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:

# ANALYSIS OF THE INFLUENCE OF OPERATING PARAMETERS ON THE EFFICIENCY OF SHIP HEAT EXCHANGERS

Performed by: Cadet Maksym Hlazyrin

Supervisor: Prof. Olga Khliyeva

Normocontrol PhD Parmenova D.M.

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 15 2025.
Head of the Department of SAP&RE
D.Sc., Professor Olga KHLIYEVA

ДОПУЩЕНИЙ ДО ЗАХИСТУ
ЗАВІДУЮЧИЙ КАФЕДРОЮ
200_ р.

Recommended for defense at the EC, Protocol No 9 dated 17.12 2025.
Head of the Department of Ship Power Plants
D.Sc., Professor Serhiy SAGIN

Reviewer (external) PhD, Hlek Ya, 18.12.25
(signature, academic title, full name, date)

Reviewer (internal) Семик ОВ
(signature, academic title, full name, date)

Odesa – 2025

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

APPROVED BY
Head of the Graduating Department
_____________ D.Sc., Prof. S.V. Sagin
"10" 09 2025.

## ASSIGNMENT
## for the completion of the master's thesis

Cadet (student) of ESIE *Maksym Hlazyrin*

1. Thesis topic: *Analysis of the Influence of Operating Parameters on the Efficiency of Ship Heat Exchangers*

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

2. Object of research: *Improvement of the efficiency of processes in shipboard heat exchangers cooled by seawater*

3. Subject of research: *Shipboard heat exchangers cooled by seawater*

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 shipboard heat exchangers, their design, operating modes, and factors leading to performance degradation; analysis of the effect of fouled heat transfer surfaces on heat transfer intensity and hydraulic losses in the heat exchanger; analysis of possible methods for mitigating fouling in seawater heat exchangers and selection of a rational fouling prevention method; evaluation of the economic savings resulting from the implementation of the proposed solution*

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:

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

9. Calendar plan of work execution

| № 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. | | |

10. Date of assignment issuance 23/09/25

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

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

Thesis Performer ____________ Maksym Hlazyrin
(signature) (full name)

# РЕФЕРАТ

Магістерська кваліфікаційна робота на тему: «Аналіз впливу режимних параметрів на ефективність роботи суднового теплообмінного обладнання»: 96 с., 23 рис., 6 табл., 51 джерел, 11 слайдів презентаційного матеріалу.

Магістерська робота спрямована на розв’язання науково-прикладного завдання – підвищення ефективності тепло-гідравлічних процесів к пластинчастого теплообміннику, а також зниження витрат суднового палива при його експлуатації за рахунок запобігання забрудненню поверхні теплообміну біоплівкою.

Висунута та підтверджена наукова гіпотеза про те, що підвищення ефективності тепло-гідравлічних процесів та зниження витрат палива забезпечуються раціональним вибором методу запобігання забрудненняя теплообмінної поверхні.

Для судна з головним двигуном 12V Wärtsilä 46F та додатковими енергетичними системами розраховано та підібрано пластинчастий теплообмінника GEA NT (Marine Duty) NT250S з площею поверхні теплообміну 160 м² для відведення теплоти від прісної води до забортної. Показано, що наявність забруднення як зі сторони прісної води, так й забортної значно знижує кількість теплоти, що передається (приблизно з 4700 кВт до 2230 кВт) та збільшує втрати тиску.

Показано, що додаткові витрати на паливо (внаслідок збільшення гідравлічних втрат та зменшення теплового навантаження завдяки забрудненню поверхні зі сторони забортної води) становлять 117 та 575 $ USD на місяць для товщини біоплівки 0,2 мм та 0,5 мм відповідно. Встановлено, що найбільш екологічно та економічно доцільним методом запобігання біообрастання є періодичне електролізне хлорування. Показано, що витрати на запобігання біообростанню ($128 USD у місяць) окуповуються вже при утворення біоплівки товщиною 0,2 мм.

ТЕПЛООБМІННЕ ОБЛАДНАННЯ, ЗАБОРТНА ВОДА ЯК ТЕПЛОНОСІЙ; БІОЛОГІЧНЕ ЗАБРУДНЕННЯ ТЕПЛООБМІННОЇ ПОВЕРХНІ, КОЕФІЦІЕНТ ТЕПЛОПЕРЕДАЧІ, ГІДРАВЛІЧНІ ВТРАТИ, ЕКОНОМІЯ ПАЛИВА

# ABSTRACT

Master’s qualification thesis on the topic: “Analysis of the Influence of Operating Parameters on the Efficiency of Ship Heat Exchangers”: 96 pages, 23 figures, 6 tables, 51 references, 11 presentation slides.

The master’s thesis is aimed at solving a scientific and applied problem, namely improving the efficiency of thermo-hydraulic processes in a plate heat exchanger, as well as reducing ship fuel consumption during its operation by preventing biofilm fouling of the heat transfer surface.

A scientific hypothesis has been proposed and confirmed, stating that an increase in the efficiency of thermo-hydraulic processes and a reduction in fuel consumption are achieved through the rational selection of a method for preventing fouling of the heat exchange surface.

For a vessel equipped with a 12V Wärtsilä 46F main engine and auxiliary power systems, a GEA NT (Marine Duty) NT250S plate heat exchanger with a heat transfer surface area of 160 $m^2$ was calculated and selected to remove heat from fresh water to seawater. It has been shown that the presence of fouling on both the fresh water side and the seawater side significantly reduces the amount of transferred heat (approximately from 4700 kW to 2230 kW) and increases pressure losses.

It has been demonstrated that additional fuel costs resulting from increased hydraulic losses and reduced thermal load due to fouling of the seawater-side heat exchange surface amount to 117 USD/month and 575 USD/month for biofilm thicknesses of 0.2 mm and 0.5 mm, respectively. It has been established that the most environmentally and economically feasible method for preventing biofouling is periodic electrolytic chlorination. It has been shown that the costs of biofouling prevention (128 USD per month) are recovered already at a biofilm thickness of 0.2 mm.

HEAT EXCHANGE EQUIPMENT, SEAWATER AS A HEAT TRANSFER MEDIUM; BIOLOGICAL FOULING OF THE HEAT EXCHANGE SURFACE, HEAT TRANSFER COEFFICIENT, HYDRAULIC LOSSES, FUEL SAVINGS

# CONTENTS

## NOMENCLATURE

AC — Air Conditioning

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

BPHE — Brazed Plate Heat Exchanger

CAPEX — Capital Expenditures

CII — Carbon Intensity Indicator

CIP — Cleaning-in-Place

$CO_2$ — Carbon Dioxide

CuNi — Copper–Nickel Alloy

ECA — Emission Control Area

EEDI — Energy Efficiency Design Index

EPS — Extracellular Polymeric Substances

FW — Fresh Water

GPHE — Gasketed Plate Heat Exchanger

HFO — Heavy Fuel Oil

HT — High Temperature (freshwater cooling circuit)

HVAC — Heating, Ventilation and Air Conditioning

HX — Heat Exchanger

IMO – International Maritime Organization

IMO — International Maritime Organization

ISS — Internal Seawater System

LO — Lubricating Oil

LT — Low Temperature (freshwater cooling circuit)

MARPOL — International Convention for the Prevention of Pollution from Ships

MCR — Maximum Continuous Rating

MDO — Marine Diesel Oil

MDPI — Multidisciplinary Digital Publishing Institute

ME — Main Engine

MEPC — Marine Environment Protection Committee

MGPS — Marine Growth Prevention System

MIC — Microbiologically Influenced Corrosion

MSD — Medium-Speed Diesel engine

$NO_x$ — Nitrogen Oxides

OPEX — Operational Expenditures

ORC — Organic Rankine Cycle

PHE — Plate Heat Exchanger

PID — Proportional–Integral–Derivative (controller)

RFR — Required Freight Rate

SCR — Selective Catalytic Reduction

SFOC — Specific Fuel Oil Consumption

$SO_x$ — Sulfur Oxides

STHE — Shell-and-Tube Heat Exchanger

SW — Seawater

VLSFO — Very Low Sulfur Fuel Oil

# INTRODUCTION

The maritime shipping industry's efficiency hinges on its internal seawater systems (ISS) and heat exchange equipment, which are vital for main engine cooling, power generation, and refrigeration. Failures in these sophisticated components—including plate and shell-and-tube exchangers—due to the harsh raw seawater environment, pose significant threats to vessel safety and economic viability [1, 2].

The primary operational challenge is fouling, the undesirable accumulation of biotic (biofouling) and abiotic deposits. Biofouling, starting with a microbial biofilm and progressing to macrofouling (mussels, barnacles), severely degrades system performance in two ways. First, deposits create thermal resistance, where a 1 mm film can reduce heat transfer efficiency by 50% [1]. Second, accumulation causes hydraulic resistance and substantial head loss due to occlusion, particularly in the narrow gaps of heat exchangers [1, 2].

The severity of this issue leads directly to the core research motivation: Untimely cleaning of heat exchange equipment and poor chemical water treatment (if heat exchangers operate with water) reduce the thermal efficiency of the apparatus by 50–70%. During the design of marine heat exchange apparatus, the critical calculations are the thermal and hydraulic system calculations, which are performed with a margin for fouling and corrosion. These same calculations can be used to assess the influence of fouling on the heat exchange surface's operational efficiency during their exploitation. Unmanaged fouling leads to chronic inefficiency that rapidly escalates to acute system failure. As fouling thresholds are exceeded, modern engine room systems (which rely on PID controllers for initial compensation) become overwhelmed, particularly in warmer waters [2]. This results in dangerous engine overheating, forcing automatic vessel slowdown and triggering significant economic losses, as the Required Freight Rate (RFR) can increase by over 80% during forced slowdown conditions [2]. Furthermore, fouling accelerates aggressive forms of corrosion, such as Microbiologically Influenced Corrosion (MIC), which compromises the structural integrity of piping and heat exchanger tubes, leading to leaks and unscheduled downtime [1, 3].

To combat this, the industry employs proactive solutions like Marine Growth Prevention Systems (MGPS) and specialized coatings, alongside physical and chemical cleaning regimes [1,3,4]. However, the efficacy and environmental impact of these solutions in complex marine systems require ongoing investigation. The ultimate challenge is developing rational, cost-effective, and environmentally compliant methods to maintain system efficiency.

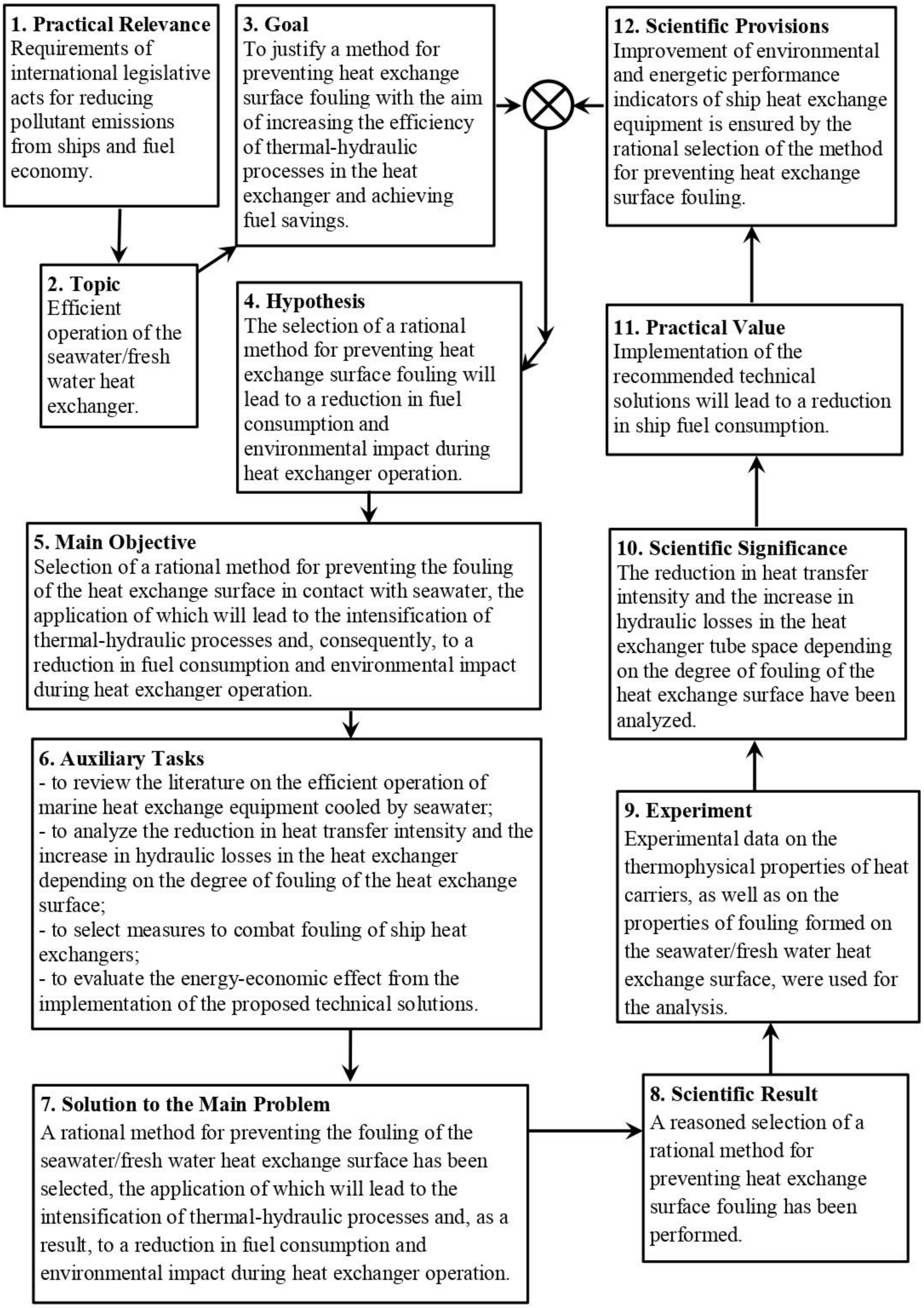
The objective of this research is to analyze the impact of heat exchange surface fouling during their operation in marine conditions on the intensity of heat transfer and head loss, and to select and justify rational methods for fouling prevention.

To achieve this objective, the following tasks have been formulated:

- to review the contemporary literature on the efficient operation of marine heat exchange equipment, the influence of various types of fouling, including biological, on its performance, and an overview of methods for preventing and removing these foulants;
- to derive calculation dependencies for heat transfer intensity and hydraulic losses in heat exchangers of various designs based on the thickness of fouling on the heat exchange surface;
- to consider promising measures for fouling control in heat exchangers and provide an economic justification for the expediency of implementing specific fouling control measures.

This project will integrate theoretical thermal-hydraulic modeling with practical material science and contemporary anti-fouling technologies to develop a comprehensive framework for optimizing marine heat exchanger performance.

# TECHNOLOGICAL CARD OF RESEARCH



**1. Practical Relevance**
Requirements of international legislative acts for reducing pollutant emissions from ships and fuel economy.

**2. Topic**
Efficient operation of the seawater/fresh water heat exchanger.

**3. Goal**
To justify a method for preventing heat exchange surface fouling with the aim of increasing the efficiency of thermal-hydraulic processes in the heat exchanger and achieving fuel savings.

**4. Hypothesis**
The selection of a rational method for preventing heat exchange surface fouling will lead to a reduction in fuel consumption and environmental impact during heat exchanger operation.

**5. Main Objective**
Selection of a rational method for preventing the fouling of the heat exchange surface in contact with seawater, the application of which will lead to the intensification of thermal-hydraulic processes and, consequently, to a reduction in fuel consumption and environmental impact during heat exchanger operation.

**6. Auxiliary Tasks**
- to review the literature on the efficient operation of marine heat exchange equipment cooled by seawater;
- to analyze the reduction in heat transfer intensity and the increase in hydraulic losses in the heat exchanger depending on the degree of fouling of the heat exchange surface;
- to select measures to combat fouling of ship heat exchangers;
- to evaluate the energy-economic effect from the implementation of the proposed technical solutions.

**7. Solution to the Main Problem**
A rational method for preventing the fouling of the seawater/fresh water heat exchange surface has been selected, the application of which will lead to the intensification of thermal-hydraulic processes and, as a result, to a reduction in fuel consumption and environmental impact during heat exchanger operation.

**8. Scientific Result**
A reasoned selection of a rational method for preventing heat exchange surface fouling has been performed.

**9. Experiment**
Experimental data on the thermophysical properties of heat carriers, as well as on the properties of fouling formed on the seawater/fresh water heat exchange surface, were used for the analysis.

**10. Scientific Significance**
The reduction in heat transfer intensity and the increase in hydraulic losses in the heat exchanger tube space depending on the degree of fouling of the heat exchange surface have been analyzed.

**11. Practical Value**
Implementation of the recommended technical solutions will lead to a reduction in ship fuel consumption.

**12. Scientific Provisions**
Improvement of environmental and energetic performance indicators of ship heat exchange equipment is ensured by the rational selection of the method for preventing heat exchange surface fouling.

# 1 REVIEW OF THE INFLUENCE OF OPERATING PARAMETERS ON THE EFFICIENCY OF SHIP HEAT EXCHANGERS

## 1.1 Heat Exchangers as a Key Component of Marine Auxiliary Systems

Heat exchangers are fundamentally integrated into virtually every critical auxiliary system aboard a commercial vessel. These devices are essential for managing and controlling thermal energy flows to ensure the safe, efficient, and reliable operation of the main propulsion system, power generation, and habitability services. The necessity for these components arises from the continuous need to dissipate waste heat or transfer heat between various fluid circuits. Without reliable heat exchange, key systems would rapidly overheat or operate outside their specified thermal limits, leading to potential catastrophic failure and loss of propulsion.

The primary function of heat exchangers (HXs) on a ship is thermal control, predominantly involving the transfer of heat from high-temperature circuits to the readily available heat sink—seawater. This function is vital across several major areas of application, ensuring operational continuity.

In Main Engine Cooling Systems, HXs are arguably the most critical component, transferring immense heat from the engine’s internal freshwater cooling circuits (High Temperature - HT and Low Temperature - LT) to the cold seawater [5]. Failure here directly necessitates an automatic engine slowdown (derating) to prevent catastrophic damage [2]. Similarly, Lubricating Oil (L.O.) Coolers must continuously remove heat absorbed by the oil as it cools and lubricates engine components, a process crucial for maintaining the oil's viscosity and preventing premature wear and serious damage.

Beyond cooling, HXs are used for heating, notably in Fuel Oil Heating and Treatment, where Heavy Fuel Oil (HFO) requires precise temperature control to reduce viscosity for effective purification and efficient combustion. Furthermore, they are indispensable for Auxiliary Machinery Cooling, managing the heat generated by essential equipment such as diesel generators, air compressors, hydraulic systems, and refrigeration units.

Heat exchangers also play a role in Habitability and Water Production. In Air Conditioning and Refrigeration (HVAC/Provisions), they function as condensers and evaporators, transferring heat to the cooling medium (seawater) and absorbing heat from the cooled spaces, respectively. For Freshwater Generation (Evaporators), plate heat exchangers often utilize waste heat from the main engine's HT circuit to evaporate seawater under vacuum, producing potable and technical freshwater for the ship's needs.

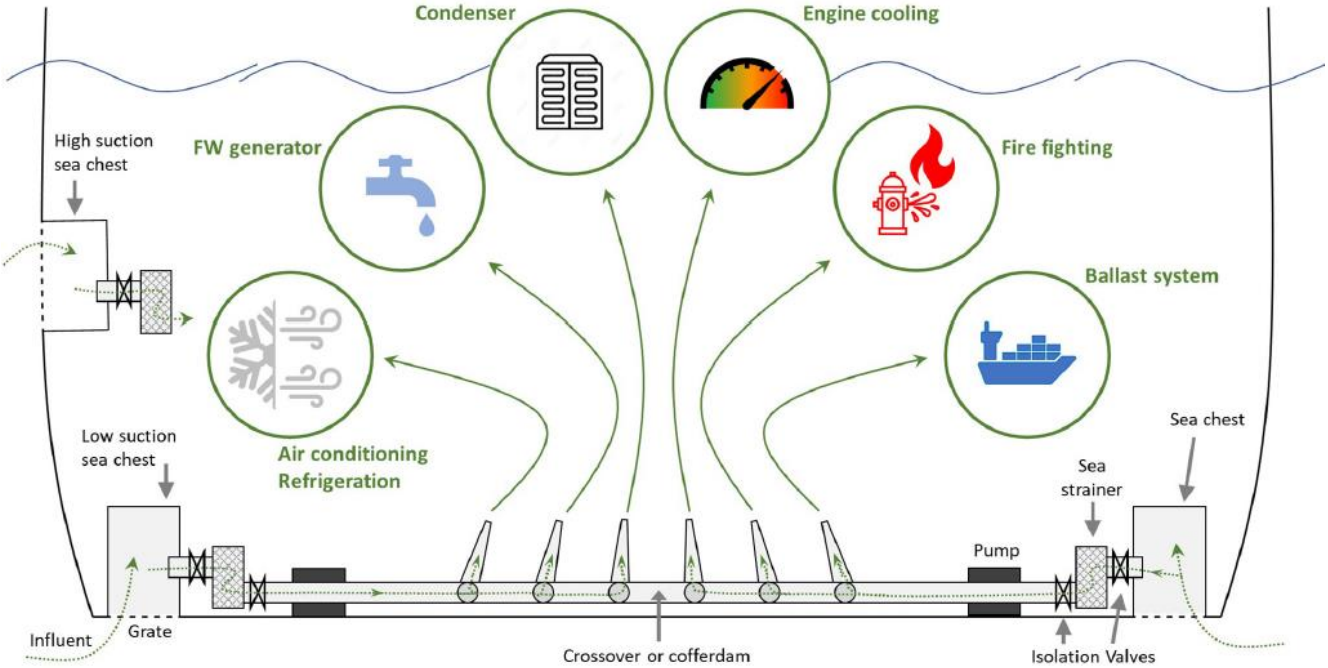


Fig 1.1 Schematic of initial sea water passage through an integrated ISS, depicting the range of uses for sea water on board a ship [1]

## 1.2 Types of Marine Heat Exchanger and the Importance of Reliable and Efficient Operation

Marine auxiliary systems primarily employ two main categories of heat exchange apparatus, selected based on capacity, pressure, and maintenance requirements. Plate Heat Exchangers (PHEs), including gasketed and brazed designs, are common in modern vessels due to their high thermal efficiency and compact nature. They consist of plates with complex patterns (like GEA’s OptiWave) creating narrow flow channels, offering high surface-to-volume ratio and modularity for easy maintenance and capacity expansion [4]. Conversely, Shell-and-Tube Heat Exchangers (STHEs) have been the traditional choice, particularly for robust design requirements dealing with higher pressures. Though generally less thermally efficient and larger than PHEs, STHEs are less susceptible to blockage by large debris. In general, heat exchangers are not mere auxiliary components but essential, high-impact elements that dictate the thermal performance and, ultimately, the safety and profitability of the vessel's operations. Their operational integrity is therefore paramount, making fouling control a non-negotiable aspect of maritime engineering.

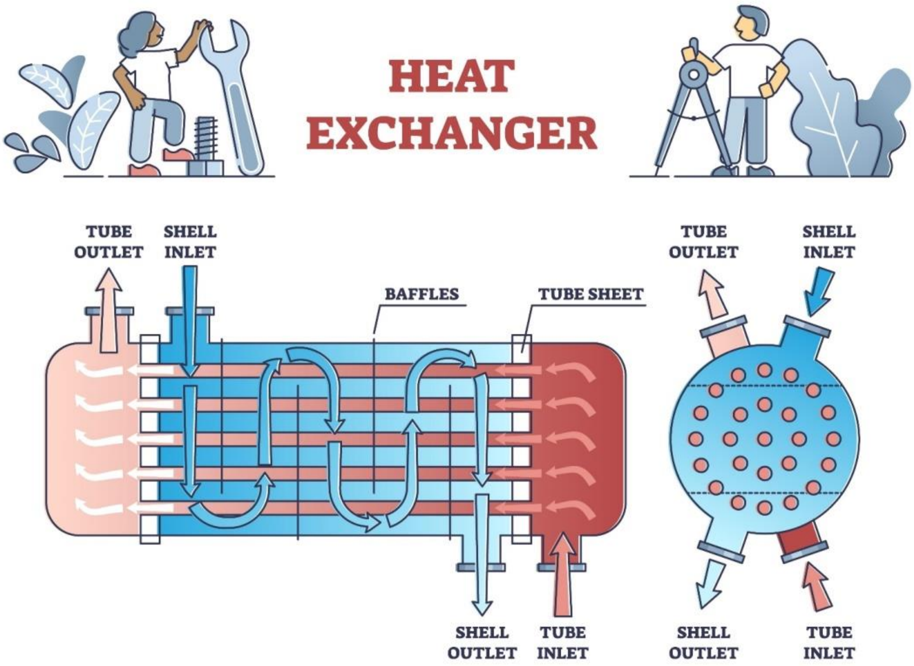


Fig 1.2. Basic Design and Flow Scheme of a Heat Exchanger

***Shell-and-Tube Heat Exchangers (STHEs)***

Shell-and-tube heat exchangers have historically been the traditional and dominant choice in marine engineering. They consist of a bundle of tubes housed within a cylindrical shell. One fluid flows through the tubes, and the second fluid flows outside the tubes, within the shell, facilitating heat transfer across the tube walls.

*Design and Structure:* STHEs are characterized by their robust construction, which makes them highly durable and resistant to high pressures and temperatures. Their design is relatively simple, allowing for easy mechanical cleaning of the tubes.

*Application:* They are typically used in applications where the operating fluid has a higher potential for clogging with large debris, such as the initial stages of seawater cooling, or where a heavy-duty, reliable design is mandatory.

*Limitations:* Despite their strength, STHEs are generally larger and heavier than modern alternatives and exhibit lower overall heat transfer coefficients due to less turbulent flow patterns compared to PHEs.

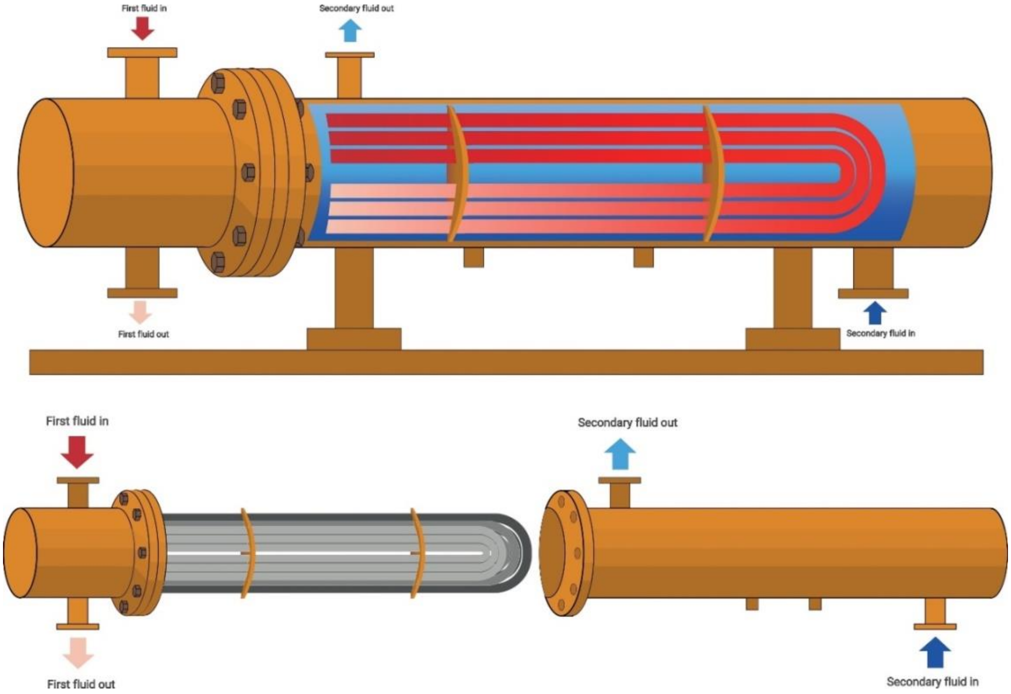


Fig 1.3 Schematic view of Shell-and-Tube Heat Exchanger

***Plate Heat Exchangers (PHEs)***

Plate heat exchangers have gained widespread acceptance and prominence in modern marine systems due to their superior thermal performance and compact footprint. PHEs, such as the gasketed and brazed variations offered by manufacturers like GEA [4], utilize a series of thin, corrugated metal plates assembled between a frame and a pressure plate. The efficiency of PHEs is highly dependent on the precision of the plate patterns, which are crucial for achieving the necessary turbulence at lower flow rates

*Design and Structure:* The fluids flow in alternate channels created by the corrugated plates. The complex patterns (e.g., GEA's OptiWave) induce high levels of turbulence, significantly increasing the heat transfer coefficient and minimizing the formation of stagnant layers [4].

*Advantages:* PHEs offer extremely high thermal efficiency and are highly compact, often requiring only a fraction of the space needed for an STHE of similar capacity. Gasketed PHEs are also easily disassembled for cleaning and inspection, which is a major advantage for maintenance in a fouling-prone environment [4].

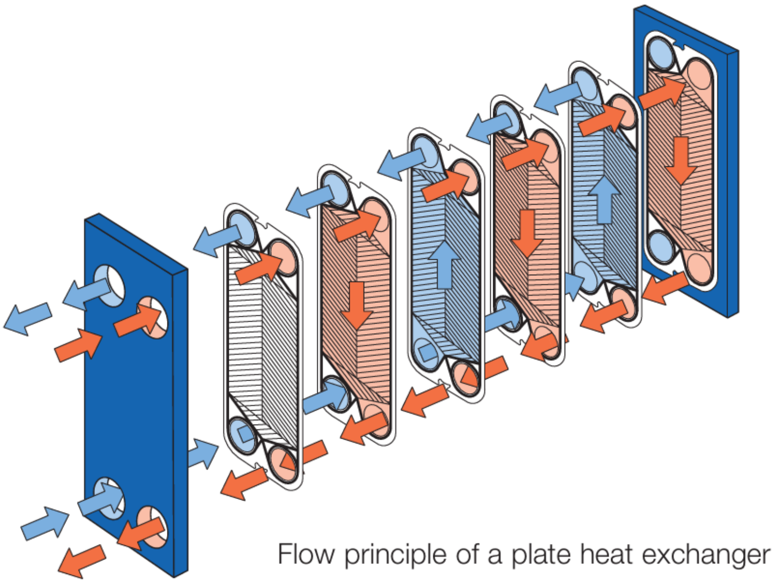


Fig 1.4 Schematic view and flow principle of a of Plate Heat Exchangers [6]

Table 1.1 - Comparison of shell-and-tube and plate heat exchangers

| Parameter | Plate Heat Exchangers (PHE) | Shell-and-Tube Heat Exchangers (S&THE) |
|---|---|---|
| Overall heat transfer coefficient | 5000–9000 W/m$^2$K – high due to turbulence and thin plates (advantage) | 800–1800 W/m$^2$K – lower due to geometry limits, tube fouling, weaker turbulence (limitation) |
| Minimum temperature approach | 1–3 °C (advantage) | 10–20 °C (limitation) |
| Maximum pressure | 16–25 bar (up to 40 bar for semi-welded) (limitation) | 40–150 bar (advantage) |
| Maximum temperature | 150–180 °C (limitation) | 300–450 °C (advantage) |
| Size & weight | Very compact; 5–10× smaller compare to S&THE (advantage) | Large footprint & heavy (limitation) |
| Pressure drop | High: 60–120 kPa (limitation) | Low: 10–25 kPa (advantage) |
| Fouling resistance | Moderate; narrow 2–5 mm channels (limitation) | High; mechanical cleaning possible (advantage) |
| Maintenance | Easy to expand; requires gasket replacement (mixed) | Difficult; tube bundle removal required (limitation) |
| Vibration resistance | Lower; gaskets sensitive (limitation) | High; robust construction (advantage) |
| Cost | Lower CAPEX & OPEX (Capital Expenditures & Operational Expenditures) (advantage) | Higher cost; expensive alloys (limitation) |
| Applications | **HT/LT cooling,** HVAC, ORC, heat recovery (neutral) | Steam, high-pressure, chemical media (neutral) |
| Thermal efficiency | 90–95% (advantage) | 60–75% (limitation) |

## 1.3 Fouling and Biofouling of Heat Exchanger Surfaces: Problems and Mitigation Strategies

The reliable and efficient operation of heat exchange apparatus in marine auxiliary systems is fundamentally and relentlessly challenged by fouling and biofouling. This pervasive phenomenon is universally recognized as the single largest operational obstacle to thermal control in all systems utilizing seawater, leading directly to reduced safety margins, accelerated component degradation, and devastating economic consequences [2, 3, 7]. Effective management of fouling is paramount to minimizing fuel consumption, reducing greenhouse gas emissions in line with IMO targets [8], and ensuring the longevity of critical onboard equipment.

### 1.3.1 The Fouling Problem: Mechanism, Consequences, and Criticality

Fouling is the undesirable deposition and accumulation of biotic and/or abiotic materials on heat transfer surfaces. This process immediately compromises the pristine design specifications of the apparatus, transforming the metal surface into a thermal barrier. The cumulative damage extends beyond mere inefficiency, encompassing severe hydraulic, structural, and economic consequences.

The core of the fouling problem lies in the formation of a thermal barrier. The fouling layer, regardless of its composition (be it microbial biofilm, mineral scale, or metallic oxides), introduces significant thermal resistance ($R_f$) between the working fluids, which drastically reduces the overall heat transfer coefficient (U) of the heat exchanger. Although marine HXs are designed with a fouling margin, this design safety factor is often quickly consumed in operation. The correlation between fouling thickness and efficiency loss is profoundly disproportionate: studies confirm that fouling layers, including biological biofilms, measuring only 1 mm in thickness can reduce the overall heat transfer efficiency by 50% or more [1, 3]**.** This thermal decline directly impairs the cooling system's capacity to dissipate heat from the main engine's freshwater circuits, causing operating temperatures to rise uncontrollably [2].

Concurrently, the physical mass and dramatically rougher structure of the accumulated material reduce the cross-sectional flow area (bore) inside tubes or the narrow channels of plate heat exchangers [1]. This severe flow restriction creates a corresponding dramatic increase in hydraulic resistance (head loss) [1]. The resultant pressure drop forces auxiliary pumps to operate at higher loads and consume substantially more electrical power, constituting a chronic source of wasted energy and increased greenhouse gas emissions (due to the associated increase in auxiliary engine load) [2,3].

The operating criticality of the system is highly dependent on ambient conditions. Research indicates a distinct vulnerability in warmer seas: while engine systems can tolerate up to 50% occlusion in cold seawater due to PID controller compensation, this tolerance drops dramatically to near zero (less than 10% occlusion) when ambient seawater temperatures are high (e.g., 25C° and above). This rapid drop in margin leads to fast performance deterioration, a pattern universally described as a "hockey-stick" curve [2].

The degradation caused by fouling links microscopic biological activity to macroscopic economic and structural failure, creating significant operational and regulatory risks. Structural Damage and Corrosion are accelerated as fouling layers create localized anaerobic zones on metallic surfaces. Microorganisms, particularly Sulfate-Reducing Bacteria, within these layers initiate and accelerate Microbiologically Influenced Corrosion (MIC) [3, 7]. MIC releases potent corrosive metabolic by-products that rapidly compromise the integrity of metallic components, leading to localized "impingement attack" on heat exchanger tubes and pipework (often Cupronickel alloys), resulting in leaks, unscheduled repair, and potential flooding [1].

The acute loss of temperature control, once the thermal capacity margin is exceeded, triggers an emergency engine slowdown (derating) by the automated monitoring system, preventing catastrophic engine failure [2]. This necessary loss of propulsion and maneuverability poses a severe safety hazard, particularly when a vessel is operating in congested or restricted waterways, or during adverse weather conditions [1, 2]. The collective consequences result in devastating economic losses: operational modeling demonstrates that problems caused by fouling occlusion drastically increase the voyage's Required Freight Rate (RFR) — a critical measure of profitability. When forced into engine slowdown mode, the RFR can increase by over 80% [2], necessitating costly unscheduled repairs and substantial revenue loss. Furthermore, the chronic inefficiency and increased fuel consumption directly impact the vessel’s compliance with the IMO EEDI (Energy Efficiency Design Index) and CII (Carbon Intensity Indicator) regulations [8]. This critical linkage translates the operational fouling problem into a high-stakes regulatory and financial risk.

### 1.3.2 Classification of Fouling on Heat Exchange Surfaces

Fouling is a complex, dynamic process involving a mixture of physical, chemical, and biological mechanisms, often occurring simultaneously. These mechanisms are broadly categorized into biological and non-biological types.

Biological Fouling (Biofouling): The Dominant Marine Threat. Biofouling is recognized as the most significant and aggressive form of fouling in seawater-cooled systems, occurring in a well-defined sequence [1, 3]. The process begins with Initial Adhesion and Conditioning where an organic layer immediately coats the surface. This is followed by Microfouling (Biofilm Formation) within hours, as motile bacteria and diatoms adhere, multiply, and secrete a sticky, highly permeable Extracellular Polymeric Substance (EPS) matrix, forming the resilient biofilm (or slime layer) [1, 7]. The EPS matrix functions as a complex hydrogel, trapping particulates and protecting underlying bacteria from biocide penetration and hydrodynamic shear stress [9]. Critically, the creation of local anaerobic zones beneath the biofilm directly facilitates MIC [10]. Finally, Macrofouling Establishment occurs as the established biofilm acts as a necessary substratum for the subsequent settlement and metamorphosis of larger organisms' larvae and spores. This includes hard-bodied species (mussels, barnacles, tube worms) and soft-bodied organisms (hydroids) [1, 7]. Macrofouling accumulation creates major physical flow restriction, severely exacerbating the hydraulic resistance problem and contributing to blockages.

Non-Biological Fouling Types: Physical, Chemical, and Cryogenic. Non-biological deposits frequently provide a base for biofouling and severely contribute to efficiency loss through insulation and flow restriction [7]. Precipitation Fouling (Scaling) occurs when inorganic salts, primarily Calcium Carbonate ($CaCO_3$), precipitate onto the heat transfer surface, forming a hard, crystalline layer. Its inverse solubility with temperature makes scaling particularly severe on hot surfaces (e.g., in freshwater generator evaporators or high-temperature condensers), following fundamental ASHRAE principles [11]. Other types include Particulate and Corrosion Fouling, where suspended solids (silt, sand) and metallic oxides accumulate, respectively [1]. Chemical Reaction Fouling deposits are formed from undesirable chemical reactions at the surface [7]. Finally, Freezing Fouling is a critical concern in dedicated refrigeration (HVAC, provision stores) and oil cooling systems. This involves the solidification of fluid constituents, such as wax in oil coolers, or frost and ice accumulation on the air side of provision refrigeration evaporators. This ice acts as a severe thermal barrier, necessitating periodic defrosting (a form of reactive maintenance) to restore capacity. The design and operation of these low-temperature systems are governed by stringent standards like ANSI/ASHRAE 26 [12].

### 1.3.3 Methods of Fouling Control and Mitigation

Controlling fouling and biofouling is a critical engineering priority, necessitating a holistic and integrated approach involving both proactive (preventative) strategies and reactive (cleaning and restoration) methods.

1) Proactive Fouling Prevention Strategies. Preventative methods aim to inhibit initial adhesion and subsequent growth, maximizing apparatus uptime and efficiency. This starts with Optimized Design, Material Selection, and Hydrodynamic Control**.** Utilizing anti-corrosive and anti-fouling materials is crucial: Cupronickel alloys (e.g., Cu-Ni 90/10) release controlled amounts of copper ions, providing an effective localized biocidal environment, provided the flow velocity is kept high enough [7, 13]. Titanium is highly favored for its exceptional corrosion and erosion resistance [13, 14]. Hydrodynamic design is equally vital; PHE manufacturers design complex corrugated patterns to promote high fluid velocity and turbulence, which actively discourages the

initial adhesion of microbial films [4]. Furthermore, maintaining high operational flow velocities generates high shear stress, which physically strips away nascent fouling layers from pipe walls. Complementing this are Marine Growth Prevention Systems (MGPS) and Chemical Treatment. MGPS are essential electrochemical systems installed at the seawater intake [4, 15]. Copper Anodes release controlled concentrations of copper ions, which act as a biocide to prevent the settlement of marine larvae and spores throughout the ISS piping and HXs [7, 15]. Iron/Aluminum Anodes dissolve to form a protective colloidal oxide film that coats the internal pipe walls, inhibiting corrosion [15]. Complementary chemical dosing via Chlorination/Electro-chlorination may also be used, generating hypochlorite *in situ* from seawater to kill microorganisms [7]. Lastly, Mechanical Online Cleaning Systems (like the Taprogge system for STHEs) circulate sponge rubber balls through the tubes periodically, which abrasive action removes soft deposits and newly formed scale/biofilm without interrupting operation.

2) Reactive Fouling Removal and Restoration. Once fouling has accumulated to a point where efficiency drops significantly, reactive removal is necessary to restore performance. Cleaning protocols are highly dependent on the exchanger's design and are labor-intensive. Mechanical Cleaning and Disassembly is mandatory for Gasketed Plate Heat Exchangers (GPHEs) and accessible STHEs. GPHEs are disassembled by unclamping the plate stack, allowing plates to be individually cleaned using soft brushes or high-pressure water jets [4, 15]. STHEs typically require removing the water box heads or extracting the entire tube bundle (in floating head designs) for mechanical cleaning using specialized brushes, scrapers, or high-pressure water lances [15, 16]. Alternatively, Chemical Cleaning (Cleaning-in-Place - CIP) involves circulating specialized chemical solutions through the exchanger without disassembling it. This method is the preferred or mandatory approach for permanently sealed units like Brazed Plate Heat Exchangers (BPHEs) and air coolers [7, 15]. Acid cleaning is used to dissolve tenacious mineral scale, while Biocidal/Dispersant cleaning is used to break up soft organic and biological films. The success of CIP must always be verified by measuring the restoration of pressure drop and temperature performance.

### 1.3.4 Conclusion on Fouling Management

The management of heat exchanger fouling is a continuous and complex engineering challenge driven by necessity, directly impacting safety, environmental compliance, and profitability. The problem links microscopic biological processes (biofilm formation) to macroscopic economic and safety consequences (engine derating and RFR increase). The optimal approach is an integrated one, combining robust initial design and anti-fouling materials (CuNi, Titanium) with proactive continuous treatment systems (MGPS, turbulence-inducing designs) and efficient, planned reactive cleaning regimes (CIP or mechanical disassembly) tailored to the specific type of apparatus. This holistic strategy is vital for maximizing the thermal efficiency required for safe vessel operation and regulatory compliance with global environmental standards.

# 2. CALCULATION AND SELECTION OF AUXILIARY SYSTEMS FOR THE ENGINE ROOM OF MV NEPTUNE KALLOS

All calculations given in this section were performed in accordance with the Rules for Classification and Construction of Seagoing Vessels of the Ukrainian Register of Shipping. [17].

The initial data for the calculations are taken according to the information for the prototype vessel MV NEPTUNE KALLOS, and the main engine 12V Wärtsilä 46F, given in sections 2.1 and 2.2.

## 2.1 Characteristics of the prototype vessel MV NEPTUNE KALLOS

The prototype is a specialized Ro-Ro vessel for transporting cars in numbers equivalent to 3,800 units. The vessel NEPTUNE KALLOS (IMO 9442122) is a vehicle built in 2010 and currently sailing under the flag of Malta.

Table 2.1 Characteristics of the prototype vessel MV NEPTUNE KALLOS

| | |
|---|---|
| Vessel name | NEPTUNE KALLOS |
| Vessel type | Ro-Ro |
| IMO number | 9442122 |
| Year built | 2010 |
| Overall length, m | 168,0 |
| Length between perpendiculars, m | 158,0 |
| Breadth, m | 28,00 |
| Height from keel to upper deck, m | 30.65 |
| Height from keel to freeboard, m | 14.00 |
| Draft to summer load line, m | 8.71 |
| Lightweight displacement, t | 11824 |
| Loaded displacement, t | 24100 |
| Deadweight, t | 12250 |
| Main engine type | 12V Wärtsilä 46F |
| Fuel type ISO 8217-2012 | IFO 380 |

Continuation of Table 2.1

| | |
|---|---|
| Specific calorific value of fuel, kJ/kg | 41200 |
| Speed at summer draft (NOR 15 % S.M.), knots | 19,6 |
| Propeller type | fixed |
| Number of blades/number of propellers | 5/1 |

## 2.2 Main engine characteristics 12V Wärtsilä 46F

*General Description and Application*

The Wärtsilä 12V46F engine is a prominent member of the Wärtsilä 46F medium-speed engine series, which has established itself as a standard for reliability and efficiency in the marine industry. This model was designed to provide high power density while maintaining fuel efficiency and compliance with strict environmental regulations.

The 12V46F engine is widely used as a prime mover on various types of vessels, including container ships, tankers, bulk carriers, cruise liners, and ferries. Due to its modular design and high reliability indicators, it is also applied on floating production storage and offloading (FPSO) units and power plants.

*Engine Type*

The Wärtsilä 12V46F is classified as:

- 4-stroke: The working cycle is completed in four piston strokes (intake, compression, power, exhaust).
- Diesel engine: Operates on the principle of compression ignition.
- Medium-speed: The rated speed is 600 rpm, which requires the use of a reduction gear to transmit torque to the propeller.
- V-configuration: Cylinders are arranged in two banks at an angle to each other (V-configuration), allowing for a reduced engine length while maintaining high power.
- Trunk piston: Uses a trunk piston design, as opposed to crosshead engines.
- Turbocharged and intercooled: Equipped with a high-efficiency turbocharging system to increase power and efficiency.

- Non-reversible: Typically operates in one direction of rotation (reversing the vessel's direction is achieved via a controllable pitch propeller or a reversing gearbox).

Technical Data is presented in Table 2.2.

Table 2.2 The main technical parameters of the Wärtsilä 12V46F engine

| Parameter | Value | Unit |
|---|---|---|
| Number of cylinders | 12 | pcs |
| Configuration | V-form | - |
| Cylinder Bore | 460 | mm |
| Piston Stroke | 580 | mm |
| Swept volume per cylinder | 96.4 | L |
| Maximum Continuous Rating (MCR) | 14,400 | kW |
| Output per cylinder | 1,200 | kW/cyl |
| Rated speed | 600 | rpm |
| Mean Effective Pressure (MEP) | 24.9 - 25.9 | bar |
| Piston speed | 11.6 | m/s |
| Specific Fuel Oil Consumption (SFOC) | 170 - 175 | g/kWh |
| Lubricating oil consumption | 0.6 | g/kWh |

*Cylinder Block and Crankshaft*

The engine block is made of nodular cast iron, providing high stiffness and strength. The crankshaft is of the underslung type, allowing for effective load distribution.

*Piston and Cylinder Liner*

- Piston: Consists of a steel crown and a nodular cast iron skirt. It is oil-cooled using the shaker cooling method.
- Anti-Polishing Ring: The cylinder liner is equipped with a specific calibration ring at the top. This ring removes carbon deposits from the piston crown, preventing "bore polishing," which significantly reduces lube oil consumption and wear.

*Fuel Injection System*

The engine can be equipped with a traditional twin injection pump system or a

Common Rail system. Twin Injection Pump uses two plungers in one pump to ensure high injection pressure and timing control at various loads. This allows for smokeless exhaust even at low loads. The engine is designed to run on Heavy Fuel Oil (HFO) with viscosity up to 700 cSt/50°C, Marine Diesel Oil (MDO/LFO), and is also adapted for alternative fuels (methanol, biofuels) in specific modifications.

Valve Mechanism and Turbocharging

A 4-valve cylinder head technology (2 inlet and 2 exhaust valves) is used. Turbochargers (typically located at the flywheel end or free end) provide the high boost pressure required for operation on the Miller cycle, which reduces combustion temperature and NOx emissions.

*Dimensions and Weight*

- Length (approx.): 11,080 - 11,150 mm
- Width (approx.): 3,800 - 4,050 mm
- Height (approx.): 3,800 - 4,050 mm
- Dry weight (approx.): 177 - 178 tonnes

*Environmental Compliance*

In its standard configuration, the engine complies with IMO Tier II requirements for nitrogen oxide (NOx) emissions. To meet IMO Tier III standards (required in ECA zones), the engine can be equipped with a Selective Catalytic Reduction (SCR) system or a scrubber to clean exhaust gases of sulfur oxides (SOx), depending on the fuel used. High thermal efficiency and an optimized combustion process ensure low $CO_2$ emissions and no visible smoke in most operating modes.

## 2.3 Hourly fuel consumption of the main engine 12V Wärtsilä 46F

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

$$Q_e = 0.164 \cdot 8600 = 1410{,}4 \text{ kg/h}$$

where $g_e$ – is the specific effective fuel consumption, kg/(kWh);

ΣNe – total effective power of the main engines, kW.

The amount of heat released during fuel combustion:

$q = Q_e \cdot Q_н$

$q = 1410{,}4 \cdot 41400 = 58390560$ kJ/h,

where $Q_н$ – is the lower calorific value of the fuel, kJ/kg.

For the fuel of the 12V Wärtsilä 46F engine, we assume $Q_н$ = 41400 kJ/kg.

## 2.4 Main Engine Auxiliary Machinery and Equipment 12V Wärtsilä 46F

The values of the flow rates of working fluids in the MV NEPTUNE KALLOS ship systems, which are necessary for calculating the diameters of pipelines:

- fuel pumping system – 1.6 m/s,
- fuel in the pumping 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, purifing, heating and supplying fuel to the main and auxiliary engines and boilers, as well as for pumping it to the shore or another vessel.

The 12V Wärtsilä 46F diesel is designed to operate on heavy fuel grades, that require ship power plants to be treated with a special fuel preparation system. The system includes heavy fuel and diesel fuel separators, steam heaters and, equipped with thermostats, settling and ventilating tanks for heavy fuel and diesel fuel, coarse and fine filters. Each separator has two paired fuel pumps (for injection and pumping). MV NEPTUNE KALLOS tanks are considered a steam heating system. The fuel is pumped to the settling tank, from where, after settling for 20-24 hours, it is pumped to the heater, then to the separators and after - to the supply tank. From the supply tank, fuel is supplied by fuel transfer pumps through a heater to the main 12V Wärtsilä 46F engine. The fuel system also includes pumps for transfering fuel from one tank to another.

**Fuel tanks.** The volume of each of the two heavy fuel oil settling tanks and two service tanks $V_{вт}$ is selected to ensure the operation of the 12V Wärtsilä 46F main engine for a duration of $\tau_1 = 24$ hours.

$V_{вт} = (Q_e \cdot \tau_1)/\rho_m$

$V_{вт} = (1410{,}4 \cdot 24)/ 882 = 38{,}4 \ m^3,$

where $\rho_m$ – fuel density, kg/ $m^3$.

For heavy fuel oil we assume $\rho_m = 960$ kg/$m^3$.

The volume of each of the two diesel fuel settling and consumption tanks is assumed to be 80% of the volume of the two consumption tanks of heavy fuel, e.g. 30.7 m3. The supply of simultaneously operating fuel separators is calculated from the condition of separation of daily fuel consumption for τs.p. from 8 to 12 hours.

$Q_c = V_{вт}/\tau_{c.п.}$

$Q_c = 38{,}4/12 = 3{,}2 \ m^3/hour$

where $\tau_{c.п.}$ assumed to be 12 hours.

**Separators.** Two heavy fuel oil (HFO) separators and one marine diesel oil (MDO) separator are installed. To ensure equipment unification, the diesel fuel separator is selected to be identical to the HFO separators. The selection of the separators is based on the required throughput capacity.

To ensure the purification of heavy fuel oil (HFO) and marine diesel oil (MDO), three identical centrifugal separators of the Alfa Laval S 841 type (or equivalent S-series model) are selected. Two units are designated for HFO treatment (one duty, one standby) and one unit for MDO treatment. This unification simplifies maintenance and spare parts management.

*Technical Description of Alfa Laval S 841 Separator*

The Alfa Laval S 841 is a high-speed centrifugal separator designed for the efficient cleaning of marine fuel oils. It utilizes the proprietary **Alcap technology**, which allows for the automatic adjustment to the nature of the oil, ensuring optimal separation efficiency even with varying fuel densities and viscosities.

*Technical Specifications:*

- Type: Self-cleaning disc-stack centrifuge.
- Max. density of feed: 1010 kg/m$^3$ at 15°C.
- Viscosity range: Up to 700 cSt at 50°C (compatible with modern VLSFO/HFO).
- Throughput capacity: Selected to exceed the engine consumption of 2,545 l/h, ensuring a multiple pass treatment if necessary or operation at optimal efficiency (recommended flow rate for HFO cleaning is typically 15-20% of the maximum hydraulic capacity).
- Motor power: Approx. 7.5 - 11 kW (depending on specific configuration).
- Sludge discharge: Automated, partial or total discharge.

Operational Role: By installing the Alfa Laval S 841, the fuel treatment system guarantees the removal of catalytic fines (cat fines), water, and other impurities from the fuel before it reaches the service tanks. This protection is critical for the Wärtsilä 12V46F injection system, preventing premature wear of fuel pumps and injectors.

**The fuel transfer pump** must ensure the pumping of fuel from the larger main supply tank $V_з$ in time $\tau_{відк}$ = 4 hours. At the same time, it must ensure the pumping of not less than the daily fuel consumption by the main engines in time $\tau_2$ = 6 hours.

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

$Q_{нп} \geq V_з / \tau_{відк}$

$Q_{нп} \geq 500/4 = 125,0$ m$^3$/h,

$Q_{нп} \geq V_{вт} / \tau_2$ .

$Q_{нп} \geq 30.7/6 \geq 5.1$ m$^3$/h,

where $Q_{нп}$ – fuel booster pump flow, m3/h.

The pressure developed by the $H_{пн}$ pump is 0.4 MPa. The power consumption of the fuel transfer pump drive motor is determined from the formula:

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

Where $Q$ – pump supply, m3/h;

$p$ – pressure created by the pump, MPa;

$\eta$ – pump efficiency;

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

$P=(125,0\cdot 0,4)/(3,6\cdot 0,8)=17,4$ kW.

For fuel transfer there must be two transfer pumps with independent drives, one of which is a backup. The fuel transfer pump for diesel fuel is assumed to be the same as the pump for heavy fuel.

The fuel injection pump flow is calculated using the formula:

$Q_{пн} = (К_{пн} \cdot Q_e)/\rho_m$

$Q_{пн}= (3\cdot 1410\cdot 4)/882 = 4,8$ m$^3$/h

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

$p_{пн}$ – the pressure developed by the pump for 12V Wärtsilä 46F engines is assumed to be from 0,25 to 0,50 МПа. We assume $H_{пн} = 0,30$ MPa.

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

$P= (4,8\cdot 0,30)/(3,6\cdot 0,8)=5.0\ kW$

We accept for the engine of the fuel injection pump drive $\eta = 0,8$.

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

The amount of heat $q_m$, supplied to bring the fuel temperature up to the level of required viscosity:

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

$q_m =4,8 \cdot 882 \cdot 1,8 \cdot (353 -310) = 327495$ kJ/h,

where $c_n$ – specific heat capacity of fuel, from 1,68 to 2,1 kJ/(kg·K);

$T_1$ – initial fuel temperature (approximately 310 K);

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

$T_2 \leq T_{cn} – 10$ °C. The flash point $T_{cn}$ is in the range from 60 to 110 °C.

Fuel heater heat exchange surface area:

$A_m = q_m/(k_m\ \Delta T_m)$,

$$A_m = 327495/(1000 \cdot 68{,}5) = 4{,}8\ m^2,$$

where $k_m$ – heat transfer coefficient, can be taken as 1000 kJ/(m2·h·K).

$\Delta T_m$ - the mean temperature difference in the heat exchanger.

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

$$\Delta T_m = 400 - (310+353)/2 = 68{,}5\ K$$

where $T_s$ – steam temperature at operating pressure, $T_s$ is approximately between 390 and 400 K. We assume $T_s = 400\ K$.

The heater is selected based on its heat exchange surface area.

### 2.4.2 Lubrication system of 12V Wärtsilä 46F

Consists of a circulating oil system and a cylinder lubrication system. The circulating lubrication system provides oil supply to the friction surfaces, as well as cooling of the pistons.

The system consists of tanks for storing oil reserves, waste tanks, oil circulating pumps, filters, separators, oil coolers.

**The volume of the main oil supply tanks** for the circulation system $V_{м.з.}$ is taken based on the specific flow rate of the circulating oil $b_{м.м.}$, which is 0.0002 kg/(kWh) for low-speed diesel engine (LSD), for medium-speed diesel engine (MSD) 0.002, with a 20% service margin.

$$V_{м.з.} = 1{,}2 \cdot b_{м.м} \cdot \Sigma N_e \cdot \tau_{гд} / \rho_м$$

$$V_{м.з.} = 1{,}2 \cdot 0{,}0002 \cdot 8600 \cdot 1200 / 867 = 2{,}9\ m^3,$$

where $\tau_{гд}$ – duration of operation of the main engine 12V Wärtsilä 46F in the calculated voyage, hours.

We assume from the prototype $\tau_{гд}$ = 1200 hours.

$\rho_м$ – density of oil, which is equal to 867 kg/m3.

$V_{м.з.}$ – with 20% reserve per voyage is equal to 3,4 $m^3$.

Volume of tanks for cylinder oil reserve:

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

$V_{ц.з}$= 0,0005 · 8600 · 1200 /867 = 6,0 $m^3$,

where $b_{цм}$ – is the specific cylinder oil consumption, which depends on the engine type, for МОД from $0{,}4 \cdot 10^{-3}$ to $0{,}7 \cdot 10^{-3}$ kg/(kWh), for СОД from $0{,}6 \cdot 10^{-3}$ to $1{,}2 \cdot 10^{-3}$ kg/(kWh). We assume for MSD 12V Wärtsilä 46F: $b_{цм}$ = 0.001 kg/(kWh).

**Circulating oil pump**

Circulating oil pump supply:

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

$$Q_{м.н} = (554700+2919528)/(3{,}0 \cdot 867 \cdot 8) = 167{,}0\ m^3/h,$$

where $c_м$ – the heat capacity of oil can be taken as 3.0 kJ/(kg.K);

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

$q_п$ – heat received by the oil from the piston, kJ/kg;

$\Delta T_м$ – difference in oil temperature at the outlet and inlet of the 12V Wärtsilä 46F engine can be taken as 8 K.

$$q_{тр}=3{,}6 \cdot 10^3 \cdot a_{тр} \cdot N_e \cdot (1- \eta_м)/\eta_м$$

$$q_{тр} =3{,}6 \cdot 10^3 \cdot 0{,}43 \cdot 8600 \cdot (1-0{,}96)/0{,}96 = 554700\ kJ/h.$$

where $a_{тр}$ – the fraction of heat released during friction and transferred by oil is from 0,4 to 0,45. We assume $a_{тр} = 0{,}43$;

$\eta_м$ – efficiency of the main engine 12V Wärtsilä 46F. equals: $\eta_м$ = 0,96.

$$q_п = a_п \cdot q$$

$q_п = 0{,}05 \cdot 58390560 = 2919528$ kJ/h.

where $a_п$ – is the part of the heat transferred from the piston to the oil for МОД from 0,04 to 0,06, for СОД 0.07–0.12. MSD Д 12V Wärtsilä 46F $a_п = 0{,}10$.

The power consumption of the motor of the circulating oil pump drive is determined from the formula:

$P=(167{,}0 \cdot 0{,}25)/(3{,}6 \cdot 0{,}85)=13{,}6$ kW

We assume for the motor of the circulating oil pump drive η = 0,85.

$p_{мм}$ – the pressure created by the pump, from 0.2 to 0.4 MPa.

We assume $p_{мм} = 0{,}25$ MPa

The amount of oil in the system:

$V_{мс} = Q_{мс}/z.$

$V_{мс} = 167{,}0/\,10 = 16{,}70$ $\mathrm{m}^3$,

where $z$ – circulation rate, $\mathrm{h}^{-1}$ at a boot higher than 40 % $z = 10$.

**Waste oil tank volume**

$V_{сц} = r \cdot V_{мс}$

$V_{сц} = 1{,}25 \cdot 16{,}70 = 20{,}9\ m^3$,

where $r$ – is the foaming coefficient, which ranges from 1,2 to 1,3. We take $r = 1{,}25$.

**Oil separator**

Oil separator feed

$Q_{м.с} = V_{мс}/\,\tau_{см}.$

$Q_{м.с} = 16{,}70\,/\,8 = 2{,}1\ m^3/h,$

where $\tau_{см}$ – is the separation time of all the oil in the system, $\tau_{см}$ is from 4 to 8 hours. We take $\tau_{см} = 8$ *год*.

We select an oil separator by feed, and also install one backup of the same type.

To ensure efficient separation of lubricating oil (LO) for the Wärtsilä 12V46F engine with a required capacity of $2{,}1\ m^3/h$, it is advisable to select a separator from the same manufacturer and series as the fuel separators to standardize maintenance procedures and spare parts inventory. Recommended Model: Alfa Laval S 831 (or S 836)

The Alfa Laval S 831 separator is designed for cleaning mineral oils from water and mechanical impurities. Like the S-series fuel models, it utilizes Alcap technology.

Technical Specifications

- Type: Self-cleaning disc-stack centrifuge.
- Application: Suitable for cleaning heavy fuel oils (densities up to 1010 kg/$m^3$ at 15°C) and lubricating oils.
- Viscosity Range: Capable of handling viscosities up to 700 cSt at 50°C.

- Max Bowl Speed: Approximately 10,683 RPM (as indicated in some specific unit data, though this can vary slightly by exact configuration).
- Motor Power: Typically recommended around 7.5 kW to 11 kW depending on the specific installation requirements.
- Control System: Operated via the EPC 60 process controller, which manages separation sequences, sludge discharge, and monitors parameters like oil temperature and water content.

Oil cooler cooling surface:

$$А_м = \frac{q_{тр}+q_п}{k_м \cdot \Delta Т_м},\ m^2,$$

$$A_м = (554700+2919528)/(1000\cdot 12)=289{,}5\ m^2,$$

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

$k_м$ – heat transfer coefficient, we take from 500 kJ/ (m2·h·K) to 1000 kJ/ (m2·h·K). We take = 1000 kJ/ (m2·h·K)

We install two coolers, the surface area of each is 60% of the total, i.e. 173.7 m2.

**Oil transfer pump**

The oil transfer pump supply must be sufficient to supply oil from the main supply tanks to the circulation system for a period of $\tau_м$ from 0.5 to 1 hour:

$$Q_{м.п} = V_{мс} / \tau_м\ ,$$

$$Q_{м.п} = 16{,}70\ /\ 1 = 16{,}7\ \ m^3/h.$$

The supply of oil from the main supply tanks to the circulation system must occur within $\tau_м$ = 1 hour.

The rated power of the oil transfer pump drive is:

$$P = (16.7\cdot 0.2)/(3{,}6\cdot 0.8)=1.2\ kW.$$

$p_{мн}$ – the pressure generated by the pump, taken from 0.2 MPa to 0.3 MPa;

We take $p_{мн} = 0{,}2$ MPa.

The efficiency of the oil transfer pump is taken as η = 0.8.

### 2.4.3 Engine cooling system 12V Wärtsilä 46F

**Seawater pumps**

Heat transferred through the bushings and covers and removed by fresh water:

The seawater pumps circulate water through the fresh water coolers (central coolers), lubricating oil coolers, and charge air coolers. The seawater removes friction heat $q_{тр}$, heat transferred through cylinder liners and cylinder covers $q_{ц}$, heat transferred from pistons $q_{п}$, and charge air heat $q_{н}$.

$q_{ц}= a_{ц}\cdot q\ ;$

$q_{ц}=0,11 \cdot 58390560 = 6422961\ kJ/h;$

where $a_{ц}$ – is the part of the heat transferred through the sleeves and covers; the value of $a_{ц}$ varies from 0.1 to 0.14 for LSD, and from 0.14–0.20 для MSD. We take 0.17.

Fresh water pump flow:

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

$Q_{п.в} = 6422961/(4,2\cdot 10\cdot 1000) = 160,6\ m^3/\ h,$

where $c_{п.в}$ – the heat capacity of fresh water is taken to be 4.2 kJ/(kg K);

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

$\Delta T_{п.в}$ – the difference in fresh water temperature at the inlet and outlet of the 12V Wärtsilä 46F engine, which is typically in the range of 6 to 10 K.

Nominal power of the fresh water circuit pump drive:

$P=(Q_{п.в}\cdot p_{п.в})/(3,6\cdot \eta),$

$P=(160,6\cdot 0,3)/(3,6\cdot 0,9)=14,9\ kW.$

where $p_{п.в}$ – the pressure created by the pump is taken from 0.2 to 0.4 MPa;

We take $p_{пв} = 0,3\ MPa$;

The efficiency of the piston pump is taken to be η = 0.9.

The flow rate of the seawater pump 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+(0.6·2919528)+2335622)/(4,2·11·1025) = 245.3 m$^3$/h.

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

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

$\Delta T_{з.в}$ – the difference in temperature of 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* – is a coefficient, accepted for turbochargers from 0,5 (with deep recycling) to 1 (without recycling). We accept r = 0.6.

$q_н = a_п \cdot q$,

$q_н = 0{,}04 \cdot 58390560 = 2335622$ kJ/h,

where $a_п$ – is a coefficient, when cooling pistons with fresh water it lies in the interval from 0.03 to 0.05; we take $a_п$ = 0,04.

$q_п = a_п \cdot q$,

$q_п$=0,05·58390560=2919528 kJ/h,

where $a_п$ – is the fraction of fuel heat removed by cooling water in the air cooler.

We assume that at $p_s$ = 0,3 We assume that at $a_н = 0{,}05$.

The correctness of the calculation is done using the approximate dependence:

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

$$\frac{Q_{з.в}}{N_e} = 245.3/8600 = 0{,}029.$$

That is, the obtained value satisfies the above-written dependence. The calculation of the sea water pump supply is correct.

Nominal power of the sea water pump drive:

$P = (245{,}3 \cdot 0.2)/(3{,}6 \cdot 0{,}9) = 15{,}1$ kW

where $p_{зв}$ – the pressure generated by the pump is from 0.2 to 0.4 MPa;

We assume $p_{зв} = 0.2$ MPa;

The efficiency of the piston pump is taken to be equal to $\eta = 0.9$.

### Water cooler

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

$A_{охл} = q_ц/(k_ц \ \Delta T_в)$,

$A_{охл} = 6422961/(5000·15) =85,6\ m^2$.

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

$k_ц$ – is the heat transfer coefficient, we assume 5000 kJ/(m2·h·K).

We take a water cooler for the heat transfer surface area.

$\Delta T_в = T_{пв} - T_{зв}$, K.

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

We take $T_{пв}$ = 320 K.

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

$\Delta T_в$ = 320…305 = 15 K.

### 2.4.4 Compressed air system

The system provides compressed air of specific pressure for starting and reversing the main engine 12V Wärtsilä 46F and starting auxiliary diesel engines. The system includes starting air compressors, starting air cylinders.

The amount of free air $V_B$ for $n_n$ diesel starts:

$V_B = n_n·b_B\,\Sigma V_s$,

$V_B = 12 \cdot 5 \cdot 2,5= 147,2\ м^3$,

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

$b_B$ – specific free air flow rate per 1 $m^3$ of cylinder volume; taken for LSD from 4 to 6, for MSD 2–3.5. We take $b_B = 3$.

$\Sigma V_s$ – working volume of the starting cylinders of the engine.

The working volume of the engine cylinders is found by the formula:

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

$\Sigma V_s = (3,14 \cdot 0,52 \cdot 2,5 \cdot 5)/4 = 2,5\ m^3$,

where $D$ – is the cylinder diameter, m;

$S$ – the piston stroke, m;

$N$ – the number of cylinders.

Total volume of cylinders:

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

$\Sigma V_б = 147.2 \cdot 0,1013/(3,0-1,0) = 7,5$ m$^3$,

where $p_{мах}$ – is the maximum air pressure in the cylinder, from 2,5 MPa to 3,0 MPa; we assume $p_{мах}$ =3,0 МПа.

$p_{мin}$ – minimum air pressure at which it is possible to start the engine, from 1.0 MPa to 1.5 MPa; we take $p_{мin}$ = 1,0 MPa;

$p_B$ – free air pressure we take as $p_B$ =0,1013 MPa.

Standard cylinders in the amount of not less than two should have a total volume close to $\Sigma V_б$, we assume $\Sigma V_б$ = 3,7 m$^3$.

The selection of cylinders for a diesel generator is done similarly.

The total supply of compressors should ensure filling of cylinders in 1 hour starting from atmospheric pressure to pressure $p_{мах}$.

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

$Q_к = 7,5 \cdot (3,0-1.0)/0,1013 = 147,2\ m^3/h.$

If it is necessary to ensure the operation of the typhoon for at least 6 minutes while maintaining the possibility of 12 consecutive starts, the volumes of cylinders are increased accordingly.

### 2.4.5 Exhaust Gas System

The system includes an exhaust pipe, muffler, spark arrester, and a waste heat boiler. The amount of gases leaving the engine:

$Q_г = 10 \cdot N_e$,

$Q_г = 10 \cdot 8600 = 86000\ m^3/h$,

where $N_e$ – effective power of the main engine, kW. The exhaust pipes are made separately for each engine. The diameter of the pipes is calculated based on the speed of the exhaust gases. For two-stroke internal combustion engines, the diameter of the pipelines is:

$d_{газ} = 12 \cdot \sqrt{N_e}$ .

$d_{газ} = 12 \cdot \sqrt{8600}$ = 1115 mm.

## 2.5 Mechanisms and devices of general ship systems

### 2.5.1 Fire extinguishing system MV NEPTUNE KALLOS

Total feed $\Sigma Q$ of fire pumps must be no less than:

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

$\Sigma Q = 0{,}008 \cdot 169{,}92 = 230{,}8$ m$^3$/h.

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

$m$=1,68·√ (177·(30+12))+25=169,9,

where $L_c$ – length of the vessel, m,

$B_c$ – width of the vessel, m,

$H_б$ – height to the bulkhead deck amidships, m,

$k$ – coefficient, for MV NEPTUNE KALLOS, we take it as k = 0.008.

Rated power of the fire pump drive:

$P=(\Sigma Q \cdot p)/(3{,}6 \cdot \eta)/2,$

$P=(230{,}8 \cdot 1{,}0)/(3{,}6 \cdot 0{,}93)/2= 68{,}9\ kW$

where $p$ – the pressure created by the pump, taken according to the requirements of the Register, MPa,

$\eta$ – the efficiency of the pump, taken equal to 0.93

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

Sanitary, ballast, drainage and sea water pumps can be used as stationary fire pumps, if their supply and pressure correspond to the design ones. These pumps cannot be used for pumping petroleum products, oil and other flammable liquids.

### 2.5.2 Drainage system MV NEPTUNE KALLOS

The drainage system pumps are used to remove water from the engine room bilges, propeller shaft corridors, from the bilges of cargo holds, and can also be used to pump out ballast from the aft peak and fore peak.

The drainage pumps can perform the role of ballast in cases specified by the Rules of the Register.

The diameter of the main line $d_{\text{ос}}$ is found by the Register formula:

$$d_{\text{ос}} = 1{,}68 \cdot \sqrt{L_{\text{с}} \cdot \left(B_{\text{с}} + H_{\text{б}}\right)} + 25,$$

$$d_{\text{ос}} = 1{,}68 \cdot \sqrt{(177 \cdot (30+12))} + 25 = 170\ mm,$$

where $L_{\text{с}}$ – length of the vessel, m,

$B_{\text{с}}$ – width of the vessel, m,

$H_{\text{б}}$ – height to the bulkhead deck amidships, m.

The total flow of the drainage pumps must be no less than the total flow of the fire pumps. The supply of drainage pumps is calculated based on the fluid flow rate in the main pipes $v_{\text{он}} \geq 2\ m/s$.

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

$$Q_{\text{ос}} = 3600 \cdot (3{,}14 \cdot 0{,}170^2)/4 \cdot 2.5 = 203.8\ m^3/h.$$

Rated power of the drainage pump drive:

$$P_{\text{ос}} = (Q_{\text{ос}} \cdot p_{\text{он}})/(3{,}6 \cdot \eta),$$

$$P_{\text{ос}} = (203.8 \cdot 0.3)/(3{,}6 \cdot 0.9) = 19.0\ \text{kW},$$

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

We accept the pump efficiency as $\eta = 0.9$.

The number of pumps is not less than two. A ballast pump or another pump with the required flow rate can be used as one of the independent pumps.

### 2.5.3 Ballast water system of MV NEPTUNE KALLOS

The system is designed to fill and drain ballast tanks. The ballast pump flow rate should be such that all ballast tanks are drained within 4 to 10 hours, depending on the size of the vessel.

According to the Rules of the Register, the internal diameter of the ballast pipe $d_{бс}$ is found by the formula:

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

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

where $V_{бв}$ - the volume of the largest ballast compartment, determined by reference to the prototype ship, $V_{бв} = 2258\ m^3$.

The supply of ballast pumps with a known pipeline diameter is determined by the water flow rate in the main pipes. $v_{бс} \geq 2\ m/s$.

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

$$Q_{бн} = 3600 \cdot (3{,}14 \cdot 0{,}236^2)/4 \cdot 2{,}5 = 394{,}0\ m^3/h.$$

$v_{бс}$ – flow speed.

For deballasting systems we accept $v_{бс} = 2{,}5$ m/s.

Rated power of ballast pump drive:

$$P_{бн} = (Q_{бн} \cdot p_{бн})/(3{,}6 \cdot \eta),$$

$$P_{бн} = (394{,}0 \cdot 0{,}3)/(3{,}6 \cdot 0{,}78) = 43{,}0\ kW.$$

where $p_{бн}$ – pressure generated by the pump, from 0,2 MPa to 0,3 MPa; we tale $p_{бн} = 0.3$ MPa.

For pump efficiency, we accept $\eta = 0{,}78$.

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

## 2.6 Ventilation system MV NEPTUNE KALLOS

The supply of air blowers that provide ventilation of living spaces is calculated based on the supply condition for each crew member or passenger from $33\ m^3/h$ to $50\ m^3/h$ *of air*. The supply of the engine room ventilation system is determined based on the calculation of ensuring the operation of the main engines in storm conditions with the engine room closed. According to the prototype of the vessel, 21 crew seats and 2 spare seats are provided. Thus, to ensure full human occupancy on the vessel with a total number of seats for 23 crew members, the total air supply is equal to:

$Q_e = 40 \cdot 23 = 920{,}0$ m$^3$/h.

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

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

$Q_{мо}=2.5 \cdot 14 \cdot 920{,}0 /1{,}29 = 24961\ m^3/h.$

where $\alpha_g$ – coefficient of residual air during fuel combustion in the engine. For the 12V Wärtsilä 46F engine, we take $\alpha_g = 2{,}5$,

$G_0$ – mass air flow rate per 1 kg of fuel,

$\rho_B$ – air density at barometric pressure.

## 2.7 Sanitary system

### 2.7.1 Fresh water system MV NEPTUNE KALLOS

Drinking water is stored in reserve tanks located outside the double bottom. From the reserve tanks, water is pumped to the hydrophores. The volume of the pressure tank is assumed to be 0.2 of the daily consumption. The supply of the drinking water pump is determined by calculating the consumption per person per day from 10 liters to 40 liters of drinking water. Thus, based on the maximum crew 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 fresh 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 $p_{пв}$ – the pressure created by the drinking water pump, which rates from 0, 5 MPa to 0,7 MPa; assumed $p_{пв} = 0{,}7\ MPa$.

For pump efficiency, we accept $\eta = 0{,}75$.

### 2.7.2 Water system for sanitary and domestic needs MV NEPTUNE KALLOS

Water consumption for sanitary and household needs is on average from 100 ltr to 200 ltr per person/day. The definition of water supply for sanitary and household needs is similar to the previous calculation. $V_{цпв} = Q_{цпв} = (200 \cdot 23) \cdot 0,2 = 9,20\ m^3$.

Rated power of the pump drive:

$P_{спп}=(9.20 \cdot 0.7)/(3,6 \cdot 0,7) = 2.56\ \ kW,$

where $p_{спп}$ – The pressure created by the pump: 0,5 MPa to 0,7 MPa; we assume $p_{спп}$ = 0.7 MPa;

For pump efficiency, we accept $\eta = 0,7$.

### 2.7.3 Seawater system for sanitary purposes MV NEPTUNE KALLOS

To calculate the pump flow, the water consumption for sanitary needs is assumed to be 20...30 liters per person/day. The pump flow is determined similarly to the previous calculation.

$V_{цзв} = Q_{цзв} = (30 \cdot 23) \cdot 0,65 = 4,49\ \ m^3$.

Rated power of the pump drive:

$P_{стп}=(4,49 \cdot 0,7)/(3,6 \cdot 0,74) = 11,78\ kW.$

where $p_{стп}$ – the pressure created by the pump is from 0.5 MPa to 0.7 MPa; we accept $p_{стп} = 0,7\ MPa$;

We take the pump efficiency to be η = 0.74.

# 3. SCHEMATIC DIAGRAM AND OPERATING PRINCIPLE OF THE CENTRAL COOLER

## 3.1 The Central Low-Temperature Cooler in the Ship's Integrated Seawater System

The central cooling system on a maritime vessel is a critical component of the ship's overall service system, ensuring temperature control for various machinery, most notably the main propulsion engine. Modern commercial vessels typically employ a robust and segregated three-circuit cooling architecture to manage waste heat effectively and mitigate the risks associated with raw seawater: the High-Temperature (HT) freshwater system, the Low-Temperature (LT) freshwater system, and the external Internal Seawater System (ISS) [2]. The central cooler, often a Plate Heat Exchanger (PHE), serves as the primary thermal interface between the LT freshwater system—the main cooling medium for auxiliary equipment—and the seawater (SW) circuit, which acts as the ultimate heat sink. This PHE is strategically positioned to cool the LT freshwater by transferring heat directly to the ambient seawater drawn from the ocean, effectively isolating the sensitive engine and auxiliary machinery from the corrosive and fouling characteristics of the external marine environment [1].

The ship's cooling architecture typically consists of three distinct circuits. The High-Temperature (HT) circuit is a closed-loop system designed to cool the main engine jacket water and cylinder heads, maintaining a high operational temperature, often around 80°C [2]. This circuit transfers its excess heat to the Low-Temperature (LT) circuit via an HT/LT heat exchanger. The LT circuit, also a closed-loop system, is cooled directly by the central cooler. It draws heat from the HT circuit as well as other equipment, such as scavenge air coolers, generator engine jackets, and lubricating oil coolers, maintaining a regulated temperature of approximately 34°C [2]. Finally, the Seawater (SW) circuit, or Internal Seawater System (ISS), is an open system that continuously draws ambient seawater from sea chests and discharges it overboard.

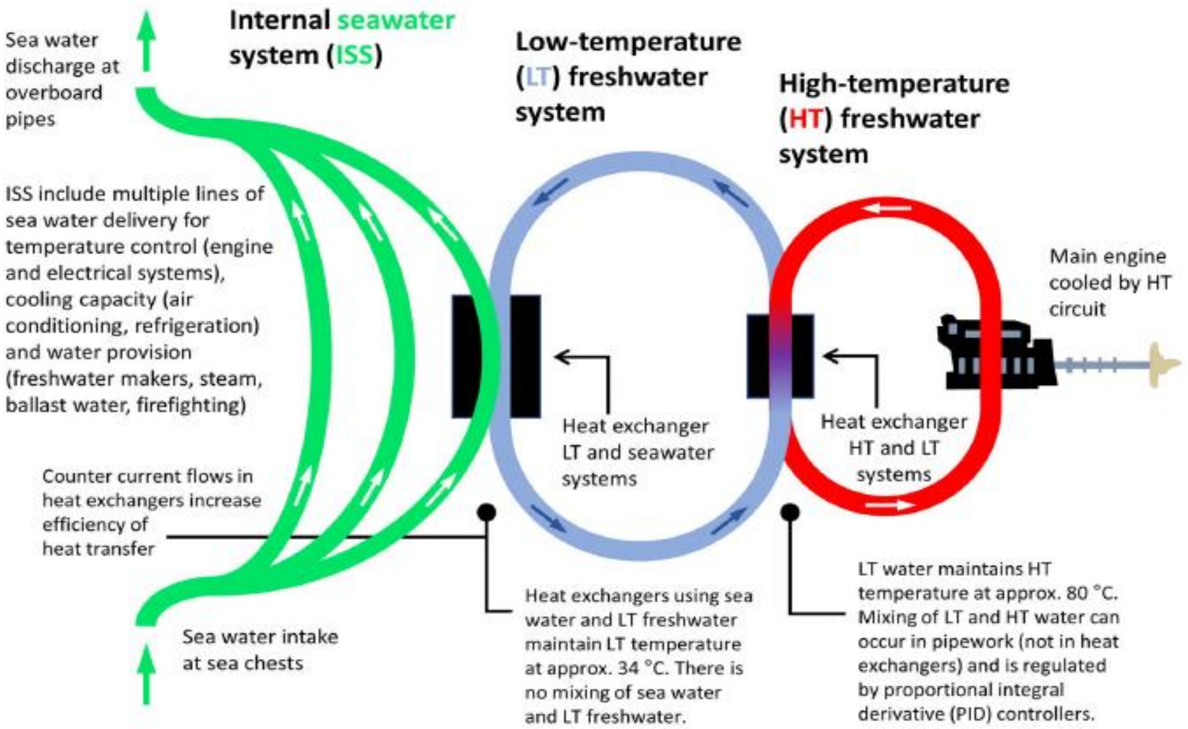


Figure 3.1 – Schematic view of a typical central cooler system layout [1]

It serves as the sole heat rejection path to the marine environment. Critically, there is no mixing of fluids between the ISS and the freshwater systems, ensuring that machinery operates with treated, non-corrosive water while utilising the immense thermal capacity of the sea [1, 2].

The PHE's operational principle in this setup is vital: it must continuously and efficiently maintain the LT freshwater set point. Failure to do so leads to an uncontrollable increase in the temperature of the LT system, which in turn leads to the overheating of engine components and auxiliary systems, eventually forcing the engine management and monitoring system to initiate an automatic engine slowdown [2].

## 3.2 Heat Exchanger Design and Main Parameters

Selection of Geometric Characteristics of Heat Exchanger Plates from the GEA NT250S Series (Marine Duty) for "Fresh Water – Seawater" Heat Transfer Media.

For the central cooling system, the GEA NT Series gasketed plate heat exchanger has been selected due to its optimized design for marine applications, offering high heat transfer rates with a compact footprint. The construction relies on a modular system comprising a frame plate, a pressure plate, and a pack of corrugated heat transfer plates clamped together by tightening bolts.

Key Design Technologies:

1. OptiWave Design: The plates feature the *OptiWave* corrugation pattern. This optimized profile ensures uniform media distribution across the entire plate width. By eliminating low-flow zones, it maximizes the heat transfer efficiency and promotes high turbulence even at lower flow rates. This high turbulence is critical for the "Seawater" circuit, as it creates a self-cleaning effect that mitigates the risk of biofouling accumulation and sedimentation [18, 19].

2. EcoLoc Gasket System: The unit utilizes the *EcoLoc* glueless gasket system. The gaskets are mechanically fixed in the groove, ensuring perfect alignment and simplifying maintenance procedures during plate cleaning or replacement. This reduces downtime and maintenance costs [19].

3. PosLoc Assembly: The *PosLoc* system ensures the self-alignment of the plate pack during tightening. This prevents misalignment of the gaskets and plates, ensuring the integrity of the seal and preventing leaks between the fresh water and seawater circuits [19].

Based on the manufacturer's data [20] the GEA NT250S series heat exchanger is adopted, and its geometric parameters are listed as follows:

- Type: NT250S (GEA NT series);
- Chevron angle: $2\beta = 124^\circ$ ($\beta \approx 62^\circ$);
- Corrugation Depth (Plate Gap): $2b = 3.2\,mm$;
- Port diameter: (corner holes): $D_p \approx 250\,mm$;
- Plate size: 1736 × 745 мм;
- Effective Heat Transfer Area (per plate): ≈0,88 м$^2$.
- Corner hole centre distance: $L = 1406\,mm$;
- Effective Plate Width: $B = 420\,mm$;
- Channel Flow Area: $A_{chan} = 0.00219\,m^2$;

- The equivalent hydraulic diameter of the channel $D_h = 6.4\,mm$.

For the marine "Fresh Water – Seawater" (FW–SW) application in this series, plates are typically manufactured from Titanium or SMO 254 material.
For this model, a standard heat exchanger plate thickness of δp=0,5 мм is applied. The thermal conductivity is λ≈14 W/(m·K); if the plate is made of high-alloy austenitic stainless steel for marine environments 254 SMO (UNS S31254), and λ≈16 W/(m·K) for Titanium Grade 1/2 plates.

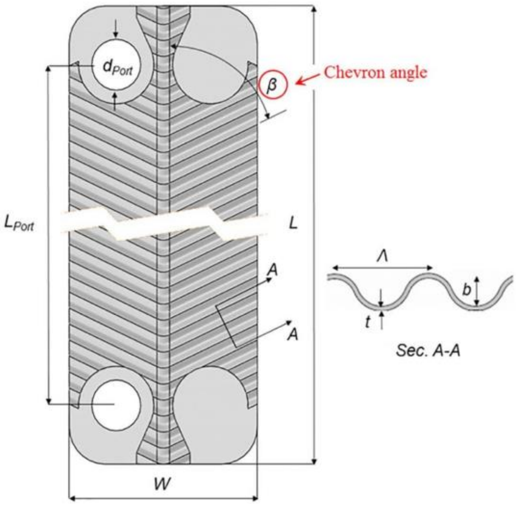


Figure 3.2. Schematic of a chevron-type plate heat exchanger plate showing the chevron angle β between the corrugation direction and the main flow direction [21]

Next, certain geometric parameters must be additionally determined to perform hydraulic and thermal calculations. Heat exchanger channel length along the coolant path from port to port. The catalogue for the NT100M provides the distance between the port centers along the flow direction (corner hole centre distance): $L = 1406\,mm$.This is taken as the direct calculated channel length from the inlet port to the outlet port. This is the geometric (direct) length along the plate. The actual flow path along the corrugated channel is slightly longer due to the chevron angle, but for hydraulic loss calculations, the direct length $L_{chan}$ is often used in conjunction with empirical correlations. Channel free flow area The catalogue for NT100M plates provides the free flow cross-sectional area of a single channel (flow cross-sectional area): $A_{chan}$ = 0.001216 $м^2$.. To estimate the effective 'thickness' of the coolant passage, this cross-section can be divided by the working width of the plate $B \approx 0.262$:

$$t_{eff} \approx \frac{A_{chan}}{B} = \frac{0.00219\, m^2}{0.429\, m} \approx 0.0051\, m \approx 5.1\, mm.$$

We find that the effective geometric thickness of the channel (taking into account the corrugated surface) is somewhat larger than the nominal corrugation depth $2b = 3.2$ mm.

The hydraulic diameter of the channel is determined by the general formula:

$$D_h = \frac{4A_{chan}}{P},$$

where $A_{chan}$ — is the flow cross-sectional area, $P$ — is the wetted perimeter of the channel.

For a flat slit between two parallel plates of width $B$ and gap $2b$ (under the condition $B \gg 2b$):

$$A \approx 2bB, \qquad P \approx 2B,$$

then the hydraulic diameter:

$$D_h = \frac{4A}{P} = \frac{4 \cdot 2bB}{2B} = 4b.$$

According to the geometry of the NT100M plate, the corrugation depth (gap between plates) is $2b = 3.2\ mm$,, which means $b = 1.6\ mm$. Then:

$$D_h = 4b = 4 \cdot 1.6\ mm = 6.4\ mm$$

This value corresponds to the catalogue «equivalent diameter 6.4 mm» for the NT250S plate, meaning the equivalent diameter provided by the manufacturer is effectively the hydraulic diameter of the channel.

## 3.3 Input Data for Analysis

For the preliminary thermal calculation of the central cooler, we assume the calculation mode is sea-going condition at 100% MCR for the 12V Wärtsilä 46F main engine, including two diesel generators and the air conditioning and refrigeration systems. In this mode, the HT circuit of the central cooling system removes heat from the following main consumers:

– Main Engine (ME) 12V Wärtsilä 46F;

– Two diesel generators (Wärtsilä Auxpac type);

– Water-cooled condensers of the refrigeration and air conditioning systems.

Based on the thermal balance of the Wärtsilä 46F series engines, the total thermal power rejected to the LT circuit water (including charge air cooling by the LT circuit and lubricating oil cooling) from the 12V Wärtsilä 46F main engine at 100% MCR nominal load is taken as approximately:

$$Q_{HT,ME} \approx 3.0\,MW.$$

For the two diesel generators connected to the HT circuit (heat rejection from charge air cooling and lubricating oil cooling), the total thermal power entering the central cooler, based on catalogue data for engines of an analogous class (Wärtsilä Auxpac type), can be estimated as:

$$Q_{LT,2DG} \approx 1.2\,MW.$$

The air conditioning and refrigeration systems have water-cooled condensers connected to the HT cooling circuit. For the maximum load condition, the total refrigeration capacity of the cooling machines is assumed to be:

$$Q_{Ref} \approx 400\,kW,$$

We tentatively assume the Coefficient of Performance (COP) of the compression refrigeration machines is equal to 3. Then, the thermal power transferred to the LT circuit in the condensers is:

$$Q_{Ref,cond} = Q_{Ref}\left(1+\frac{1}{COP}\right) \approx 400\cdot\left(1+\frac{1}{3}\right) \approx 0.5\,MW.$$

The total thermal power that must be rejected through the central cooler in the design mode is determined as:

$$Q_{HT,total} = Q_{HT,ME} + Q_{HT,2DG} + Q_{Ref,cond} \approx 3.0 + 1.2 + 0.5 \approx 4.7\,MW.$$

The resulting value of $Q_{HT,total} \approx 4.7\ MW$ is used as the design thermal power for determining the fresh water flow rate $V_{IIB}$, selecting the heat transfer area, plate count, and calculating fresh and seawater flow rates in subsequent thermal and hydraulic calculations. The seawater flow rate is taken from the condition of ensuring the required temperature regime and is $V_{SW} \approx (1.3\pm0.2)\, V_{FW}$. This is consistent with Wärtsilä recommendations: the recommendation from the Wärtsilä 46F project guide is that the capacity of the sea water pump is determined by the type of cooler and the amount of heat to be dissipated; the sea water flow is (1,2…1,5) the fresh water flow. The central cooler is designed for the sea-going condition at a seawater inlet temperature of $t_{SWin} = 32\ °C$ (the most worst conditions). The outlet temperature of the HT fresh water from the central cooler (before the HT consumers) is taken as $t_{FWout}$=36 °C [22]. The return temperature of the fresh water is $t_{FWin}$≈45 °C.

The calculated thermal load value for the heat exchanger, as well as other input data for the calculations, are entered into Table 3.1.

Table 3.1. Input Data for Heat Exchanger Calculation

| | |
|---|---|
| Max. Allowable FW Pressure Drop | $\Delta p_{FW} \leq$ 0,2…0,3 (till 0,4) bar |
| Max. Allowable SW Pressure Drop | $\Delta p_{SW} \leq$ 0,3…0,5 (till 0,7) bar |
| Seawater Inlet Temperature | $t_{SW,in} = 32\ °C$ |
| Fresh Water Outlet Temperature | $t_{FW,out}$ =36 °C |
| Fresh Water Inlet Temperature | $t_{h,in}$=45 °C |
| SW Temperature Difference (In/Out) | $\Delta t_{SW} \approx$ 4…7 K |
| FW Temperature Difference (In/Out) | $\Delta t_{FW} \approx$ 8…10 K |

**The exact values must be confirmed based on the specific characteristics of the fresh water and seawater pumps.*

*** Corresponds to typical values for the HT water circuits of ship central cooling systems (HT water is maintained at ≈35…36 °C after the central cooler and has a temperature of ≈45 °C at the inlet).*

Additionally, the flow configuration options for the plate heat exchanger can be considered. For fresh water, manufacturers typically specify 2–3 passes (or channels) because a larger pressure drop is acceptable, which allows for higher velocity in the channels. This higher velocity is desired as it reduces scaling (fouling) and increases the intensity of heat transfer. For seawater, the primary goal is to limit pressure losses, thus often resulting in 1 pass, and sometimes 2 passes being utilized.

## 3.4 Calculation Methodology for Water-to-Water Plate Heat Exchanger

Input Data for Calculation

For the "Fresh Water – Seawater" plate heat exchanger (central cooler), the following input data are specified:

– Thermal Load: $Q$ (kW);

– Temperatures:

– Hot fluid (Fresh Water inlet/outlet): $t_{FW,in}$, $t_{FW,out}$;

– Cold fluid (Seawater inlet/outlet): $t_{SW,in}$, $t_{SW,out}$;

– Thermophysical Properties: Properties of the fluids at corresponding average temperatures;

– Allowable Pressure Drops:

– Plate Materials: (Titanium or high-alloy steels for seawater) and their geometric characteristics. The calculation is performed according to the standard approach outlined in [23-25].

*Thermal Balance*

If not all temperatures / flow rates are known, some values are found from the thermal balance: We check if the thermal load matches for both circuits:

$$Q = \dot{m}_{FW} c_{p,FW} (t_{FW,in} - t_{FW,out})$$

$$Q = \dot{m}_{SW} c_{p,SW} (t_{SW,out} - t_{SW,in})$$

If only one flow rate is known, for example, the fresh water flow rate $\dot{m}_{FW}$ is known, then:

1) For the hot circuit (fresh water):

$$Q = \dot{m}_{FW} c_{p,FW} (t_{FW,in} - t_{FW,out})$$

$$t_{FW,out} = t_{FW,in} - \frac{Q}{\dot{m}_{FW} c_{p,FW}}$$

2) For the cold circuit (seawater):

$$Q = \dot{m}_{SW} c_{p,SW} (t_{SW,out} - t_{SW,in})$$

$$\dot{m}_{SW} = \frac{Q}{c_{p,SW} (t_{SW,out} - t_{SW,in})}$$

Similarly, one can specify the «cold» circuit parameters and find the «hot» circuit parameters.

*Temperature Head in the Heat Exchanger*

For a plate heat exchanger (which is close to ideal counterflow in terms of fluid arrangement), we define the limiting temperature differences:

$$\Delta T_1 = t_{FW,in} - t_{SW,out}$$

$$\Delta T_2 = t_{FW,out} - t_{SW,in}$$

Mean Logarithmic Temperature Head:

$$\Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{ln\left(\frac{\Delta T_1}{\Delta T_2}\right)}$$

For a single-pass heat exchanger operating in counterflow, the correction factor F usually lies within the range F ≈ 0,95…1,0; for initial calculation, F = 1.

*Overall Heat Transfer Coefficient*

For a water-to-water plate heat exchanger operating in a turbulent regime in the channels, the overall heat transfer coefficient (U) can be tentatively assumed:

$$U \approx 3000 \dots 6000\ \text{W}/(m^2 \cdot \text{K})$$

For the first approximation (preliminary calculation), it is convenient to take $U \approx 3500 \dots 4500\ \text{W}/(m^2 \cdot \text{K})$ assuming normal surface cleanliness and sufficient flow turbulence (not very low velocities).

However, for a more accurate calculation, it is appropriate to use the formula for the overall heat transfer coefficient U:

$$\frac{1}{U}=\frac{1}{\alpha_{FW}}+R_{f,FW}+\frac{\delta}{\lambda_{pl}}+R_{f,SW}+\frac{1}{\alpha_{SW}}$$

where $\alpha_{FW}$, $\alpha_{SW}$ – are the heat transfer coefficients from the fluid to the wall on the hot/cold sides;

$R_{f,FW}$, $R_{f,SW}$ – are the thermal resistances due to fouling; $\delta$ – is the plate thickness;

$\lambda_{pl}$ – is the thermal conductivity of the plate material.

Here, the heat transfer coefficients are usually determined by empirical equations of the type $Nu = a\,\mathrm{Re}^{b}\,\mathrm{Pr}^{c}$ correlations, which are provided in reference literature.

For calculating the heat transfer coefficient in the chevron channels of a plate heat exchanger with single-phase fluid flow (water-to-water), the generalized correlations by Muley & Manglik [26] for the Nusselt number are convenient. These correlations were derived based on experiments for chevron angles $30^{\circ} \le \beta \le 60^{\circ}$, Reynolds number range $Re \gtrsim 10^{3}$, and area enlargement factor $\varphi$ in the range $1 \le \varphi \le 1.5$.

First, the Reynolds number and Prandtl number for the given fluid in the channel are calculated:

$$Re=\frac{\rho w D_h}{\mu}, \qquad Pr=\frac{c_p\,\mu}{\lambda}.$$

where $\rho$ is the density, $w$ is the mean velocity in the channel, $D_h$ is the hydraulic diameter of the channel, $\mu$ is the dynamic viscosity, $c_p$ is the isobaric specific heat capacity, and $\lambda$ is the thermal conductivity.

The Nusselt number in the plate heat exchanger channels:

$$Nu=\left[\,0.2668-0.006967\beta+7.244\times10^{-5}\beta^{2}\,\right]\times$$

$$\times\left[\,20.7803-50.9372\varphi+41.1585\varphi^{2}-10.1507\varphi^{3}\,\right]Re^{\,0.728+0.0543\sin\left(\frac{2\pi\beta}{90}+3.7\right)}\,Pr^{1/3}.$$

where $\beta$ is the chevron angle (in degrees), $\varphi$ is the plate area enlargement factor (ratio of developed area to projected area). For typical industrial chevron plates, $\varphi \approx 1.1\ldots1.3$.

Next, the heat transfer coefficient in the channel is calculated from the Nusselt number:

$$\alpha = \frac{Nu\,\lambda}{D_{FW}}.$$

Since the channels for fresh water and seawater are identical, the same correlation for the heat transfer coefficients $\alpha_{FW}$ and $\alpha_{SW}$ is used.

It is impossible to calculate $\alpha_{FW}$ and $\alpha_{SW}$ immediately because the velocity in the channels is unknown, and the velocity depends on the number of channels. Therefore, in the first step, the calculation is performed using the average $U \approx 3500 \ldots 4500$ W/($m^2$·K), the required number of plates is selected, and then $\alpha_{FW}$, $\alpha_{SW}$, and $U$ are checked.

*Required Heat Transfer Surface Area*

Once $Q$, $\Delta T_{lm}$, and $U$ are known (taking into account the correction factor F), the required heat exchange surface area is determined by the formula:

$$A_{th} = \frac{Q}{U\,F\,\Delta T_{lm}}$$

where $A_{th}$ – is the required surface area of the plates participating in the heat transfer, m$^2$.

*Plate Type and Number of Plates Selection*

The technical data provided by the manufacturer lists the area of a single plate $A_{pl}$ (m$^2$), the plate size (height×width), pitch, corrugation depth, chevron angle, etc. The preliminary number of heat exchange plates is calculated as:

$$N_{th} = \frac{A_{th}}{A_{pl}}$$

The resulting value $N_{th}$ is rounded up to the next even number to ensure the correct number of channels for both flows, and two blank frame plates are added - $N_{pl}$.

For a single-pass scheme 1/1 (one pass on both sides), the total number of channels is determined as:

$$N_{ch,tot} = N_{pl} - 1$$

where $N_{pl}$ – is the total number of plates (including the end frame plates). The number of channels for each circuit is approximately equal to:

The number of channels for each circuit is approximately equal to:

$$N_{ch,FW} \approx N_{ch,SW} \approx \frac{N_{pl}-1}{2}.$$

Then, the flow rate in a single channel will be:

$$\dot{m}_{ch,FW} = \frac{\dot{m}_{FW}}{N_{ch,FW}}, \qquad \dot{m}_{ch,SW} = \frac{\dot{m}_{SW}}{N_{ch,SW}}.$$

The velocity in the channel is:

$$w = \frac{\dot{m}_{ch}}{\rho A_{cs}},$$

where $A_{cs}$ is the free flow cross-sectional area of the channel, which depends on the corrugation depth, pitch, and effective channel width (determined by the geometric characteristics for the selected heat exchanger). Recommended water velocities in plate heat exchanger channels typically lie within the 0,3…1,5 m/s.

Heat Exchanger Pressure Drop Check

$$\Delta p_{ch} = \left( f \frac{L}{D_h} + \sum \zeta \right) \frac{\rho w^2}{2},$$

where $L$ – the length of the channel, $f$ – the Darcy friction factor, $\sum \zeta$ – is the sum of local resistances (inlet to the channel, flow turns in distribution zones, channel outlet, etc.).

For calculating hydraulic losses in the chevron channels of a plate heat exchanger with single-phase fluid flow (water-to-water), the generalized correlations by Muley & Manglik [26] for the Darcy friction factor along the length. Similar to the Nusselt number, this correlation was obtained based on experiments for chevron angles $30^\circ \le \beta \le 60^\circ$, Reynolds number range $Re \gtrsim 10^3$ and area enlargement factor $\varphi$ within the range $1 \le \varphi \le 1.5$.

$$f = \left[ 2.917 - 0.1277\beta + 2.016 \times 10^{-3} \beta^2 \right] \times$$

$$\times \left[ 5.474 - 19.02\varphi + 18.93\varphi^2 - 5.3413\varphi^3 \right] Re^{-\left(0.2 + 0.0577 \sin\left(\frac{\beta}{45} + 2.1\right)\right)}.$$

The Darcy friction pressure drop is calculated using:

According to [27], the total coefficient of local losses in one plate heat exchanger channel was taken as $\sum \zeta_{chan} \approx 3$.

The hydraulic losses in the plate package are equal to the pressure drop in one pass (since all channels are parallel), i.e., equal to the pressure drop in one channel:

$$\Delta p_{HE} = \Delta p_{ch}$$

If the heat exchanger has $N_{pass}$ passes sequentially for the fluid:

$$\Delta p_{HE} = N_{pass} \cdot \Delta p_{ch}$$

The resulting value $\Delta p$ is compared with the allowable pressure drop for each circuit.

Calculation Sequence:

1. Calculate the flow rates of both fluids if some data are unknown;
2. Determine the logarithmic mean temperature difference $\Delta T_{lm}$;
3. Estimate the overall heat transfer coefficient U for the "fresh water – seawater" pair;
4. Calculate the required heat transfer area $A_{th}$;
5. Select a realistic plate heat exchanger configuration (plate type, number of plates, flow scheme);
6. Calculate the pressure drop; if it exceeds the allowable limit, select a heat exchanger with different geometric parameters and repeat the calculations.

# 4. ANALYSIS OF THE INFLUENCE OF FOULING ON THE FRESH WATER – SEAWATER HEAT EXCHANGER PERFORMANCE

The objective of this section is to perform the design calculation for the central cooler (FW–SW PHE) using the methodology and input data established in Section 3, and subsequently to analyze the operational regime of the heat exchanger under conditions of varying seawater temperature and the presence of fouling on the heat exchange surfaces.

## 4.1 Design Calculation of the Fresh Water – Seawater Plate Heat Exchanger

The calculation was performed according to the methodology described above using Excel software. The GEA NT250S heat exchanger (plate area 0.88 $m^2$) was adopted as the prototype.

The heat transfer surface area is regulated by the number of plates, which was determined in the calculations. However, in addition to the heat transfer surface area, it is necessary to create certain hydrodynamic conditions for the flow of heat medium, determined by velocity, so that the pressure losses do not exceed the allowable limits and the heat transfer intensity is sufficient. For this purpose, the possibility of arranging passes for the heat medium in the plate heat exchanger is additionally considered. For fresh water, manufacturers usually design 2–3 passes, as a large pressure drop is available and it is necessary to create a relatively high velocity in the channels (this reduces formation of scale deposits and increases heat transfer intensity). For seawater, efforts are made to limit pressure losses, so 1 or 2 passes are more commonly used.

The calculation according to the methodology described above consisted of initially assuming a certain number of plates (approximately for the average value of the heat transfer coefficient), for which the calculation of heat transfer coefficients and the overall heat transfer coefficient was performed, the area was determined, and a new number of plates was calculated. Thus, the calculation was repeated multiple times (using the Excel "Solver" add-in) until the required number of plates (a specific heat exchange area) was achieved. Subsequently, the pressure losses for both heat carriers were evaluated.

The calculations showed that the 1/1 configuration (1 pass for fresh water and 1 pass for seawater), as well as the 2/1 and 2/2 configurations, do not allow for the necessary heat transfer intensity, thereby leading to the need to select a very large number of plates. Therefore, the 3/2 heat exchanger pack configuration (3 passes for fresh water and 2 passes for seawater) was finally selected. Such a configuration allows for increasing the flow velocity of the heat carriers, thereby increasing the heat transfer intensity and making the heat exchanger compact with allowable pressure losses.

The thermophysical properties of water were taken according to [28] depending on the temperatures of the heat carriers at the inlet to the heat exchanger.

The calculation results are presented in Table 4.1.

Table 4.1 Calculation Results of the GEA NT250S Plate Heat Exchanger with Determination of Plate Count and Pressure Head Losses

| | Cold Fluid – Seawater (SW) | Hot Fluid – Fresh Water (FW) |
|---|---|---|
| Inlet Temperature, °C | 32 | 45 |
| Density $\rho$, kg/m$^3$ | 995.0 | 968.6 |
| Specific Heat $c_p$, kJ/(kg·°C) | 4.180 | 4.180 |
| Kinematic Viscosity ν, m$^2$/s | 0.00000077 | 0.00000060 |
| Thermal Conductivity Coefficient λ, W/(m K) | 0.06174 | 0.06348 |
| Total Heat Load Q, kW | 4700 | |
| Outlet Temperature, °C | 38.9 | 36.0 |
| Number of passes | 2 | 3 |
| Number of Channels per Fluid (Assumed) | 92 | |
| Single Channel Length per Fluid, m | 1.406 | |

Continuation of Table 4.1

| | | |
|---|---|---|
| Area of One Plate, $m^2$ | 0.88 | |
| Plate Thickness, m | 0.0005 | |
| Plate Material Thermal Conductivity, W/(m·K) | 15 | |
| Number of Channels | 91 | 91 |
| Channel Flow Cross-Sectional Area, $m^2$ | 0.00219 | 0.00219 |
| Equivalent Channel Diameter, m | 0.0064 | 0.0064 |
| Mass Flow Rate of Fluid, kg/s | 162.4 | 124.9 |
| Volumetric Flow Rate of Fluid, $m^3$/s | 0.1632 | 0.1290 |
| Fluid Velocity in Channel, m/s | 1.631 | 1.933 |
| Re | 13586 | 20563 |
| Pr | 51.75 | 38.38 |
| Nu | 855.2 | 675.1 |
| Heat Transfer Coefficient, W/($m^2$·K) | 16500 | 13393 |
| Logarithmic Mean Temperature Difference, K | 4.97 | |
| Overall Heat Transfer Coefficient, W/($m^2$·K) | 5931 | |
| Heat Transfer Surface Area, $m^2$ | 160 | |
| Number of Plates | 184 | |
| Number of Channels per Fluid (Verification) | 92 | |
| Hydraulic Loss Coefficient along Channel Length | 0.0246 | 0.0226 |
| Pressure Drop, bar | 0.3657 | 0.1086 |

The determined number of plates (including two blind plates), 184 units, is a realistic value for heat exchangers of this type.

## 4.2 Fouling types of Heat Exchange Surfaces, common for "Fresh Water – Seawater" Coolers

### Fresh Water Side

In a central plate cooler of the "fresh water – seawater" type, mixed complex deposits form on the fresh water side, characterized simultaneously by corrosive, mineral, organic, and biological origins [29–31]. Although modern ship cooling systems utilize desalinated or specially treated water, the fresh water circuit is never chemically pure. It always retains certain concentrations of $Ca^{2+}/Mg^{2+}$, ions, dissolved oxygen, trace amounts of chlorides/sulfates, corrosion products, and traces of organic substances, creating the potential for the formation of various types of fouling [29, 30, 32].

One of the most common types of fouling involves mineral salt deposits (scaling), which form due to the thermal destabilization of the water's carbonate and bicarbonate system. At temperatures of 70–90 °C, calcium carbonate ($CaCO_3$), magnesium hydroxide ($Mg(OH)_2$), sulfates, and silica compounds can precipitate onto the plate surfaces. Although these deposits form a thin passivating film in normal operation that partially protects the metal, excessive alkalinity, overdosing of inhibitors, or uncontrolled quality of make-up water can cause scale to grow rapidly and significantly increase thermal resistance [29, 31].

A second major group consists of deposits of corrosion products entering the fresh water circuit from jacket cooling system, pipelines, etc. Typical components of such deposits are iron oxides and hydroxides ($Fe_3O_4$, $Fe_2O_3$) [30, 32, 33].These deposits can form a dense dark-brown layer that easily adheres to plates and serves as a base for subsequent scale growth. Cases of oxygen ingress into fresh water, as well as the formation of air pockets where intense local aeration corrosion is observed, are particularly dangerous.

A separate group is constituted by precipitates of chemical additives formed under improper water treatment regimes. Corrosion inhibitors based on nitrites, molybdates, phosphates, or combined packages, described in MAN and Wärtsilä recommendations [29, 30], can react with hardness ions or precipitate due to excessive concentration under certain conditions. Such deposits often have a powdery or crystalline structure, deposit as "islands" in areas with reduced flow velocity, and significantly complicate cleaning, especially if combined with corrosion products [32].

In cases of even minor oil leakage from "oil–water" type heat exchangers, organo-mineral fouling occurs. The oil film adheres well to metal and attracts rust particles, carbonates, and organics, forming a layer with low thermal conductivity.

Another significant type of fouling involves microbiological formations (biofilms). Although the fresh water circuit is closed and operates at relatively high temperatures, microorganism development is possible in zones with reduced flow velocities, "dead zones," or in the LT circuit with temperatures of 30–40 °C, forming slimy colonies. Biofilm not only impairs heat transfer but also creates a risk of corrosion [33, 34].

All listed types of fouling act synergistically. Thin carbonate films become a substrate for the accumulation of iron oxides, the latter serving as a base for organic matter adhesion, while organic components promote microorganism development and form multilayered complex deposits [31–34].

### Seawater Side

On the seawater side, mixed deposits predominantly form, comprised of biological and mineral fouling with admixtures of corrosion products and suspended particulates. Unlike the closed fresh water circuit, seawater is supplied to the cooler almost always without chemical softening: at most through mesh filters or basket strainers, and occasionally with biocide dosing (most often chlorine) in sea chests. Such water contains high concentrations of salts, dissolved oxygen, nutrients, zooplankton, phytoplankton, macroorganism larvae, as well as suspended solid particles, creating favorable conditions for intensive fouling and contamination of heat exchange surfaces [35–37].

Biological fouling (biofouling) plays a vital role. Even with the relatively short residence time of seawater within the vessel's internal systems, a large volume of "fresh," biologically active water containing microorganisms and macroorganism larvae passes through the heat exchangers. They are capable of attaching to plate surfaces and forming bacterial biofilms, diatom films, etc. [35, 36]. In the narrow channels of a plate cooler (typical inter-plate gap of 3–7 mm), even a thin biological layer leads to a noticeable increase in hydraulic resistance due to localized cross-sectional narrowing and increased roughness, as well as a significant increase in thermal resistance [38, 39]. Furthermore, internal seawater circuits can transport marine organisms, thus being considered an additional pathway for the spread of invasive species to new regions [35, 37].

A second key type of fouling involves inorganic deposits. Seawater has high concentrations of $Ca^{2+}$, $Mg^{2+}$, $HCO_3^-$, $SO_4^{2-}$, $SiO_2$, as well as suspended particles. On well-heated surfaces of the seawater side, $CO_2$ degassing and a shift in carbonate equilibrium occur, promoting the precipitation of $CaCO_3$, $Mg(OH)_2$, $CaSO_4$ [37, 39]. Simultaneously, a large amount of suspended solid particles is introduced, which mechanically settle in the "dead zones" of the channels, forming a dense layer of mixed mineral-mechanical deposits [37, 39].

A separate group consists of corrosion product deposits. Seawater is characterized by a high content of dissolved oxygen and chloride salts, creating an aggressive environment for carbon steels and even stainless steels in the presence of crevices, stagnant zones, and high temperatures. Corrosion products can additionally settle on plates, mixing with biofilms and carbonate scale [35, 37, 40].

For plate heat exchangers, the situation is complicated by the fact that they have very narrow channels, and seawater flow velocities (around 0,6–0,8 m/s) often prove to be optimal specifically for biofilm formation rather than for its removal [38].

Typical fouling for both surfaces and their average thermal conductivities are presented in Table 4.1. These data are necessary for the subsequent analysis of the impact of heat exchange surface fouling on the efficiency of the heat exchanger operation.

Table 4.2. Summary Table of Typical Fouling and Their Thermal Conductivity

| Heat Exchange Side | Characteristic Fouling Type | Average Value of Effective Thermal Conductivity λ |
|---|---|---|
| Fresh Water (FW) | Mixed corrosion-mineral layer: iron oxides + $CaCO_3$/$CaSO_4$ + sludge | ≈ 1.0 W/m·K |
| Seawater (SW) | Biofilm (diatoms) with an admixture of mineral scale and silt | ≈ 0.8–1.0 W/m·K (biofilm ~0.6; mixed layer up to ~1.5) |

## 4.3 Analysis of the Influence of Heat Exchange Surface Fouling on the Heat Flux and Hydraulic Losses of the Heat Exchanger

It should be noted that the quantity of heat transferred through the heat exchange surface $(Q = U \cdot \Delta\bar{t} \cdot A)$ is "subtracted" from the "hot" heat carrier $(Q = \dot{m}_{FW} c_{p,FW} (t_{FW,in} - t_{FW,out}))$ and transferred to the "cold" heat carrier $(Q = \dot{m}_{SW} c_{p,SW} (t_{SW,out} - t_{SW,in}))$. Thus, in a heat exchanger with a fixed heat transfer surface area and a specific overall heat transfer coefficient, a thermal flux $Q$ will be established, which will act as the determinant for the outlet temperatures of both heat carriers.

Consequently, a reduction in the overall heat transfer coefficient $U$ due to fouling of the heat exchange surface will lead to a decrease in the amount of heat transferred within the heat exchanger and a corresponding alteration in the outlet temperatures of the fluids.

### 4.3.1. Fouling of the Heat Exchange Surface on the Seawater Side

To demonstrate the trend of fouling influence, an additional calculation was performed for the heat transfer surface area determined above, comparing a clean surface against a surface fouled on the seawater side. The fouling scenario assumes the presence of a biofilm (diatoms) admixed with mineral scale and silt**,** characterized by a thermal conductivity of $\lambda \approx 1.0$ W/(m·K) and a layer thickness of 0.5 mm.

Figure 4.1 illustrates the approach for determining the heat load $Q$ of the heat exchanger. For this purpose, two dependencies $Q=f(t_{SW,out})$ were determined and plotted on a single graph. The intersection point of these dependencies determines the actual thermal load $Q$ and the seawater outlet temperature.

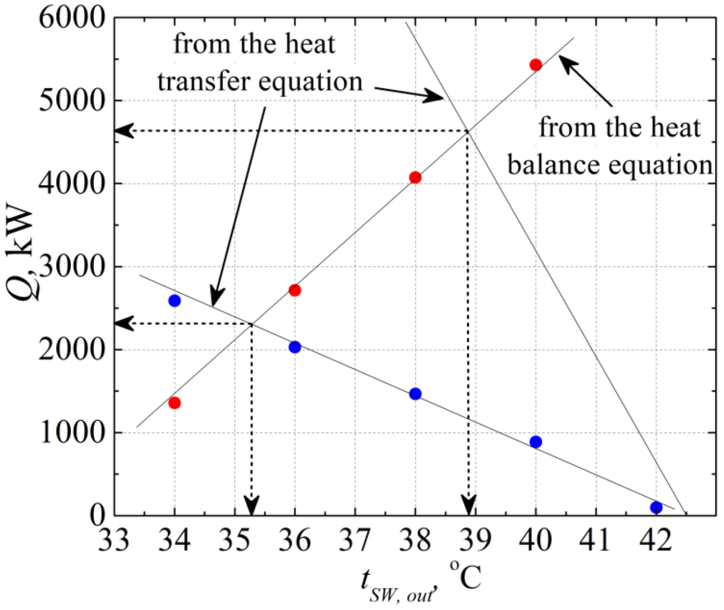


Figure 4.1 – Determination of the seawater (cold coolant) temperature at the heat exchanger outlet ($t_{SW,out}$) and the heat flux value $Q$ in the heat exchanger for a clean surface and for a surface with a scale layer on the seawater side; blue round markers – clean surface, blue square markers – heat exchange surface with fouling (thickness of 0.5 mm)

It is evident from Fig. 4.1 that fouling of the heat exchange surface on the seawater side very significantly reduces the heat flux transmitted in the heat exchanger. As a consequence, the temperature to which the seawater is heated and to which the fresh water is cooled is also reduced (this is not shown in Fig. 4.1).

Since all calculations were performed using Excel, the values of $Q$ and the temperature were estimated more accurately using the "Solver" function:

- For a clean surface: $Q$ = 4700 kW and temperature $t_{SW,out}$ = 38,9 °C;
- For a fouled surface: $Q$ = 2230 kW and temperature $t_{SW,out}$ = 35,3 °C.

As can be seen, the amount of heat transmitted in the heat exchanger in the presence of a fouling layer 0.5 mm thick decreases by almost half.

In addition, pressure losses (and, as a consequence, electricity consumption by pumps) increase significantly from 0.366 bar for a clean surface on the seawater side to 0.588 bar for a fouled surface.

### 4.3.2. Fouling of the heat exchange surface on the fresh water side

Similarly, we analyze the influence of surface fouling on the fresh water side. We assume that deposits of a mixed corrosion-mineral layer (iron oxides + $CaCO_3$/$CaSO_4$ + sludge) are formed on the fresh water side with a thermal conductivity of $\lambda \approx 1.0$ W/(m·K) and a thickness of 0.5 mm on the heat exchanger surface.

The calculation methodology is similar. Fig. 4.1 visually presents the obtained results. At the intersection point of the dependencies $Q=f(t_{FW,out})$, the thermal load $Q$ and the fresh water temperature at the heat exchanger outlet are determined.

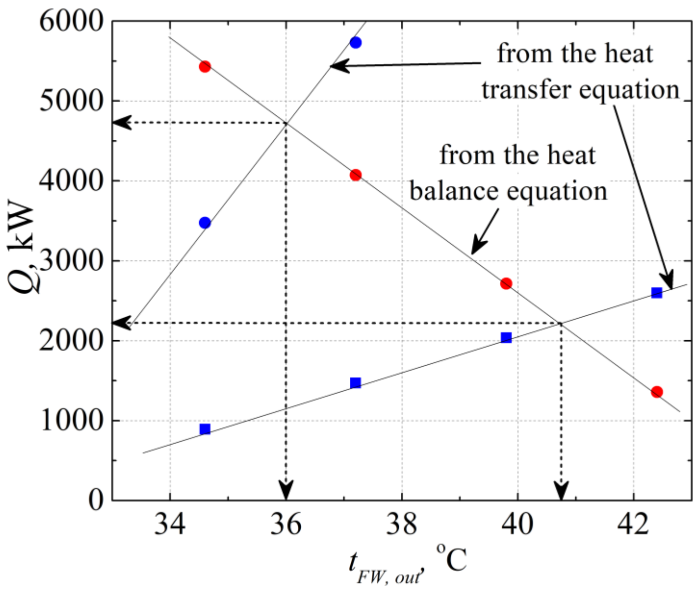


Figure 4.2 – Determination of the fresh water (hot coolant) temperature at the heat exchanger outlet $t_{FW,out}$ and the heat flux value $Q$ in the heat exchanger for a clean surface and for a surface with a scale layer on the fresh water side; blue round markers – clean surface, blue square markers – heat exchange surface with fouling (thickness of 0.5 mm)

It is evident from Fig. 4.2 that, by analogy with the previous analysis, fouling of the heat exchange surface on the fresh water side very significantly reduces the heat flux transmitted in the heat exchanger. As a consequence, the temperature to which the fresh water is cooled and to which the seawater is heated is also reduced (this is not shown in Fig. 4.1).

Since all calculations were performed using Excel, the values of $Q$ and the temperature were estimated more accurately using the "Solver" function:

- For a clean surface: $Q$ = 4700 kW and $t_{SW,out}$ = 38,9 °C; $t_{FW,out}$ = 36,0 °C;
- For a fouled surface: $Q$ = 2236 kW and $t_{SW,out}$ = 35,3 °C; $t_{FW,out}$ = 40,7 °C.

As can be seen, the amount of heat transmitted in the heat exchanger in the presence of a fouling layer 0.5 mm thick on the fresh water side decreases by almost half. Moreover, this value differs insignificantly from the heat flux with similar fouling on the seawater side.

The small difference in the value of $Q$ given the same thickness of fouling on the fresh water side and the seawater side (thermal conductivity coefficients are also approximately the same, despite the different composition of the fouling) is explained by the different velocities of the coolants and the influence of this velocity on the heat transfer coefficient.

In addition, pressure losses (and, as a consequence, electricity consumption by pumps) increase significantly from 0.648 bar for a clean surface on the fresh water side to 1.051 bar for a fouled surface. Such a significant pressure drop during fouling of the surface on the fresh water side (compared to the seawater side) is explained by the higher velocity of fresh water in the channels and the greater number of flow passes for fresh water compared to seawater (3 versus 2).

Thus, a general conclusion can be drawn: fouling on either side of the heat exchange surface affects the intensity of heat transfer and the heat flux transmitted in the heat exchanger approximately equally; however, fouling of the heat exchange surface on the fresh water side has a greater impact on pressure losses.

### 4.3.3 Analysis of the influence of fouling thickness on the seawater side on hydraulic losses in the heat exchanger

According to the methodology described above, the relationships between the pressure losses in the heat exchanger and the thickness of the fouling layer on the seawater side and on the fresh water side were obtained

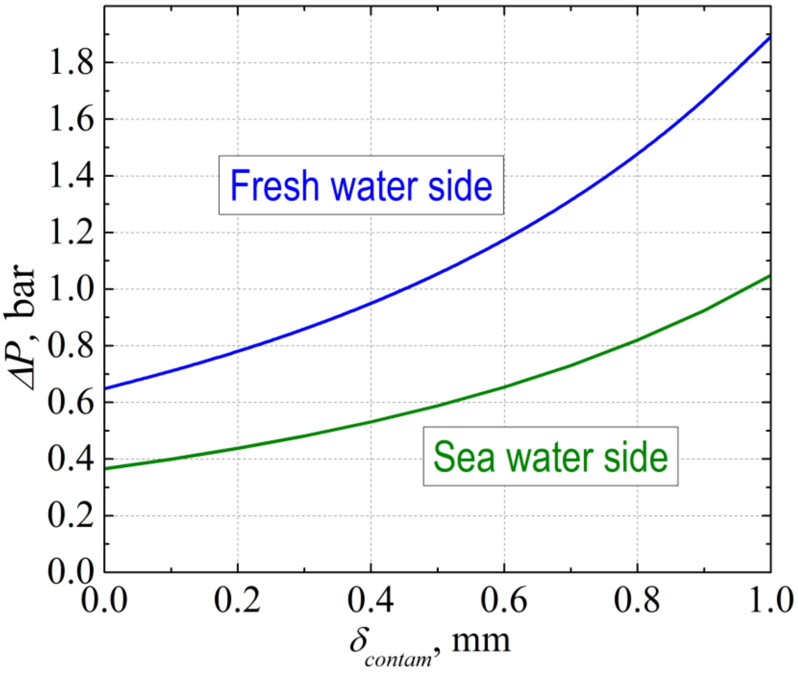


Figure 4.3 – Influence of fouling layer thickness on pressure drop on the fresh water and seawater sides of a plate heat exchanger

The influence of film of contamination on the hydraulic processes in a heat exchanger manifests itself primarily through a reduction of the effective flow cross-sectional area available for water passage. This leads either to an increase in flow velocity (if the volumetric flow rate remains constant) or to a decrease in the flow rate, which is associated with the operating characteristics of centrifugal pumps within a given hydraulic network.

Both the reduction in the hydraulic diameter and the increase in flow velocity, in accordance with the Darcy–Weisbach equation, result in an increase in pressure losses, as illustrated in Figure 4.3. It can be observed that an increase in the thickness of the fouling layer leads to a corresponding increase in pressure drop. As a consequence, this results in higher power consumption by the circulation pumps. Alternatively, beyond a certain fouling thickness, system operation must be interrupted to allow for cleaning of the heat exchangers in order to restore acceptable hydraulic performance.

The figure illustrates the effect of fouling layer thickness on the pressure drop (ΔP) in a plate heat exchanger for both the fresh water side and the seawater side. The pressure losses are shown as a function of the fouling thickness $\delta_{contam}$ in the range from 0 to 1.0 mm.

An increase in fouling thickness leads to a nonlinear growth of pressure drop on both sides of the heat exchanger due to the reduction of the effective flow cross-sectional area and the increase in hydraulic roughness of the channels. However, the rate and magnitude of this increase differ significantly between the two circuits.

On the fresh water side, the pressure drop increases more rapidly with fouling thickness. This is associated with the typically higher design velocity and the sensitivity of the closed-loop freshwater circuit to channel narrowing caused by deposits. According to the graph, the pressure drop approximately doubles when the fouling thickness reaches about 0.6 mm, increasing from roughly 0.65 bar to about 1.3 bar.

On the seawater side, the pressure drop also increases with fouling thickness but at a lower rate compared to the fresh water side. This behavior is attributed to lower design flow velocities and a greater initial hydraulic tolerance of the seawater circuit. From the graph, a doubling of pressure losses is observed at a fouling thickness of approximately 0.7 mm, where the pressure drop rises from about 0.35 bar to around 0.7 bar.

Overall, the figure demonstrates that even relatively small fouling thicknesses (less than 1 mm) can lead to a significant increase in hydraulic losses in plate heat exchangers. The results highlight the critical importance of fouling control, particularly for maintaining acceptable pressure losses and minimizing additional pumping power in marine cooling systems.

# 5. ANALYSIS OF METHODS FOR PREVENTING BIOFOULING OF THE HEAT EXCHANGE SURFACE

## 5.1 Legislative and Environmental Aspects of Applying Anti-Fouling Systems on Ships

The anti-fouling system is one of the primary means for preventing biofouling and controlling the underwater surfaces of ships, including the hull and niche areas [41]. An anti-fouling system can be:

- A coating system**,** where materials resistant to biofouling are applied to surfaces exposed to the marine environment;
- A Marine Growth Prevention System (MGPS) for sea chests and internal seawater cooling systems, applicable pipelines, and other unpainted components; or
- Other novel measures for biofouling control.

Design measures against biofouling during the design and construction phases regarding internal seawater systems recommend the following [41]: "Internal seawater cooling systems should be designed and manufactured from appropriate materials to minimize biofouling, and they should have a design with a minimal number of bends, loops, and flanges in the seawater piping."

Thus, in Part VI "Requirements for Anti-Fouling Systems" [41], primary attention is paid to anti-fouling coatings and the control of pollutant content within them, while anti-fouling systems for protecting internal seawater cooling systems are not described in detail.

## 5.2 Selection and Justification of a Rational Method for Removing Biological Fouling from the Heat Exchange Surface

*Design Aspects Minimizing Biofouling Adhesion*

The design of the heat exchanger can be a crucial factor in minimizing biological fouling on its surfaces. The main design and operational parameters influencing the biofouling growth process include [3]:

- Fluid velocity
- Material and roughness of the heat exchange surface
- Geometry of the flow path of the coolant (seawater)
- Surface temperature, coolant (seawater) temperature
- Physicochemical properties of the coolant (seawater)
- Biological variables in the coolant

It is difficult to link the biofilm formation process on the heat exchange surface to only one of these characteristics; however, it can be asserted that all of them influence the biofilm formation process on the surface in contact with seawater. It is recommended to maintain a minimum flow velocity of at least 0,9 m/s, use smooth heat exchange surfaces, and high temperatures (>90°C). Otherwise, biofouling will inevitably form on the heat exchange surface.

Obviously, a heat exchange surface that has been in operation for a certain period cannot remain perfectly smooth. It is also impossible to significantly alter the operating temperature regime of the heat exchanger. Therefore, there is a need to control biofouling through other methods.

*Chemical Anti-Fouling Treatment*

Chemical compounds used as anti-fouling agents in heat exchangers are called biocides. Biocides contain one or more active substances whose function is to destroy, counteract, neutralize, prevent the action of, or exercise another type of control over any harmful organism by chemical or biological means. Correct dosing drastically reduces the number and activity of microorganism cells in the water and the growth of biofilms. Due to their toxic nature, certain doses of biocides can harm the environment and are therefore subject to strict usage regulations [42] (regulation of this issue is discussed in the previous section). These regulations are becoming increasingly strict, leading to situations where the permitted biocide concentration becomes lower than the effective one, necessitating the search for alternative biofilm control methods.

There is a wide range of chemical substances that have been used as biocides to prevent the formation of biofouling deposits in heat exchange systems using seawater as a coolant. According to their general principle of action, they are divided into oxidizing and non-oxidizing agents. They act by oxidizing the organic matter of the microorganism and, consequently, causing its death. Their use is widespread, and their efficacy as an anti-fouling agent is widely proven.

*Biological Anti-Fouling Agents*

Biological anti-fouling biocides are active substances created by the organisms themselves (algae, bacteria, fungi, etc.) for self-protection against other organisms in their natural environment. These substances are usually proteinaceous in nature and are called enzymes.

Biological biocides are in the early stages of research and development. Their use requires in-depth analysis of the organisms targeted for destruction and the environment in which they are located. Recent studies have shown the inhibition of marine biofouling by various types of bacteria, cyanobacteria, and marine fungi [3].

*Physical Methods for Combating Biofouling of Heat Exchange Surfaces*

These represent an alternative to the use of biocides and are actively being studied, with some being actively applied. Such methods can be applied during the operation of the installation with heat exchange equipment (so-called online methods) or during the decommissioning of heat exchange equipment (offline methods) [43].

*Methods Applied During Heat Exchanger Operation (Online Methods)*

*Taprogge System.* A mechanical cleaning system using sponge balls or elastomeric abrasives involves removing biological fouling mechanically by periodically introducing special balls into the heat exchanger with the water flow to clean the tubes. Balls of appropriate roughness are introduced into the water box, from where they enter the tubes. The balls have a diameter larger than the tube diameter; they are elastic and pass through the tubes to clean the internal surface, avoiding the settlement of organic or inorganic substances.

*Brush Cleaning System.* Special baskets with a brush are installed at the ends of the tubes. The brush moves in both directions through the internal part of the tube due to the reversal of the cooling water flow. Reversal is carried out using a valve system. Usually, no more than three daily cycles are required to maintain good surface cleaning conditions.

*Thermal Treatment.* The approach involves heating the cooling water by recirculating it in a closed loop of the heat exchanger without mixing with cold water or by installing a secondary hot water circuit with additional heating. The water recirculates until it reaches a temperature of 50 to 70 °C, which leads to the death of microorganisms. The frequency of this treatment is established taking into account the biological activity of the seawater. The method's effectiveness depends mainly on water temperature, exposure time, and repetition frequency.

Changing Cooling Water Flow Velocity. The water flow velocity through the heat exchanger tubes is one of the determining factors in the biofouling rate. Biofilm that "sticks" to the internal surface of the heat exchanger tubes can be removed as a result of the impact of a high-velocity water flow on it. The heat exchanger can be designed to operate at high flow rates (at a velocity of 1 to 2 m/s) or so that the water flow velocity can be changed over a short period (for example, from 1 to 3 m/s) at regular intervals. This increase in flow velocity is possible only provided that the system pump power is increased.

*Pulse Injections.* Pulse injections can also be made by injecting gas (compressed air or nitrogen) at a very high velocity (3 m/s) periodically at short intervals (5, 10, and 60 minutes). The effect is to enhance turbulence, which creates pressure fluctuations inside the tube and increases shear stress on the tube surface. As a result, biofilm detachment occurs with greater efficiency than that achieved by simply increasing the water flow velocity [43]. However, in some cases, the biofilm becomes so compacted that removing it by this method is difficult.

*Electric Current Anti-Fouling.* The operating principle of this system is based on generating voltage between copper anodes (Cu) and built-in steel cathode plates, such that when Cu dissolves, it promotes the repulsion of marine organisms and prevents their deposition on the surface. The level of copper dissolution is regulated by a control unit, adapting the system to factors influencing the biofouling rate (for example, the season of the year).

*Ultraviolet Radiation Treatment.* Water sterilization by ultraviolet (UV) radiation is a potentially useful method for preventing the formation of a biological fouling film. Its principle of action is based on attacking the deoxyribonucleic acid (DNA) of cells, causing their death. Thus, the number of microorganisms present in the cooling water flow is minimized. This treatment is very effective against microorganisms, and its environmental impact is zero. Conversely, the useful life of a UV lamp is limited (approximately 8 months), and the cost of operation and maintenance is high, so they are usually not used in industrial installations with a large water flow. To reduce treatment costs and expand the spectrum of organisms against which treatment occurs, this method is usually combined with one of the other chemical or physical methods.

*Ultrasonic Treatment.* The mechanism of action of ultrasonic waves is based on creating vibration with the subsequent formation of cavitation bubbles. These bubbles form and then collapse with the release of a large amount of energy. These micro-explosions cause disturbances on the surface, disrupting the adhesion forces of bio-deposits to the surface, causing their detachment. The effect of ultrasonic treatment is greater the higher the frequency of ultrasonic vibrations. Such treatment is quite effective against fouling and does not affect the environment. However, the cost of ultrasonic installations is quite high both during their production and during operation.

*Filtration systems.* These systems are considered an additional method for eliminating biofouling in any seawater facility. The filtration prevents the early stages of various organisms, such as mussels, crustaceans, and others, from entering the seawater circuits. Filters partially reduce the formation of biofilm.

*Fresh Water Injection into Seawater Systems.* Microorganisms present in the biofilm feed on nutrients available in the seawater through osmotic processes (via a semi-permeable membrane). When fresh water is introduced, the saline concentration in the water changes, and the pressure inside the cells increases, causing their death. The holding time of the system when filled with fresh water must be high, reaching 48 hours or more. This method is used in piping systems that are not in operation for a long time.

*Tube Inserts for Flow Turbulization.* Usually, this is a helical element installed inside the tubes, which induces flow turbulence and promotes the detachment of biological fouling adhering to the inner surface of the tubes. An example of such a system is SPIRELF®. A metal spiral element attached to the ends of the tube and fitted to its length. When fluid circulates inside the pipe, it vibrates in the axial and radial directions, eliminating biofouling adhering to its walls. Inserting metal elements inside pipes has proven to be an effective method for preventing biofouling formation in heat exchangers of various industrial processes. However, the use of this type of device means an increase in equipment cost and maintenance, as well as higher energy costs for pumping due to increased pressure losses in the heat exchanger.

*Circulation of Polymers in the Heat Exchanger Circuit.* The method involves introducing polymer fibers entrained by the fluid flow to scour and detach the layer of adhering deposits. The effectiveness of the method depends on the concentration of polymer fibers and the cooling water flow velocity. The biggest disadvantage is the constant loss of polymer fibers with discharge waters and their release into the natural environment.

*Periodic methods for removing surface fouling during heat exchanger shutdown (off-line methods)*

The application of these anti-fouling methods is based on periodic scheduled shutdowns, during which aggressive cleaning techniques are used to restore the efficiency of the heat transfer surface. The frequency of scheduled cleaning is determined primarily by operational experience, while the cost–benefit ratio depends on the specific cleaning method selected for implementation.

Mechanical cleaning of heat exchanger surfaces using brushes, either in dry or wet conditions, is a highly labor-intensive process. The cost of such cleaning is relatively high and requires direct access to the heat exchanger surfaces. Moreover, not all surfaces can be effectively cleaned using this method, particularly complex geometries such as U-shaped tubes, as is the case for the heat exchanger analyzed in this master’s thesis.

Despite the wide variety of available cleaning techniques for industrial heat exchangers, the most commonly applied methods for cleaning the internal surfaces of tubes involve the use of air, water, and steam.

The selection of a cleaning agent depends on the type of fouling and the degree of adhesion of the deposits to the heat transfer surface.

*Pressurized air* is used to remove deposits that exhibit relatively weak adhesion to the surface.

*High-pressure water* is effective in removing biological fouling and calcium-based deposits in tubular heat exchangers.

*Steam* cleaning is applied to eliminate scale deposits and other substances with high hardness.

To enhance cleaning efficiency, detergents or abrasive materials may be added to pressurized water or air. The nature of the abrasive materials depends on the hardness and material of the heat transfer surface. Commonly used abrasives include sand, crushed mollusk shells, and similar materials.

### 5.3 Selection of methods for preventing and cleaning heat exchanger surfaces from biofouling

In practice, even when methods for preventing fouling of heat transfer surfaces are applied during operation, they are generally unable to eliminate the inherent need for periodic shutdown of the heat exchanger for cleaning scale and biological deposits [1]. When comparing mechanical and chemical cleaning methods, the latter offers certain advantages and is therefore widely used. Chemical cleaning, in particular, makes it possible to remove deposits from surfaces that are difficult or impossible to access using mechanical means. Moreover, the application of chemical methods facilitates the mechanization and automation of cleaning processes, significantly reducing labor requirements. At the same time, mechanical cleaning methods may be applied as a supplementary measure in combination with chemical cleaning.

On average, heat transfer surfaces in contact with seawater require shutdown and cleaning after approximately 2,500 hours of continuous operation (with specific requirements applying to the evaporator of a desalination unit). However, this interval may be adjusted as necessary depending on operating conditions.

At present, *the most widely used chemical methods for controlling biofouling in onboard seawater systems* include the following [1]:

- the use of copper anodes, which corrode at a controlled rate and release copper ions into the seawater flow;

- the application of electrolytic chlorination systems, during the operation of which a biocide containing chlorine is released into the seawater stream.

The widespread use of chlorination is associated with its high effectiveness at low concentrations, well-established technologies, and the relatively low cost of this fouling control method. In order to avoid the transportation and storage of chlorine, it is generated on site by means of electrolytic chlorination [44]. Seawater is used as the raw material for this process. At a pH value of 8.0–8.5, hypochlorous acid and hypochlorite ions are formed. When chlorinated water enters the ocean in large quantities, it may to some extent inhibit marine organisms; however, the environmental consequences are not considered catastrophic. It has been established [44] that the main biocidal components produced during chlorination—hypochlorous acid and hypochlorite ions—exhibit high chemical reactivity and therefore decompose rapidly, which prevents their accumulation in the marine environment.

Chlorination for the protection of marine seawater pipelines against fouling is recommended by many researchers. Studies have shown that even low chlorine concentrations (0.5 mg/L) are capable of preventing fouling. In this case, the larvae of marine organisms are not destroyed; however, their settlement process is effectively inhibited. According to some sources, a residual chlorine concentration of 0.25 parts per million in water provides complete protection against fouling. To remove existing fouling, the use of chlorine at concentrations of 3–4 mg/L is recommended, while for fouling prevention it is suggested to apply chlorine at concentrations of 2–3 mg/L for 15 minutes after every 6 hours of operation of the seawater system. Other authors recommend the use of higher chlorine concentrations (up to 12 mg/L) for 50 minutes every 8 hours. The application of lower chlorine concentrations (approximately 0.1 mg/L) is possible when continuous chlorination is employed.

A significant disadvantage of periodic chlorination is its low effectiveness against bivalve mollusks, which represent one of the primary fouling organisms in onboard seawater systems.

Another disadvantage of chlorination is the intensification of corrosion processes. For this reason, chlorine is sometimes replaced with sodium pentachlorophenolate at a concentration of 1 mg/L or sodium hypochlorite at a concentration of 0.3–0.5 mg/L, which is injected into the pipeline once per day for a duration of one hour.

Some researchers recommend the use of active oxygen compounds instead of chlorine [1], due to their complete environmental and human safety. Ozone or 35% hydrogen peroxide is used as a source of active oxygen. The use of carbon dioxide at a concentration of 170 mg/L has also been proposed as an alternative to chlorination.

Another widely applied biocidal additive for fouling protection is copper sulfate. It is recommended to be used at concentrations of 1–2 mg/L with treatment intervals of 1–2 months for preventive fouling control, or under more severe conditions at concentrations of 5–6 mg/L for one hour every 2.5 days. In cases of intensive fouling development, it is advised to increase the copper sulfate concentration to 10–15 mg/L. Unlike chlorine, copper sulfate is highly effective in destroying mollusks.

The disadvantages of using copper sulfate are associated with the harmful environmental impact of copper salts and with the fact that, under intensive application, they can significantly accelerate corrosion processes. As an alternative to copper sulfate, soluble copper salts can be generated on site by means of anodic dissolution of copper.

Other substances used to prevent biofouling in shipboard seawater systems include ferric chloride, calcium dimethyldithiocarbamate, Busan-90, Pyror-400, sodium ethylenebisdithiocarbamate, organochlorine compounds, phenol-containing compounds, copper–ammonia complexes, acrolein, quaternary ammonium compounds, and others. In some cases, organotin compounds are also applied, despite their high environmental hazard; therefore, their use is subject to strict regulatory control.

*The most widespread physical method currently used to combat biofouling in shipboard seawater systems* is the application of heated water. This method is particularly cost-effective in cases where waste heat is available. The required water temperature and exposure time vary depending on the type of bioorganisms involved. Studies have shown that the cleaning of water pipelines, primarily from mussel fouling, can be effectively achieved by flushing with water heated to 37 °C for 3–4 hours at 21-day intervals; however, other research indicates that a temperature of 42 °C is necessary. Additional sources recommend water temperatures of 44 °C or 50 °C with shorter exposure cycles. Nevertheless, in practice, it is generally advisable to use water at a temperature of approximately 70 °C.

### 5.4 Electrolytic Seawater Chlorination System

The electrolytic chlorination system (electrochlorination equipment, electrolyzer) prevents the accumulation of marine organisms, such as microorganisms and mollusks, through the direct electrolysis of seawater, resulting in the formation of sodium hypochlorite, which acts as a biocide.

Electrolyzers are considered the most effective electrochlorination technology for the on-site production of sodium hypochlorite solutions through the electrolysis of seawater (or sodium chloride brine in the case of freshwater applications) [45-48].

The electrochlorination process primarily consists of the partial electrolysis of a solution containing sodium chloride, which is present in large quantities in seawater. The product generated as a result of the electrolysis is sodium hypochlorite (NaOCl). Sodium hypochlorite is a chemical compound widely recognized as an effective disinfectant and biocide due to its strong oxidizing properties. Its primary applications include the disinfection of drinking water sources and the control of marine biological fouling in industrial and marine systems that use seawater in their cooling circuits. The effective operating dosage of sodium hypochlorite (NaOCl) is extremely low, typically ranging from 0.3 to 6 ppm.

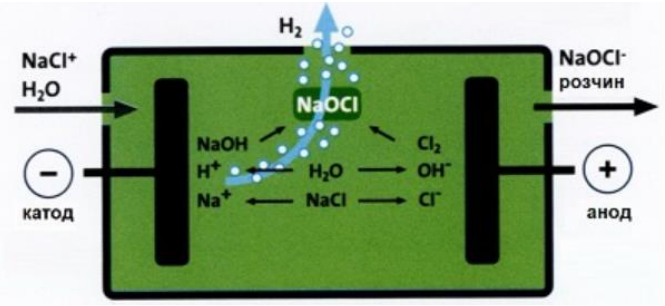


Figure 5.1. Principle of sodium hypochlorite generation from seawater [45]

A typical electrochlorination system is a standalone unit, generally mounted on steel skids within the vessel's engine room. It comprises one or more electrolytic cells containing titanium anodes and cathodes. Mixed Metal Oxide (MMO) electrodes serve an electrocatalytic function and are essential for chlorine production via electrolysis. The direct current (DC) supplied to the titanium electrodes is generated by an AC/DC transformer/rectifier and control circuitry that regulates the DC supply to the electrolytic cells. The system may incorporate dosing pumps, sampling points, and a hypochlorite storage tank, and can be configured for both marine and onshore applications.

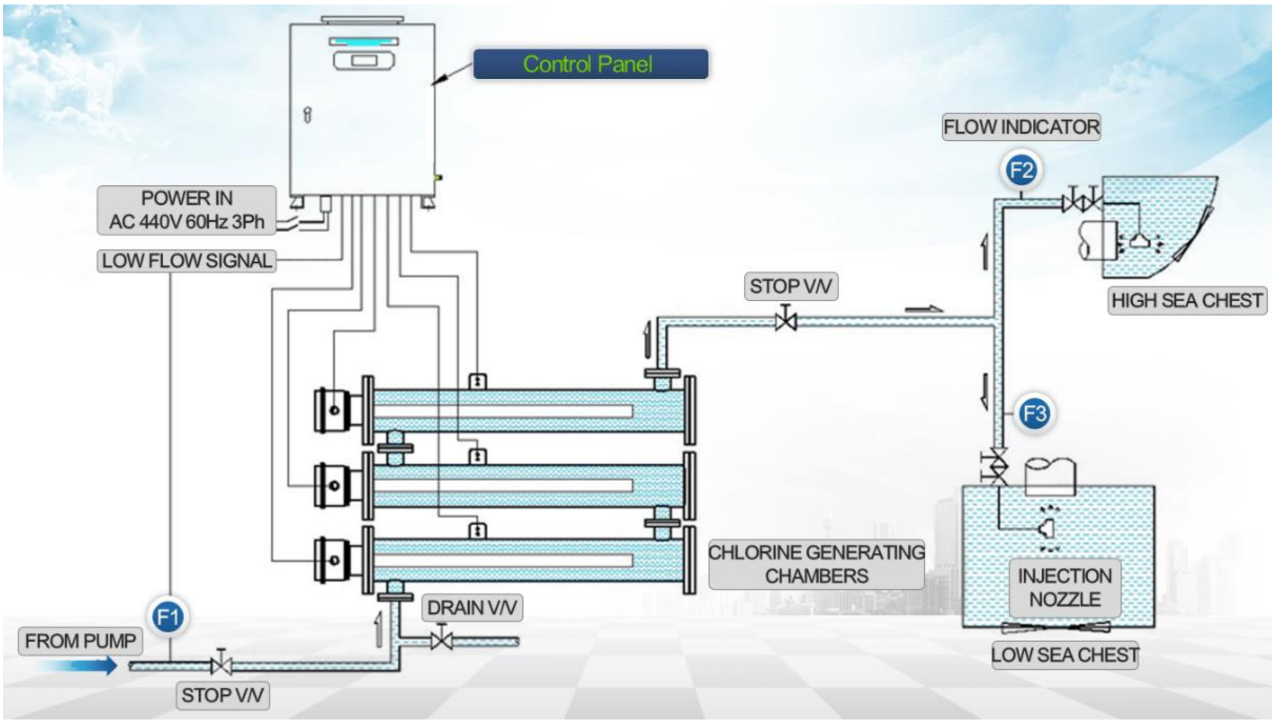


Figure 5.2. Schematic representation of a typical electrochlorination system in a vessel's engine room [48]

*Typically, the system operates as follows:*

Following seawater intake via pumps, the water passes through a filter before undergoing a treatment stage to remove marine organisms and sediments. This stage varies across vessels, as different methods may be employed for microbial water treatment.During overboard discharge, the Total Residual Oxidant (TRO) level is monitored. If the concentration is excessive, the water is neutralized—for instance, by the addition of sodium bisulfite—to prevent the discharge of significant quantities of sodium hypochlorite into the marine environment.

### 5.5 Economic Aspects of Biofouling Prevention on Heat Exchange Surfaces in Contact with Seawater

Fouling of heat exchange equipment compromises both the hydraulic and thermal efficiency of the equipment. The economic aspects of fouling encompass both design and operational stages [49]. Costs at the heat exchange equipment design stage are associated with increased capital expenditures. The heat transfer area of the exchanger is increased (oversized) to compensate for fouling effects anticipated during the design phase. Operational stage costs include: additional maintenance expenses for cleaning operations, including materials and reagents, and costs incurred to compensate for the loss of heat exchanger efficiency (such as additional pumping energy). Data presented in [1] indicates that for shipboard seawater system operating conditions over 2,500 hours, savings of $16,500 USD can be achieved by implementing the recommended cleaning schedule for all systems (every 600 hours). Similarly, under different conditions, an analysis of the impact of biofouling on a vessel's cooling system determined a monthly cost of excessive fuel consumption amounting to $464,000 USD due to performance losses in the main engine heat exchangers caused by biofouling. Therefore, the mitigation of biofouling in internal vessel systems presents significant economic justification.

Thus, the total costs associated with the fouling of heat exchange surfaces can be categorized into four main components [49]:

- *Additional capital expenditures* for the fabrication of larger heat exchangers, accounting for excess surface area, costs for additional space requirements, and increased transportation and installation costs.
- *Additional energy costs* due to reduced thermal efficiency and increased pressure losses.

•*Losses incurred during technological processes* due to scheduled and unscheduled shutdowns for cleaning operations (typically less characteristic for marine equipment, as critical equipment is generally redundant).

•*Maintenance costs,* including the cleaning of heat exchange equipment and the utilization of antifoulants.

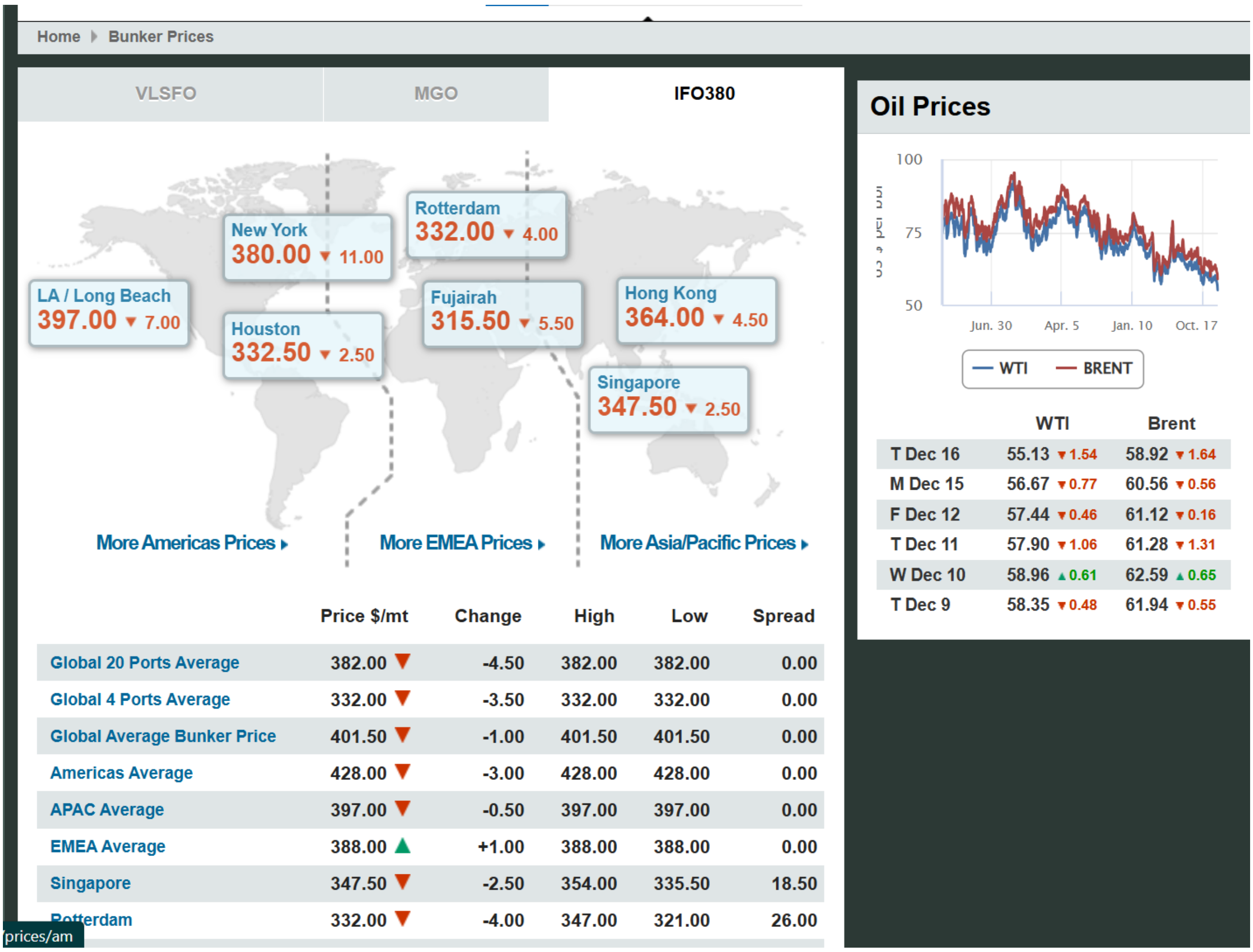
Additional fuel consumption and monetary costs resulting from reduced thermal efficiency and increased pressure losses can be estimated as follows. The increase in pump power requirement due to fouling of the heat exchange surface is:

$$N_p = \frac{G_x\left(\Delta p_{забр} - \Delta p\right)}{\eta_p}\left(\frac{Q}{Q_{забр}}\right), \tag{5.1}$$

where $\Delta p$ and $\Delta p_{забр}$ - are the pressure drops across the heat exchanger under clean and fouled conditions, respectively (Pa); $Q$ and $Q_{забр}$ - are the heat transfer rates (amount of heat transferred) in the heat exchanger under clean and fouled conditions, respectively (kW); $\eta_p$ - is the pump efficiency; for centrifugal pumps, an average value of 0.75 is assumed. Pumps consume electrical energy generated onboard by diesel generators. The specific consumption of marine diesel fuel per 1 kWh of electricity onboard averages $g_{MDO}$=0,25 L/kWh [50]. Consequently, the additional diesel fuel consumption (in kg/h) associated with the increased energy consumption of the pumps due to heat exchange surface fouling (assuming an average marine diesel fuel density of 0.82 kg/L) is:

$$G_{MDO} = N_p \cdot g_{MDO} \cdot \rho_{MDO}\,. \tag{5.2}$$

To calculate monetary expenses, the average market price of IFO380 fuel on the global market is adopted — Fig. 4.3.



| | Price $/mt | Change | High | Low | Spread |
|---|---|---|---|---|---|
| Global 20 Ports Average | 382.00 ▼ | -4.50 | 382.00 | 382.00 | 0.00 |
| Global 4 Ports Average | 332.00 ▼ | -3.50 | 332.00 | 332.00 | 0.00 |
| Global Average Bunker Price | 401.50 ▼ | -1.00 | 401.50 | 401.50 | 0.00 |
| Americas Average | 428.00 ▼ | -3.00 | 428.00 | 428.00 | 0.00 |
| APAC Average | 397.00 ▼ | -0.50 | 397.00 | 397.00 | 0.00 |
| EMEA Average | 388.00 ▲ | +1.00 | 388.00 | 388.00 | 0.00 |
| Singapore | 347.50 ▼ | -2.50 | 354.00 | 335.50 | 18.50 |
| Rotterdam | 332.00 ▼ | -4.00 | 347.00 | 321.00 | 26.00 |

| | WTI | Brent |
|---|---|---|
| T Dec 16 | 55.13 ▾ 1.54 | 58.92 ▾ 1.64 |
| M Dec 15 | 56.67 ▾ 0.77 | 60.56 ▾ 0.56 |
| F Dec 12 | 57.44 ▾ 0.46 | 61.12 ▾ 0.16 |
| T Dec 11 | 57.90 ▾ 1.06 | 61.28 ▾ 1.31 |
| W Dec 10 | 58.96 ▴ 0.61 | 62.59 ▴ 0.65 |
| T Dec 9 | 58.35 ▾ 0.48 | 61.94 ▾ 0.55 |

Figure 5.3. World Bunker Prices for IFO380 [51]

Based on the data presented in Fig. 5.3, the average cost of IFO380 marine fuel is assumed to be $C_{MDO}$= 382 USD per metric ton, i.e.$C_{MDO}$ = 0.382 USD/kg

$$C_1 = G_{MDO} \cdot C_{MDO} \cdot \tau , \qquad (5.3)$$

Where τ is the calculation period in hours, which is assumed to be one month for the purpose of this comparative analysis.

Additional monetary costs for maintenance, including heat exchanger cleaning and the utilization of additional reagents, are estimated according to the selected maintenance method.

Here, we account for biofouling prevention costs using sodium hypochlorite according to the following scheme: dosing chlorine at a concentration of 2 mg/L for 15 minutes every 6 hours of seawater system operation. As data on the operational costs of electrolytic chlorination are currently unavailable for this stage of the study, calculations are performed based on the cost of a ready-made sodium hypochlorite solution with an active chlorine content of at least 190 g/L (an option also feasible onboard). The average cost of such a solution is $C_{реаг}$=2,05 USD/L.

Table 5.1 Dependence of additional fuel consumption for seawater pump operation on the fouling thickness of the heat transfer surface

| $\delta_{contam}$, mm | $\Delta P$, *bar* | *Q, kW* | $N_p$, *kW* | $G_{MDO}$, *kg/hout* | $C_1$, *USD per monht* |
|---|---|---|---|---|---|
| 0 | 0.3657 | 4700 | 0.0 | 0.0 | 0.0 |
| 0.1 | 0.3995 | 4098 | 0.8 | 0.2 | 47.6 |
| 0.2 | 0.4378 | 3550 | 2.1 | 0.4 | 117.1 |
| 0.3 | 0.4814 | 3056 | 3.9 | 0.8 | 218.2 |
| 0.4 | 0.5311 | 2616 | 6.5 | 1.3 | 364.6 |
| 0.5 | 0.5882 | 2230 | 10.2 | 2.1 | 575.4 |
| 0.6 | 0.6541 | 1898 | 15.5 | 3.2 | 876.2 |
| 0.7 | 0.7305 | 1620 | 23.0 | 4.7 | 1298.8 |
| 0.8 | 0.8198 | 1396 | 33.3 | 6.8 | 1876.1 |
| 0.9 | 0.9248 | 1226 | 46.6 | 9.6 | 2630.0 |
| 1 | 1.0491 | 1110 | 63.0 | 12.9 | 3550.6 |

The duration of seawater treatment per month, according to the described scheme, is $\tau_{обр}$=30 hours.

The consumption rate of sodium hypochlorite solution with an active chlorine content of at least $x_{рег}$ =190 g/L required to achieve a water concentration of $x_{обр}$=2 mg/L (0,002 g/L) at a water flow rate of $G_x$=54,4 kg/s ~ approx 55 L/s is:

$$G_{реаг} = G_x \frac{x_{обр}}{x_{рег}}, \tag{5.4}$$

$$G_{реаг} = 55\frac{0,002}{190} = 0,000579 \text{ L/s.}$$

The monetary cost of the sodium hypochlorite solution per month for seawater treatment to prevent biofouling is:

$$C_2 = C_{реаг} \cdot G_{реаг} \cdot \tau_{обр}, \tag{5.5}$$

$C_2 = 2{,}05 \cdot 0{,}000579 \cdot 30 \cdot 3600 = 128{,}3$ $ USA на місяці.

The results of the calculation of the two components of total monetary costs associated with heat exchange surface fouling—specifically, additional fuel costs and biofouling prevention costs—are presented in Fig. 5.4.

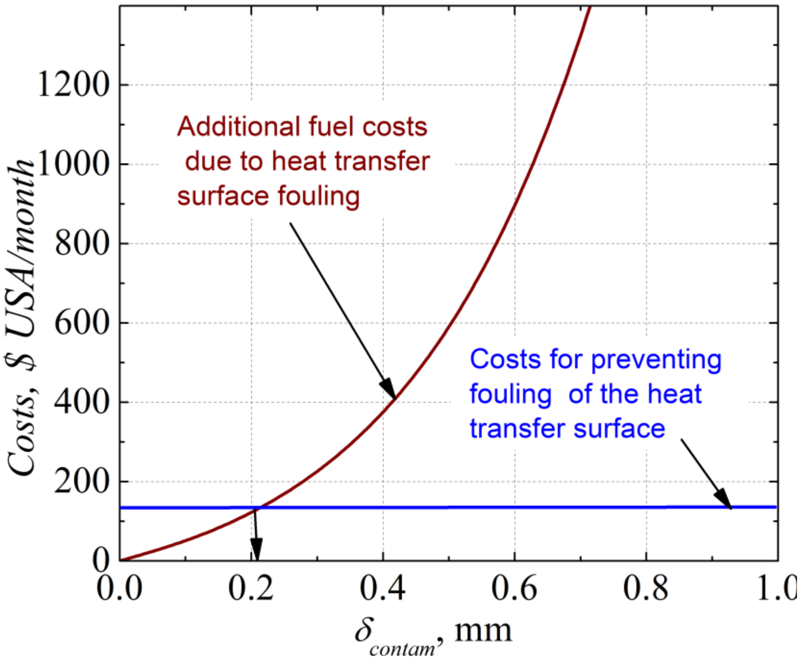


Figure 5.4. Comparison of monetary costs for additional fuel due to heat exchange surface biofouling versus biofouling prevention measures for a single plate heat exchanger with a surface area of 160 m².

The figure 5.4 presents a quantitative comparison between the additional fuel costs caused by heat transfer surface biofouling and the costs associated with fouling prevention measures for a plate heat exchanger with a heat transfer area of 160 m².

The cost of fouling prevention measures is approximately 130 USD per month. In contrast, the additional fuel costs resulting from biofouling increase nonlinearly with increasing fouling layer thickness due to the rapid growth of hydraulic losses and pump power demand.

At a fouling thickness of approximately 0.2 mm, the additional fuel costs reach about 130 USD per month, which is comparable to the cost of fouling prevention. This point represents the economic break-even fouling thickness. For fouling thicknesses below 0.2 mm, the additional fuel costs remain lower than the prevention costs, whereas beyond this value, further operation without preventive measures becomes economically inefficient.

As the fouling thickness increases further, the additional fuel costs rise sharply. At a fouling thickness of approximately 0.4 mm, the additional fuel expenses increase to around 400 USD per month, which is roughly three times higher than the cost of prevention. At 0.6 mm, the additional fuel costs reach approximately 900 USD per month, exceeding the prevention costs by nearly a factor of seven.

Based on this analysis, it can be concluded that fouling prevention measures should be implemented once the fouling layer thickness reaches approximately 0.2 mm. Beyond this threshold, preventive treatment is not only technically justified but also economically advantageous, as it prevents a rapid escalation of operating costs caused by biofouling-induced hydraulic losses.

Additionaly, it should be noted that this analysis was performed for a single heat exchanger unit. In contrast, biofouling prevention via the periodic addition of sodium hypochlorite to the seawater intake will affect the entire seawater hydraulic system while maintaining the same reagent costs. Consequently, the actual additional fuel costs attributable to biofouling will exceed those shown in Fig. 5.4, thereby further increasing the economic feasibility of seawater treatment.

## 5.6 Chapter Conclusions

Following a detailed analysis of biofouling control methods for internal shipboard seawater systems—and heat exchangers as a critical component thereof—it has been established that the most environmentally and economically viable method is the application of periodic electrolytic chlorination of seawater. The frequency and duration of treatment should be selected based on seawater temperatures and the operational region. However, even this method cannot entirely eliminate the need for the periodic decommissioning of the heat exchanger to perform mechanical, or combined mechanical and chemical, cleaning of the internal tube surfaces. It has been established that sodium hypochlorite degrades rapidly upon discharge overboard; furthermore, excessive concentrations in the discharge water can be neutralized. Therefore, it does not exert a detrimental impact on marine organisms (hydrobionts), unlike tin and copper compounds, for instance, which are also utilized as biocides.

Calculations indicate that additional fuel costs (resulting from increased hydraulic losses and reduced thermal load on the heat exchanger during surface fouling) amount to $117 and $575 USD per month for biofilm thicknesses of 0.2 mm and 0.5 mm, respec-tively, for a plate heat exchanger with an area of 160 m$^2$. Meanwhile, the monthly costs for biofouling prevention using sodium hypochlorite amount to $128 USD.

Thus, the simplified calculation demonstrates that biofouling prevention costs are recovered only once a biofilm of 0.2 mm thickness has formed. However, it must be emphasized that this analysis was conducted for a single heat exchanger unit. The implementation of biofouling prevention via the periodic addition of sodium hypochlorite to the seawater will protect the entire seawater hydraulic system while incurring the same reagent costs.

# CONCLUSION

The master's scientific research is aimed at solving a scientific and applied problem, namely improving the efficiency of thermo-hydraulic processes in a plate heat exchanger designed for cooling fresh water by seawater, as well as reducing fuel consumption during its operation by preventing fouling of the heat exchange surface on the seawater side.

The reliability and efficiency of operation of seawater heat exchangers are of great importance for the functioning of the entire vessel. With the increasing focus on improving ship energy efficiency, the role of these heat exchangers will become even more significant. The formation of fouling on heat exchange surfaces represents a serious economic and environmental problem.

For the purpose of the calculations, a ship power plant equipped with a 12V Wärtsilä 46F main engine with an effective power output of 8600 kW was adopted. The thermal loads from diesel generators and auxiliary systems cooled by the low-temperature fresh water circuit were also taken into account. Based on the determined amount of heat to be removed, the fresh water flow rate in the low-temperature circuit was calculated as 124.9 $m^3/h$, while the seawater flow rate amounted to 162.4 $m^3/h$. Based on the obtained thermal capacity, a thermal design calculation of the central cooler was performed and the required heat transfer surface area was determined to be 160 $m^2$. A standard GEA NT (Marine Duty) NT250S plate heat exchanger was selected.

The study determined the influence of fouling on the heat exchange surfaces in contact with both fresh water and seawater on the operating efficiency of the heat exchanger.

The analysis showed that the presence of even a small fouling layer leads to deterioration in heat exchanger performance. At a fouling layer thickness of 0.5 mm on both the fresh water side and the seawater side, the amount of transferred heat decreases approximately equally from Q = 4700 kW to about 2230 kW. At the same time, pressure losses increase significantly from 0.366 bar for a clean surface on the seawater side to 0.588 bar for a fouled surface. In addition, pressure losses increase from 0.648

bar for a clean surface on the fresh water side to 1.051 bar for a fouled surface. Such a significant pressure drop during fouling of the surface on the fresh water side (compared to the seawater side) is explained by the higher velocity of fresh water in the channels and the greater number of flow passes for fresh water compared to seawater (3 versus 2).

The reduction in the heat flux transferred from fresh water to seawater in the presence of a biofilm is explained by its low thermal conductivity, which is close to that of water. Thus, the biofilm creates a significant thermal resistance to heat transfer. The increase in pressure losses during seawater flow is explained by the increase in flow velocity due to the reduction in the effective cross-sectional area.

Thus, it has been confirmed that the main direction for improving the efficiency of seawater heat exchange equipment is the prevention of biofouling and the timely cleaning of heat exchange surfaces.

After a detailed analysis of methods for controlling biofouling in onboard seawater systems and heat exchangers as one of their components, it was established that the most environmentally and economically feasible method is the application of periodic electrolytic chlorination. However, even this method cannot eliminate the need for periodic removal of the heat exchanger from operation and the application of mechanical or combined mechanical and chemical cleaning of the internal surface of the tubes.

The analysis of available literature sources showed that when sodium hypochlorite is discharged overboard, it decomposes rapidly, and at high concentrations in discharge water it can be neutralized. Therefore, its application does not have a detrimental impact on the environment.

Calculations indicate that additional fuel costs (resulting from increased hydraulic losses and reduced thermal load on the heat exchanger during surface fouling) amount to $117 and $575 USD per month for biofilm thicknesses of 0.2 mm and 0.5 mm, re-spec-tively, for a plate heat exchanger with an area of 160 m$^2$. Meanwhile, the monthly costs for biofouling prevention using sodium hypochlorite amount to $128 USD.

Thus, the simplified calculation demonstrates that biofouling prevention costs are recovered only once a biofilm of 0.2 mm thickness has formed. However, it must be emphasized that this analysis was conducted for a single heat exchanger unit. The implementation of biofouling prevention via the periodic addition of sodium hypochlorite to the seawater will protect the entire seawater hydraulic system while incurring the same reagent costs.

# REFERENCES

1. Davidson I., Cahill P., Hinz A., Kluza D., Scianni C., Georgiades E. A review of biofouling of ships’ internal seawater systems. *Frontiers in Marine Science*. 2021. Vol. 8. Article 761531.

2. Davidson I., Cahill P., Hinz A., Major R., Kluza D., Scianni C., Georgiades E. Biofouling occlusion of ships' internal seawater systems: operational, economic, and biosecurity consequences. *Biofouling.* 2023. Vol. 39, No. 4. P. 410–426.

3. García S., Trueba A. Fouling in Heat Exchangers. In: Inverse Heat Conduction and Heat Exchangers. London: IntechOpen, 2019.

4. GEA PHE Systems. GEA Plate Heat Exchangers. Product brochure. Germany: GEA Group, 2011.

5. Kelvion. Industry Solutions: Transportation – Marine On-Board Systems. https://www.kelvion.com/industry-solutions/transportation/marine-on-board-systems (Avalaible: 12.10.2025).

6. Alfa Laval. *M30 Plate heat exchanger: Product brochure*. Available at: https://alfalaval.si/uploads/gia2/public/document/78-giaflex_m30_sl.pdf (Accessed: 22 November 2025).

7. Unnikrishnan A., Mulky L. Overview of fouling – an industrial jeopardy. ChemBioEng Reviews. 2023. Vol. 10, No. 4. P. 597–610.

8. International Maritime Organization. Fourth IMO GHG Study 2020. Executive summary. London: IMO, 2021.

9. Lotti T., Carretti E., Berti D., Montis C. Hydrogels formed by anammox extracellular polymeric substances: structural and mechanical insights. *Scientific Reports*. 2019. Vol. 9. Art. 11633.

10. Beech I., Bergel A., Mollica A., Flemming H.-C., Scotto V., Sand W. Simple methods for the investigation of the role of biofilms in corrosion. *Microbially Influenced Corrosion of Industrial Materials – Biocorrosion Network (Brite-Euram III Thematic Network № ERB BRRT-CT98-5084), Task 1: Biofilms Publication*. September 2000.

11. ASHRAE. ASHRAE Handbook. Refrigeration. Atlanta: American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc., 2019.

12. ANSI/ASHRAE 26-2010 (R2020). Mechanical Refrigeration and Air-Conditioning Installations Aboard Ship. Atlanta: ASHRAE, 2020.

13. Admiralty Industries. Best tube materials for shell and tube heat exchangers. URL: https://admiraltyindustries.com/blog/best-tube-materials-for-shell-and-tube-heat-exchangers/ (Avalaible: 12.10.2025).

14. Titanium Marine. Shell & Tube Heat Exchangers. URL: https://titaniummarine.com/titanium-heat-exchangers/ (Avalaible: 12.10.2025).

15. GeneralCargoShip.com. Maintenance of marine heat exchangers. URL: http://generalcargoship.com/maintenance-of-heat-exchangers.html (Avalaible: 12.10.2025).

16. MarineEngineeringOnline.com. Maintenance of cooler or heat exchanger on ships. URL: https://marineengineeringonline.com/maintenance-of-cooler-or-heat-exchanger-on-ships/ (Avalaible: 12.10.2025).

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

18. Alfa Laval. Plate heat exchangers for marine applications: Product brochure. Lund : Alfa Laval Corporate AB, 2020. [https://www.alfalaval.com/globalassets/documents/products/heat-transfer/plate-heat-exchangers/brochures/plate-heat-exchangers-for-marine-applications.pdf (Avalaible: 17.11.2025).

19. Alfa Laval. Gasketed plate heat exchangers – Operating principles and design features. Lund : Alfa Laval Corporate AB, 2019. https://www.alfalaval.com/globalassets/documents/products/heat-transfer/plate-heat-exchangers/gasketed-plate-heat-exchangers-operating-principles.pdf (Avalaible: 17.11.2025).

20. GEA PHE Systems. GEA PHE Systems – Tailor-made plate heat exchanger solutions. Product brochure, GEA Heat Exchangers, Düsseldorf, Germany

21. Amalfi R. L., Marcinichen J. B., Thome J. R. Towards Green Technology: Modeling of a Compact Plate Heat Exchanger Condenser for Thermosyphon Cooling of Entire High Power Datacenter Racks. *Electronics Cooling*, 2019.

22. MAN Diesel & Turbo. MAN B&W S50MC-C8-TII Project Guide: Camshaft Controlled Two-Stroke Engines. Copenhagen SV, Denmark: MAN Diesel & Turbo; 2010

23. Alfa Laval. Plate heat exchanger calculation method. https://www.alfalaval.com/microsites/gphe/tools/calculation-method/?utm_source=chatgpt.com (Avalaible: 02.11.2025)

24. VDI Heat Atlas. VDI-Gesellschaft Verfahrenstechnik und Chemieingenieurwesen (VDI-GVC). 2nd international ed. Berlin: Springer-Verlag, 2010. 1544 p.

25. Bhattad A., Atgur V., Rao B.N., Banapurmath N.R., Manavendra G., Sajjan A.M., Badruddin I.A., Tambrallimath V., Kamangar S., Hussien M. A simplified LMTD approach to assess the effectiveness of a chevron-type plate heat exchanger. *Journal of Thermal Analysis and Calorimetry*. 2024. Vol. 149, No. 21. P. 12205–12217.

26. Muley A., Manglik R. M. Experimental Study of Turbulent Flow Heat Transfer and Pressure Drop in a Plate Heat Exchanger With Chevron Plates. *Journal of Heat Transfer*. 1999. Vol. 121(1). P. 110–117.

27. Idelchik I. E. *Handbook of Hydraulic Resistance*. 3rd ed., Begell House, 1994.

28. RefProp: Reference fluid thermodynamic and transport properties, NIST standard reference database. Version 9.1 mini (teaching tool for the introduction of thermodynamics to students). https://refprop-mini.software.informer.com/9.1/

29. Wärtsilä. *Wärtsilä 46F Product Guide. Section 9.1 "Cooling water system – Water quality"*. 2008.

30. MAN Diesel & Turbo. *Service Letter SL2016-623: Cooling water treatment and periodical test of cooling water*. 2016.

31. Arsenyeva O.P., Crittenden B.D., Yang M., Kapustenko P.O. Accounting for the thermal resistance of cooling water fouling in plate heat exchangers. *Chemical Engineering and Processing: Process Intensification*. 2013.

32. Chardon Laboratories. *Corrosion & Scale in Closed Loops – Prevention & Treatment*. Technical article. Chardon Labs.

33. Water Technologies (Veolia / Betz). Deposit and Scale Control – Cooling System; Corrosion Control – Cooling Systems. In: *Handbook of Industrial Water Treatment*. Watertechnologies.com.

34. Heat Exchanger World. Fouling and plate heat exchangers. *Heat Exchanger World Magazine*. 2020.

35. Davidson I., Cahill P., Hinz A., Kluza D., Scianni C., Georgiades E. A review of biofouling of ships' internal seawater systems. *Frontiers in Marine Science*. 2021. Vol. 8. Art. 761531.

36. Davidson I., Cahill P., Hinz A., Major R., Kluza D., Scianni C., Georgiades E. Biofouling occlusion of ships' internal seawater systems: operational, economic, and biosecurity consequences. *Biofouling*. 2023. Vol. 39, No. 4. P. 410–426.

37. Pugh S. Fouling during the use of seawater as coolant. In: *Fouling of Heat Exchangers*. ECI Conference Proceedings. 2003.

38. Murthy P.S., Venkatesan R., Nair K.V.K. Biofilm control for plate heat exchangers using surface treatment and megasonic cleaning. *Biofouling*. 2004. Vol. 20, Nos. 4–5. P. 247–257.

39. Arsenyeva O.P., Kapustenko P.O., Demirskiy O.V., Khavin G.L. The water fouling development in plate heat exchangers. *Chemical Engineering Research and Design*. 2022. Vol. 181. P. 364–377.

40. Mathew N.T., Kronholm J., Bertilsson K., Despeisse M., Johansson B. Environmental and economic impacts of biofouling on marine and coastal heat exchangers. 2021. (Conference / technical report).

41. Регістр судноплавства України. Правила запобігання забрудненню з суден. Бюлетень №1. Вводиться в дію 01.01.2022 року.

42. Eguía E, Trueba A. Application of marine biotechnology in the production of natural biocides for testing on environmentally innocuous antifouling coatings. *Journal of Coatings Technology and Research*. 2007. Vol 4. P. 191- 202.

43. Bott TR. Industrial Biofouling. Elsevier. 2011. 220 p.

44. The NALCO Water Handbook, Fourth Edition. An Ecolab Company. NALCO Water. New York: McGraw-Hill Education, 2017/2018

45. Selcoperm Electrolysis Units (Grundfos) https://aquapolymer.com.ua/obladnannya/systemy-elektrolitychnogo-hloruvannya-selcoperm-ses-195-selcoperm/ (Accessed 28 November 2025)

46. ELECTROCHLORINATION SYSTEMS, https://www.acgmarine.com/water-treatment/electrochlorination/?lang=en (Accessed 28 November 2025)

47. How does an electrochlorination system work? https://kraftpowercon.com/news/how-does-an-electrochlorination-system-work (Accessed 28 November 2025)

48. Electrochlorination Hypochlorite System https://cathodicme.com/services/electrochlorination-hypochlorite-system/ (Accessed 28 November 2025)

49. Ezgi C., and Ozbalta N. Optimization of heat exchanger cleaning cycle on a ship. *J. Naval Sci. Eng*. 2012. Vol. 8. P. 33–46.

50. A Practical Guide to the Selection of Energy Efficiency Technologies for Ships. GreenVoyage2050 Project Coordination Unit, IMO (2022) https://greenvoyage2050.imo.org (Accessed 12 November 2025

51. World Bunker Prices. https://shipandbunker.com/prices#IFO380 (Accessed 23 November 2025)