Ministry of Education and Science of Ukraine
National University "Odesa Maritime Academy"
Educational and Scientific Institute of Engineering
Department of Ship Auxiliary Plants and Refrigeration Equipment

MASTER'S THESIS

on the topic:

EFFICIENCY ANALYSIS OF NEW-GENERATION REFRIGERANTS IN SHIP'S REFRIGERATION SYSTEMS

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The work was heard at the meeting of the Department of Ship Auxiliary Plants and Refrigeration Technology.

Recommended for defense at the EC, ~~Protocol No.~~ 6 ~~dated~~ Dec 16 2025.
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Odesa – 2025

Ministry of Education and Science of Ukraine
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Department of Ship Auxiliary Plants and Refrigeration Equipment

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"10" ___09___ 2025.

ASSIGNMENT
for the completion of the master's thesis

Cadet (student) of ESIE *Kharchenko Kostiantyn*

1. Thesis topic: *Efficiency analysis of new-generation refrigerants in ship's refrigeration systems*

Approved by the order of the Rector of NU "OMA" No. 1414 dated 24/11/ 2025.

2. Object of research: *Ship refrigeration systems.*

3. Subject of research: *Methods for improving energy efficiency and reducing greenhouse gas emissions during the operation of ship refrigeration systems.*

4. Volume of the explanatory note: *approx. 80 pages of printed text.*

5. Structure of the explanatory note:: *Five chapters of the main part: one review, four – calculation and analytical with conclusions and development of recommendations, other elements of the explanatory note - in accordance with the requirements for the master's qualification work.*

6. Content of the main part of the explanatory note (list of issues to be developed): *Description of the prototype vessel and its main engine, basic calculation of the vessel's auxiliary equipment, requirements for ship refrigeration systems, diagram and operating principle of the provision chamber refrigeration machine and air conditioner, analysis of energy and environmental characteristics of refrigeration machines when operating on different refrigerants, analysis of technological factors and safety requirements for the use of different refrigerants in ship conditions, development of recommendations for refrigerant selection.*

7. List of graphic material: *slides designed as a presentation (MS Office Power Point).*

The work must be performed in accordance with the "Methodological Instructions for the Master's Thesis", approved by the Academic Council of ESIE on June 27, 2023, Protocol No. 11.

8. Consultants:

8. Consultants:

Type of consulting	Consultant (academic title, position, full name)	Signature, date	
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Reviewing of the English language content	PhD, Tsynova M.V.	23.09.25	15.12.25

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№ з/п	Name of the thesis stage	Supervisor's mark on stage completion (date, signature)
1.	*Problem analysis, review of literary sources*	
2.	*Review and analysis of the system diagram and structure, selection of technical solutions for further analysis*	
3.	*Calculation part*	
4.	*Analysis of obtained results*	
5.	*Energy efficiency*	
6.	*Formatting of the explanatory note, preparation of the work presentation, preparation for the work defense*	
7.		

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Deadline for submission of the thesis to the graduating department 15/12/25

Thesis Supervisor ________ (signature) Prof. Olga Khliyeva (academic title, position, full name)

Thesis Performer ________ (signature) Kostiantyn Kharchenko (full name)

РЕФЕРАТ

Дипломна робота магістра на тему: «Аналіз ефективності холодоагентів нового покоління у судновій холодильній техніці»: 101 с., 22 рис., 17 табл., 36 джерел, 12 слайдів презентаційного матеріалу.

Магістерське наукове дослідження спрямоване на розв’язання науково-прикладного завдання – зниження впливу на навколишнє середовище суднового холодильного обладнання під час його експлуатації шляхом вибору раціонального холодоагенту з використанням методів енергетичного та екологічного аналізу.

Висунута та підтверджена наукова гіпотеза про те, що підвищення енергоефективності та екологічності роботи суднового холодильного обладнання можливе за рахунок вибору раціонально холодоагенту нового покоління з низьким потенціалом глобального потеплення.

Встановлено, що заміна холодоагенту R404A на холодоагент з низьким потенціалом глобального потеплення R1233zd(E) у холодильній машині суднового кондиціонеру повітря призведе до підвищення холодильного коефіцієнту на 24 %. Питомі викиди парникових газів на одиницю виробленого холоду при застосуванні холодоагенту R1233zd(E) на 38 % менше, а при застосуванні холодоагенту R1234ze(E) на 33 % менше, ніж для холодоагенту R404A. Аналогічний аналіз був виконаний для холодильної машини провізійних камер та показане, що заміна холодоагенту R404A на R290 призведе до підвищення холодильного коефіцієнту на 15 % та до зниження викиди парникових газів на 33 %

СУДНОВА ХОЛОДИЛЬНА МАШИНА, РОБОЧЕ ТІЛО ХОЛОДИЛЬНОЇ МАШИНИ, ПОТЕНЦІАЛ ГЛОБАЛЬНОГО ПОТЕПЛЕННЯ, ХОЛОДИЛЬНИЙ КОЕФІЦІЕНТ, ЕКОНОМІЯ ПАЛИВА, ЕНЕРГОЗБЕРЕЖЕННЯ

ABSTRACT

Master's diploma thesis on the topic: “ Efficiency analysis of new-generation refrigerants in ship’s refrigeration systems”: 101

pages, 22 figures, 17 tables, 36 sources, 12 presentation slides.

The master's scientific research is aimed at solving a scientific and applied problem – reducing the environmental impact of ship refrigeration equipment during its operation by selecting a rational refrigerant using energy and environmental analysis methods.

A scientific hypothesis has been put forward and confirmed that it is possible to increase the energy efficiency and environmental friendliness of ship refrigeration equipment by selecting a rational new-generation refrigerant with low global warming potential.

It has been established that replacing the R404A refrigerant with the low global warming potential refrigerant R1233zd(E) in a ship air conditioner refrigeration machine will increase the refrigeration coefficient by 24%. Specific greenhouse gas emissions per unit of cooling produced are 38% lower when using R1233zd(E) refrigerant and 33% lower when using R1234ze(E) refrigerant than for R404A refrigerant. A similar analysis was performed for the refrigeration machine of the provision chambers and showed that replacing refrigerant R404A with R290 will increase the refrigeration coefficient by 15% and reduce greenhouse gas emissions by 33%.

Keywords: SHIP REFRIGERATION MACHINE, WORKING FLUID OF THE REFRIGERATION MACHINE, GLOBAL WARMING POTENTIAL, COOLING COEFFICIENT, FUEL ECONOMY, ENERGY SAVINGS

CONTENTS

NOMENCLATURE

COP – Coefficient of Performance;

GWP – global warming potential;

HFC – hydrofluorocarbon refrigerant;

CFC – chlorofluorocarbon refrigerant;

IMO – International Maritime Organization;

MEPC – Marine Environment Protection Committee;

MARPOL - International Convention for the Prevention of Pollution from Ships;

SOLAS - International Convention for the Safety of Life at Sea;

EEDI – Energy Efficiency Design Index;

GHG – Greenhouse gases;

TEWI – Total Equivalent Warming Impact;

ODP – Ozone Depletion Potential;

HFO – Hydrofluoroolefin;

HCFC – Hydrochlorofluorocarbon;

EEXI – Energy Efficiency Existing Ship Index;

CII – Carbon Intensity Indicator;

HVAC – Heating, Ventilation, and Air Conditioning;

RSW – Refrigerated Sea Water;

DX – Direct Expansion;

ECR – Engine Control Room.

INTRODUCTION

Refrigeration systems are essential components of shipboard plant that ensure the preservation of perishable cargos, secure storage of provisions, and controlled environment conditions within various ship compartments. The reliability and efficiency of these systems will directly affect cargo quality, crew well-being, and the safety of ships under operation. In particular, the operation of the refrigeration plant determines the ability of refrigerated cargo vessels and fishing vessels to maintain required cargo temperatures throughout long voyages and under variable ambient sea conditions. The operation of marine refrigeration plant is therefore an essential requirement for commercial and technical success within the maritime industry.

Traditionally, refrigerating plants onboard ships are based on the vapor-compression cycle, and this remains the most prevalent and energy-saving method of producing low temperatures onboard ships. The major working fluid in such systems is the refrigerant — a substance that undergoes the phase changes to absorb and release heat in the thermodynamic cycle. The choice of refrigerant determines the operating parameters of the compressor, condenser, and evaporator and system-wide Coefficient of Performance (COP) and reliability. The refrigerant is thus the focal point that determines both energy and environmental effectiveness of maritime refrigeration appliances.

Over the last decades, halogenated hydrocarbon refrigerants such as R22, R404A, and R134a have been used in the maritime industry because they have optimal thermodynamic properties, chemically stable, and compatible with existing plant. They possess high Global Warming Potential (GWP) and, in some cases, Ozone Depletion Potential (ODP). With the enforcement of international environmental laws — specifically the Montreal Protocol and its Kigali Amendment — manufacturing and use of such high-GWP refrigerants have been subjected to progressively tightening phase-out timetables. The refrigeration sector now faces the challenge of moving to new-generation refrigerants that can satisfy both performance and environmental requirements.

In recent years, various families of alternative refrigerants have emerged, including hydrofluoroolefins (HFOs), natural refrigerants, and HFC/HFO blends. HFOs, such as R1234yf and R1234ze(E), possess very low GWP values (< 10) with zero ODP and are some of the most promising candidates for future marine refrigeration technologies. Natural refrigerants, such as carbon dioxide (R744) and ammonia (R717), are also being considered with their higher thermodynamic efficiency and lower environmental footprint. However, marine applications require particular attention to safety considerations — ammonia toxicity and high CO_2 operating pressure, for instance — and material compatibility with the system and resistance to corrosion in the marine environment [2].

The use of hydrofluorocarbons (HFCs) as an alternative for refrigeration units has grown over the past decades as a replacement to chlorofluorocarbons (CFCs), banned by the Montreal Protocol because of their effect on the depletion of the ozone layer. However, HFCs are known to be greenhouse gases with considerable global warming potential (GWP), thousands of times higher than carbon dioxide. The Kigali Amendment to the Montreal Protocol has promoted an active area of research toward the development of low GWP refrigerants to replace the ones in current use, and it is expected to significantly contribute to the Paris Agreement by avoiding nearly half a degree Celsius of temperature increase by the end of this century.

Environmental and regulatory drivers for change and accompanying economic and technical incentives exist. The global transition to low-GWP refrigerants will reduce direct GHG emissions due to refrigerant loss and improve the energy efficiency of onboard equipment with better thermophysical properties of the novel working fluids. Furthermore, application of these refrigerants is in accordance with the IMO Initial GHG Strategy that has specific decarbonization targets for the shipping industry until 2050 [1]. From the thermodynamic aspect, the selection of a refrigerant having appropriate vapor pressure characteristics, greater latent heat of vaporization, and improved heat transfer coefficients can enhance the system COP and reduce compressor power consumption.

Accordingly, modernization of ship refrigerating equipment by using new-generation refrigerants is not only an environmental need but also a technical possibility to enhance the energy efficiency of ships in general. Thermodynamic performance evaluation of such refrigerants in performing under marine conditions — fluctuating seawater temperature, motion loads on thermal performance, and compact heat-exchanger sizes — is essential to ensure their feasible application.

The purpose of the research is to analyze the feasibility of using new-generation refrigerants with low global warming potential in marine refrigeration equipment for various purposes.

- analysis of the requirements for working fluids in marine refrigeration equipment and current legislation regulating the use of these substances;
- determination of the energy characteristics of refrigeration machines in air conditioning systems and food storage chambers using alternative refrigerants;
- calculate the specific greenhouse gas emissions to identify the environmental impact of using the refrigerants under consideration in marine refrigeration equipment;
- perform an economic justification of the feasibility of introducing the refrigerants under consideration;
- based on the data obtained, develop recommendations for the use of certain working fluids.

TECHNOLOGICAL CARD OF RESEARCH

1. Request for practice
Requirements of international legislation on reducing air emissions from ships and fuel economy

2. Subject
The conversion of marine refrigeration equipment to refrigerants with low global warming potential

3. Objective
To justify the sustainability of retrofitting ship refrigeration equipment to alternative refrigerants with low global warming potential.

4. Hypothesis
The selection of a rational refrigerant for ship refrigeration equipment will lead to a reduction in energy consumption and environmental impact during its operation.

5. Main task
Selection of a rational refrigerant for ship refrigeration equipment, the introduction of which will improve the environmental and energy characteristics of this equipment.

6. Additional tasks
- review of promising alternative refrigerants for ship refrigeration equipment, selection of rational options;
- thermodynamic analysis of the refrigeration cycle when operating on different refrigerants with low global warming potential for different refrigeration systems;
- environmental analysis and justification of the selection of a rational refrigerant for various ship refrigeration systems;
- evaluation of the energy and environmental effect of implementing the proposed technical solutions.

7. Solving the main problem
Choosing a rational refrigerant for ship refrigeration equipment, the application of which will improve the environmental and energy characteristics of this equipment.

8. Scientific result
A rational choice of a refrigerant with low global warming potential for ship refrigeration equipment has been made.

9. Experiment
Experimental data on the energy characteristics of the refrigeration machine, as well as experimental data on the thermophysical properties of refrigerants, were used for analysis.

10. Scientific value
Alternative refrigerants for ship refrigeration equipment have been proposed, the use of which will contribute to improving the environmental and energy performance of its operation.

11. Practical value
The implementation of the recommended technical solutions will lead to a reduction in fuel consumption on ships and a decrease in the amount of pollutants emissions.

12. Scientific provisions
Improvements in the environmental and energy performance of vessel refrigerating systems are achieved through the rational selection of refrigerants and additional retrofitting upgrades.

1 MARINE REFRIGERATION: ENVIRONMENTAL AND ENERGY SAVING PROBLEMS

1.1 Refrigeration equipment on merchant ships and the basic refrigerants

According to the International Maritime Organisation the total use of HCFC/HFC as refrigerant in the world merchant fleet is estimated to consist of 70% R22, 26% R134a and 4% R404A [3]. The release of refrigerants from global shipping (reefer containers excluded) is estimated at 8,400 tons, which corresponds to around 15 million tons CO2 equivalent emissions. If these numbers are compared to the CO2 emissions of shipping the refrigerant emissions constitute about 2% of the GHG emissions of shipping [1, 4].

In Europe, refrigerants that are legal to use in marine refrigeration units can be grouped as follows: saturated hydrofluorocarbons (HFCs), unsaturated hydrofluorocarbons (HFOs), and natural working fluids [4, 5].

Saturated Hydrofluorocarbons (HFCs): R134a: This saturated HFC refrigerant is property wise comparable to R22 and is widely used as an alternative in medium temperature refrigeration systems. It has a GWP of 1.430. R404A: A blend (CF3CH3 / CF3CHF2 / CF3CH2F) with a GWP of 3.922. R407C: A blend (CH2F2/ CF3CHF2 / CF3CH2F) with a GWP of 2.107. Due to high GWP, R404A and other high-GWP HFC blends are restricted under the EU F-gas Regulation No 517/2014.

Unsaturated Hydrofluorocarbons (HFOs): In general, unsaturated hydrofluorocarbons react more rapidly with OH radicals and their atmospheric lifetimes are significantly shorter compared to traditional HFCs. R1234yf: Has a GWP of 4. It is so-called mildly flammable and classified in the newly developed safety group A2L. R1234ze(E): Has a GWP of 7. It exhibits flame limits at temperatures in excess of 28°C. These working fluids are increasingly used in blends (e.g., R513A) to reduce flammability while maintaining performance [5].

Natural Working Fluids: R717 (Ammonia): Has a GWP of 0. It has very high latent heat and the refrigeration capacity per unit mass flow is the highest of all refrigerants used in traditional vapour compression systems. It is flammable and toxic (safety group B2L). Carbon dioxide R744 (GWP = 1) is a non-flammable A1 refrigerant with excellent volumetric capacity and is increasingly used in transcritical marine systems due to favourable COP and TEWI results [6, 7]. Propane R290 (GWP = 3) demonstrates high thermodynamic efficiency but, being an A3 refrigerant, is mainly used in small factory-sealed units on ships [5].

Cargo vessels typically utilize refrigeration for air conditioning and provision refrigeration. Traditionally, the standard air conditioning system in cargo ships is a split system with direct evaporation. The condenser is placed near the engine room and the generator, while the evaporator is placed near to the living area of the crew, connected by refrigerant lines that could be of considerable length. Many ships have an AC system divided in smaller aggregates, in order to decrease the refrigerant pipe length, and thus the refrigerant charge and risk for leakages. Many modern ships reduce refrigerant charges by installing multiple smaller AC units or adopting chilled-water systems [4].

For RoRo ships today, R134a and R407C are dominating with about an equal share (Fig. 1.1 and Fig.1.2) [4].

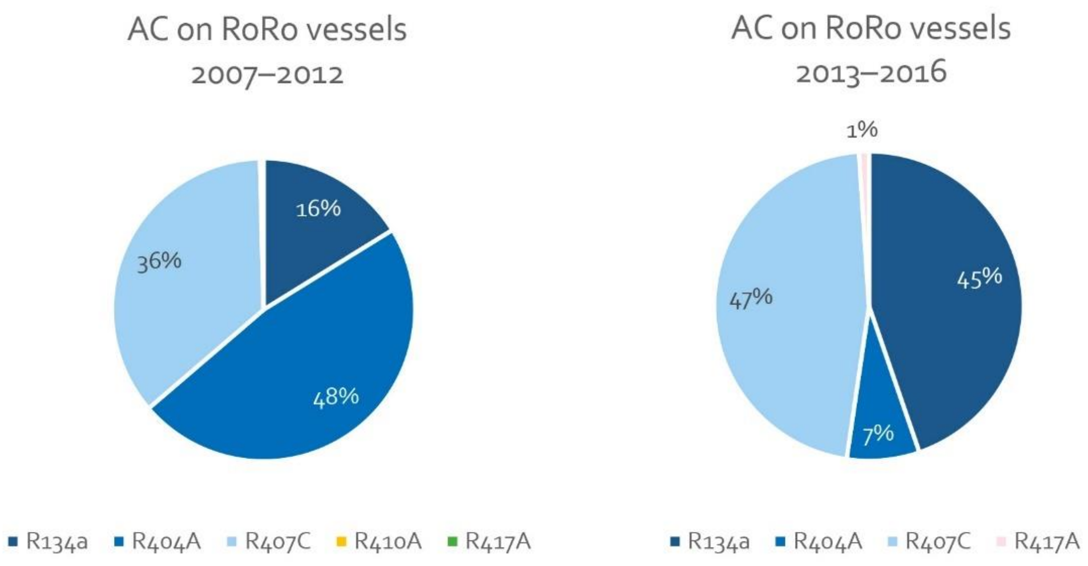


Figure 1.1 Refrigerants for AC units on RoRo vessels [4]

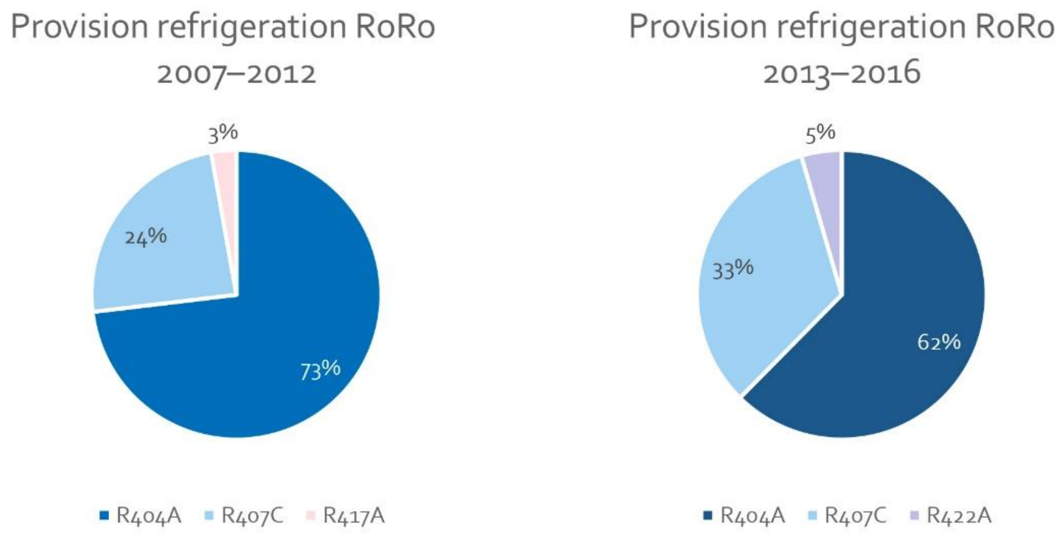


Figure 1.2 Refrigerants for provision refrigeration units on RoRo vessels [4]

The provision refrigeration on a cargo ship is a direct system. Most often the freezing and cooling plant use the same compressor and condenser. For provision refrigeration on cargo ships, the share of R404A has increased during the time period investigated and it is today the dominating refrigerant [4, 5].

The release of refrigerants from global shipping is estimated at 8,400 tons. According to correspondence with vendors, the annual leakage rate is 35% for standard direct systems on cargo vessels (bulk and tankers). Surveys of Scandinavian ships have shown an average leakage rate of 38% for direct AC and provision refrigeration [1, 8]. The main reasons for the high level of emissions from marine vessels compared to land-based systems include the permanent exposition of the entire system to vibrations from sea-waves, and the fact that there are few crewmembers on board skilled in refrigeration, meaning leakages are often not repaired while at sea.

Current research and industrial development focus on two parallel strategies:

• reduction of refrigerant charge and leakage through indirect cooling loops, chilled-water AC systems, and compact factory-sealed units [4, 5];

• substitution of high-GWP HFCs with low-GWP alternatives such as R717, R744, R290, and modern HFO/HFC blends without compromising safety or efficiency [5-7].

System-level studies for passenger ships show that R407F can serve as an interim low-GWP replacement for R404A, offering a balance between efficiency, volumetric capacity, and safety [5]. At the same time, CO_2-based refrigeration systems have demonstrated competitive or superior COP and drastically reduced TEWI compared with HFC systems in several studies [6, 7]. Natural refrigerants such as ammonia and CO_2 are increasingly adopted in fishing vessels, passenger ships, LNG carriers and newbuild cargo ships, provided that safety systems and crew training are adequate [5, 6].

The choice of refrigerant in marine systems is increasingly constrained by environmental legislation (Montreal Protocol, Kigali Amendment, EU F-gas Regulation) and the decarbonisation strategies of shipowners. Therefore, newbuild and retrofit designs must consider not only thermodynamic performance but also refrigerant GWP, leak potential, TEWI, integration with other onboard systems, operational safety, and maintainability.

1.2 Legislative acts regulating the use of refrigerants with a high global warming potential

Technological progress in the refrigeration sector is governed by a number of international legal instruments related to the need to reduce greenhouse gas (GHG) emissions. The most important of these at present is the Paris Agreement (2015). Most of the working fluids (refrigerants) used in refrigeration equipment are responsible for significant GHG emissions due to their high global warming potential (GWP). Specifically aimed at reducing emissions of GHGs such as refrigerants, the Kigali Amendment (2016) to the Montreal Protocol requires countries to reduce the production and consumption of hydrofluorocarbons (HFCs) by 85% by the year 2047. In parallel, regional and national legislative frameworks have been developed to regulate the use of HFCs. For example, the EU F-Gas Regulation (Regulation (EU) No 517/2014) imposes strict limitations on the use of HFCs and encourages the transition to low-GWP refrigerants. Addressing this issue proves to be particularly challenging for marine refrigeration systems.

Although IMO's GHG reduction strategies [1] aim at full decarbonization of international shipping by 2050, they do not explicitly address emissions from refrigerants, which are instead subject to the Montreal Protocol and its Kigali Amendment. At present, HFCs are the most widely used working fluids in marine refrigeration systems, along with ammonia. Ammonia (R717) is primarily applied on large fishing and refrigerated cargo vessels, where high cooling capacity is required. Concerning shipping, the GHG emissions reduction is governed by MARPOL Annex VI. However, Annex VI limits emissions of CO_2, NOx, SO2, and other harmful substances, but does not include HFCs. This means that MARPOL does not prohibit the use of high-GWP refrigerants and does not regulate their release into the atmosphere. Nevertheless, this is only a matter of time.

Leading classification societies, such as Lloyd's Register (LR), Bureau Veritas (BV), Det Norske Veritas (DNV), American Bureau of Shipping (ABS), and others, formulate guidelines and requirements for the design, installation, and operation of onboard refrigeration systems that take into account both technical safety and the environmental impact of refrigerants. These societies are already establishing specific requirements for refrigerants used in marine equipment, aimed at a gradual transition toward using low-GWP substances.

Refrigerant manufacturers have already responded to the changing requirements for working fluids - Fig. 1.3. Blend refrigerants specifically designed for use in marine systems – without the need for equipment retrofitting – have emerged on the market. For example, Wilhelmsen Ships Service offers refrigerants that can be used as drop-in replacements for R404A (GWP = 3920) [2] without modernising the marine refrigeration equipment, and which are classified as non-flammable substances (Wilhelmsen, 2022). Wilhelmsen Ships Service recommends, as a first step, the transition to refrigerants with a GWP below 2500 (in accordance with the F-Gas Regulation, substances with a GWP greater than 2500 have been banned since 2020). In such cases, R-407F (GWP = 1670) is considered the most suitable replacement. In addition, refrigerants with even lower GWP values have been developed specifically for marine applications.

They are R448A (GWP=1360), R449A (GWP=1280), R513A (GWP=573) and R407H (GWP=1588) [2]. All of these blends contain flammable components (such as R32 or R1234yf), although the mixtures themselves are classified as non-flammable. However, their use represents an intermediate step toward the modernisation of refrigeration systems for the adoption of refrigerants with ultra-low GWP. Currently, the GWP of refrigerants used in large commercial refrigeration systems must not exceed 150 according to the EU F-Gas Regulation (Regulation (EU) No 517/2014). Although this requirement does not yet apply to the maritime sector, future regulations are expected to impose stricter limits on refrigerants for onboard applications.

Table 1.1 Summary of Legislative acts that regulate the usage of High-GWP refrigerants

Legislative act/ Protocol	Main provisions	Date of implementation
Montreal Protocol	Ban on CFCs and HCFCs due to ozone depletion; gradual transition to HFCs	1987
Kyoto Protocol	Inclusion of HFCs as greenhouse gases; restrictions on their production and use	1997
Kigali Amendment to the Montreal Protocol	Global phase-out of HFCs; target reduction of 15% from 2011-2013 levels by 2036	2016
Paris Agreement	General targets for reducing greenhouse gas emissions, including refrigerants	2016
EU Regulation No. 517/2014 (F-Gas)	Strict restrictions on fluorinated gases with GWP > 150; phase-out of new installations with HFCs	2014

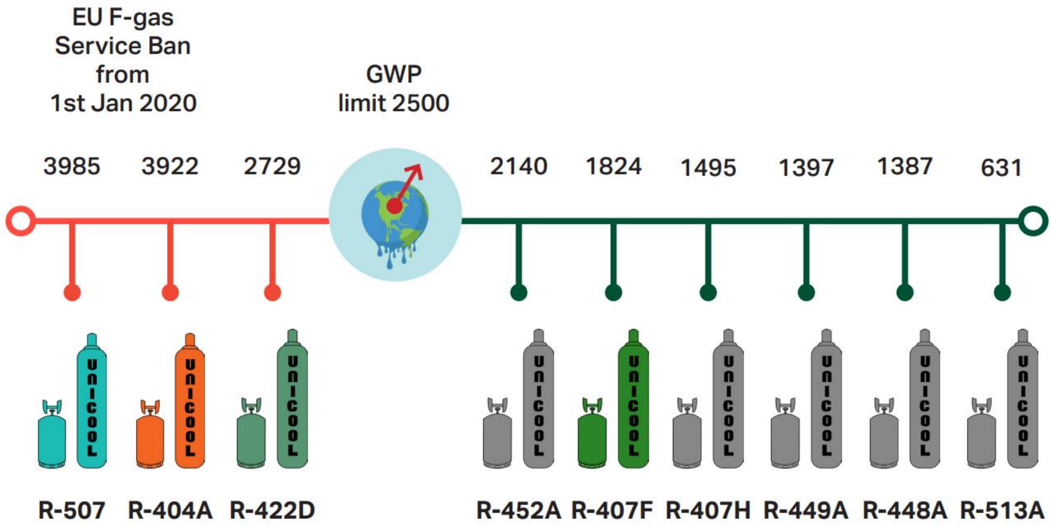


Figure 1.3. Low GWP refrigerant for marine application [9]

1.3 Next-generation low-GWP alternative refrigerants for marine refrigeration systes

Ultra-low GWP refrigerants suitable for both land-based and marine refrigeration systems are currently the subject of active discussion and evaluation [4, 8, 10-12]. They can conventionally be divided into two groups. The first group includes the so-called 'natural' refrigerants (ammonia, carbon dioxide, and hydrocarbons), as these are substances that naturally occur in the environment. The second group comprises the so-called 'synthetic' refrigerants, whose presence in the environment is not of natural origin.

Ammonia (R717). It provides high efficiency, simple leak detection and oil management, but is toxic and flammable. Used in larger vessels and fishing ships. Ammonia has long been used on ships, and design and operational guidelines for marine refrigeration systems using ammonia have already been developed.

Carbon Dioxide (R744). Refrigerant is used in transcritical and subcritical refrigeration systems that operate at high pressures that vary from 30 to 90 bar and even higher when transcritical (Heinen & Hopman, Natural refrigerants). Increasingly considered for use in marine refrigeration and air conditioning systems [13, 14].

Hydrocarbons (HCs, e.g., propane R290, isobutane R600a, propylene R1270) are flammable. Used in small refrigeration units (up to 500 g of refrigerant charge without specific safety measures). They are increasingly considered for larger systems with safety measures.

Hydrofluoroolefins (HFOs, e.g., R1234yf, R1234ze) have low flammability. Unlike the refrigerants discussed above, they are not classified as 'natural'. They are considered emerging alternatives to HFCs in both refrigeration and air conditioning. They can be used in newer systems designed for low-GWP refrigerants as well as for retrofits. Nevertheless, the GHG emissions from their production, the environmental impact of their decomposition products and their impacts on human health and safety need to be understood [4, 13].

Table 1.2 Classification of refrigerants by flammability according to standards ISO 817 and ASHRAE Standard 34

Class	Flammability	Explanation	Examples of ultra-low GWP refrigerants
1	Non-flammable	Do not exhibit flame propagation when tested in air at 60°C and 101.3 kPa (normal conditions)	R744 (CO_2), R1233zd(E)
2	Lower flammability	Exhibit flame propagation with moderate burning velocity; flammable in air under normal conditions	-
2L	Lower flammability, low burning velocity	Exhibit flame propagation but have a low burning velocity (<10 cm/s); slow to ignite and spread	R717 (ammonia), R1234yf, R1234ze(E)
3	Highly flammable	Readily ignite and propagate flame rapidly in air; high burning velocity	R290 (propane), R600a (isobutane), R1270 (propylene)

The transition to "eco-friendly" refrigerants has not yet been fully resolved, either for land-based or transport applications, including marine systems. Ammonia, HCs, and

HFOs are characterised by excellent environmental properties (ultra-low GWP), but they are flammable (Table 1), which limits their use onboard ships. Moreover, HFOs are expensive and often less efficient than HCs and ammonia. CO_2 is a non-flammable refrigerant. However, high working pressures are inherent to it. It requires refrigeration systems with specific design solutions and components; the cost and weight are significantly higher than those of other systems, a critical factor in marine applications. Regarding efficiency, CO_2 systems also often fall short compared to systems using other refrigerants. Thus, transitioning shipboard refrigeration systems to low-GWP refrigerants is significantly more challenging than for stationary systems.

The main limitation hindering the adoption of low-GWP flammable refrigerants (except R717) in maritime applications is the lack of established standards and experience in designing and maintaining onboard refrigeration equipment with such refrigerants. Comprehensive standards have been developed and practically tested only for R717. However, in recent years, the other working fluids discussed above have also gradually started to appear in standards, regulations, rules and guidelines related to marine refrigeration systems. This indicates a gradual adaptation of maritime refrigeration equipment to the new realities dictated by the Paris Agreement.

2. CALCULATION AND SELECTION OF AUXILIARY EQUIPMENT FOR THE ENGINE ROOM OF THE PROTOTYPE VESSEL

All calculations presented in this section have been performed in accordance with the Rules for Classification and Construction of Sea-going Ships of the Shipping Register of Ukraine [15].

The initial data for calculations are taken from the data of the prototype vessel.

2.1 Main parameters of the prototype-vessel and ship auxiliary plants as object of study

The prototype is a cargo ship designed to transport wheeled vehicles (ro-ro).

The vessel type is a single-propelled ro-ro vessel with double boards, double bottom, with the engine and steering compartments located at the aft, and living and service quarters located longitudinally, with a straight bow and a cruiser stern, cut off in the above-water part like a transom, and a flow-directing wing installed at the stern on the port side. The main characteristics are given in Table 2.1.

Table 2.1 - Main characteristics of the vessel

Type	Ro-Ro
Deadweight	20.586
Displacement when loaded	75.036 tons
Cargo	7539 CEU
Speed (knots)	21.5
Maximum length	200 m
Width	38 m
Main engine	KOBE DIESEL-MITSUBISHI
Nominal effective power of main engine	13.200 kW
Diesel generators (power, number)	2850 kW x 3 (YANMAR 6EY22(A)LW)

A ro-ro ship (roll-on/roll-off) is a type of cargo ship specially designed to transport wheeled cargo such as cars, motorcycles, trucks, buses, and railway cars. Cargo can drive onto and off the ship on its own wheels or with the help of a platform. Ro-Ro ships have a horizontal cargo handling system and are widely used in maritime trade to transport motor vehicles and other wheeled cargo. The ship we use as a prototype has 13 enclosed cargo decks, two cargo ramps, and a capacity of 7,539 CEU.

The ship's power plant consists of a complex of equipment (heat engines, mechanisms, devices, pipelines, systems) designed to convert fuel energy into mechanical, electrical, and thermal energy and transport it to consumers. These types of energy ensure: the movement of the ship at a given speed; safety and reliability of navigation; the operation of engine room mechanisms, deck mechanisms and devices; electric lighting; the operation of navigation, control, signaling, and automation equipment; general ship and domestic needs of the crew; the performance of various production operations on the vessel (cargo operations, pumping of working liquids, operation of fire-fighting equipment, etc.).

The ship's power plant is diesel-powered with a main engine that transmits power directly to the shaft line. The power plant consists of diesel-driven generators. All auxiliary mechanisms are electrically driven. Two types of fuel are used for the engine:

1) light diesel for auxiliary diesel generators and emergency use; for the main engine during start-up, manoeuvres, navigation in narrow areas and before stopping;

2) heavy fuel for operation in the main modes of the main engine and auxiliary boiler.

While the ship's power plant is substantial, the focus of this thesis is on the primary consumers of this electrical power that relate to refrigeration, as defined in the thesis topic. On this prototype vessel, two such independent systems are critical: the air conditioning (AC) plant and the refrigeration provision plant. These systems represent a continuous, 24/7 "hotel load" and are essential for both crew welfare and operational safety.

Marine Air Conditioning (HVAC) System. The HVAC plant is essential for maintaining a habitable thermal environment in the accommodation block, galley, and critical spaces like the Engine Control Room (ECR) and navigation bridge. On this prototype, the system is a Direct Expansion (DX) plant. This design uses the refrigerant to cool the air directly within one or more Air Handling Units (AHUs). The cooled air is then distributed via ductwork. This medium-temperature application (maintaining air at approx. +18°C to +22°C) has utilized refrigerants such as R134A. The energy efficiency of this system is a key focus of this research.

Provision Refrigeration Plant Separate from the HVAC, the provision plant is a critical system for crew sustenance and welfare, essential for long voyages. This plant serves the ship's galleys and food stores, typically comprising several insulated rooms with different temperature requirements, such as a freezer room and a chiller/vegetable room (Table 2.2). These systems are universally direct-expansion (DX) vapor-compression cycles. Due to the low temperatures required for the freezer room, the chosen refrigerant must have suitable thermodynamic properties at low evaporating pressures. For decades, the industry standard for this application has been R404A.

Table 2.2 – Air parameters in food storage chambers and volumes of these chambers

Chamber	Volume, m^3	Temperature, °C
Meat	22.6	-18
Fish	12.5	-18
Vegetable	22.6	+4

As outlined in the approved research task, this research will specifically analyze the performance of the provision refrigeration plant as it transitions away from the high-GWP refrigerant R404A. Furthermore, the efficiency of the air conditioning plant will be analyzed. Both systems are prime candidates for modernization due to the legislative pressures described in the introduction and their direct impact on the vessel's fuel consumption and (CII) rating. The analysis will compare the baseline performance (using R404A) against new-generation alternatives to provide a clear efficiency analysis.

2.2 General information about the KOBE DIESEL-MITSUBISHI main engine

The main engine is a Mitsubishi 7UEC60 type, 7-cylinder, crosshead turbocharged engine with a built-in thrust bearing. The engine is equipped with an integrated control system and automation devices that ensure unattended operation. The main engine is controlled remotely from the CPU and the wheelhouse.

Table 2.3 Main characteristics of the KOBE DIESEL-MITSUBISHI 7UEC60LSE-Eco-A2 diesel engine

Cylinder diameter (mm)	600
Piston stroke (mm)	2400
RPM	104
Average effective pressure (bar)	15
Maximum pressure (bar)	160
Number of cylinders	7
Power of single cylinder (kW)	1885
Engine power (kW)	13200

Support plate and main bearing. The frame is made with a thrust bearing at the rear of the engine. The base consists of high, welded, longitudinal and welded transverse beams with cast load-bearing supports. Elastic clamping bolts and hydraulic tightening tools are used to install the engine on the ship. The base is manufactured without a cone for engines and is mounted on epoxy pads. The oil pan, which is made of sheet steel and welded to the base plate, collects return oil from the forced lubrication and oil cooling system. The oil outlet from the oil pan is usually vertical and equipped with a grate. The main bearings consist of a thin-walled steel housing lined with bearing metal. With the help of special tools in combination with hydraulic tools, it is possible to repair the crankshaft and other structural elements.

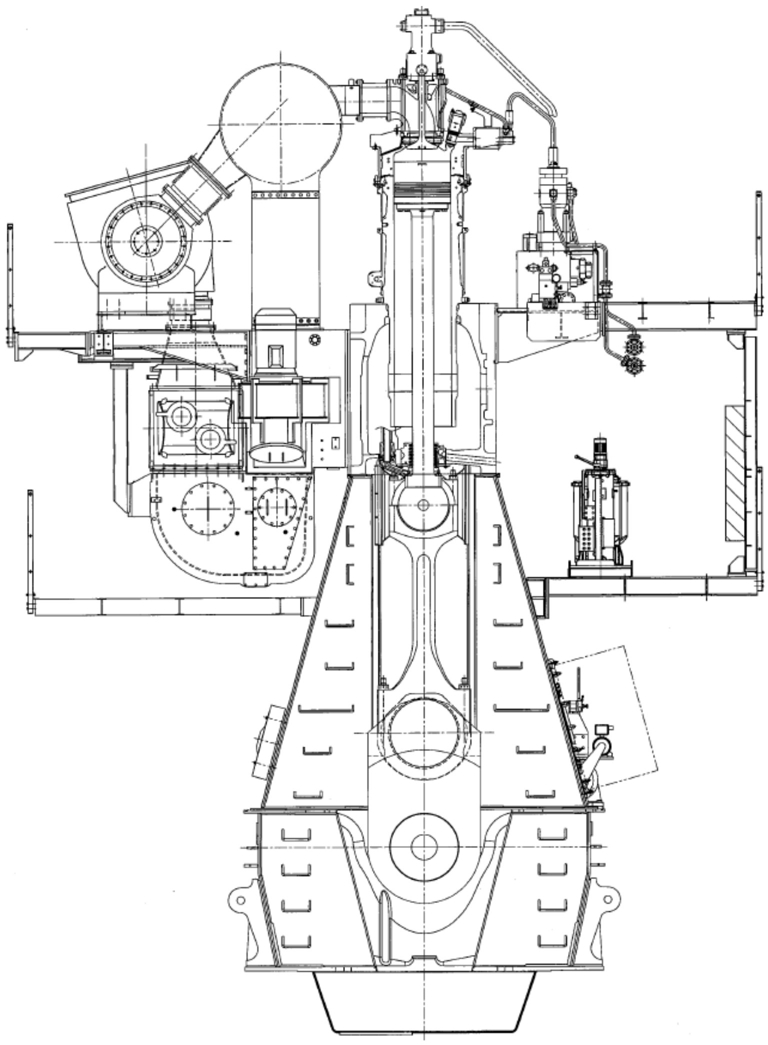


Figure 2.1 - Cross-section of the KOBE DIESEL-MITSUBISHI 7UEC60LSE-Eco-A2 engine

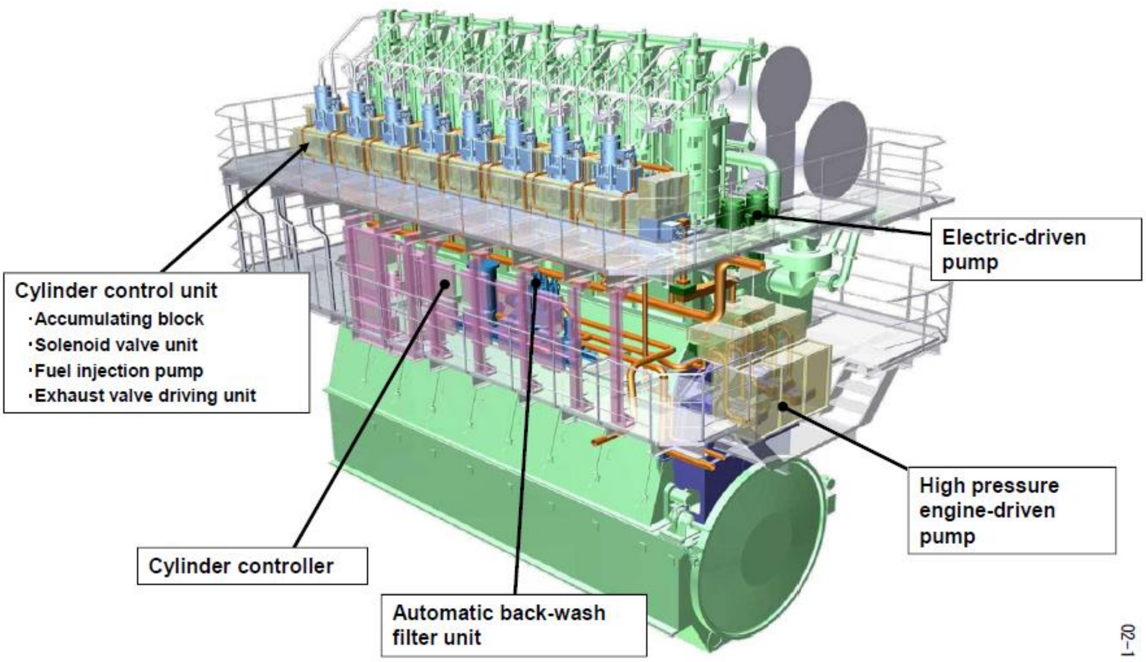


Figure 2.2- Equipment of the hydraulic engine control system

Frame box. Frame box of welded construction. On the exhaust side, it is equipped with safety valves for each cylinder, and on the other side, it is equipped with large hinged doors for each cylinder. The guide crosspieces are welded to the box frame. The frame box is bolted to the base plate. The support, frame box, and cylinder frame are bolted together.

Cylinder frame and oil seal. The cylinder frame is cast, welded, and comes with access covers for cleaning the blow-by air space, if necessary, and for checking piston rings, etc. Together with the cylinder liner, it forms the blow-by air space. Oil inlet pipes for cooling the piston are installed on the cylinder frame. The blow-by air receiver, turbocharger, air cooling box, and gallery brackets are located on the cylinder frame. At the bottom of the cylinder frame, there is a piston rod seal equipped with sealing rings for air removal and oil scraper rings that prevent oil from rising from the crankcase into the air space.

Cylinder liner. The cylinder liner is made of alloyed cast iron and is suspended in the cylinder frame by means of low-mounted flanges. The upper part of the cylinder liner is equipped with a cooling jacket. The cylinder liner has blow-off holes and drilled holes for cylinder lubrication.

Cylinder head. The cylinder head is forged, made in one piece, and has holes for cooling water. It has a central hole for the exhaust valve and holes for the fuel valves, starter valve, and indicator valve. The cylinder cover is attached to the cylinder frame with pins and nuts.

Crankshaft. Semi-assembled crankshaft made of forged or cast steel. For engines with 9 cylinders or more, the crankshaft is supplied in two parts. At the rear end, the crankshaft is equipped with a clamp, a flange for a fitting, a gear wheel for increasing the transmission to the hydraulic power unit, if installed on the engine, a flange for a return wheel, and bolts for a coupling to the intermediate shaft. At the front end, the crankshaft is equipped with a clamp for an axial vibration damper and a flange for installing a tuning wheel. The flange can also be used for power take-off if necessary. Clamping bolts and nuts for connecting the crankshaft to the intermediate shaft are not normally supplied.

Thrust bearing. The thrust of the screw is transmitted through the thrust collar, segments, and thrust plate to the end stops and engine mount, and thus to the ship's hull. The thrust bearing is located at the rear of the engine. The thrust bearing is of the Michell type and consists mainly of a thrust collar on the crankshaft, a bearing support, and segments made of steel lined with white metal.

Connecting rod. The connecting rod is forged or cast from steel and equipped with bearing caps for the crosshead and connecting rod bearings. The crosshead and connecting rod bearing caps are attached to the connecting rod with pins and nuts and tightened using hydraulic devices. The crosshead bearing consists of a set of thin-walled steel housings lined with bearing metal. The crosshead bearing cap is solid, with an angular cutout for the piston rod. The connecting rod bearing is thin-walled steel, lined with bearing metal. Lubricant is supplied through channels in the crosshead and connecting rod.

Piston. The piston consists of a piston and a skirt. The piston body is made of heat-resistant material. The piston cleaning ring is located at the very top of the cylinder liner and cleans excess ash and carbon deposits from the top of the piston. The piston has four ring grooves with a hard chrome-plated upper and lower surface coating. The top ring is of the CPR (controlled pressure relief) type, while the other three piston rings have a slanted cut. The top piston ring is higher than the others. All four rings are coated with aluminum. The piston skirt is made of cast iron with bronze.

Piston rod. The piston rod is made of forged steel and hardened on the running surface. The piston rod is connected to the crosshead with four bolts. The piston rod has a central hole which, in combination with the cooling oil pipe, forms the inlet and outlet for the cooling oil.

Crosshead. The crosshead is made of forged steel and comes with cast steel guide shoes. A telescopic oil inlet pipe and oil pipe are mounted on the guide shoes.

Air cleaning system. Air is collected in the turbocharger directly from the engine room compartment through the turbocharger intake silencer. From the turbocharger, the air is fed through the supercharger pipe, air cooler, and blow-by air receiver to the blow-by ports of the cylinder liners.

Air cooler. Each turbocharger is equipped with a monoblock air cooling system designed for cooling with seawater. Alternatively, it can be cooled with a central cooling system using fresh water. Working pressure up to 4.5 bar.

Auxiliary blower. The engine is equipped with an auxiliary electric blower. The suction side of the blower is connected to the blow-by air space after the air cooler. Check valves are installed between the air cooler and the purge air receiver, which automatically close when the auxiliary blowers supply air. The auxiliary blowers will start up sequentially before the engine starts to ensure sufficient purge air pressure for a safe start.

Exhaust system. Through the exhaust valves, the exhaust gases are directed to the exhaust receiver, where the pressure in the individual cylinders is equalized and the total volume of gas is reduced to the turbocharger(s). After the turbocharger(s), the gas is fed to the external exhaust system. Compensators are installed between the exhaust valves and the receiver, as well as between the receiver and the turbochargers. The exhaust gas receiver and exhaust pipes are insulated. A protective grille is installed between the exhaust gas receiver and the turbocharger.

Exhaust turbocharger. The engine is equipped with two ME-Turbo turbochargers.

Reverse. Reverse is performed electronically and controlled by the engine management system by changing the fuel injection timing, activating the exhaust valves and start valves.

Hydraulic power source. The hydraulic power supply (HPS) filters and generates oil pressure for use in the hydraulic system. Depending on the type of engine, the HPS consists of 2-4 pumps that are driven either mechanically by the engine or electrically. The hydraulic pressure is 300 bar. An electrically driven HPS can be installed on the engine, usually at the stern or in the engine room.

Hydraulic cylinder unit. The hydraulic cylinder unit (HCU), one per cylinder, consists of a base plate on which the distributor is mounted. The distribution unit is installed with one or more accumulators to ensure that the required peak hydraulic oil flow is available during the fuel injection sequence. The distribution unit serves as a mechanical support for the hydraulically activated fuel pressure and the hydraulic drive of the exhaust valve.

Fuel pressure booster. The engine is equipped with one hydraulic fuel pressure booster for each cylinder. Fuel injection is activated by a multi-valve (FIVA) controlled electronically by the cylinder control unit (CCU).

Nozzles and start-up air valve. The cylinder cap is equipped with two nozzles, a start-up air valve, and an indicator valve. The opening of the injectors is controlled by a high-pressure pump created by pressure increase, and the valves are closed by a spring. An automatic ventilation flap ensures fuel circulation through the injector and high-pressure pipes when the engine is stopped. The pipes are insulated but not heated. Slow rotation before start-up is a program that is included in the basic engine management system. The start valve is opened by control air and closed by a spring.

Exhaust valve. The exhaust valve consists of a valve body and a spindle. The valve body is made of cast iron and equipped with water cooling. The exhaust valve spindle is manufactured by Nimonica. The exhaust valve is bolted to the cylinder with studs and nuts. The exhaust valve is opened hydraulically by an electronic valve activation system and closed by air pressure. The operation of the exhaust valve is controlled by a proportional valve, which also activates fuel injection. During operation, the valve spindle rotates slowly under the influence of exhaust gases acting on small fixed blades. to the spindle. The seal of the exhaust valve guide spindle is ensured by a controlled oil level (COL).

Pipeline installation. All pipelines are made of steel pipes, except for the control air and fuel steam heating pipelines, which are made of copper. The pipes are equipped with connections for local devices, alarms, and safety equipment, as well as several sockets for additional signaling equipment.

2.3 Hourly fuel consumption of the main engine KOBE DIESEL-MITSUBISHI 7UEC60LSE

$$Q_e = g_e \cdot \Sigma N_e$$

$$Q_e = 0.164 \cdot 8600 = 2164.8 \text{ kg/hour}$$

where g_e – specific effective fuel consumption, kg/(kW·hour); ΣNe – total effective power of main engine, kW .

The amount of heat released during fuel combustion:

$$q = Q_e \cdot Q_н$$

$$q = 2164.8 \cdot 41400 = 89622720 \text{ kJ/kg}$$

where $Q_н$ – the lowest heat of combustion of fuel, kJ/kg.

We assume the following for the fuel of the KOBE DIESEL-MITSUBISHI 7UEC60LSE engine:

$Q_н = 41400$ *kJ/kg.*

2.4 Machinery and equipment that serve the main engine KOBE DIESEL-MITSUBISHI 7UEC60LSE

The flow rates of working fluids in the MV HORIZON HIGHWAY ship systems required for calculating pipe diameters:

- fuel transfer system – 1.6 m/s,
- fuel in the transfer system – 1.3 m/s,
- oil in the circulation system – 2.6 m/s,
- water in cooling systems – 2.7 m/s,
- water in sanitary systems – 3.2 m/s.

2.4.1 Fuel System

Designed for receiving, storing, pumping, cleaning, heating, and supplying fuel to main and auxiliary engines and boilers, as well as for pumping it ashore or to another vessel.

The KOBE DIESEL-MITSUBISHI 7UEC60LSE diesel engine is designed to operate on heavy fuel grades, for which the ship's power plants are equipped with a special fuel preparation system. The system includes heavy fuel and diesel fuel separators, steam heaters equipped with thermostats, heavy fuel and diesel fuel storage and consumption tanks, and coarse and fine filters. Each separator has two paired fuel pumps (for injection and pumping). The tanks of the prototype vessel are equipped with a steam heating system. The fuel is pumped to the settling tank, where, after settling for 20 to 24 hours, it is pumped to the heater, then to the separators, and then to the consumption tank. From the consumption tank, the fuel is fed by fuel transfer pumps through the heater to the main engine KOBE DIESEL-MITSUBISHI 7UEC60LSE. The fuel system also includes pumps for transferring fuel from one tank to another.

Fuel tanks. The capacity of each of the two settling tanks and two heavy fuel service tanks $V_{вт}$ is selected based on the calculation of ensuring the operation of the main engine KOBE DIESEL-MITSUBISHI 7UEC60LSE over time τ_1 = 24 hours.

$$V_{вт} = (Q_e \cdot \tau_1)/\rho_т$$

$$V_{вт} = (2164.8 \cdot 24)/\ 960 = 54.12\ m^3,$$

where ρ_m – fuel density, kg/ m^3.

For heavy fuel assumed ρ_m = 960 kg/m^3.

The volume of each of the two diesel fuel storage tanks is assumed to be equal to 80% of the volume of the two heavy fuel tanks, i.e., 42.3 m^3.

The supply of simultaneously operating fuel separators is calculated based on the condition of separating the daily fuel consumption for $\tau_{c.n.}$ from 8 to 12 hours.

$$Q_c = V_{вт}/\tau_{c.n.}$$

$$Q_c = 54.12/12 = 4.5\ m^3/hour$$

where $\tau_{c.n.}$ assumed as 12 hours.

Purifiers. We install two heavy fuel purifiers and one diesel fuel purifier. The diesel fuel purifier is accepted as the same for standardization. We select separators based on supply.

Based on the calculated required throughput of 4.5 m^3/h for heavy fuel oil (HFO) with a density of 960 kg/m^3 (at 15°C), the Mitsubishi Selfjector SJ-50H (Hercules Series) was selected.

This model was chosen because the required flow rate of 4.5 m^3/h is close to the capacity limit of the smaller model (SJ-40H). To ensure efficient separation quality and to provide a necessary capacity margin (approximately 15-20%) for handling lower quality fuels or peak loads, the SJ-50H model is the technically optimal solution.

Technical Specifications of Mitsubishi Selfjector SJ-50H:

-Model: Mitsubishi SJ-50H (Hercules Series);

-Type: Disc type centrifugal separator with self-cleaning (total discharge) mechanism;

-Application: Purification of Heavy Fuel Oil (HFO) and Marine Diesel Oil (MDO);

Throughput Capacity: ~5.250 L/h (for HFO 380 cSt 50°C), which covers the required 4.500 L/h with a sufficient efficiency margin;

-Electric Motor Power: 11 kW (Standard);

-Power Supply: 440V / 60Hz;

-Bowl Speed: Approx. 8000 – 9000 rpm;

-Weight (Dry): Approx. 950 kg;

Drive System: Belt drive.

This purifier ensures stable and high-quality purification of heavy fuel oil with a density of up to 1010 kg/m³ at a separation temperature of 98°C, meeting all operational requirements of the vessel's power plant.

The fuel transfer pump must ensure that fuel is pumped out of the larger main storage tank $V_з$ for the time $\tau_{відк}$ = 4 hours. At the same time, it must ensure the transfer of at least the daily fuel consumption by the main engines for the time τ_2 = 6 hours.

For the selected prototype vessel, the storage tank has a volume of $V_з$ = 500 m^3.

$$Q_{нп} \geq V_з / \tau_{відк}$$

$$Q_{нп} \geq 500/4 = 125{,}0 \text{ m}^3\text{/hour},$$

$$Q_{нп} \geq V_{вт} / \tau_2 .$$

$$Q_{нп} \geq 54.12/6 \geq 9.02 \text{ m}^3\text{/hour},$$

where $Q_{нп}$ – fuel transfer pump supply, m^3/hour.

Pressure delivered by the pump $H_{пн}$ equals 0.4 MPa.

The power consumption of the fuel transfer pump drive motor is determined by the formula:

$$P = \frac{Q \cdot p}{3{,}6 \cdot \eta}, \text{ kW}$$

where Q – pump supply, m^3/hour;

p – pump delivery pressure, MPa;

η – Pump efficiency.

For screw pump η from 0.75 to 0.85. We assume η = 0.8.

P=(125·0.4)/(3.6·0.8)=17.4 kW

There should be two fuel transfer pumps with independent drives, one of which is a backup.

The diesel fuel transfer pump is the same as the heavy fuel pump.

The fuel transfer pump flow rate is calculated using the following formula:

$$Q_{пн} = (K_{пн} \cdot Q_e)/\rho_m$$

$Q_{пн}$= (3·2164.8)/960 = 6.77 m^3/hour

where $K_{пн}$ assumed from 2 to 5. $K_{пн}$ = 3.

$p_{пн}$ – delivery pressure of the pump for the KOBE DIESEL-MITSUBISHI 7UEC60LSE, assumed from 0.25 to 0.50 MPa. $p_{пн}$ = 0.30 MPa

The power consumption of the fuel pump drive motor is determined by the formula:

P= (6.77·0.30)/(3.6·0.8)=0.7 kW

For the fuel booster pump drive motor η = 0.8.

Heavy fuel heaters provide heating to the required viscosity. Steam shell-and-tube heaters are used.

Heat amount q_m, which is supplied to the fuel to reach the temperature at which the fuel will have the required viscosity:

$$q_m = Q_{пн} \cdot \rho_m \cdot c_n \cdot (T_2 - T_1)$$

$q_т$ =6.77 · 960 · 1.8 · (353 -310)=503038.08 kJ/hour,

where c_n – specific heat capacity of fuel, from 1.68 to 2.1 kJ/(kg·К);

T_1 – initial fuel temperature (approximately 310 К);

T_2 – final fuel temperature, corresponding to the viscosity of the fuel used, required for this engine (approximately from 2 to 2.5 ° E), according to the rules of the Registry.

$T_2 \leq T_{cn}$ – 10 °C. Flash point T_{cn} for viscous fuels is in the range from 60 to 110 °C.

Heat exchange surface area of the fuel heater:

$$A_m = q_т/(k_т \, \Delta T_m)$$

$$A_m = 503038.08/(1000 \cdot 68.5) = 7.34 \text{ m}^2,$$

where $k_т$ – heat transfer coefficient, can be accepted 1000 kJ/(m^2·hour·К).

ΔT_m – thermal head in the heat exchanger.

$\Delta T_m = T_s - (T_1 + T_2)/2.$

ΔT_m= 400 – (310+353)/2 = 68.5 К

where T_s – steam temperature at operating pressure, T_s approximately from 390 to 400 К. We assumed T_s = 400 К.

We choose a heater based on its heat exchange surface area. According to the calculations performed, the required heat exchange surface area is 7.34 m^2. To ensure efficient heat transfer and account for the fouling factor typical for heavy fuel oil heating, a shell-and-tube heat exchanger with a surface area of 8.0 m^2 was selected.

Consistent with the choice of auxiliary equipment manufacturers, the Sasakura Engineering (Series FH) heater was chosen.

Technical Specifications:

Manufacturer: Sasakura Engineering Co., Ltd.;

Type: Horizontal shell-and-tube heater with U-tube bundle;

Heat Transfer Area: 8.0 m^2;

Heating Medium: Saturated steam (Shell side);

Heated Medium: Heavy Fuel Oil (Tube side);

Material: Carbon steel shell, Seamless steel tubes;

Design Pressure: 16 bar;

The U-tube design allows for free thermal expansion of the tubes, which is essential given the high temperature differences during HFO heating operations. The selected surface area margin compensates for potential fouling during extended operation.

2.4.2 Lubrication system of the KOBE DIESEL-MITSUBISHI 7UEC60LSE engine

It consists of a circulating oil system and a cylinder lubrication system. The circulating lubrication system supplies oil to the rubbing surfaces and also cools the pistons.

The system consists of oil storage tanks, waste tanks, oil circulation pumps, filters, separators, and oil coolers.

Capacity of main lube oil storage tanks for circulation system $V_{м.з.}$ is taken based on the specific consumption of circulating oil $b_{м.м.}$, which is compiled for the LSE 0.0002 kg/(kW·hour), with a 20% voyage reserve.

$$V_{м.з.} = 1.2 \cdot b_{м.м} \cdot \Sigma N_e \cdot \tau_{гд} / \rho_м$$

$$V_{м.з.} = 1.2 \cdot 0.0002 \cdot 13200 \cdot 1200 / 867 = 4.38 \text{ m}^3,$$

where $\tau_{гд}$ – Duration of operation of the main engine KOBE DIESEL-MITSUBISHI 7UEC60LSE in a calculated voyage, hours. Assume for the prototype: $\tau_{гд}$ = 1200 hours.

$\rho_м$ – oil density equal to 867 kg/m^3.

$V_{м.з.}$ with a 20% reserve for the voyage equals 5.26 m^3.

Cylinder oil storage tanks capacity:

$$V_{ц.з} = b_{цм} \cdot \Sigma N_e \cdot \tau_{гд} / \rho_м$$

$$V_{ц.з} = 0.0005 \cdot 13200 \cdot 1200 / 867 = 9.13 \text{ m}^3.$$

where $b_{цм}$ – specific consumption of cylinder oil, depending on the type of engine, for LSE KOBE DIESEL-MITSUBISHI 7UEC60L from $0.4 \cdot 10^{-3}$ to $0.7 \cdot 10^{-3}$ kg/(kW·hour). We assume $b_{цм}$ = 0.0005 kg/(kW·hour).

Circulation oil pump

Circulation oil pump flow rate:

$$Q_{м.н} = \frac{q_{тр} + q_п}{c_м \cdot \rho_м \cdot \Delta T_м}$$

$$Q_{м.н} = (554700 + 2919528)/(3 \cdot 867 \cdot 8) = 167.0 \text{ m}^3/\text{hour},$$

where $c_м$ – heat capacity of oil, can be assumed to be 3 kJ/(kg·K);

$q_{тр}$ – heat of friction removed by oil, kJ/kg;

$q_п$ – heat that oil receives from the piston, kJ/kg;

$\Delta T_м$ – difference between oil temperature at the outlet and inlet of the engine can be assumed 8 K.

$$q_{тр}=3.6\cdot 10^3\cdot a_{тр}\cdot N_e\cdot(1-\eta_м)/\eta_м$$

$$q_{тр}=3.6\cdot 10^3\cdot 0.43\cdot 13200\cdot(1-0.96)/0.96 = 851400 \text{ kJ/kg.}$$

where $a_{тр}$ – the proportion of heat generated by friction and carried away by oil from 0.4 to 0.45. We assume $a_{тр}$ = 0.43;

$\eta_м$ – The mechanical efficiency of the main engine KOBE DIESEL-MITSUBISHI 7UEC60LSE is equal to: $\eta_м$ = 0.96.

$$q_п = a_п\cdot q$$

$$q_п = 0.05\cdot 58390560= 2919528 \text{ kJ/hour.}$$

where $a_п$ – part of the heat transferred from the piston to the oil for KOBE DIESEL-MITSUBISHI 7UEC60LSE, from 0.04 to 0.06. We assume $a_п$ = 0.05.

The power consumption of the circulation oil pump drive motor is determined by the formula:

$$P= (167.0\cdot 0.25)/(3.6\cdot 0.85)= 13.6 \text{ kW}$$

We assume for the drive motor of the circulation oil pump η = 0.85.

$p_{мм}$ – pump delivery pressure, from 0.2 to 0.4 MPa.

We assume $p_{мм}$ = 0.25 MPa.

Lube oil volume in the system:

$$V_{мс} = Q_{мс}/z.$$

$$V_{мс} = 167.0/10 = 16.70 \text{ m}^3.$$

where z – circulation rate, hour^{-1} when turbocharge above 40% z= 10.

Waste oil tank capacity

$$V_{сц}=r\cdot V_{мс}$$

$$V_{сц}=1.25\cdot 16.70 = 20.9 \text{ m}^3,$$

where r – foaming coefficient, which is from 1.2 to 1.3. We assume r = 1.25.

Lube oil purifier

Lube oil purifier feed

$$Q_{м.с} = V_{мс} / \tau_{см}.$$

$$Q_{м.с} = 16.70 / 8 = 2.1\ m^3/hour,$$

where $\tau_{см}$ – time required to treat all oil in the system, $\tau_{см}$ from 4 to 8 hours. We assume $\tau_{см}$ = 8 hours.

We select an oil separator based on the supply rate and also install one spare of the same type. Based on the calculated required feed rate of 2.1 m^3/h for the main engine lubricating oil system, the Mitsubishi Selfjector SJ-20H (Hercules Series) was selected.

Lubricating oil purification requires precise flow control to effectively remove water, sludge, and oxidation products without depleting the oil's additives. The SJ-20H model provides an optimal effective capacity range (approx. 2000 – 2400 L/h for detergent oils), perfectly matching the calculated demand of 2.1 m^3/h while ensuring high separation efficiency.

Technical Specifications:

Model: Mitsubishi SJ-20H (Hercules Series);

Type: Disc type centrifugal separator with self-cleaning (total discharge) mechanism;

Application: Purification of Main Engine System Oil and Auxiliary Engine Lubricating Oil;

Throughput Capacity: ~2200 L/h (Rated for detergent lubricating oil purification),

Electric Motor Power: 5.5 kW

Power Supply: 440V / 60Hz;

Bowl Speed: Approx. 9000 – 10000 rpm;

Weight (Dry): Approx. 700 kg;

Drive System: Belt drive;

Equipped with a "Multi-Monitor" for continuous monitoring of oil inlet temperature, discharge pressure, and leakage detection, ensuring automated and safe operation.

Oil cooler heat exchanger surface:

$$A_м = \frac{q_{тр}+q_п}{k_м \cdot \Delta T_м},$$

$$A_м = (554700+2919528)/(1000 \cdot 12) = 289.5\ m^2,$$

where $\Delta T_м$ - difference between the average temperature of oil and seawater in the cooler, K ($\Delta T_м$ = 12 K);

$k_м$ – heat transfer coefficient, from 500 kJ/ (m^2·hour·K) to 1000 kJ/ (m^2·hour·K). We assume $k_м$ = 1000 kJ/ (m^2·hour·K).

We install two coolers, each with a surface area of 60% of the total, i.e., 173.7 m^2.

Based on the calculated heat transfer surface area of 173.7 m^2, a Plate Heat Exchanger (PHE) was selected. For this capacity, a shell-and-tube heat exchanger would be excessively large and heavy. A PHE offers a compact design, high heat transfer coefficient, and ease of maintenance suitable for main engine lubrication systems. The Hisaka Works, Ltd. (Series LX) heat exchanger was chosen.

Technical Specifications:

Manufacturer: Hisaka Works, Ltd. (Japan);

Type: Gasketed Plate Heat Exchanger;

Heat Transfer Area: 190.0 m^2;

Hot Medium: Lubricating Oil;

Cooling Medium: Seawater;

Plate Material: Titanium;

Gasket Material: NBR (Nitrile Butadiene Rubber), suitable for oil applications;

Design Pressure: 10 bar;

The plate design promotes turbulent flow, reducing fouling tendencies and maximizing thermal efficiency within a minimal footprint. The unit is designed for easy disassembly for cleaning and inspection.

Lube oil transfer pump

The oil transfer pump capacity must be sufficient to transfer oil from the main storage tanks to the circulation system within $\tau_м$ from 0.5 to 1 hour:

$$Q_{м.п} = V_{мс} / \tau_м,$$

$$Q_{м.п} = 16.70 / 1 = 16.7 \text{ m}^3\text{/hour.}$$

Oil should be supplied from the main storage tanks to the circulation system continuously. $\tau_м$ = 1 hour.

The nominal power of the oil transfer pump drive is equal to:

$$P = (16.7 \cdot 0.2)/(3.6 \cdot 0.8) = 1.2 \text{ kW.}$$

$p_{мн}$ – pump delivery pressure, taken from 0.2 MPa to 0.3 MPa; We assume $H_{мн}$ = 0.2 MPa.

We assume the efficiency of the oil transfer pump η = 0.8.

2.4.3 Cooling system of the KOBE DIESEL-MITSUBISHI 7UEC60LSE engine

Sea water pumps

Sea water pumps are used to pump seawater through water coolers, oil coolers, and charge air coolers. Seawater is used to dissipate frictional heat $q_{тр}$, heat transferred through liners, covers $q_ц$, heat transferred from pistons $q_п$, and the heat of the turbocharged air $q_н$.

Heat transferred through liners and covers and cooled by fresh water:

$$q_ц = a_ц \cdot q;$$

$$q_ц = 0.11 \cdot 58390560 = 6422961 \text{ kJ/hour;}$$

where $a_ц$ – part of the heat transferred from liners and covers; the value $a_ц$ varies from 0.1 to 0.14 for LSE.

Fresh water pump delivery:

$$Q_{п.в} = \frac{q_ц}{c_{п.в} \cdot \rho_{п.в} \cdot \Delta T_{п.в}}$$

$$Q_{п.в} = 6422961/(4.2 \cdot 10 \cdot 1000) = 160.6 \text{ m}^3\text{/ hour,}$$

where $c_{п.в}$ – heat capacity of fresh water, we assume 4.2 kJ/(kg·K);

$\rho_{п.в}$ – The density of fresh water is approximately equal to 1000 kg/ m^3;

$\Delta T_{п.в}$ – the difference in temperature between the fresh water at the inlet and outlet of the engine, which is usually in the range of 6 to 10 K.

Nominal power of the fresh water countour water pump drive:

$$P=(Q_{п.в}\cdot p_{п.в})/(3.6\cdot \eta),$$

$$P=(160.6\cdot 0.3)/(3.6\cdot 0.9) =14.9 \text{ kW}.$$

where $p_{п.в}$ – delivery pressure, assumed from 0.2 to 0.4 Mpa;

We assume $p_{пв}$ = 0.3 Mpa;

We assume the efficiency of the piston pump to be equal to $\eta = 0.9$.

The sea water pump delivery is determined by the formula:

$$Q_{з.в} = \frac{q_ц + q_{тр} + r \cdot q_н + q_п}{c_{з.в} \cdot \rho_{з.в} \cdot \Delta T_{з.в}},$$

$$Q_{з.в} = (6422961+554700+(0.6\cdot 2919528)+2335622)/(4.2\cdot 11\cdot 1025) = 245.3 \text{ m}^3/\text{hour}.$$

where $c_{з.в}$ – heat capcity of seawater= 4.2 kJ/(kg·K);

$\rho_{з.в}$ – density of seawater, assumed 1025 kg/m^3;

$\Delta T_{з.в}$ – the difference in temperature between the fresh water at the inlet and outlet of the engine, from 10 K to 15 K; we assume $\Delta T_{з.в}$ = 11 K.

r – coefficient, assumed for turbochargers from 0.5 (with deep utilization), to 1 (without utilization). We assume *r* = 0.6.

$$q_н = a_п \cdot q,$$

$$q_н = 0.04\cdot 58390560 = 2335622 \text{ kJ/hour},$$

where $a_п$ – The coefficient, during cooling of the pistons with fresh water, lies in the range from 0.03 to 0.05; we assume $a_п$ = 0.04.

$$q_п = a_п \cdot q,$$

$$q_п = 0.05\cdot 58390560 = 2919528 \text{ kJ/hour},$$

where $a_п$ – the proportion of fuel heat removed by cooling water in air coolers.

We assume p_s = 0.3 Mpa and $a_н = 0.05$.

We verify the correctness of the calculation of the sea water pump delivery using an approximate relationship:

$$\frac{Q_{з.в}}{N_e} = (0,035...0,045),$$

$$\frac{Q_{з.в}}{N_e} = 245.3/8600 = 0.029.$$

That is, the value obtained satisfies the above dependence. The calculation of the seawater pump delivery is correct.

Nominal power of the sea water pump drive:

$$P = (245.3 \cdot 0.2)/(3.6 \cdot 0.9) = 15.1 \text{ kW}$$

where $p_{зв}$ – pup delivery pressure, from 0.2 to 0.4 MPa;

We assume $p_{зв}$ = 0.2 Mpa;

We assume the efficiency of the piston pump to be equal to $\eta = 0.9$.

Water cooler

The surface area of a water cooler is calculated using the formula:

$$A_{охл} = q_ц/(k_ц\ \Delta T_в),$$

$$A_{охл} = 6422961/(5000 \cdot 15) = 85.6 \text{ m}^2.$$

where $\Delta T_в$ – difference between the average temperature of fresh water and sea water in the cooler, K;

$k_ц$ – heat transfer coefficient, we assume 5000 kJ/(m^2·hour·K).

We choose a water cooler for the surface plane.

$$\Delta T_в = T_{пв} - T_{зв}, \text{ K}.$$

$T_{пв}$ – feed water temperature, accepted from 310 K to 350 K;

We assume $T_{пв} = 320$ K.

$T_{зв}$ – temperature of sea water, assumed $T_{зв} = 305$ K.

$$\Delta T_в = 320\text{-}305 = 15 \text{ K}.$$

2.4.4 Compressed air system

The system provides compressed air at specific pressure for starting and reversing the main engine KOBE DIESEL-MITSUBISHI 7UEC60LSE and starting auxiliary diesel engines. The system includes starting air compressors and starting air cylinders.

Amount of compessed air V_B for n_n engine starts:

$$V_B = n_n \cdot b_B \, \Sigma V_s,$$

$$V_B = 12 \cdot 5 \cdot 4.74 = 284.86 \text{ m}^3,$$

where n_n – minimum number of consecutive starts in forward and reverse that the system must provide; accepted for reversible motors $n_n \geq 12$.

b_B – specific consumption of free air per 1 м^3 cylinder’s volume; assumed for LSE from 4 to 6, we assume $b_B = 5$.

ΣV_s – working volume of the engine's starter cylinders.

The working volume of the engine cylinders is calculated using the following formula:

$$\Sigma V_s = (\pi \cdot d^2 \cdot s \cdot n_ц)/4,$$

$$\Sigma V_s = (3.14 \cdot 0.6^2 \cdot 2.4 \cdot 7)/4 = 4.74 \text{ m}^3,$$

where D – cylinder’s diameter, m;

S – piston stroke, m;

N – number of cylinders.

Total volume of air cylinders:

$$\Sigma V_б = V_B \cdot p_B/(p_{мах} - p_{мin})$$

$$\Sigma V_б = 284.86 \cdot 0.1013/(3.0-1.0) = 14.281 \text{ m}^3,$$

where $p_{мах}$ – maximum air pressure in the cylinder, from 2.5 MPa to 3.0 MPa; we assume $p_{мах}$ =3.0 MPa.

$p_{мin}$ – minimum air pressure at which the engine can be started, from 1.0 MPa to 1.5 MPa; we assume $p_{мin}$ = 1.0 MPa;

p_B – free air pressure; we assume p_B =0.1013 MPa.

At least two standard air cylinders must have a total volume close to $\Sigma V_б$, we assume $\Sigma V_б = 14.3$ m^3.

The selection of air cylinders for a diesel generator is done in a similar manner.

The total output of the compressors must ensure that the cylinders are filled in 1 hour, starting from atmospheric pressure to the pressure $p_{мах}$.

$$Q_к = \Sigma V_б (p_{мах} - p_{мin})/p_B,$$

$$Q_к = 14.3 \cdot (3.0-1.0)/0.1013 = 282.33 \text{ m}^3\text{/hour}.$$

If it is necessary to ensure that the vessel's horn operates for at least 6 minutes while maintaining the ability to perform 12 consecutive starts, the cylinder volumes are increased accordingly.

2.4.5 Gas release system

The system includes an exhaust pipe, muffler, spark arrestor, and waste heat boiler.

Amount of gases emitted from the engine:

$$Q_г=10\cdot N_e,$$

$$Q_г=10\cdot 13200 = 132000\ м^3/год,$$

where N_e – effective power of the main engine, kW.

Exhaust pipes are made separately for each engine. The diameter of the pipes is determined based on the speed of the exhaust gases.

For two-stroke internal combustion engines, the diameter of the pipes is equal to:

$$d_{газ}=12\cdot\sqrt{N_e}\ .$$

$$d_{газ}=12\cdot\sqrt{13200}=1380\text{ mm}.$$

2.5 Mechanisms and devices of general ship systems

2.5.1 Firefighting system MV HORIZON HIGHWAY

The total flow rate ΣQ of stationary fire pumps must be at least:

$$\Sigma Q = k\cdot m^2,$$

$$\Sigma Q = 0.008\cdot 220.43^2 = 388.71\text{ m}^3/\text{hour}.$$

$$m=1{,}68\cdot\sqrt{L_c\cdot\left(B_c+H_б\right)}+25,$$

$$m=1.68\cdot\sqrt{(199\cdot(38+30))}+25=220.43,$$

where L_c – ship length, m,

B_c – ship width, m,

$H_б$ – height of the side to the bulkhead deck at midship, m, k – coefficient, for MV HORIZON HIGHWAY, we take k = 0.008.

Nominal power of the fire pump drive:

$$P = (\Sigma Q \cdot p)/(3.6 \cdot \eta)/2,$$

$$P = (388.71 \cdot 1.0)/(3.6 \cdot 0.93)/2 = 53.89 \text{ kW}$$

where p – pump delivery pressure, accepted in accordance with the requirements of the Register, MPa;

η – Pump efficiency, we assume equal 0.93.

There are two stationary fire pumps. The pressure at the location of any hydrant is assumed to be p = 0.30 MPa.

Fire pumps can be sanitary, ballast, bilge, and seawater pumps, provided that their flow rate and pressure meet the design requirements. These pumps cannot be used to pump petroleum products, oil, or other flammable liquids.

2.5.2 Bilge system MV HORIZON HIGHWAY

Bilge system pumps are used to remove water from the engine room bilges, propeller shaft corridors, cargo hold bilges, and can also be used to pump ballast from the aft peak and forepeak.

Bilge pumps can serve as ballast pumps in cases specified by the Register's Rules.

Pipeline diameter d_{oc} is calculated using the formula of Register:

$$d_{oc} = 1{,}68 \cdot \sqrt{L_c \cdot \left(B_c + H_б\right)} + 25,$$

$$d_{oc} = 1.68 \cdot \sqrt{(199 \cdot (38+30))} + 25 = 220 \text{ mm},$$

where L_c – ship length, m,

B_c – ship width, m,

$H_б$ – height of the side to the bulkhead deck at midship, m,

The total flow rate of drainage pumps must be no less than the total flow rate of fire pumps.

The flow rate of drainage pumps is calculated based on the flow velocity of fluid in the pipeline. $v_{он} \geq 2$ *m/s*.

$$Q_{ос} = 3600 \cdot \frac{\pi \cdot d_{ос}^2}{4} \cdot v_{он}.$$

$$Q_{ос} = 3600 \cdot (3.14 \cdot 0.220^2)/4 \cdot 2 = 273.56 \text{ m}^3\text{/hour}.$$

Nominal power of the bilge pump drive:

$$P_{ос} = (Q_{ос} \cdot p_{он})/(3.6 \cdot \eta),$$

$$P_{ос} = (273.56 \cdot 0.3)/(3.6 \cdot 0.9) = 25.33 \text{ kW},$$

where $p_{он}$ – the pump delivery pressure is in the range from 0.2 MPa to 0.3 MPa; we assume $p_{он}$ = 0.3 MPa;

Pump efficiency $\eta = 0.9$.

The number of pumps shall be no less than two. A ballast pump or other pump with the required flow rate may be used as one of the independent pumps.

2.5.3 Ballast water system MV HORIZON HIGHWAY

The system is designed to fill and drain ballast tanks.

The ballast pump delivery must be such that all ballast tanks are drained within 4 to 10 hours, depending on the size of the vessel.

According to the Register Rules, the internal diameter of the ballast pipeline dбс is calculated using the formula:

$$d_{бс} = 18 \cdot \sqrt[3]{V_{бв}}.$$

$$d_{бс} = 18 \cdot 3\sqrt{2258} = 236 \text{ mm},$$

Where $V_{бв}$ - The volume of the largest ballast compartment is determined based on the data of the prototype vessel. $V_{бв}$ = 2258 m^3.

The flow rate of ballast pumps for a given pipe diameter is determined by the speed of water flow in the pipeline. $v_{бс} \geq 2$ *m/s*.

$$Q_{бн} = 3600 \cdot \frac{\pi \cdot d_{бс}^2}{4} \cdot v_{бс}$$

$$Q_{бн} = 3600 \cdot (3.14 \cdot 0.236^2)/4 \cdot 2.5 = 394.0 \text{ m}^3\text{/hour}.$$

Where $v_{бс}$ – working flow speed.

For ballast water system $v_{бс}$ = 2.5 м/с.

Nominal power of ballast pump drive :

$$P_{бн}= (Q_{бн} \cdot p_{бн})/(3.6 \cdot \eta),$$

$$P_{бн} = (394.0 \cdot 0.3)/(3.6 \cdot 0.78)= 43.0 \text{ kW}.$$

where $p_{бн}$ – pump delivery pressure, from 0.2 MPa to 0.3 MPa; we assume $p_{бн}$ = 0.3 MPa.

Pump efficiency equals to $\eta = 0.78$.

Reserve cooling pumps, fire pumps, and drainage pumps can be used as ballast pumps.

2.6 Ventilation system MV HORIZON HIGHWAY

The supply of air blowers that provide ventilation of living quarters is calculated based on the condition of supplying each crew member or passenger with 33 m^3/hour to 50 m^3/hour of air.

The ventilation system for the engine room is designed to ensure the operation of the main engines in storm conditions with the engine room closed.

According to the prototype, the ship has 21 crew berths and 2 spare berths. Thus, to ensure full human occupancy of the ship with a total of 23 crew berths, the total air supply is equal to:

$$Q_e = 40 \cdot 23 = 920.0 \text{ m}^3\text{/hour}.$$

Air supply required for the operation of machines and mechanisms in the engine room:

$$Q_{мо}=\alpha_g \cdot G_0 \cdot Q_e/\rho_B,$$

$$Q_{мо}=2.5 \cdot 14 \cdot 920.0 /1.29 = 24961 \text{ m}^3\text{/hour}.$$

Where α_g – air residue coefficient during fuel combustion in the engine. For the KOBE DIESEL-MITSUBISHI 7UEC60LSE engine, we assume α_g = 2.5,

G_0 – mass air flow per 1 kg of fuel,

ρ_B – air density at barometric pressure.

2.7 Sanitary systems

2.7.1 Drinking water system MV HORIZON HIGHWAY

Drinking water is stored in reserve tanks located outside the double bottom. Water is pumped from the reserve tanks to the hydrophores. The volume of the pressure tank is assumed to be 0.2 times the daily consumption.

The drinking water pump supply is determined based on the calculation of consumption per person per day from 10 l to 40 l of drinking water.

Thus, based on the maximum human capacity of the vessel, the daily consumption from the drinking water tank is equal to:

$$Q_{цпв} = (23 \cdot 40) \cdot 0.2 = 1.84\ m^3.$$

Nominal power of the drinking water system pump drive:

$$P_{пв} = (Q_{цпв} \cdot p_{пв})/(3.6 \cdot \eta),$$

$$P_{пв} = (1.84 \cdot 0.7)/(3.6 \cdot 0.75) = 0.48\ kW.$$

Where $p_{пв}$ – pump delivery pressure, from 0.5 MPa to 0.7 MPa; we assume $p_{пв}$ = 0.7 MPa

Pump efficiency equals to η = 0.75.

2.7.2 MV HORIZON HIGHWAY water system for sanitary and domestic needs

Water consumption for sanitary and domestic needs averages between 100 and 200 liters per person per day. Determining the water supply for sanitary and domestic needs is similar to the previous calculation.

$$V_{цпв} = Q_{цпв} = (200 \cdot 23) \cdot 0.2 = 9.20\ m^3.$$

Nominal power of the pump drive:

$$P_{спп} = (9.20 \cdot 0.7)/(3.6 \cdot 0.7) = 2.56\ kW,$$

where $p_{спп}$ – pump delivery pressure, from 0.5 MPa to 0.7 MPa; we assume $p_{пв}$ = 0.7

MPa;

Pump efficiency equals to $\eta = 0.7$.

2.7.3 Seawater system for sanitary and technical needs MV HORIZON HIGHWAY

To calculate the pump flow rate, the water consumption for sanitary and technical needs is taken as 20–30 liters per person per day. The pump flow rate is determined in the same way as in the previous calculation.

$$V_{цзв} = Q_{цзв} = (30 \cdot 23) \cdot 0.65 = 4.49\ m^3.$$

Nominal power of the pump drive:

$$P_{стп}=(4.49 \cdot 0.7)/(3.6 \cdot 0.74) = 11.78\ kW.$$

where $p_{стп}$ pump delivery pressure, from 0.5 MPa to 0.7 MPa; we assume $p_{пв}$ = 0.7 MPa;

Pump efficiency equals to $\eta = 0.74$.

3. ANALYSIS OF THE EFFICIENCY OF REFRIGERATION MACHINE FOR SHIP AIR CONDITIONING SYSTEM OPERATING ALTERNATIVE REFRIGERANTS

3.1 Diagram and operating principle of refrigeration machine for ship air conditioning system

A typical marine air conditioner manufactured by HI AIR KOREA Co., Ltd., which uses R404A as a refrigerant, was selected as the object of analysis (baseline option). The global warming potential (GWP) for R404A is 3920, which is sufficiently high, so it will need to be replaced with an alternative refrigerant in the future. The ship air conditioner diagram and its specifications are shown in Figures 3.1 and 3.2.

Main characteristics of the air conditioning system:

- Cooling system - direct evaporation of R404A refrigerant in air coolers (without intermediate coolant);
- Condenser cooling with fresh water at a temperature not exceeding +60°C;
- Electrical sources: Main circuit 3 x 440 Volt 60 Hz, control circuit 1 x 220 Volt 60 Hz

System operation

Compressor capacity is regulated by suction pressure. An OT-1 pressure sensor in the suction line controls the compressor cylinders' start-up and shutdown using three-way solenoid valves. A horizontal shell-and-tube condenser condenses the compressed refrigerant using water. The suction line for the refrigerant vapor into the compressor is located in the lower part of the condenser below the pipes. This design protects the compressor from refrigerant vapor containing moisture droplets and at the same time reduces the need for external liquid supercooling. The space between the suction gas heat exchanger tubes and the condenser tubes acts as a receiver.

Two solenoid valves are located in the liquid line before the air cooler, and a temperature sensor is located on the fresh air supply. When the cooled air temperature is reached, the sensor will signal the solenoid valve to shut off the liquid refrigerant supply through the throttle valve to the air cooler, and then the low pressure relay will stop the compressor.

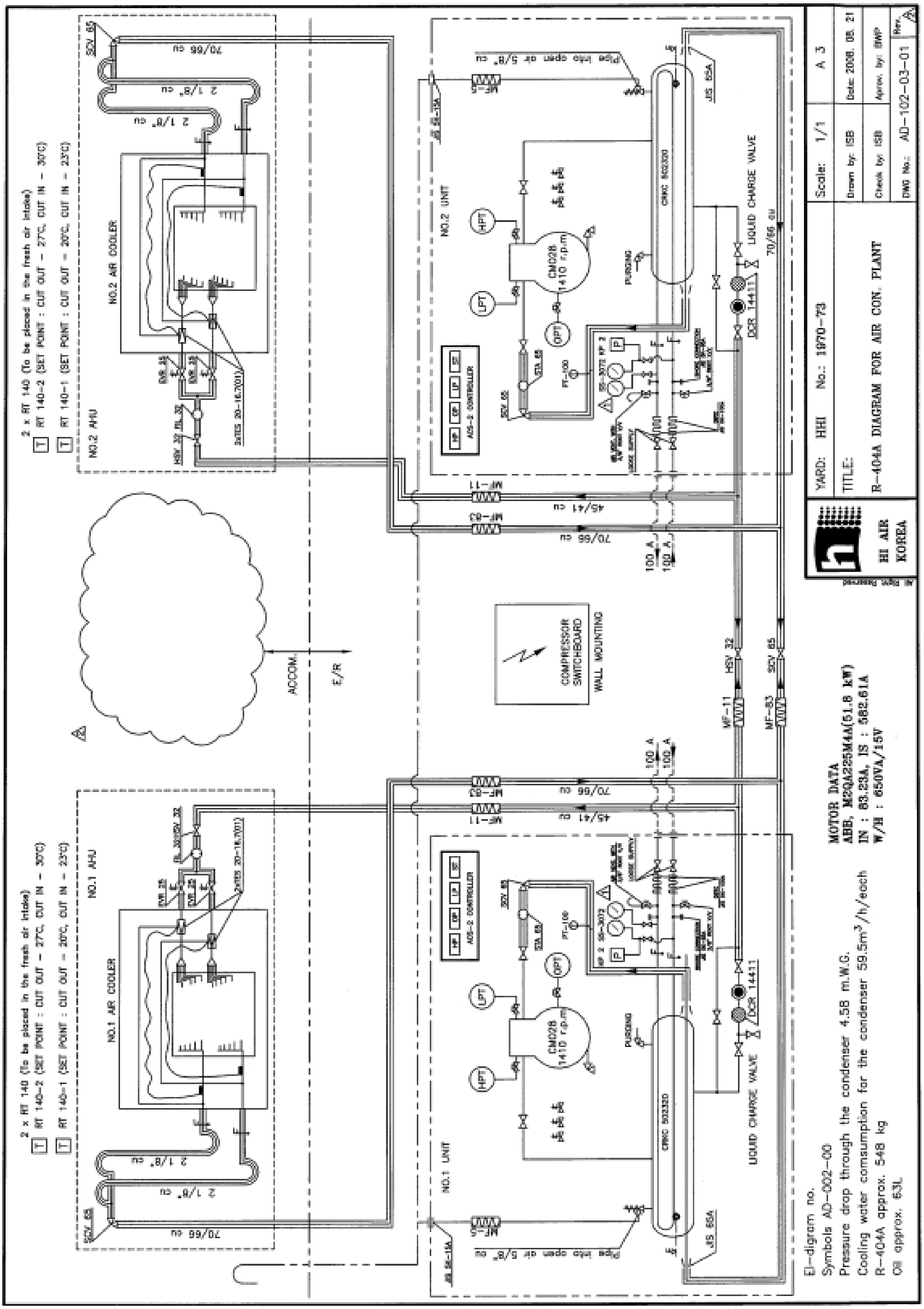


Figure 3.1 - Diagram of a marine air conditioner refrigeration system

	Запірний клапан		KP 98 High temp. control oil & disch.		Масляний сепаратор
	Запірний клапан		KP 1 low pressure control		Масловіддільник з поплавком
	Запобіжний клапан		KP 2 low cooling water control		Компресор
	Ручний регулюючий клапан		RT термостат		Конденсатор
	Незворотний клапан		Термометр		Ресівер
	Соленоїдний клапан		Скло рівня рідини		Повітряохолоджувач
	Клапан постійного тиску		Оглядове скло		Вентилятор
	Терморегулюючий вентиль		Фільтр-осушувач		Відцентровий насос
	Автоматичний водяний клапан		Фільтр		Ізольована труба
	MP 55 oil pressure control		Манометр		3-ходовий випробувальний клапан
	KP 15 high & low pressure control		Розподільник рідини		Диференційний манометр
	KP 5 high pressure control		Сепаратор рідини		Гнучкі труби

Figure 3.2 - General specifications for ship air conditioning

Air conditioning unit specifications

Single-stage compressor operating on R404A refrigerant, with the following specifications:

- Model – SM028
- Number of cylinders – 8
- Piston diameter – 70 mm
- Piston stroke – 70 mm
- Rotation speed – 1410 rpm
- Refrigerant condensation temperature – 42.0 °C
- Refrigerant evaporation temperature – 7.0 °C
- Cooling capacity – 174.1 kW.

Based on the geometric characteristics of the compressor and the rotation speed, the volume described by the compressor pistons (the volumetric capacity of the compressor) was determined. V_h = 0.0507 m^3/s;

All the information provided was used in the further calculation of the refrigeration cycle and determination of the air conditioner's efficiency parameters.

The following operating parameters of the air conditioning system's refrigeration machine were adopted for the energy analysis (according to the manual):

- boiling temperature of the refrigerant in the evaporator t_0= +7 °C;
- superheating of vapors after the evaporator by 10 °C due to regenerative heat exchange;
- condensation temperature t_k=42.0 °C;
- supercooling after the condenser due to regenerative heat exchange, where Δt is determined from the heat balance equation.

3.2 Methodology for calculating the energy characteristics of a refrigeration machine

A schematic diagram of the cycle of a single-stage vapor compression refrigeration machine, which is used in ship air conditioning refrigeration units, is shown in Fig. 3.3.

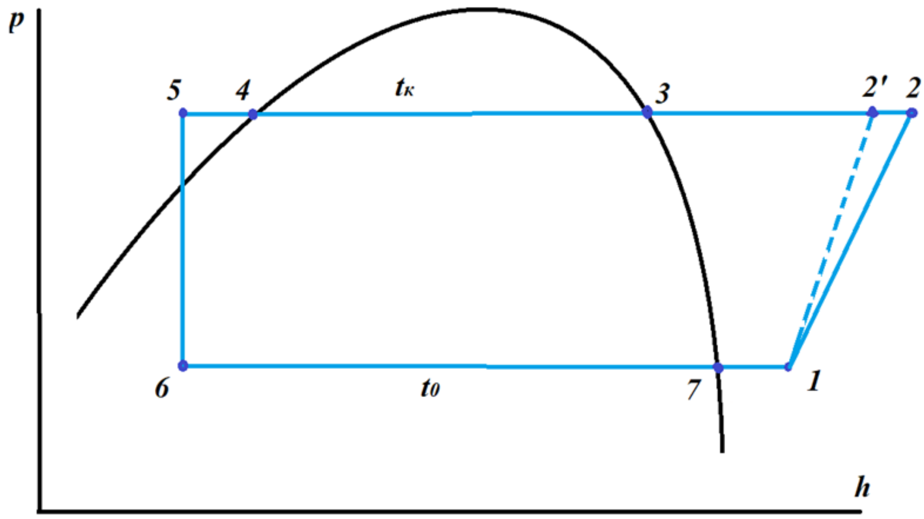


Fig. 3.3 - Cycle of a single-stage vapor compression refrigeration machine in a p-h diagram: process **1-2'** – adiabatic (ideal) compression of refrigerant vapor in the compressor (s=const); process **1-2** – actual compression of refrigerant in the compressor; process **2-3** – cooling of refrigerant vapor in the condenser (p=const); process **3-4** – condensation of refrigerant vapor in the condenser (p=const); process **4-5** – supercooling of

liquid refrigerant in the condenser or in the regenerative heat exchanger (p=const); process **5-6** - expansion of the refrigerant (h=const); process **6-7** - evaporation of the refrigerant in the evaporators (p=const); process **7-1** - superheating of the refrigerant vapors in the regenerative heat exchanger (p=const)

The enthalpy at point 5 is determined as follows:

$h_5=h_4-(h_1-h_7)$.

After plotting the cycle in the p-h diagram of the refrigerant, we find the enthalpy at the cycle nodes using the diagram or the RefProp program [16].

The energy characteristics of the refrigeration machine of the vessel air conditioning system were calculated using the standard method described in [17], taking into account the internal irreversibility of the refrigerant compression process in the compressor, the mechanical and electrical efficiency of the compressor, and the compressor's delivery ratio.

Specific cooling capacity (kJ/kg):

$$q_0=(h_7-h_6). \tag{3.1}$$

Specific adiabatic compression work (kJ/kg):

$$l_a=(h_{2'}-h_1). \tag{3.3}$$

Theoretical refrigeration coefficient:

$$\varepsilon_{TEOP}=q_0/l_a. \tag{3.4}$$

The compressor delivery coefficient (taking into account the reduction in actual compressor delivery compared to theoretical delivery) can be approximately calculated using the following formula:

$$\lambda=\lambda_i\cdot\lambda_w. \tag{3.5}$$

Heating coefficient

$$\lambda_w=T_0/T_\kappa, \tag{3.6}$$

where T_0 and T_k are the boiling and condensation temperatures, K.

Coefficient of volumetric losses in the compressor:

$$\lambda_i=\frac{(p_0-\Delta p_{BC})-c(p_K+\Delta p_H-p_0+\Delta p_{BC})}{p_0}, \tag{3.7}$$

where Δp_{BC} - is the pressure drop at the compressor inlet, which can be taken as $\Delta p_{BC}=0.005$ *MPa*; Δp_H - pressure drop at the discharge, can be taken as $\Delta p_H=0.01$ *MPa*; c - relative dead space in the compressor, can be taken as $c=0.03$.

The compressor's indicator efficiency, which takes into account the difference between the actual working process and the theoretical (isentropic) one – the deviation of processes 1-2 and 1-2' in Fig. 3.3, approximately for fluorocarbon refrigerants (freons), can be calculated using the empirical formula:

$$\eta_i=\lambda_w+0.0025\cdot t_0, \tag{3.8}$$

The total efficiency of the compressor:

$$\eta=\eta_i\cdot \eta_{мех}\cdot \eta_п\cdot \eta_д, \tag{3.9}$$

where η_i - compressor's indicator efficiency; $\eta_{мех}$ - compressor's mechanical efficiency, which takes into account losses caused by friction; $\eta_п$ - is the transmission efficiency; $\eta_д$ - is the efficiency of the compressor motor. For approximate calculations, the following values can be assumed: $\eta_{мех} = 0.8 – 0.9;\ \eta_п = 0.95;\ \eta_д = 0.95$.

Mass flow rate of refrigerant in a refrigeration machine (kg/s):

$$G = \lambda \cdot V_h \cdot \rho_1, \tag{3.10}$$

where λ - is the compressor delivery coefficient; V_h - is the volume characterized by the compressor pistons (compressor volumetric capacity), which for the air conditioning system analyzed in the work is equal to 0.0507 m^3/s; ρ_1 - is the density of refrigerant vapors at the compressor inlet (at point 1 of the cycle).

Cooling capacity (kW):

$$Q_0=q_0\, G. \tag{3.11}$$

Actual power consumed by the refrigeration machine compressor (kW):

$$N = \frac{l_a\cdot G}{\eta}, \tag{3.12}$$

The temperature of the refrigerant at the compressor outlet is determined taking into account the indicator efficiency:

$$\eta_i = \frac{l_a}{l_д} = \frac{h_{2\prime}-h_1}{h_2-h_1}. \tag{3.13}$$

The expression above is used to determine h2, and t2 is determined from the diagram based on the found values of enthalpy and condensation pressure.

Actual refrigeration coefficient

$$\varepsilon_Д = \frac{Q_0}{N}. \tag{3.14}$$

3.3 Determination of energy characteristics of air conditioning systems operating on different refrigerants

A diagram of refrigerant R404A showing the ideal (adiabatic compression in the compressor) cycle of a marine air conditioner is shown in Fig. 3.4. Based on the parameters taken from this diagram and the methodology described above, it is possible to determine the actual refrigeration coefficient when operating an air conditioner refrigeration machine on R404A. Similarly, it is possible to evaluate the efficiency of this refrigeration machine when operating on different refrigerants.

To simplify the analysis, we will use CoolTools v1.1.1 software, which is available for free and can be downloaded from the official website. https://www.ipu.dk/products/cooltools/ [18].

Below is an example of a calculation using CoolTools v1.1.1 for refrigerant R404A (as a base for comparison).

After launching the program, the following image will appear on the computer screen:

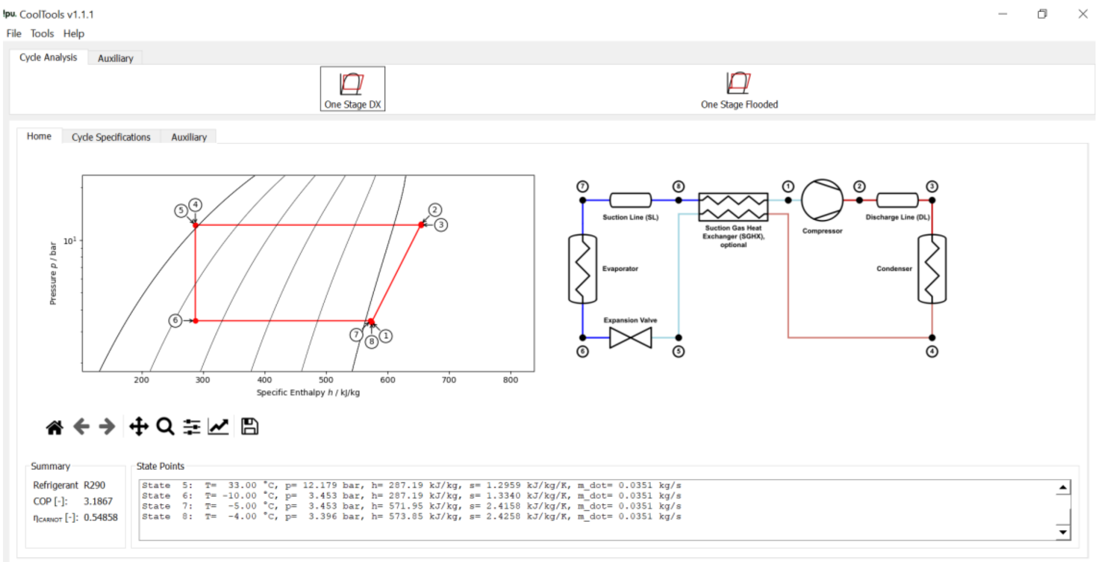


Fig. 3.5 Screenshot of the software CoolTools v1.1.1 interface

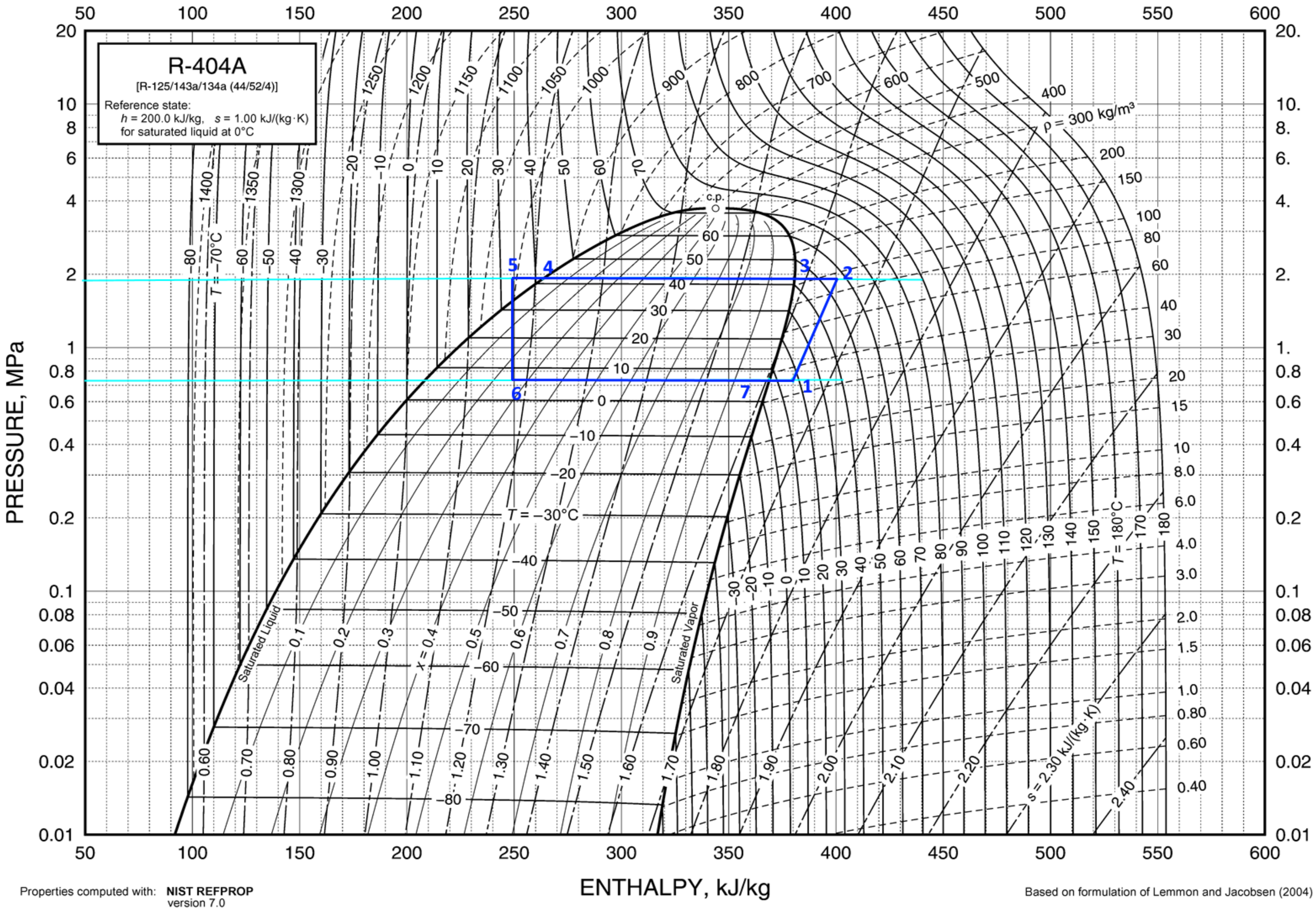
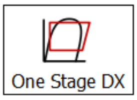
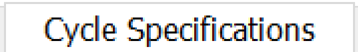


Figure 3.4 - Diagram of refrigerant R404A showing the ideal cycle of a refrigeration machine

Next, select 

 and 

. In the window, enter the following data (according to the operating parameters of the air conditioner refrigeration machine): refrigerant boiling temperature t_0= +7 °C; condensation temperature t_c=42.0 °C; vapor superheat after the evaporator 10 °C due to regenerative heat exchange, however, this software does not specify the superheat temperature, but rather the efficiency of regenerative heat exchange, which we accept according to the recommendations of CoolTools v1.1.1 as equal to 0.3 (which will approximately correspond to a superheat degree of 10 K); compressor volumetric capacity V_h = 0.0507 m3/s = 182.5 m3/h. All these input data are marked in orange in Fig. 3.6.

At the same time, select additional input data (marked in blue in Fig. 3.6).

ΔT_{SH} - refrigerant superheat in the evaporator, with a regenerative heat exchanger ΔT_{SH} = 0 K (superheat occurs in the regenerative heat exchanger, not in the evaporator).

ΔT_{CC} - refrigerant supercooling in the condenser, with a regenerative heat exchanger ΔT_{CC} = 0 K (supercooling occurs in the regenerative heat exchanger, not in the condenser).

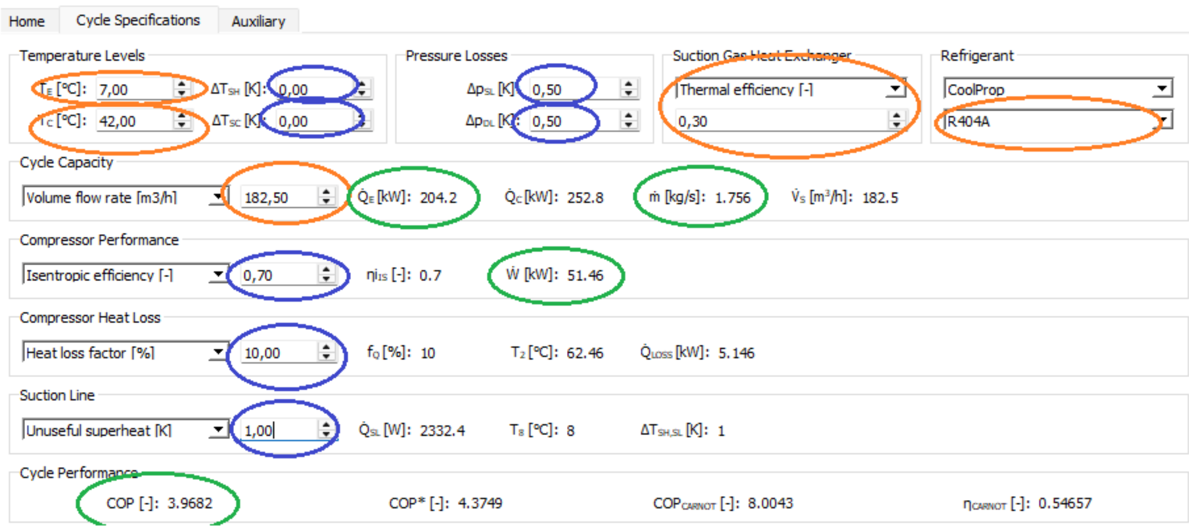


Fig. 3.6 Screenshot of the software CoolTools v1.1.1 interface at input of initial data

Isentropic efficiency – isentropic efficiency of the refrigerant compression process in the compressor, determines internal energy losses during refrigerant compression in real conditions; The calculation is performed for a value of 0.7 (approximate value), since it depends little on the type of refrigerant and depends on the type of compressor and the compression ratio.

Heat loss factor – the proportion of heat loss in the compressor; assumed to be 10%.

Unusual superheat – overheating in the suction line before the compressor, i.e., cold loss on suction without cold production. Here we take ΔTSH,SL=1 K as the recommended CoolTools v1.1.1 value.

Δp_{SL} and Δp_{DL} – pressure losses on the suction line and discharge line of the compressor. We assume these pressure losses to be equal to the drop in refrigerant temperature in the saturated state by 0.5 K.

The values obtained from the calculation results in Fig. 3.6 are marked in green. We write them down:

- cooling capacity Qe=204.2 kW;
- mass flow rate of refrigerant m=1.756 kg/s;
- power consumption of the compressor W=51.46 kW;
- coefficient of performance COP=3.9682.

Additionally, you can switch to Home (upper right corner of the program window) and view all characteristic points of the cycle and state parameters at these points, based on which the COP refrigeration coefficient was calculated – Fig. 3.7.

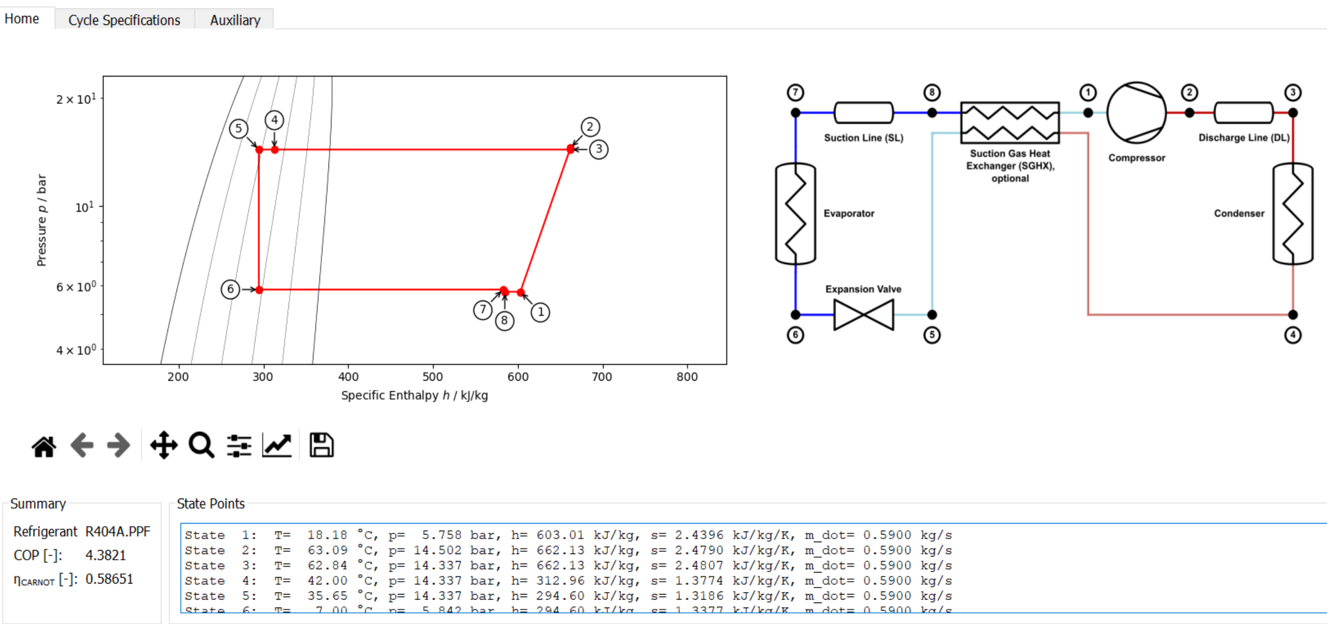


Fig. 3.7 Screenshot of the software CoolTools v1.1.1 interface with refrigeration cycle parameters

Here we can obtain the refrigerant state parameters at the cycle nodes in the following sequence: temperature, pressure, enthalpy, entropy, density:

State 1: T= 18.36 °C, p= 7.346 bar, h= 381.11 kJ/kg, s= 1.6473 kJ/kg/K, m_dot= 1.7562 kg/s
State 2: T= 62.46 °C, p= 19.420 bar, h= 407.48 kJ/kg, s= 1.6649 kJ/kg/K, m_dot= 1.7562 kg/s
State 3: T= 62.13 °C, p= 19.190 bar, h= 407.48 kJ/kg, s= 1.6656 kJ/kg/K, m_dot= 1.7562 kg/s
State 4: T= 42.00 °C, p= 19.190 bar, h= 263.51 kJ/kg, s= 1.2116 kJ/kg/K, m_dot= 1.7562 kg/s
State 5: T= 35.75 °C, p= 19.190 bar, h= 252.93 kJ/kg, s= 1.1776 kJ/kg/K, m_dot= 1.7562 kg/s
State 6: T= 6.65 °C, p= 7.458 bar, h= 252.93 kJ/kg, s= 1.1891 kJ/kg/K, m_dot= 1.7562 kg/s
State 7: T= 7.00 °C, p= 7.458 bar, h= 369.19 kJ/kg, s= 1.6045 kJ/kg/K, m_dot= 1.7562 kg/s
State 8: T= 8.00 °C, p= 7.346 bar, h= 370.52 kJ/kg, s= 1.6103 kJ/kg/K, m_dot= 1.7562 kg/s

3.4 High-potential refrigerants

R404A is a hydrofluorocarbon (HFC) blend with a very high GWP (approx. 3922). In this work, it serves as the baseline refrigerant for low-temperature provision chambers. (And also could used in AC systems for utilization of only one refrigerant onboard). For decades, R404A was the industry standard for freezer applications. Calculations confirm it is energetically inefficient: it shows the lowest COP (3.968) (excluding R407C) and requires the highest compressor power (42.66 kW) among the main HFC/HFO analogs. Its use in new equipment and for service is banned in the EU by the F-Gas Regulation.

R134a is a pure HFC refrigerant with a medium GWP (approx. 1430). In this work, it serves as the baseline refrigerant for air conditioning (AC) systems.

This was the standard for automotive and marine AC systems. Calculations show it has very good efficiency, demonstrating a high COP (4.462) and significantly lower power consumption (37.94 kW) than R404A. Despite its good performance, it is also being phased down under the Kigali Amendment and F-Gas Regulation due to its GWP.

R1234yf is a hydrofluoroolefin (HFO) and a "new generation" refrigerant with an ultra-low GWP (<10). It was designed as the primary replacement for R134a in car air conditioning. Calculations show it has a good COP (4.316). Its main drawback for marine applications is its A2L safety class (lower flammability), which requires additional safety measures onboard and complicates system design compared to non-flammable A1 refrigerants. It is also a key component in modern blends like R449A.

R407C is a zeotropic HFC blend (R32/R125/R134a) that was one of the first "transitional" refrigerants to replace R22 in air conditioning systems. This refrigerant has a GWP (approx. 1774) that is now considered high. Calculations clearly demonstrate its main flaw: it has the lowest COP (3.663) and the highest power consumption (46.22 kW) of the entire group. This makes it a highly undesirable option from an energy efficiency standpoint (and consequently, for the ship's CII rating).

R290 (Propane) is a natural refrigerant (Hydrocarbon, HC) with excellent thermodynamic properties and a negligible GWP (approx. 3). The calculations show its high efficiency (COP 4.382) and very low mass flow rate (0.588 kg/s). Its use on ships is extremely limited due to its A3 safety class (highly flammable). SOLAS and classification society rules place strict restrictions on using highly flammable substances in accommodation and machinery spaces, making its practical use in provision plants or AC very difficult.

R1233zd(E) is an HFO refrigerant with an ultra-low GWP designed for low-pressure applications. This refrigerant is primarily used in large centrifugal chillers (as a replacement for R123). The calculations show its unique feature: the lowest compressor power consumption (35.58 kW) of the entire group. It is also one of the few "new generation" refrigerants with an A1 safety class (non-flammable). This makes it very promising, but mainly for large chiller applications, not the direct expansion (DX) systems we are analyzing.

R1234ze(E) is another HFO with an ultra-low GWP (<10), similar to R1234yf. The calculations show this to be the most energy-efficient refrigerant in this specific analysis, with the highest COP (4.487) and very low power consumption (37.73 kW). Like R1234yf, its main barrier to widespread adoption on ships is its A2L safety class (lower flammability).

R32 is a pure HFC refrigerant with a medium GWP (approx. 675). R32 is itself an A2L refrigerant and is popular in residential air conditioning. In the context of our work, it is most important as a key component in A1 non-flammable blends like R448A and R449A. Manufacturers use R32 in these blends to boost their performance and efficiency, balancing its flammability with other non-flammable components

3.5 Energy efficiency parameters of ship air conditioners operating on different refrigerants

Using these parameters and the methodology described above, it is possible to calculate the refrigeration coefficient of the cycle under consideration. We perform the calculation in the same way for all refrigerants selected for analysis and enter the data obtained in Table 3.1.

Table 3.1 - Results of calculating the energy characteristics of ship air conditioners operating on different refrigerants

Cooling agent	Cooling capacity Q_e, kW	Mass flow rate of refrigerant m, kg/s	Compressor power consumption W, kW	Cooling coefficient COP
R404A	204.2	1.756	51.46	3.968
R134a	204.2	1.338	45.76	4.462
R1234yf	204.2	1.706	47.31	4.316
R407C	204.2	1.283	55.75	3.663
R290	204.2	0.71	46.6	4.382
R1233zd(E)	204.2	1.233	42.91	4.758
R1234ze(E)	204.2	1.455	45.51	4.487
R32	204.2	0.818	48.9	4.176

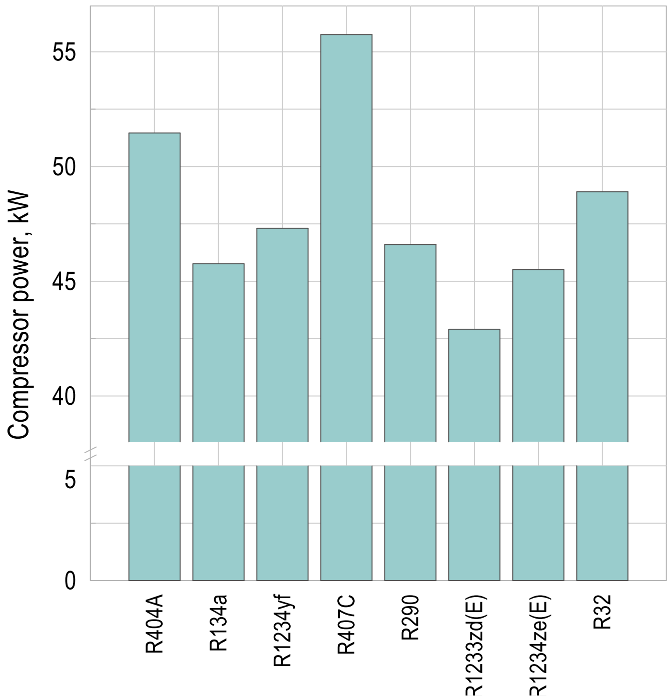


Fig. 3.7 Compressor power consumption when the air conditioner is operating on different refrigerants

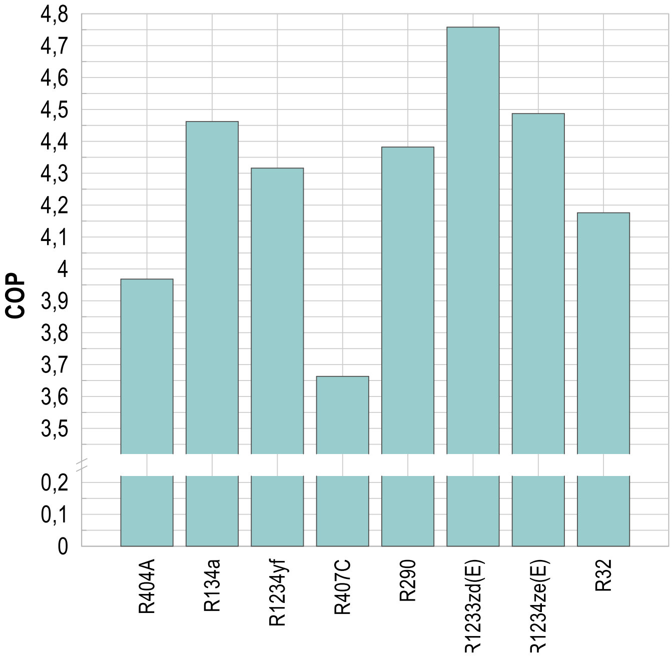


Fig. 3.8 COP refrigeration coefficient values when the air conditioner is operating on different refrigerants

An analysis of the coefficient of performance (COP) values for ship air conditioners operating on different refrigerants shows that R1233zd(E) provides the highest energy efficiency, confirming its suitability for use in modern ship air conditioning systems. R134a, R1234ze(E), and R290 also have fairly high COP values, indicating their good energy performance. The lowest COP is observed for R407C, making it the least efficient of the refrigerants considered. Refrigerants R404A and R32 occupy an intermediate position in terms of energy efficiency. Thus, from the point of view of energy consumption, it is advisable to orient new marine air conditioners towards the use of R1233zd(E), R1234ze(E) or R290, taking into account safety requirements and regulatory restrictions.

3.6 Environmental impact of ship air conditioners operating on different refrigerants

The purpose of this stage of the master's research was to conduct an ecological and energy analysis [19] of air conditioner operation on board ships using different refrigerants. The analysis consisted of calculating the total impact of refrigeration equipment operation on global warming over a certain period of its operation – TEWI (Total Equivalent Warming Impact)

When calculating TEWI for the analysis of ship air conditioners, the following contributions should be taken into account:

– indirect CO2 emissions caused by electricity consumption during the operation of ship air conditioners;

– direct GHG emissions caused by refrigerant leaks.

$$TEWI = N_{el} \cdot g_F \cdot \mathrm{C}F \cdot \tau_h + m_R \cdot L_{annual} \cdot GWP \cdot \tau_{an} \tag{3.15}$$

where N_{el} - power consumed by the air conditioner compressor, kW; g_F – specific fuel consumption of diesel generator, kg $(kW \cdot h)^{-1}$; CF – fuel mass conversion factors CO_2, (t of CO_2)·(t of fuel^{-1}); τ_h – calculation period in hours (for calculation purposes, we assume 1 year = 8760 hours); m_R – mass of refrigerant charged into the refrigeration system, kg; L_{annual} – annual refrigerant leaks during the operation of refrigeration equipment as a percentage of m_R; GWP – global warming potential of refrigerants, (kg CO_2)/kg; τ_{an}– calculation period in years (we assume 1 year for the calculation)

The specific fuel consumption of a diesel generator depends on many factors (generator power, load), but for comparative analysis, we can assume g_F=0.23 kg·(kW·hour)$^{-1}$ as an average value for generators with a capacity of 500 to 3000 kW [20].

The CF value for heavy fuel is 3.114 (t CO2)·(t fuel)$^{-1}$, for diesel fuel – 3.206 (t CO2)·(t fuel)$^{-1}$ [21, 22].

The proportion of refrigerant leaks from ship refrigeration equipment during operation is the highest of all types of refrigeration systems, accounting for 20 to 40% of the annual charge. [23, 25]. In [4] high variability in individual refrigerant losses for each vessel analyzed was noted – from 1% to 62%.

The global warming potential of refrigerants (GWP) can be found in many sources, for example [17].

The analysis should be performed in specific units (the calculations show that when switching to a new refrigerant, the cooling capacity changes). The specific CO2 emissions for a given period of operation of a ship refrigeration system, relative to 1 kWh of cooling produced on board the ship, can be used to compare ship refrigeration systems and air conditioning systems with different cooling capacities:

$$\mathrm{em}_{refr} = \frac{\mathrm{TEWI}}{Q_0 \cdot \tau}, \text{ (kg CO}_2\text{)/kJ}, \qquad (3.15)$$

where Q_0 – cooling capacity of the system, kW; τ – equipment operating time in seconds (assumed to be 1 year = $31.536 \cdot 10^6$ s).

Below are the results of a comparison of specific CO2 emissions per 1 kWh of artificial cooling. em_{refr} during operation of the ship's air conditioning system, which operates on selected refrigerants.

Some input data for the environmental and energy analysis are given in Table 3.1. The electrical power for the air conditioner's refrigeration machine when operating on the specified refrigerants at t_c=42 °C and t_0=7 °C accepted based on the results of preliminary calculations. The same table shows the results of TEWI calculations.

Table 3.2 – Results of the ecological and energy analysis of the refrigeration machine of a ship air conditioner

Parameter	R404A	R134a	R1234yf	R407C	R290	R1233zd(E)	R1234ze(E)	R32
Cooling capacity Q_0, kW	204.2	204.2	204.2	204.2	204.2	204.2	204.2	204.2
Compessor power consumed N_{el}, kW	51.46	45.76	47.31	55.75	46.6	42.91	45.51	48.9
Refrigerant charge mass m_R	30 kg							
GWP of refrigerant (kg CO_2)/kg	3940	1300	4	1774	3	1	1-7	675
CF for heavy fuel	3.206							
g_F, kg·(kW·hour)$^{-1}$	0.23							
L_{annual}	0.3							
TEWI, kg CO_2	$358 \cdot 10^3$	$298 \cdot 10^3$	$296 \cdot 10^3$	$365 \cdot 10^3$	$292 \cdot 10^3$	$286 \cdot 10^3$	$286 \cdot 10^3$	$313 \cdot 10^3$
em$_{refr}$, (kg CO_2)/kJ of cold	$5.55 \cdot 10^{-5}$	$4.62 \cdot 10^{-5}$	$4.59 \cdot 10^{-5}$	$5.67 \cdot 10^{-5}$	$4.53 \cdot 10^{-5}$	$4.44 \cdot 10^{-5}$	$4.44 \cdot 10^{-5}$	$4.86 \cdot 10^{-5}$

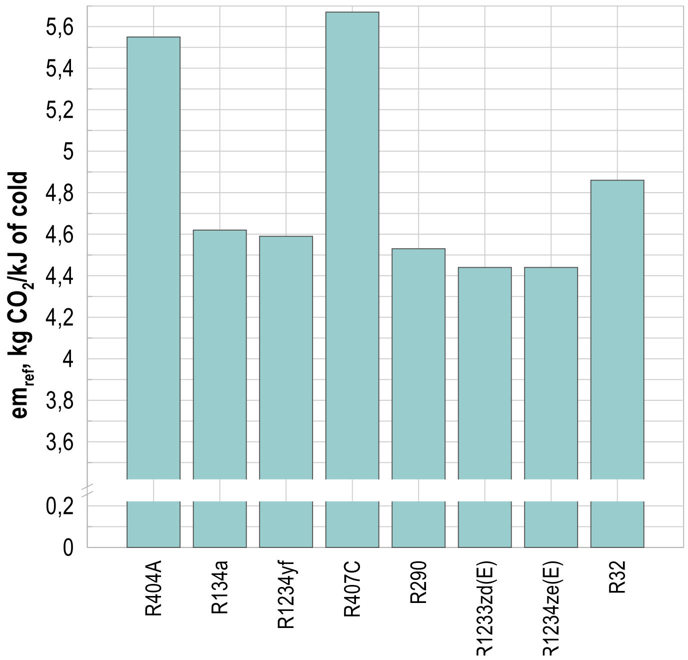


Fig. 3.9 Comparison of the environmental performance of a marine air conditioner refrigeration machine when operating on different refrigerants.

Fig. 3.9 shows the results of a comparative assessment of specific CO_2 emissions per 1 kJ of cooling produced for a marine air conditioner using different refrigerants.

Analysis of the data shows that the highest specific CO_2 emissions are characteristic of refrigerants R407C and R404A, which is due to both the lower energy performance of the refrigeration machine and the high global warming potential (GWP) of these substances.

The lowest specific CO_2 emissions are observed for R1233zd(E), R1234ze(E), and R290, confirming their high environmental benefits and compliance with current IMO requirements and regulations on limiting the use of refrigerants with high GWP.

The refrigerants R134a, R1234yf, and R32 occupy an intermediate position in terms of environmental impact, combining acceptable energy performance with moderate CO_2 equivalent emissions.

4 ANALYSIS OF THE EFFICIENCY OF THE SHIP REFRIGERATION PLANT OF PROVISION CHAMBERS OPERATING ALTERNATIVE REFRIGERANTS

4.1 Diagram and operating principle of ship refrigeration plant for provision chambers

The subject of analysis is a marine refrigeration unit designed to serve storage chambers, the diagram and specifications of which are shown in the figure. This unit is a compact condenser unit (6) assembled on a single frame, specially designed for marine operation. The system operates on the principle of direct expansion (DX) and is designed to work with the high-potential (in the context of GWP) hydrofluorocarbon (HFC) refrigerant R-404A.

A key feature of this design, dictated by the reliability requirements of marine registries, is 100% redundancy of the main components. As can be seen from the P&ID schematic and confirmed in the technical description, the unit consists of two completely independent and parallel refrigeration circuits, each of which includes a compressor and a 100% capacity condenser. In normal operation (temperature maintenance mode), only one circuit (lead) is active, and its daily operation, according to the specification, should not exceed 16 hours. The second circuit is in hot standby mode (lag). During the initial cooling of loaded chambers (pull-down mode) or under extreme thermal loads, both circuits can be activated simultaneously to achieve the set temperature within 24 hours.

The system uses “open” type compressors, which is visually confirmed in the diagram by the presence of a separate Compressor (1) and Electric Motor (2) connected by a drive shaft.

The refrigerant flow in each circuit is as follows: low-temperature, low-pressure gaseous R-404A is sucked in and compressed in the compressor (1). The hot, high-pressure gas enters the oil separator (4), which returns the compressor oil to the crankcase. The gas then enters the condenser (3), which, as described, is cooled by fresh water. This is a standard solution for marine vessels, where a central low-temperature fresh water circuit (LT FW) is used, which in turn is cooled by seawater through the main heat exchanger. In the condenser, the refrigerant releases heat and condenses, turning into a liquid phase. The high-pressure liquid refrigerant passes through the filter-dryer (5), after which it enters the common liquid line and is distributed to consumers.

The storage chambers are cooled by Fan Coil Units (8, 9, 10), which are evaporators with forced air circulation. The refrigerant supply to each evaporator is individually regulated by evaporator control valves (15), which include a thermostatic valve (TRV/TEV) and a solenoid valve. The system is fully automated: a digital temperature controller (11) in each chamber monitors the temperature and sends a signal to open or close the solenoid valve, thus maintaining the set temperature.

An automatic electric defrosting system is provided for low-temperature chambers (meat, fish). The diagram on the evaporator (8) clearly shows the symbol for the electric heating element. The technical description adds that heating elements are also installed in the drain pan and drain pipe to prevent melt water from freezing during and after the defrost cycle. The defrost cycle is controlled by a timer.

Control and safety of the entire unit is provided by the control panel (7), which, according to the specifications, includes a high/low pressure controller (13) (pressure relay) for emergency shutdown of the compressors, as well as a Panel with dial thermometers (12) for visual monitoring of key system parameters.

The provided diagram (Figure 4.2) illustrates a common method for connecting multiple evaporators operating at different evaporation temperatures to a single centralized compressor station (not shown). The diagram displays two air coolers: one low-temperature (LT) unit (7) for a freezer room (operating at -18°C) and one medium-temperature (MT) unit (8) for a chiller room (operating at 0°C). Both evaporators are fed from a common liquid line (green) equipped with standard components for each circuit, including ball valves (1), a filter-drier (2), a sight glass (3), and a solenoid valve (4). Refrigerant expansion into each evaporator is controlled by an individual thermostatic expansion valve (TEV) (6).

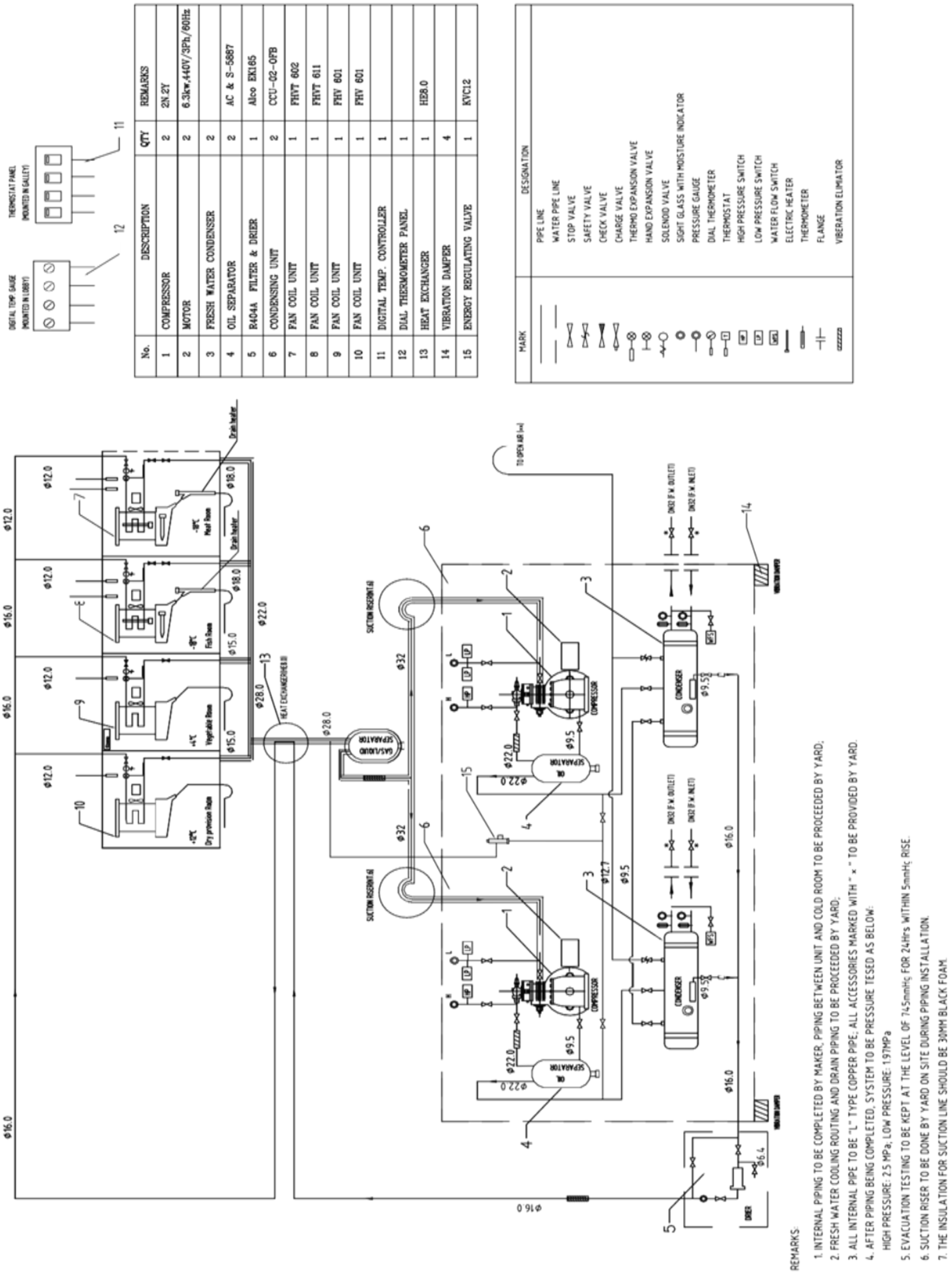


Figure 4.1 - Diagram of the refrigeration unit and location of the storage chambers

An example of connecting two evaporators at different evaporation temperatures (similar to the diagram of the refrigeration machine of the storage chamber) with Expansion valve is shown in Fig. 4.2.

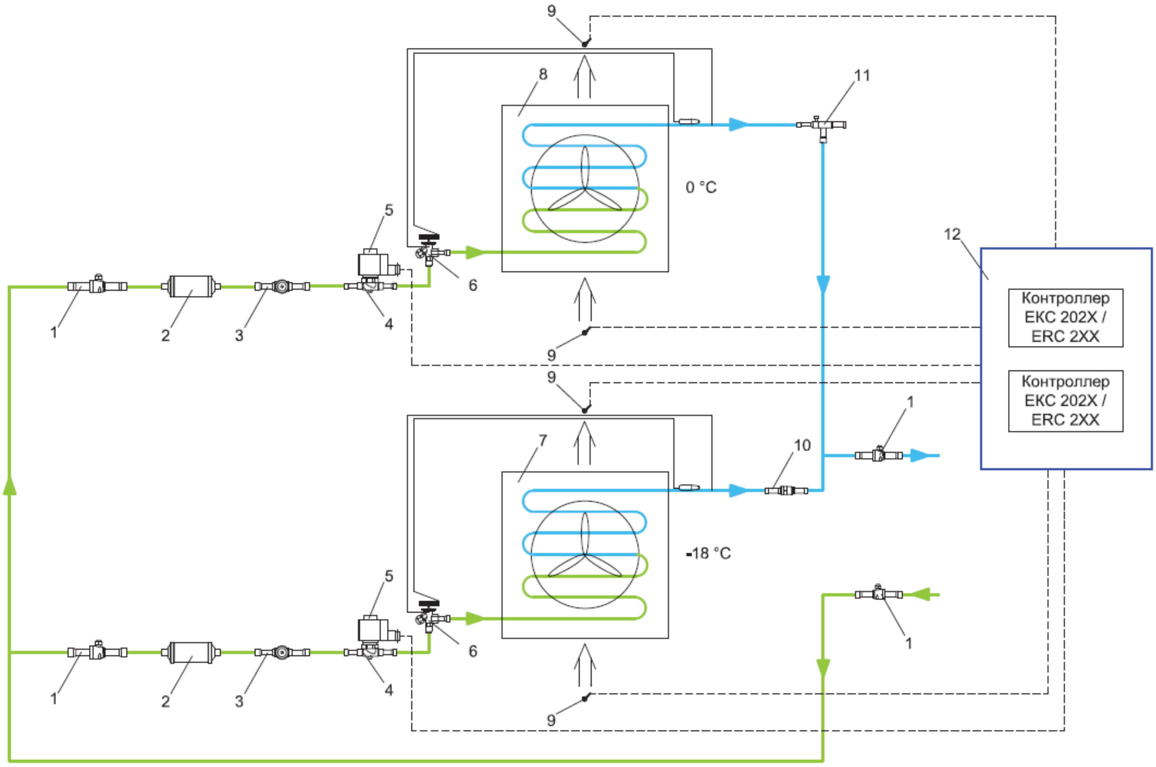


Figure 4.2 Example of connecting two evaporators at different evaporation temperatures with Expansion valve [28]: 1. GBC ball valve; 2. DML/DCL filter-dryer; 3. SG sight glass; 4. EVR solenoid valve; 5. Coil for solenoid valve; 6. TE thermostatic valve; 7. Low-temperature air cooler; 8. Medium-temperature air cooler; 9. EKS/AKS air temperature sensor; 10. NRV check valve; 11. KVP boiling pressure regulator; 12. Electrical panel with EKC 202X/ERC 2XX controllers

The key feature of this circuit is the method used to manage the different suction pressures. The central compressor station will maintain a common suction pressure corresponding to the lowest required temperature (i.e., -18°C). To maintain a higher boiling temperature (0°C) in the medium-temperature evaporator (8), a KVP boiling pressure regulator (11) is installed on its outlet suction line. This valve throttles the flow, creating an artificial back-pressure to maintain the desired higher pressure (and thus 0°C temperature) within the MT evaporator. To prevent reverse flow from the higher-pressure MT line into the LT line during operation or defrost, a check valve (10) is installed on the outlet of the low-temperature evaporator (7). Temperature control for each room is managed by electronic controllers (12), which receive input from temperature sensors (9) and cycle the respective solenoid valves (4).

4.2 Calculation of energy parameters of refrigeration machines for storage chambers when operating on different refrigerants

The main parameters of the air in the storage chambers, as well as the operating parameters of the refrigeration machine, are given below. All of the information below was used in the subsequent calculation of the refrigeration cycle and determination of the efficiency parameters of the refrigeration machine.

Table 4.1 – Air parameters in food storage chambers and volumes of these chambers

Chamber	Volume, m^3	Temperature, °C
Meat	22,6	-18
Fish	12,5	-18
Vegetable	22,6	+4

Temperature of the ambient air: +35°C;

Temperature of the cooling water: +36°C.

After cooling the premises, the unit will be able to maintain the specified temperature:

- one compressor operates for all rooms approx. 18/24 hours
- one compressor as a backup

Table 4.2 - Compressor power and energy consumption under different operating modes

Cooling water temperature, °C	Evaporation temperature, °C	Power, kW	Energy consumption, kW
+36	-10	9.7	4.9
+36	-26.9	4.4	3.5

The pressure in the evaporators is set so that the boiling temperature is 8-10 °C lower than the temperature in the chamber. Therefore, the boiling temperature of the refrigerant in the evaporator of the medium-temperature chamber (vegetable room) is -5

°C. The evaporation temperature of the refrigerant in the evaporator of the low-temperature chamber (meat and fish rooms) is -23 °C.

The thermal expansion valve was adjusted so that the superheat was maintained in the range of 8-10 °C for the specified design conditions.

The temperature difference of the water cooling the condenser between its inlet and outlet must be maintained within the range of 3-5 °C.

The compressor delivery pressure must be within the range of 1.3-2.0 MPa during normal operation.

The suction pressure of the compressor should be within 0.05-0.2 MPa during normal operation.

For further plotting and calculation of the thermodynamic cycle, it is necessary to find the mass ratios in which the liquid refrigerant flows are separated after the condenser for further throttling to their boiling point.

The ratio between the refrigerant flows (approximately): 0.75 (low-temperature chamber -18 °C) to 0.25 (medium-temperature chamber -5 °C).

That is, after the condenser, the refrigerant flow is divided, where 75% of the mass flow goes to the thermal expansion valves of the chambers with a temperature of -18 °C, and 25% goes to the thermal expansion valves of the chamber with a temperature of +4 °C.

A diagram of the R404A refrigerant showing the ideal (adiabatic compression in the compressor) cycle of a cold storage chamber refrigeration machine is shown in Fig. 4.3.

To draw up the cycle, the following assumptions were made:

- evaporation point of the refrigerant in the evaporator of the chamber with a temperature +4 °C – t_{01}= -5 °C, superheating of vapors in evaporators 1 °C;

- evaporation point of the refrigerant in the evaporator of the chamber with a temperature -18 °C – t_{02}= -23 °C, superheating of vapors in evaporators 1 °C;

- condensation temperature $t_к$= 42 °C, supercooling calculated after the condenser due to regenerative heat exchange from the heat balance.

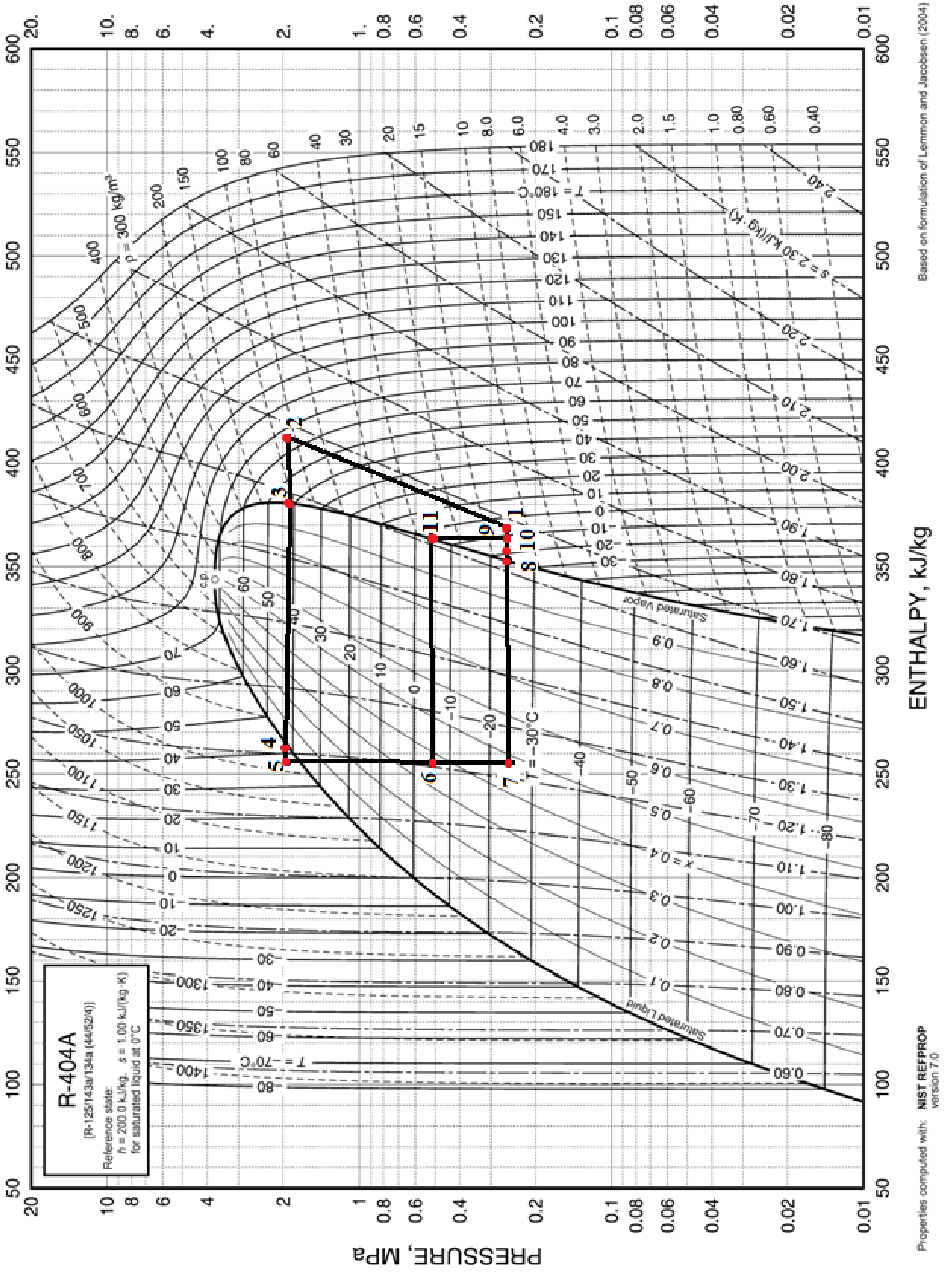


Figure 4.3 - Diagram of refrigerant R404A showing the ideal cycle of a cold storage room refrigeration machine

The calculation of the energy parameters of the refrigeration cycle, the results of which are given below, are based on standard methods described in [25-27].

On the Fig. 4.3 process 1-2 – compression in the compressor (adiabatic); 2-3-4 – cooling and condensation of the refrigerant in the condenser (isobaric); 4-5 – supercooling of liquid refrigerant in a regenerative heat exchanger by heating refrigerant vapors in process 9-1 (isobaric); 5-6 and 5-7 – throttling of the refrigerant in the control valve to certain pressures in the condensers (isenthalpic); 6-11 – boiling of the refrigerant in the evaporator of the high-temperature chamber (isobaric); 7-8 – boiling of the refrigerant in the evaporator of the low-temperature chamber (isobaric); 8-9 – superheating of liquid refrigerant in the evaporator of the corresponding chamber (isobaric); 11-9 – throttling of refrigerant in the boiling pressure regulator (isenthalpic); processes 11-9 and 8-9 – mixing of two refrigerant flows to obtain a flow whose state is characterized by the parameters at point 10 (isobaric); 10-1 – superheating of refrigerant vapors in a regenerative heat exchanger before suction into the compressor (isobaric).

After plotting the cycle, we write it out using the RefProp program [16] and at the same time calculate the enthalpy at the cycle nodes using the diagrams in the Table 4.3. In the process of determining the parameters, it is necessary to additionally find the enthalpy at points 10 and 1 from the heat balance equations.

The enthalpy at point 10 (the point that determines the state of the refrigerant after mixing the refrigerant flows from different evaporators) is determined as follows:

$$h_{10}=0.75 \cdot h_8+0.25 \cdot h_9. \qquad (4.1)$$

The enthalpy at point 1 is determined from the diagram when overheating relative to the refrigerant parameters at point 10 by 10 °C.

Point 5 is determined from the heat balance of the regenerative heat exchanger:

$$h_5=h_4-h_1+h_{10}. \qquad (4.2)$$

Additionally, we have to write down the density of the refrigerant at point 1.– ρ_1.

Table 4.3 - Parameters at cycle nodes for refrigerant R404A (points correspond to those marked in Fig. 4.3)

Point	Temmperature, °C	Pressure, MPa	Enthalpy, kJ/kg
1	-9.8910	0.27091	364.86
2	61.669	1.9115	407.04
3	42.164	1.9115	380.73
4	41.838	1.9115	263.26
5	36.693	1.9115	254.50
6	-5.1353	0.51482	254.50
7	-23.120	0.27091	254.50
8	-22.688	0.27091	353.65
9	-11.532	0.27091	363.42
10	-19.891	0.27090	356.10
11	-4.7313	0.51482	363.42

The method for determining the power consumed by the compressor is given below.

Specific cooling capacity of the evaporator chamber at a temperature of +4 °C:

$$q_{01}=h_{11}-h_6. \tag{4.3}$$

Similarly, for the evaporator in a chamber with a temperature of -18 °C:

$$q_{02}=h_8-h_7. \tag{4.4}$$

Specific adiabatic compression work of the compressor:

$$l_a=h_2-h_1. \tag{4.5}$$

Refrigeration coefficient (ideal):

$$\varepsilon=\frac{q_{01}+q_{02}}{l_a} \tag{4.6}$$

Compressor delivery coefficient (takes into account the reduction in actual compressor delivery compared to theoretical delivery):

$$\lambda=\lambda_i\cdot\lambda_w. \tag{4.7}$$

Heating coefficient:

$$\lambda_w = T_0/T_к, \tag{4.8}$$

where T_0 and $T_к$ - evaporation and condensation temperatures, *K*.

Volume loss coefficient in the compressor:

$$\lambda_i = \frac{\left(p_0 - \Delta p_{BC}\right) - c\left(p_K + \Delta p_H - p_0 + \Delta p_{BC}\right)}{p_0}, \tag{4.9}$$

Where Δp_{BC} - compressor inlet depression, can be assumed $\Delta p_{BC}=0.005$ *MPa*; Δp_H - compressor discharge depression, can be accepted $\Delta p_H=0.01$ *MPa*; c - relative dead space in the compressor, can be assumed $c=0.05$.

Mass flow rate of refrigerant in a refrigeration machine

$$G = \lambda \cdot V_h \cdot \rho_1, \tag{4.10}$$

where λ - compressor delivery ratio; V_h - compressor volumetric capacity – specified in the compressor's technical data; ρ_1 - density of refrigerant vapour at the compressor inlet (at point 1 of the cycle).

Since the refrigerant is divided into several streams (25% goes to the medium-temperature evaporator and 75% to the low-temperature evaporator), the refrigerant flow rate through these evaporators and their cooling capacities are calculated separately.

Mass flow rate of refrigerant in the medium-temperature chamber evaporator:

$$G_c=0.25 \cdot G, \text{ kg/s}.$$

Mass flow rate of refrigerant in the evaporator of a low-temperature chamber:

$$G_н=0.75 \cdot G, \text{ kg/s}.$$

Cooling capacity of the evaporator in the medium-temperature chamber Q_0

$$Q_{0c} = G_c \cdot q_{01} \tag{4.11}$$

Cooling capacity of the evaporator in a low-temperature chamber Q_0

$$Q_{0н} = G_н \cdot q_{02} \tag{4.11}$$

Since the refrigeration machine has several evaporators with different temperature levels, the cooling capacity is calculated separately for each of them, taking into account the mass flow rate of the refrigerant through the evaporators.

The indicator efficiency of the compressor, which takes into account the difference between the actual working process and the theoretical (isentropic) one – the deviation of processes 1-2 and 1-2' in Fig. 4.3, can be approximately calculated for fluorocarbon refrigerants using the empirical formula

$$\eta_i=\lambda_w+0.0025\cdot t_0, \qquad (4.12)$$

Total compressor efficiency

$$\eta=\eta_i\cdot \eta_{мех}\cdot \eta_n\cdot \eta_\partial, \qquad (4.13)$$

where η_i – indicator efficiency of the compressor; $\eta_{мех}$ – mechanical efficiency of the compressor, which takes into account losses caused by friction; η_n - Transmission efficiency; η_∂ - Compressor motor efficiency. For approximate calculations, the following can be assumed: $\eta_{мех} = 0.8 – 0.9$; $\eta_n = 0.95$; $\eta_\partial = 0.95$.

Actual power consumed by the refrigeration machine compressor (kW):

$$N_{компр}=\frac{l_a\cdot G}{\eta}, \qquad (4.14)$$

Actual refrigeration coefficient

$$\varepsilon_Д=\left(Q_{0c}+Q_{0н}\right)/N_{компр}. \qquad (4.15)$$

The calculations were based on the volumetric capacity of the compressor, taking into account the compressor parameters given in Table 4.4.

Table 4.4 - Technical specifications of the installed compressor

ONE-STAGE R-404A COMPRESSOR, pressure lubricated, marine type	
Model	SB022
Number of cylinders	2
Piston diameter, mm	60
Piston stroke, mm	57
RPM	1095

$$V_h=\frac{3.14\cdot 0.06^2}{4}0.057\cdot\frac{1095}{60}\cdot 2=0.00588\ \text{m}^3/\text{s},$$

where 0.06 m – piston diameter, 0.057 м – piston stroke, 1095 RPM – pistons rotational speed, 2 – number of cylinders.

Using the method described above, the parameters of the refrigeration cycle were determined at constant boiling and condensation temperatures and using R404A as the working fluid. In addition, the main energy characteristics of the refrigeration machine were calculated. The calculation results are shown in Table 4.5.

The cycle of the refrigerating machine of the storage chamber, which was calculated using the above method, is quite specific (due to two evaporators with different boiling temperatures). CoolTools v1.1.1 does not calculate such cycles. Manual calculation of this cycle is quite complicated. If we analyze the calculated cycle with two evaporators with different evaporation temperatures and take into account that the purpose of the work is to compare the efficiency of different refrigerants, it can be replaced by a simple regenerative cycle, where the lower evaporation temperature is taken for the evaporator. The results of calculating such a cycle for a storage chamber using CoolTools v1.1.1 are shown in Fig. 4.4. For the calculations, a condensation temperature of 42 °C (as for an air conditioner), a refrigerant boiling temperature in a low-temperature evaporator of 23 °C, and a compressor volumetric capacity (of the compressor described above) of 21.17 m3/h were assumed. All other data were taken similarly to the calculation of the refrigeration machine of the air conditioning system, except for the isothermal efficiency, which was taken to be lower – 0.6 (instead of 0.7 as for the air conditioner) due to the significantly higher pressure drop in the compressor of the refrigeration machine of the storage chambers.

Table 4.5 Results of calculating the energy characteristics of a refrigeration machine

Cooling agent	R404A
Compressor volumetric capacity V_h , m^3 /s	0.00588
Adiabatic compression work l_a , kJ/kg	42.18
Specific cooling capacity of the high-temperature chamber $q_{0в}$, kJ/kg	108.62
Specific cooling capacity of the low-temperature chamber $q_{0н}$, kJ/kg	99.15
Heating coefficient λ_w	0.794
Compressor volumetric loss coefficient λ_i	0.675
Compressor delivery coefficient λ	0.536
Total compressor efficiency η	0.632
Mass flow rate of refrigerant G, kg/s	0.023
Actual compressor power N_{comp} , kW	1.535
Mass flow rate of refrigerant in the evaporator of the high-temperature chamber $G_в$, kg/s	0.0058
Mass flow rate of refrigerant in the evaporator of the low-temperature chamber $G_н$, kg/s	0.0172
Cooling capacity of the high-temperature chamber $Q_{0в}$, kW	0.632
Cooling capacity of the low-temperature chamber $Q_{0н}$, kW	1.705
Actual cooling coefficient	1.522

Fig. 4.4 Screenshot of the software CoolTools v1.1.1 interface at input of initial data

Here we can obtain the refrigerant state parameters at the cycle nodes in the following sequence: temperature, pressure, enthalpy, entropy, density:
State 1: T= -2.82 °C, p= 2.625 bar, h= 371.22 kJ/kg, s= 1.6909 kJ/kg/K, m_dot= 0.0650 kg/s
State 2: T= 89.00 °C, p= 19.420 bar, h= 438.85 kJ/kg, s= 1.7549 kJ/kg/K, m_dot= 0.0650 kg/s
State 3: T= 88.76 °C, p= 19.190 bar, h= 438.85 kJ/kg, s= 1.7557 kJ/kg/K, m_dot= 0.0650 kg/s
State 4: T= 42.00 °C, p= 19.190 bar, h= 263.51 kJ/kg, s= 1.2116 kJ/kg/K, m_dot= 0.0650 kg/s
State 5: T= 31.93 °C, p= 19.190 bar, h= 246.76 kJ/kg, s= 1.1576 kJ/kg/K, m_dot= 0.0650 kg/s
State 6: T= -23.36 °C, p= 2.677 bar, h= 246.76 kJ/kg, s= 1.1939 kJ/kg/K, m_dot= 0.0650 kg/s
State 7: T= -23.00 °C, p= 2.677 bar, h= 353.44 kJ/kg, s= 1.6210 kJ/kg/K, m_dot= 0.0650 kg/s
State 8: T= -22.00 °C, p= 2.625 bar, h= 354.47 kJ/kg, s= 1.6266 kJ/kg/K, m_dot= 0.0650 kg/s

The values obtained from the calculation results in Fig. 4.4 are marked in green. We write them down:

- cooling capacity Qe=6.937 kW;
- mass flow rate of refrigerant m=0.065 kg/s;
- power consumption of the compressor W=4.887 kW;
- coefficient of performance COP=1.42.

As can be seen from the calculation results shown in the table and obtained using CoolTools v1.1.1, the refrigeration coefficients are quite close: 1.522 and 1.42. This confirms the possibility of using CoolTools v1.1.1 to calculate the efficiency of refrigeration machines in storage chambers using different refrigerants.

We perform the calculation in the same way for all refrigerants selected for analysis and enter the data obtained in Table 4.6.

Table 4.6 - Results of calculating the energy characteristics of ship provision chambers refrigeration mashine operating on different refrigerants

Refrigerant	Boiling pressure, bar (abs)	Cooling capacity Q_e, kW	Mass flow rate of refrigerant *m*, kg/s	Compressor power consumption *W*, kW	Coefficient of performance COP
R404A	2.667	6.937	0.065	4.887	1.420
R134a	1.164	6.937	0.049	4.27	1.625
R1234yf	1.335	6.937	0.065	4.523	1.534
R407C	1.891	6.937	0.046	4.924	1.409
R290	2.192	6.937	0.026	4.322	1.605
R1233zd(E)	0.156	6.937	0.046	3.945	1.758
R1234ze(E)	0.847	6.937	0.055	4.288	1.618
R32	3.618	6.937	0.028	4.525	1.533

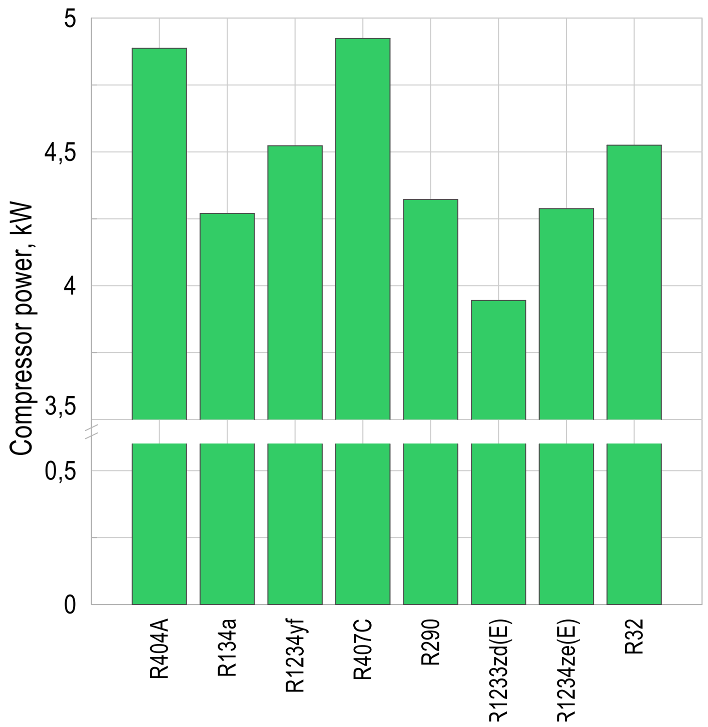


Fig. 4.5 Power consumption by the compressor during operation of the refrigeration machine of storage chambers using different refrigerants

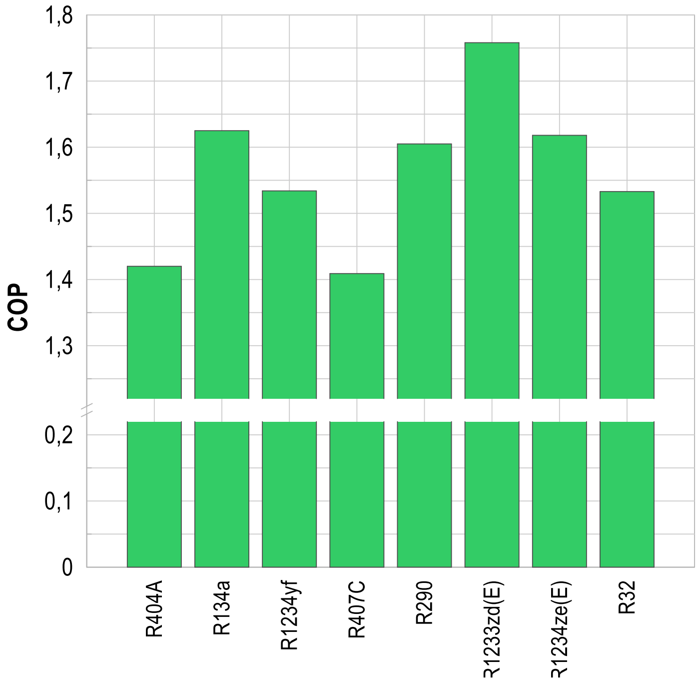


Fig. 4.6 COP refrigeration coefficient values during operation of the refrigeration machine of storage chambers using different refrigerants

An analysis of COP values for cold storage room refrigeration machines operating on different refrigerants shows that R1233zd(E) provides the highest energy efficiency, demonstrating its significant potential in terms of reducing energy consumption. R134a, R290, and R1234ze(E) also have fairly high COP values, demonstrating stable and energy-efficient operation in low-temperature modes. The lowest COP is observed for R407C and R404A, confirming their energy inefficiency for modern marine refrigeration systems. Refrigerants R1234yf and R32 occupy an intermediate position in terms of energy efficiency.

4.3 Environmental impact of refrigeration machines in storage chambers using different refrigerants

The purpose of this stage of the master's research was to conduct an ecological and energy analysis of the operation of a refrigeration machine for food storage chambers in ship conditions using various refrigerants. The analysis was performed in a manner similar to the approach described for ship air conditioners. Some of the input data for the ecological and energy analysis are given in Table 4.6. The electrical power for the refrigeration machine when operating on the specified refrigerants at t_c=42 °C and t_0=-23 °C accepted based on the results of preliminary calculations. The same table shows the results of TEWI calculations.

Fig. 4.7 shows the results of a comparative analysis of specific CO2 emissions per 1 kJ of cooling produced for a shipboard refrigeration machine for food storage rooms using refrigerants.

Table 4.7 – Results of the ecological and energy analysis of the ship's refrigeration machine for food storage chambers

Parameter	R404A	R134a	R1234yf	R407C	R290	R1233zd(E)	R1234ze(E)	R32
Cooling capacity Q_0, kW	6.937	6.937	6.937	6.937	6.937	6.937	6.937	6.937
Compressor power consumption N_{el}, kW	4.887	4.27	4.523	4.924	4.322	3.945	4.288	4.525
Refrigerant charge mass m_R	8 kg							
GWP of refrigerant (kg CO_2)/kg	3940	1300	4	1774	3	1	1-7	675
CF for heavy fuel, (t CO_2)·(t fuel)$^{-1}$	3.206							
g_F, kg·(kW·hour)$^{-1}$	0.23							
L_{annual}	0.3							
TEWI, kg CO_2	$401 \cdot 10^3$	$299 \cdot 10^3$	$283 \cdot 10^3$	$351 \cdot 10^3$	$271 \cdot 10^3$	$247 \cdot 10^3$	$269 \cdot 10^3$	$300 \cdot 10^3$
em_{refr}, (kg CO_2)/kJ of cold	$1.833 \cdot 10^{-6}$	$1.366 \cdot 10^{-6}$	$1.29 \cdot 10^{-6}$	$1.6 \cdot 10^{-6}$	$1.23 \cdot 10^{-6}$	$1.13 \cdot 10^{-6}$	$1.23 \cdot 10^{-6}$	$1.37 \cdot 10^{-6}$

The results show that the highest specific CO_2 emissions are observed when using R404A and R407C refrigerants, which is due to their high global warming potential and less favorable energy characteristics of the refrigeration cycle.

The lowest specific CO_2 emissions are characteristic of refrigerants R1233zd(E), R290, and R1234ze(E), which indicates their environmental advantages and high energy efficiency when used in ship refrigeration machines for food storage rooms.

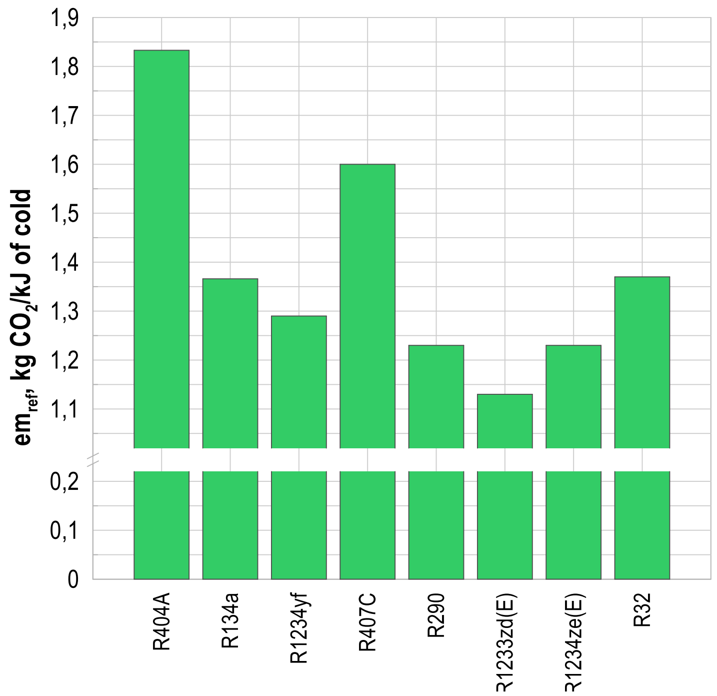


Fig. 4.7. Comparison of the environmental performance of ship refrigeration machines for food storage rooms when operating on different refrigerants

Refrigerants R134a, R1234yf, and R32 occupy an intermediate position, combining moderate CO_2 emissions with sufficiently high performance characteristics.

Thus, the results obtained confirm the feasibility of a gradual transition of ship refrigeration units in food storage rooms to environmentally friendly refrigerants with low GWP, primarily R1233zd(E), R1234ze(E), and natural refrigerant R290, in accordance with current environmental legislation requirements and IMO recommendations.

5. ANALYSIS OF RESULTS AND FORMULATION OF RECOMMENDATIONS FOR THE SELECTION OF REFRIGERANTS FOR SHIP REFRIGERATION EQUIPMENT

5.1. Refrigerant selection for ship air conditioning

5.1.1. Energy performance

The research examined R404A, R134a, R1234yf, R407C, R290, R1233zd(E), R1234ze(E), and R32. For a marine air conditioner in typical operating mode, it was shown that:

- R134a has a high coefficient of performance (COP ≈ 4.46) and lower compressor power (≈ 37.9 kW) compared to R404A (≈ 51.5 kW).

- R407C has the lowest COP and the highest power consumption among the agents considered, so it is undesirable from an energy point of view.

- R1234yf and R290 have a COP only slightly lower or close to that of R134a (a moderate difference of a few percent), while having a significantly lower GWP.

- R1233zd(E) has the lowest compressor power (≈ 35.6 kW) among all options, which provides the best conditions for energy saving.

Therefore, in terms of energy efficiency for air conditioning, R1233zd(E), R134a, R290, R1234yf / R1234ze(E) are the most advantageous, while R404A and especially R407C are the worst.

5.1.2. Environmental indicators

The results of calculating specific greenhouse gas emissions for ship air conditioners are presented in Table 3.2. The following general conclusions can be drawn from the results obtained:

- R404A has the highest GWP (~3920) and, accordingly, the highest TEWI and specific CO_2 equivalent emissions.

- R134a reduces total emissions, but still has an average GWP (~1430) and is subject to strict phase-out in the EU and under the Kigali Agreement.

- HFO refrigerants R1234yf and R1234ze(E), as well as R1233zd(E), have ultra-low GWP (<10 for R1234yf/ze, ≈1–2 for R1233zd(E)), so their TEWI is significantly lower even compared to R134a.

- R290 (GWP ≈3) has a very low TEWI, but belongs to class A3 (high flammability).

- R32 has a moderate GWP (~675) and can be considered a transitional option, but not an end goal in terms of future requirements (150...300).

5.1.3. Operational safety

R404A, R134a, R1233zd(E) – class A1, non-flammable. This is an important advantage for shipboard conditions (especially for air conditioning in living quarters).

R1234yf, R1234ze(E), R32 – A2L (low to moderate flammability, low flame spread rate). They require additional safety measures, but are considered acceptable for modern systems with proper layout and additional safety measures.

R290 – A3 (high flammability), which significantly limits its use in residential air conditioning systems on ships under SOLAS and classification society rules.

5.1.4. General conclusion on the choice of refrigerant for ship air conditioning

Taking into account all three groups of criteria (energy efficiency, ecology, and safety), the following recommendations can be made for ship air conditioning:

Basic option for existing systems:

R134a – justified for existing systems, as long as it is permitted by legislation. Provides high COP and significantly lower emissions than R404A, while being fully compatible with the equipment. However, this is a transitional option, not a long-term one.

Target low GWP options:

- R1233zd(E) – the most energy-efficient and non-flammable. Realistically, it is the optimal candidate for marine chillers and centralized air conditioning systems where large dimensions and low boiling pressures are acceptable.

- R1234ze(E) / R1234yf – promising for new systems with limited refilling and implemented safety measures (possible as working agents in chiller systems or as part of mixtures).

Limitedly suitable options:

- R32 – can serve as an interim solution with reduced GWP and decent COP, but requires A2L safety measures and does not meet long-term goals (GWP is still significantly >150).
- R290 – thermodynamically very advantageous, but its use in air conditioning systems for living quarters on large ships is only advisable in low-power systems (such as split systems) manufactured in a factory and located outside the living quarters.

Not recommended options:

R404A and R407C – have the worst combination of “high TEWI + low COP”. Their use in new ship air conditioners should be considered inadvisable.

Table 5.1 – Comparative characteristics of refrigerants for marine air conditioners

Refriger-ant	GWP	Safety class	COP in relation to R134a	Environmental assessment (TEWI)	Advantages	Disadvantages
R404A	Very high	A1	Lower	The worst	Fully compatible with older systems	Very high GWP, low COP, prohibited by regulations
R134a	Average	A1	Base line/ relatively high	Better than R404A	Solid energy, well-developed equipment	Phased restrictions due to GWP
R1234yf	Very low	A2L	Slightly lower	Significantly lower than R134a	Low GWP, similar operating parameters	A2L, higher price, safety requirements
R407C	High	A1	The lowest	Worse than R134a	Transition agent for R22	Low COP, environmentally outdated

Continuation of Table 5.1

R290	Very low	A3	Close to R134a	Very low	Excellent thermodynamic properties	High flammability
R1233zd(E)	Very low	A1	Higher	Very low	Non-flammable, high energy efficiency	Low pressure, larger proportions
R1234ze(E)	Very low	A2L	Close to R134a	Very low	Energy/environment balance	A2L, more expensive
R32	Average	A2L	Higher	Better than R134a	High COP, reduced charging volume	A2L, GWP above long-term requirements

5.2. Refrigerant selection for frigeration machine for storage chambers

For a two-chamber provision unit (medium- and low-temperature evaporators), detailed energy and environmental indicators are given in the thesis (Tables 4.6, 4.7).

5.2.1. Energy analysis

The thesis examined R404A, R134a, R1234yf, R407C, R290, R1233zd(E), R1234ze(E), and R32. For a ship refrigeration machine for food storage chambers under typical operating conditions, it was shown that:

- The base agent R404A gives a COP of ≈ 1.42.
- R1233zd(E) has the highest COP – ≈ 1.76, which is ≈ 24% higher than R404A.
- R134a, R290, and R1234ze(E) also provide a COP of 1.60–1.63, which is ≈ 13–15% higher than the base.
- R1234yf and R32 have a COP of around 1.53 (≈ +8% to the baseline).
- R407C has a COP very close to or even lower than R404A and has no significant energy advantages.

5.2.2. Environmental analysis (TEWI and specific emissions)

The results of calculating specific greenhouse gas emissions for ship air conditioners are presented in Table 3.7. The following general conclusions can be drawn from the results obtained:

- R404A: TEWI ≈ $4.01 \cdot 10^5$ kg CO_2 and specific emissions of $1.83 \cdot 10^{-6}$ kg CO_2/kJ of cooling – the worst indicators.

- R407C: TEWI ≈ $3.51 \cdot 10^5$ kg CO_2 (only ~12% reduction compared to R404A) and fairly high specific emissions – about $1.60 \cdot 10^{-6}$ kg CO_2/kWh.

- R134a / R32: TEWI ≈ $2.99–3.00 \cdot 10^5$ kg CO_2 (approximately 25% less than R404A), specific emissions are reduced by the same ~25%.

- R1234yf: TEWI ≈ $2.83 \cdot 10^5$ kg CO_2 (~–29% compared to R404A).

- R290 and R1234ze(E): TEWI ≈ $2.69–2.71 \cdot 10^5$ kg CO_2 (a reduction of 32–33%), specific emissions are ~33% lower than for R404A.

- R1233zd(E): TEWI ≈ $2.47 \cdot 10^5$ kg CO_2, i.e. ≈ 38 % less than R404A, specific emissions are reduced by approximately 38 % as well.

Therefore, for a provisional installation, the combination of “highest COP + minimum environmental impact” is observed for R1233zd(E), followed by the R290 / R1234ze(E) / R1234yf group.

5.2.3. Operational and safety aspects

According to the classification of refrigerants according to ISO 817 and ASHRAE 34 standards, the working fluids considered belong to the following safety groups:

- R404A, R134a, R1233zd(E) — class A1, non-toxic and non-flammable;

- R1234yf, R1234ze(E) — class A2L, refrigerants with low explosion and fire hazard and limited flammability;

- R290 — class A3, chemically safe, but with high flammability.

In refrigeration machines for storage chambers, compressor-condenser equipment is usually located in the machine room or in a special compartment directly adjacent to the food storage chambers. On the one hand, this allows the use of refrigerants with an increased level of fire hazard, but on the other hand, it requires mandatory installation of forced ventilation, gas control, emergency shut-off, explosion-proof fans, and electrical equipment.

At the same time, when choosing a refrigerant for food refrigeration machines, it is necessary to take into account not only the COP and TEWI indicators, but also the thermodynamic suitability of the agent for low-temperature modes. In particular, it has been established that refrigerants R1233zd(E) and R1234ze(E) have a relatively high normal boiling point, as a result of which, when operating in low-temperature evaporators, the boiling pressure drops to values below atmospheric pressure, which leads to:

- complications in compressor shaft sealing;
- an increased risk of air and moisture suction;
- deterioration of heat transfer stability and mode control;
- an increased likelihood of corrosion processes.

In this regard, R1233zd(E) and R1234ze(E) should be considered unsuitable for refrigeration machines in storage chambers operating in the negative boiling temperature range, despite their excellent environmental performance.

Based on practical experience in operating ship refrigeration systems and in accordance with the requirements of classification societies, the following are currently realistic areas for modernization:

- short-term solutions without major reconstruction — transition from R404A to medium-GWP HFC/HFO blends (e.g., R448A, R449A) or use of R134a and R407F;
- medium- and long-term solutions — introduction of ultra-low-GWP refrigerants that are thermodynamically suitable for low-temperature modes, in particular R290, R1234yf, as well as new specialized HFO mixtures.

5.2.4. Recommendations for selecting refrigerants for cold storage refrigeration machines

Taking into account the results of thermodynamic calculations, analysis of COP and TEWI indicators, as well as operational and safety restrictions, the feasibility of using refrigerants for ship refrigeration units in provision rooms can be summarized as follows.

The best options based on a bunch of indicators

R290 (propane) — gives a high COP, one of the lowest specific CO_2 emissions, and the lowest TEWI among those that are actually suitable for low-temperature systems. Its use is technically feasible in systems with a small refrigerant charge, hermetic factory units located outside residential premises and equipped with gas control and emergency ventilation systems.

R1234yf is an environmentally friendly refrigerant with low GWP, acceptable energy performance, and safety class A2L. It can be considered a promising alternative for marine refrigeration machines, but only if that explosion and fire safety requirements are met.

Acceptable but transitional options

R134a and R32 provide a significant reduction in TEWI compared to R404A and an acceptable COP, but due to their relatively high GWP, they do not meet long-term environmental goals and can only be considered as a temporary solution when upgrading existing installations.

Inappropriate and limited options

R404A — has the worst combination of high specific CO_2 emissions and low energy efficiency, currently is a subject to strict F-Gas regulations, and is not eligible for further use in new installations.

R407C — does not provide a significant reduction in environmental impact compared to R404A and is inferior to most modern alternatives in terms of energy performance.

R1233zd(E) and R1234ze(E) — despite their extremely low GWP and TEWI values, are not recommended for cold storage refrigeration machines due to their unsuitability for low-temperature conditions and evaporator operation at sub-atmospheric pressures. They should only be considered for chiller systems with a secondary heat transfer medium in air conditioning systems.

Table 5.2 – Comparative characteristics of refrigerants for cold storage refrigeration machines

Refrigerant	GWP	Safety class	COP rela-tively to R404A	Environmental assessment (TEWI)	Advantages	Disadvantages
R404A	Very high	A1	Base line (≈1.42)	The high-est	Widely used in marine installations, compatible with existing equipment	Very high GWP, worst TEWI, low COP, F-Gas bans
R407C	High	A1	≈ the same / slightly lower	High	Transition agent for R22, compatible with part of existing systems	Does not provide real benefits in terms of COP and TEWI, environmentally obsolete
R134a	Average	A1	Higher by ≈15 %	Average	Better COP and lower TEWI than R404A; non-flammable, established equipment	GWP is still quite high; only a transitional option
R32	Moderate	A2L	Higher by ≈8 %	Average/ moderate	Higher COP, reduced charge mass, lower TEWI than R404A	A2L (limited flammability), GWP above target values

Continuation of Table 5.2

R1234yf	Very low	A2L	Higher by ≈8 %	Low	Low GWP, significant reduction in TEWI, acceptable COP, suitable for HT mode	A2L, higher cost, increased sealing requirements
R1234ze(E)	Very low	A2L	Higher by ≈14 %	Very low	Very low TEWI, high COP, excellent environmental profile	High boiling point → low-temperature evaporator operation at subatmospheric pressure; limited suitability
R290	Very low	A3	Higher by ≈13 %	Very low	Excellent thermodynamic properties, one of the lowest TEWI values, ideal for systems with low charge volume	High flammability (A3); strict safety requirements
R1233zd(E)	Very low	A1	Higher by ≈24 %	The lowest	Low GWP, very high COP, non-flammable	High boiling point, vacuum in low-temperature evaporator; practically unsuitable for storage chambers

CONCLUSION

The diploma research was devoted to the actual problem of reducing the impact of ship refrigeration technology on the environment in the context of growing energy efficiency requirements (EEXI/CII indices) and environmental responsibility of the shipping industry. The main focus was on analyzing the use of the traditional refrigerant R404A, which no longer meets modern standards due to its high global warming potential (GWP=3922) and low energy efficiency.

The study considered alternative refrigerants to replace it, including R134a, R407C, R1234yf, R1234ze(E), R1233zd(E), R32, and R290 (propane), in order to determine the most optimal modernization option. The analysis was performed for two key consumers: a marine air conditioner with a cooling capacity of 204.2 kW and a refrigeration machine for food storage rooms with a cooling capacity of 6.9 kW.

The results of the energy analysis of the air conditioning system showed that the base refrigerant R404A is the least efficient (COP=3.97). Switching to the standard refrigerant R134a provides a 12.4% higher coefficient of performance (COP=4.46). The lowest pressure refrigerant R1233zd(E) demonstrated the highest energy efficiency, with a COP=4.76 (20% higher than R404A). However, the use of R1233zd(E) requires a complete replacement of the compressor with a larger positive displacement unit due to the specific nature of low suction pressure.

Particular attention was paid to environmental analysis using the TEWI (Total Equivalent Warming Impact) value, which takes into account both direct (leaks) and indirect (energy) greenhouse gas emissions. The results of the calculations showed that HFO refrigerants (R1233zd(E) and R1234ze(E)) and natural R290 provide the lowest TEWI values. Specific greenhouse gas emissions per unit of cooling produced are 28% lower when using R1234ze(E) and 31% lower when using R1233zd(E) than for the base refrigerant R404A.

Refrigerant R407C, which is often offered as a drop-in replacement, showed the worst results in terms of both energy and environmental performance, so its use is considered inadvisable.

For a cold storage machine (low-temperature mode), analysis of refrigerant properties revealed important technological limitations. Although R1233zd(E) showed the highest theoretical COP (1.76), it proved to be technically unsuitable for freezers because at a boiling temperature of -23°C, the pressure in the evaporator drops to a deep vacuum (0.156 bar), creating an unacceptable risk of air suction. Therefore, based on a combination of factors, R290 (propane) with a COP of 1.61 (13% higher than R404A) and R134a with a COP of 1.63 (15% higher than R404A) were determined to be the optimal options for this system.

The analysis showed that each of the proposed options has its own advantages and limitations. Refrigerants R1234ze(E) and R1233zd(E) have the lowest GWP (<10), making them ideal for the long term, but they require capital investment to upgrade equipment due to their A2L (low flammability) safety class or design features. R290 is thermodynamically excellent but requires strict safety measures (class A3). R134a remains a reliable compromise for existing systems, providing efficiency gains without design changes, but has a limited service life due to its GWP=1430.

From all of the above, we can conclude that for ship air conditioning systems, it is most expedient to switch to HFO refrigerants (in particular, R1234ze), and for provisioning units, to R290 or R134a. This transition will contribute to both energy savings (reducing the load on diesel generators by 4-6 kW for each unit) and a 25-30% reduction in total greenhouse gas emissions.

The results of the thesis research allow us to conclude that abandoning R404A in favor of alternative refrigerants is not only environmentally necessary but also energy-efficient. The proposed recommendations allow shipowners to choose a modernization strategy that will ensure the ship's compliance with IMO requirements and reduce operating costs.

REFERENCES

1. Fourth IMO Greenhouse Gase Study 2020. Publ. by the International Maritime Organization. 4 Albert Embankment, London, SE1 7SR. 2021. https://wwwcdn.imo.org/localresources/en/OurWork/Environment/Documents/Fourth%20IMO%20GHG%20Study%202020%20Executive-Summary.pdf

2. Wilhelmsen, 2022. Reduce Emissions with Responsible Refrigeration. https://www.wilhelmsen.com/globalassets/ships-service/refrigeration-solutions/documents/responsible-refrigeration-brochure_062022.pdf (Accessed 8 Oct. 2025)

3. International Maritime Organization. Third IMO GHG Study 2014. Executive Summary and Final Report. 2014. https://www.imo.org/en/ourwork/environment/pages/greenhouse-gas-studies-2014.aspx (Accessed 12 Oct. 2025)

4. Hafner I. A., Gabrielii C. H., Widell K. Refrigeration units in marine vessels: Alternatives to HCFCs and high GWP HFCs. Nordic Council of Ministers, Denmark. 2019.

5. European Commission. Analysis of emissions of fluorinated gases from refrigeration and air-conditioning equipment. Maritime sector report. Brussels, 2007.

6. D’Onofrio M., Petruzziello F., Grilletto A., Aprea C., Maiorino A. Exploring the potential of R744 as a sustainable refrigerant for marine applications. *Energies.* 2025. Vol. *18*(23). 6211.

7. Wajs J., Mrozek M., Fornalik-Wajs E. Combined cold-supply system for ship application based on low-GWP refrigerants: Thermo-economic and ecological analyses. *Energy Conversion and Management*. 2022. Vol. 258. 115518.

8. Glavan I., Poljak I., Mustać M., Lonić I. Natural Refrigerant on Board Marine Vessels. *Pomorski zbornik*. 2022. Vol. 62(1). P. 43-56.

9. Wilhelmsen Ships Service. Refrigerants and Refrigeration equipment. Marine Products Brochure. 2022.

10. Minetto S., Fabris F., Marinetti S., Rossetti A. A review on present and forthcoming opportunities with natural working fluids in transport refrigeration. *Int. J. Refrig.* 2023. Vol. 152. P. 343-355.

11. Pigani L., Boscolo M., Pagan N. Marine refrigeration plants for passenger ships: Low-GWP refrigerants and strategies to reduce environmental impact. *Int. J. Refrig.* 2016. Vol. 64. P. 80-92.

12. Abas N., Kalair A. R., Khan N., Haider A., Saleem Z., Saleem M. S. Natural and synthetic refrigerants, global warming: A review. *Renewable Sustainable Energy Rev.* 2018. Vol. 90. P. 557-569.

13. Söylemez E., Widell K. N., Gabrielii C. H., Ladam Y., Lund T., Hafner A. Overview of the development and status of carbon dioxide (R-744) refrigeration systems onboard fishing vessels. *Int. J. Refrig.* 2022. Vol. 140. P. 198-212.

14. Heinen & Hopman. Natural refrigerants: a good alternative to high-GWP refrigerants? https://www.heinenhopman.com/knowledge/whitepaper-natural-refrigerants/ (Accessed 8 Oct. 2025)

15. Регістр судноплавства України. Правила класифікації та побудови морських суден. Київ: Регістр судноплавства України, чинна редакція. Доступ: https://shipregister.ua

16. REFPROP: Reference fluid thermodynamic and transport properties, NIST standard reference database 23, Version 8.0 / E. W. Lemmon, M. L. Huber, M. O. McLinden. Gaithersburg: National Institute of Standard and Technology, 2007.

17. ASHRAE Handbook – Fundamentals. American society of heating, refrigerating and air-conditioning engineers. Peachtree Corners, GA, USA, 2021. 1013 p.

18. CoolTools. v1.1.1. Copenhagen: IPU. 2023. https://www.ipu.dk/products/cooltools/

19. Shestopalov K., Khliyeva O., Ierin V., Konstantinov O., Khliiev N., Neng G., Kozminykh M. Novel marine ejector-compression waste heat-driven refrigeration system: technical possibilities and environmental advantages. *International Journal of Refrigeration*, 2024. Vol. 158, P. 202-215

20. Chai M., Reddy B.D., Sobrayen L., Panda S.K., Die W., Xiaoqing C. Improvement in efficiency and reliability for diesel-electric propulsion based marine vessels using genetic algorithm. *2016 IEEE Transportation Electrification Conf. and Expo, Asia-Pacific (ITEC Asia-Pacific),* 2016. P. 180-184

21. EEDI Calculation Guidelines. 2022 Guidelines on the Method of Calculation of the Attained Energy Efficiency Design Index (EEDI) for New Ships. Resolution MEPC.364(79). *International Maritime Organization.* 2022

22. Wang S., Psaraftis H. N., Qi J. Paradox of international maritime organization's carbon intensity indicator, *Communications in Transportation Research*. 2021. Vol. 1. 100005

23. Schwarz W. Rhiemeier J.M. The analysis of the emissions of fluorinated greenhouse gases from refrigeration and air conditioning equipment used in the transport sector other than road transport and options for reducing these emissions Maritime, Rail, and Aircraft Sector, Brussels: European Commission. 2007. https://ec.europa.eu/clima/system/files/2016-11/2_maritime_rail_aircraft_en.pdf

24. Methods of calculating total equivalent warming impact (TEWI). Best practice guidelines, Australian Institute of Refrigeration, Air-conditioning and Heating. AIRACH. 2012. https://www.airah.org.au/Content_Files/BestPracticeGuides/Best_Practice_Tewi_June2012.pdf

25. Шестопалов К.О., Хлієва О.Я., Козьміних М.А. Суднова холодильна техніка: методичні вказівки до практ. занять. Одеса:НУ«ОМА», 2024. 30 с.

26. Колиев И.Д. Суднові холодильні установки: навчальний посібник. Одеса: Фенікс, 2009. 264 с.

27. Радченко М. І., Лехмус О. О. Суднові холодильні машини: навчальний посібник. Миколаїв: НУК, 2015. 392 с.

28. Danfoss. Refrigeration and Air Conditioning Controls: Application Guide for EKC 202X / ERC 2XX. Technical Literature RC.8A.A1.02. Nordborg: Danfoss A/S, 2019. https://assets.danfoss.com/documents/121833/RC8AA102.pd

29. EU F-gas Regulation Guidance Information Sheet 31: Marine Refrigeration and Air Conditioning. Gluckmann Consulting. 2016

http://www.gluckmanconsulting.com/wp-content/uploads/2014/12/IS-31-Marine-Applications.pdf (accepted: 17.11.2025)

30. DNV GL AS. Rules for classification. Ships. Edition July 2019. *Part 6 Additional class notations. Chapter 7 Environmental protection and pollution control*

31. Rules for the classification and construction of sea-going ships Part XII Refrigerating Plants ND No. 2-020101-138-E. 2021

32. Refrigerants and Refrigeration equipment. Marine Products. Wilh. Wilhelmsen Holding ASA**.** https://www.wilhelmsen.com/ (accepted: 19.11.2025)

33. Marine air conditioning and ventilation plant. Marine provision refrigeration plant // Hyundai heavy industries Co., Ltd. // Hull #1973 / P. 263-276.

34. Refrigerant Changeover HFC R404A or R507 to HFC R407A, R407F, R448A or R449A. Tecnical Information. Emerson. Application Engineering Europe Ref: CC7.26.3/1012-0417/E. 2017.

35. A *handbook* for *engineers*, *technicians*, *trainers* and *policy-makers* - For a *climate-friendly cooling. Guidelines* for the *safe use.* Deutsche Gesellschaft für Internationale Zusammenarbeit (GIZ) GmbH. *2012.*

36. A Practical Guide to the Selection of Energy Efficiency Technologies for Ships. GreenVoyage2050 Project Coordination Unit, IMO (2022) https://greenvoyage2050.imo.org (accepted: 02.12.2025)