equipments inside the cold room must be included in the internal heat load. ASHRAE provides the following data for heat equivalent of electric motor.

				Location of Motor and Driven Equipment with Respect to Conditioned Space or Airstream		
				A	B	C
Motor Rated, kW	Motor Type	Nominal rpm	Full Load Motor Effi-ciency, %	Motor in, Driven Equip-ment in, W	Motor out, Driven Equip-ment in, W	Motor in, Driven Equip-ment out, W
0.04	Shaded pole	1500	35	105	35	70
0.06	Shaded pole	1500	35	170	59	110
0.09	Shaded pole	1500	35	264	94	173
0.12	Shaded pole	1500	35	340	117	223
0.19	Split phase	1750	54	346	188	158
0.25	Split phase	1750	56	439	246	194
0.37	Split phase	1750	60	621	372	249
0.56	3-Phase	1750	72	776	557	217
0.75	3-Phase	1750	75	993	747	249
1.1	3-Phase	1750	77	1453	1119	334
1.5	3-Phase	1750	79	1887	1491	396
2.2	3-Phase	1750	81	2763	2238	525
3.7	3-Phase	1750	82	4541	3721	817
5.6	3-Phase	1750	84	6651	5596	1066
7.5	3-Phase	1750	85	8760	7178	1315
11.2	3-Phase	1750	86	13 009	11 192	1820
14.9	3-Phase	1750	87	17 140	14 913	2230
18.6	3-Phase	1750	88	21 184	18 635	2545
22.4	3-Phase	1750	89	25 110	22 370	2765
30	3-Phase	1750	89	33 401	29 885	3690
37	3-Phase	1750	89	41 900	37 210	4600
45	3-Phase	1750	89	50 395	44 829	5538
56	3-Phase	1750	90	62 115	55 962	6210
75	3-Phase	1750	90	82 918	74 719	8290
93	3-Phase	1750	90	103 430	93 172	10 342
110	3-Phase	1750	91	123 060	111 925	11 075
150	3-Phase	1750	91	163 785	149 135	14 738
190	3-Phase	1750	91	204 805	186 346	18 430

APPENDIX D

Selected evaporator characteristics (*Type: LS50H 5810 G 7)*

Inlet air temp. (room)	[ºC]	-18,0	
Refrigerant		R134a	
Altitude	[m]	721	
Residual static pressure	[Pa]	0	
DT1 needed for unit cooler	[K]	7,0	
Frost thickness on each fin side	[mm]	0,3	
Connection **400V-3PH-50Hz**		△	
Actual capacity	**[W]**	**19.000**	
Capacity without frost	[W]	20.000	
Air flow	[m3/h]	13.400,0	
Air throw	[m]	36	
Outlet air temperature	[ºC]	-20,9	
Evaporating temp	[ºC]	-24,9	
DT superheating	[K]	5,0	
Temp. before expansion valve	[ºC]	14,1	
DT1	[K]	6,89	
DTmlg	[K]	5,30	
Fluid pressure drop	[K]	1,2	
RC factor (Sensible capacity/Total capacity)	[%]	90,0	
Energy efficiency class		C	**(2014 thresholds)**
Fan motor consumpt.	[W]	1.460	
Power draw	[A]	2,8	
Max fan absorbed current	[A]	2,9	
Fan speed	[1/min]	1330	
Sound pressure level (5m)	[dB(A)]	58	
Sound Power Level	[dB(A)]	83	
Electric defrost (230 V)	[W]	1.570	
COMPRESSOR CAPACITY	**[W]**	**19.000**	(-24,9 / 48,0 ºC)
Compressor working time	[hh:mm]	20:00	
Required condenser capacity	**[W]**	**33.100**	(Type Semi - hermetic)

Nº Fans	[mm]	2 x 500	Weight	[kg]	327
Poles	[n]	4	Connections in	[n] x [mm]	1 x 28
Fin spacing	[mm]	7,5	Connections out	[n] x [mm]	1 x 64
Internal volume	[dm3]	37,30	Drain tray connection	["]	2"
Surface	[m2]	238,8	Overall dimensions	[mm]	2.250 x 1.245 x 930
Max Working pressure	**[bar]**	**24,0**			

Casing material	Powder coated galvanized steel RAL 9003	Fin material	Alupaint
Header material	Cu	Tube material	Cu

Accessories:

Q.ty	Code	Type:	Description
1	ALUP	ALUPAINT	ALUPAINT

APPENDIX E

Selected condenser Characteristics *(EAV9X 1111 V 1VENT (1X1) - SPECIAL EC FANS)*

Air inlet temperature	[°C]	40,0		
Condensing temperature	[°C]	47,0		
Hot gas temperature	[°C]	82,0		
Subcooling	**[K]**	**3**		
Refrigerant		R134a		
Altitude	[m]	721		
Version		Vertical		
Motor connection	**400V-3PH-50Hz**	**SPECIAL EC FANS**		
Capacity	**[kW]**	**35,71**		
Air flow	[m3/h]	19.500,0		
Energy efficiency class		D	**(2014 thresholds)**	
Power consumption	[W]	1.450		
Motor consumption	[A]	2,5		
Max absorbed current	[A]	2,4		
Fan speed	[1/min]	800		
Sound level (at distance) 10 [m])	[dB(A)]	49		
Sound Power Level	[dB(A)]	80		

Fans :	[mm]	1 x 910	Volume	[dm3]	23,00
Poles	[n]	**EC FANS**	Surface	[m2]	131,3
Fin spacing:	[mm]	2,1	Weight	[kg]	238
Coil operating maximal pressure	**[bar]**	**30,0**	Overall dimensions	[mm]	2.623 x 905 x 1.210

Casing material	Powder coated galvanized steel RAL 9003	Fin material	Alupaint
Headers material	Cu	Tube material	Cu

Accessories:

Q.ty	Code	Type:	Description
1	30087024	SPR 15	PRESSURE SENSOR
1	30108567C	A3G910-AS22--400V-800rpm	EC FANS
1	30189872	ESMC 1/32A	ELECTRICAL PANEL WITH WMC2 MODBUS BOARD
1	ALUP	ALUPAINT	ALUPAINT
1	CABLEC	CABLAGGIO REG. EC FANS	WIRING FOR EC FANS
1	SCOOL0001	XXXX	SUBCOOLING

APPENDIX F

Selected compressor performance data with arbitrary operating points

Compressor model	**6FE-40Y**	Operating mode	Auto
Mode	Refrigeration and Air condi	Power supply	400V-3-50Hz
Refrigerant	R134a	Capacity Control	100%
Reference temperature	Dew point temp.	Useful superheat	100%
Liquid subcooling(in Cond.)	3K		
Suction gas temperature	15°C		

Cooling capacity [W]

tc \ to	20.0	15.0	12.5	10.0	5.0	0.0	-5.0	-10.0	-15.0	-20.0	-25.0	-30.0
20	--	--	--	--	--	102188	82442	65718	51631	39851	30085	--
30	--	--	--	--	--	91806	73860	58634	45784	35016	26070	--
40	--	--	--	--	--	80853	64858	51252	39740	30068	22010	--
50	--	--	--	--	--	69589	55654	43760	33663	25149	18028	--
55	--	--	--	--	--	63885	51011	39999	30631	22715	16077	--
60	--	--	--	--	--	--	46331	36213	27585	20276	14128	--
65	--	--	--	--	--	--	--	--	--	--	--	--
70	--	--	--	--	--	--	--	--	--	--	--	--
80	--	--	--	--	--	--	--	--	--	--	--	--

Evaporator capacity [W]

tc \ to	20.0	15.0	12.5	10.0	5.0	0.0	-5.0	-10.0	-15.0	-20.0	-25.0	-30.0
20	--	--	--	--	--	102188	82442	65718	51631	39851	30085	--
30	--	--	--	--	--	91806	73860	58634	45784	35016	26070	--
40	--	--	--	--	--	80853	64858	51252	39740	30068	22010	--
50	--	--	--	--	--	69589	55654	43760	33663	25149	18028	--
55	--	--	--	--	--	63885	51011	39999	30631	22715	16077	--
60	--	--	--	--	--	--	46331	36213	27585	20276	14128	--
65	--	--	--	--	--	--	--	--	--	--	--	--
70	--	--	--	--	--	--	--	--	--	--	--	--
80	--	--	--	--	--	--	--	--	--	--	--	--

Power input [kW]

tc \ to	20.0	15.0	12.5	10.0	5.0	0.0	-5.0	-10.0	-15.0	-20.0	-25.0	-30.0
20	--	--	--	--	--	15.26	14.67	13.83	12.79	11.59	10.27	--
30	--	--	--	--	--	18.37	17.26	15.90	14.36	12.70	10.97	--
40	--	--	--	--	--	21.2	19.55	17.67	15.65	13.56	11.46	--
50	--	--	--	--	--	23.7	21.4	19.08	16.62	14.15	11.72	--
55	--	--	--	--	--	24.7	22.2	19.64	16.98	14.33	11.76	--
60	--	--	--	--	--	--	22.9	20.1	17.24	14.43	11.73	--
65	--	--	--	--	--	--	--	--	--	--	--	--
70	--	--	--	--	--	--	--	--	--	--	--	--
80	--	--	--	--	--	--	--	--	--	--	--	--

Selected compressor performance data with arbitrary operating points (Cont...)

Compressor model	6FE-40Y	Operating mode	Auto
Mode	Refrigeration and Air condi	Power supply	400V-3-50Hz
Refrigerant	R134a	Capacity Control	100%
Reference temperature	Dew point temp.	Useful superheat	100%
Liquid subcooling(in Cond.)	3K		
Suction gas temperature	15°C		

Current (400V) [A]

tc \ to	20.0	15.0	12.5	10.0	5.0	0.0	-5.0	-10.0	-15.0	-20.0	-25.0	-30.0
20	--	--	--	--	--	35.5	34.9	34.1	33.3	32.3	31.4	--
30	--	--	--	--	--	38.6	37.4	36.1	34.6	33.2	31.8	--
40	--	--	--	--	--	41.8	39.9	37.8	35.8	33.9	32.2	--
50	--	--	--	--	--	44.7	42.0	39.4	36.8	34.4	32.4	--
55	--	--	--	--	--	46.0	43.0	40.0	37.1	34.6	32.4	--
60	--	--	--	--	--	--	43.8	40.5	37.4	34.7	32.4	--
65	--	--	--	--	--	--	--	--	--	--	--	--
70	--	--	--	--	--	--	--	--	--	--	--	--
80	--	--	--	--	--	--	--	--	--	--	--	--

Condenser capacity (with HR) [W]

tc \ to	20.0	15.0	12.5	10.0	5.0	0.0	-5.0	-10.0	-15.0	-20.0	-25.0	-30.0
20	--	--	--	--	--	116688	96379	78857	63781	50860	39846	--
30	--	--	--	--	--	109260	90255	73741	59429	47080	36491	--
40	--	--	--	--	--	101012	83426	68036	54609	42951	32898	--
50	--	--	--	--	--	92071	76031	61886	49455	38590	29165	--
55	--	--	--	--	--	87345	72131	58654	46762	36328	27248	--
60	--	--	--	--	--	--	68074	55296	43967	33987	25269	--
65	--	--	--	--	--	--	--	--	--	--	--	--
70	--	--	--	--	--	--	--	--	--	--	--	--
80	--	--	--	--	--	--	--	--	--	--	--	--

COP/EER

tc \ to	20.0	15.0	12.5	10.0	5.0	0.0	-5.0	-10.0	-15.0	-20.0	-25.0	-30.0
20	--	--	--	--	--	6.70	5.62	4.75	4.04	3.44	2.93	--
30	--	--	--	--	--	5.00	4.28	3.69	3.19	2.76	2.38	--
40	--	--	--	--	--	3.81	3.32	2.90	2.54	2.22	1.92	--
50	--	--	--	--	--	2.94	2.59	2.29	2.03	1.78	1.54	--
55	--	--	--	--	--	2.59	2.29	2.04	1.80	1.59	1.37	--
60	--	--	--	--	--	--	2.02	1.80	1.60	1.40	1.20	--
65	--	--	--	--	--	--	--	--	--	--	--	--
70	--	--	--	--	--	--	--	--	--	--	--	--
80	--	--	--	--	--	--	--	--	--	--	--	--

Mass flow [kg/h]

tc \ to	20.0	15.0	12.5	10.0	5.0	0.0	-5.0	-10.0	-15.0	-20.0	-25.0	-30.0
20	--	--	--	--	--	1952	1565	1242	971	747	562	--
30	--	--	--	--	--	1898	1517	1198	931	709	527	--
40	--	--	--	--	--	1826	1454	1143	882	664	485	--
50	--	--	--	--	--	1737	1378	1077	824	613	438	--
55	--	--	--	--	--	1685	1335	1039	791	584	412	--
60	--	--	--	--	--	--	1287	998	756	553	384	--
65	--	--	--	--	--	--	--	--	--	--	--	--
70	--	--	--	--	--	--	--	--	--	--	--	--
80	--	--	--	--	--	--	--	--	--	--	--	--

APPENDIX G

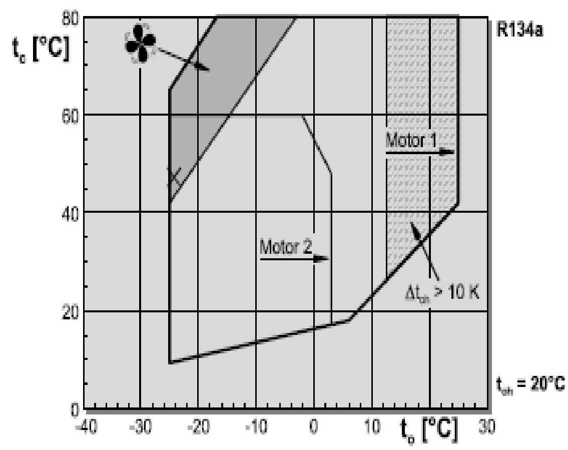
Selected compressor output characterstics

Compressor Selection: Semi-hermetic Reciprocating Compressors

Input Values

Compressor model	**6FE-40Y**
Refrigerant	R134a
Reference temperature	Dew point temp.
Evaporating SST	-24.8°C
Condensing SDT	47°C
Liquid subcooling(in Cond.)	3K
Suction gas temperature	15°C
Operating mode	Auto
Power supply	400V-3-50Hz
Useful superheat	100%
Capacity Control	100%

Application Limits (R134a)



Output

Compressor model	**6FE-40Y-40P**
Cooling capacity	19.48 kW
Cooling capacity *	19.16 kW
Evaporator capacity	19.48 kW
Power input	11.76 kW
Current (400V)	32.4 A
Voltage range	380-420V
Condenser capacity (with HR)	30.7 kW
COP/EER	1.66
COP/EER *	1.63
Mass flow	459 kg/h
Operating mode	Standard
Discharge gas temp. w/o cooling	123.5 °C

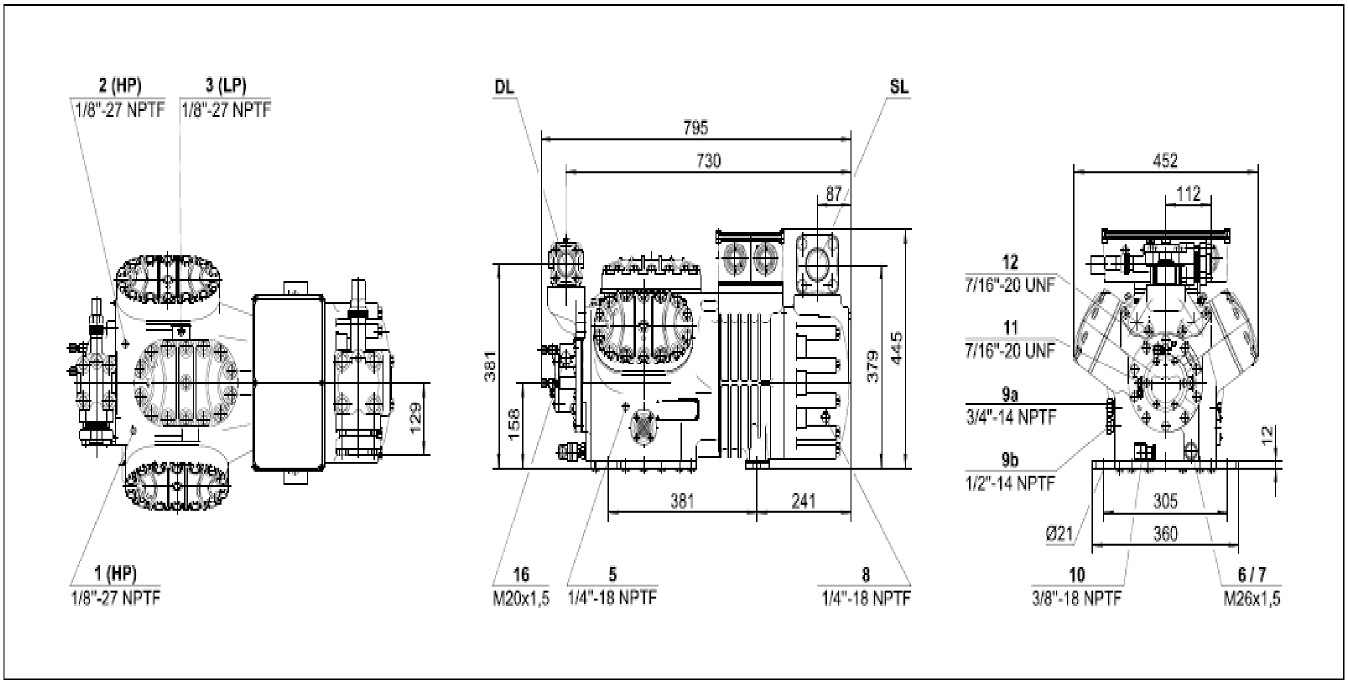
APPENDIX H

Selected compressor technical data

Displacement (1450 RPM 50Hz)	151,6 m³/h
Displacement (1750 RPM 60Hz)	183,0 m³/h
Frequency range	30..70 Hz
No. of cylinder x bore x stroke	6 x 82 mm x 55 mm
Motor voltage (more on request)	380-420V PW-3-50 Hz
Max operating current	48.7 A
Winding ratio	50/50
Starting current (Rotor locked)	180.0 A Y / 323.0 A YY
Weight	239 kg
Max. pressure (LP/HP)	19 / 28 bar
Connection suction line	54 mm - 2 1/8"
Connection discharge line	42 mm - 1 5/8"
Oil type R134a/R407C/R404A/R507A/R407A	tc<60°C: BSE32 / tc>60°C: BSE55 (Option)
Oil charge	4,75 dm³
Crankcase heater	140 W (Option)
Oil pressure monitoring	MP54 (Option), Delta P2 (Option, not for R290/R1270)
Oil service valve	Option
Discharge gas temperature sensor	Option
Motor protection	SE-B2
Enclosure class	IP54 (Standard), IP66 (Option)
Start unloading	Option
Capacity control	100-66-33% (Option)
Additional fan	Option
Vibration dampers	Standard
Sound power level (-10°C / 45°C)	81,0 dB(A) @ 50Hz
Sound pressure level @ 1m (-10°C / 45°C)	73,0 dB(A) @ 50Hz

APPENDIX I

Selected compressor dimension



APPENDIX J

Refrigerant Receiver output characterstics

Horizontal receivers F302H

Input Values

Compressor model	**1x 6FE-40Y**
Refrigerant	R134a
Reference temperature	Dew point temp.
Evaporating SST	-24.8 °C
Condensing SDT	48 °C
Mass flow	454 kg/h

Output

Types	**F302H**
Receiver volume	30.0 dm³
max refrigerant charge	33.1 kg
receiver load	91%
Receiver unit	mounted compl.
lower fixing rails	327301-05
upper fixing rails	327301-10
upper fixing plate	320366-02

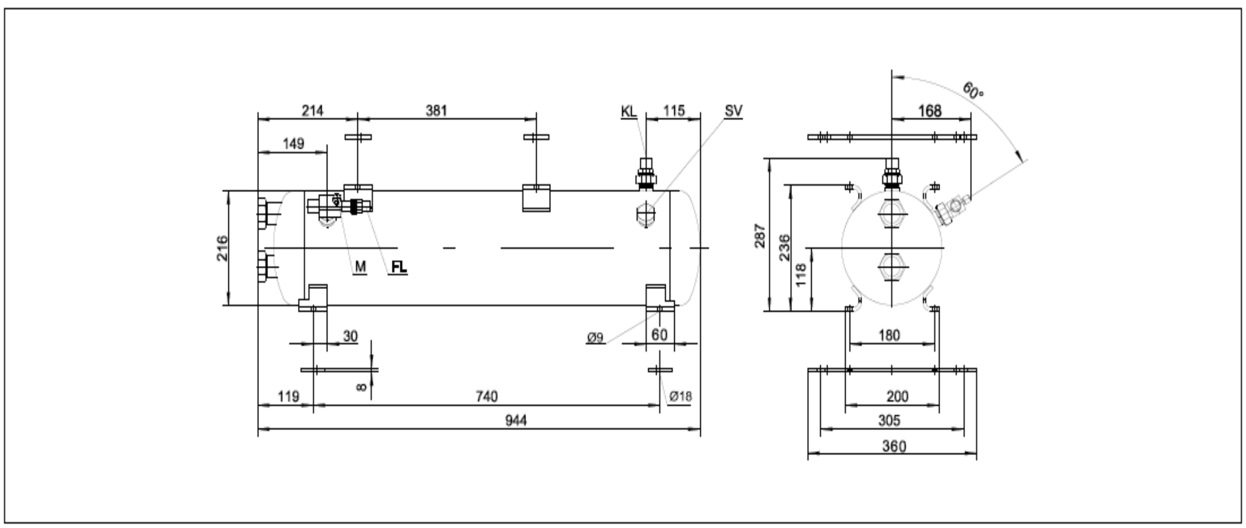
APPENDIX K

Technical Data for the Horizontal Receiver

Weight	27,2 kg
Total width	944 mm
Total depth	276 mm
Total height	287mm
Receiver volume refrigerant	30,0 l
Max. refrigerant charge 90% at 20°C	20°C
R22	32,7 kg
R134a	33,1 kg
R407C	31,3 kg
R404A/R507A	28,8 kg
Connection inlet KL	22mm - 7/8"
Connection thread/ -flange	1 1/4" - 12 UNF
Connection outlet FL	22mm - 7/8"
Connection thread/ -flange	1 1/4" - 12 UNF
Gauge	7/16" 20UNF
Connection for pressure relief valve	1 1/4"-12UNF
Adapter for pressure relief valve	Option
Minimum level control	Option
Maximum level control	Option
*According PED 97/23/EC	Standard
Special Approvals (on request)	Option

APPENDIX L

Horizontal Refrigerant Receiver Dimension



APPENDIX M

Assembly of the Body, Sealing and Bonding in Interior and Exterior of the car envelope

Source: - www.sika.com/ refrigerated trailers and containers

APPENDIX N

Technical Specification of Refrigerated Car

Vehicle Length:	21m
Load Weight:	60t
Load Capacity:	120m³
Inside Temperature:	-24 ~15℃
Outside Temperature:	40~-40 ℃
Generator Power:	15kW
Design Speed:	120km/h
Railway Gauge:	1435mm

APPENDIX O

Average summer time weather conditions in AIYSHA (the hottest region on the railway line to Djibouti), Latitude = 104524, Longitude = 0423439 and Elevation = 721 m (*Source: - National Meteorology Agency*)

Year: - 2012					
Month	**Temperature (ºC)**		**Relative Humidity (%)**		**Average Sunlight Hours**
	Min.	**Max.**	**Min.**	**Max.**	
May	20.0	41.0	53	93	8.35
June	23.0	41.0	55	93	7.00
July	22.0	41.0	51	100	7.85
August	21.0	39.0	57	100	7.50
Year: - 2013					
Month	**Temperature (ºC)**		**Relative Humidity (%)**		**Average Sunlight Hours**
	Min.	**Max.**	**Min.**	**Max.**	
May	21.0	41.0	53	93	7.70
June	25.5	41.0	49	93	9.30
July	20.0	39.5	62	93	7.25
August	19.5	37.5	67	93	7.10