Energy efficiency improvement for the older reefer vessel by the combined compression-ejector refrigeration machine

Viktor Yalama¹, Mykhailo Khmelniuk²
¹²Odesa National University of Technology, 1/3, Dvoryanska str., Odesa, 65082, Ukraine
² e-mail: ²hmel_m@ukr.net
ORCID: ¹https://orcid.org/0000-0002-8827-4537; ²https://orcid.org/0000-0002-9310-1286

A marine combined compression-ejector refrigeration machine that comprises conventional refrigeration and air-conditioning system using single-stage refrigeration machine and an ejector refrigeration machine is proposed and studied. The ejector refrigeration machine is driven by the waste heat from the single-stage refrigeration machine and acts as the bottom cycle of the single-stage refrigeration machine. A system analysis shows that the COP of a compression-ejector refrigeration machine is significantly higher than a one-stage refrigeration system. Improvement in COP can be as high as 19.3% for evaporating temperature of the single-stage refrigeration machine $T_c$ at -40°C. Influence of flowing part of ejector profile, operating parameters, scheme and cycle of the marine combined compression-ejector refrigeration machine on its performance characteristics is shown. Rational methods to increase efficiency of marine combined compression-ejector refrigeration machine, operating with environmentally friendly low-boiling refrigerant R245fa, are offered and investigated. The present study shows that the combined compression-ejector refrigeration machine using the ejector refrigeration machine as the bottom cycle of the one-stage refrigeration machine is viable. The opportunities of implementing solar panels, raised challenges for older reefer vessels (20-40 years) force to analyse situation. While opportunities exist for efficiency improvement and environmental alignment, challenges in retrofiting older vessels, especially those nearing the end of their operational life, make economic viability a critical consideration. The analysis of the possible solar panel application estimates a cost range of 398,269.20 to 426,717.00 euros, emphasizing the need for careful evaluation. The conclusion advocates against implementing solar panels on the researched vessel, highlighting the lack of justification in both economic and operational efficiency aspects.

**Keywords:** Older reefer vessel; Energy efficiency; Ejector refrigeration machine; Combined compression-ejector refrigeration machines; Eco-friendly refrigerant

doi: [https://doi.org/10.15673/ret.v59i3.2664](https://doi.org/10.15673/ret.v59i3.2664)
\( \beta \) – ratio of \( f_2/f_3 \);
\( \varphi_1, \varphi_2, \varphi_3, \varphi_4 \) – velocity coefficients;
\( \theta \) – dimensionless temperature;
\( \zeta \) – heat coefficient;
\( \eta \) – efficiency coefficient of the feed pump.

1. Introduction

The refrigerated ship market is currently experiencing active development, with numerous types and models of these vessels available. Modern refrigerated ships are typically equipped with advanced technologies that ensure precise and stable temperatures inside the vessel. They can operate on both diesel and gas fuel, making them more environmentally friendly. Additionally, refrigerated ships can be equipped with special control and monitoring systems that track temperature, humidity, gas levels, and other parameters inside the ship.

Among the manufacturers of refrigerated ships, notable companies include Maersk Line, CMA CGM, Hapag-Lloyd, MSC, NYK Line, and others. Furthermore, there are many smaller companies specializing in the production of refrigerated containers and vessels for smaller shipments.

The overall carrying capacity of refrigerated ships can vary from a few thousand to over 30,000 tons. These vessels come in various types and sizes, including container ships, bulk carriers, and roll-on/roll-off (ro-ro) ships. The refrigerated ship market continues to grow as more countries show interest in exporting their food and seafood products to other regions of the world.

Currently, vapor compression refrigeration machines (VCRM) driven by electric motors are predominantly used in the air conditioning systems of ships.

The electrical power consumed by the ship’s refrigeration equipment reaches significant levels, leading to additional fuel costs and the need to increase the power of the ship’s power station. The average energy consumption of a refrigeration system on a ship can range from 50 to 300 kW, depending on the type of refrigeration system and its operating mode.

Our research on the existing single-stage ship refrigeration system has shown that installing additional compressors and transitioning to two-stage compression can increase system performance and ensure a more uniform load on compressors when operating conditions change on the ship [1].

One way to improve energy consumption for existing and newly constructed ships is to utilize the heat from the main and auxiliary engine exhaust gases. This is relevant in light of new regulations on energy efficiency [2] and greenhouse gas emissions [3,4,5]. Heat losses from exhaust gases for ship power plants can reach from 30% to 50% and higher, depending on the type of installation and operating conditions.

A comparative analysis of various refrigeration machines shows that ejector refrigeration machines (ERM) utilizing low-boiling-point working fluids are the most promising for ship air conditioning and refrigeration systems [1,6-9]. Numerous studies have demonstrated that ERM has several advantages, including reliability, simplicity of design and maintenance, cost-effectiveness, and compact size [9-14].

However, despite several advantages over other refrigeration machines, ERMs have drawbacks, with the most significant being the comparatively low relative heat loss ratio. Additionally, until now, they have predominantly used low-boiling refrigerants such as R11, R12, R141b, R142b, etc., which do not meet modern environmental requirements [10-14].

Therefore, further research aiming to develop shipboard waste heat recovery ERMs that meet modern requirements for energy, economic, and environmental performance and are competitive in the global refrigeration machine market is of significant interest [15-19].

The analysis (Clarksons Research, Shipping Intelligence Network timeseries, 2023) indicates a complex landscape for the shipping industry, marked by challenges, resilience, and the need for sustainable practices amidst global uncertainties. The key aspects of the global shipping industry according to the main points: post-pandemic trends, maritime trade volume, decarbonization challenges, war in Ukraine impact, oil and gas trade, regional variations and future challenges. The shipping industry is navigating post-pandemic trends, with the legacy of the 2021–2022 global supply chain crunches and a softening in the reefer shipping market. There are shifts in shipping and trading patterns due to the war in Ukraine. Maritime trade volume contracted marginally by 0.4% in 2022, but UNCTAD (United Nations Conference on Trade and Development) projects a 2.4% growth in 2023, indicating resilience in the industry. Continued but mod-
Opportunities for solar panel use in the older reefer vessel, fishing carrier (20-40 years) do not cover challenges

Opportunities for solar panel use can be presented as: efficiency improvement, technological modifications, policy initiatives, hybrid system use, environmental compliance, cost saving over time, sustain-ability goals completing. Efficiency Improvements due upgrading older vessels with more energy-efficient technologies, alongside solar panels, can enhance overall efficiency and reduce reliance on conventional fuel sources. By retrofitting older vessels provides an opportunity to integrate modern, more efficient electrical systems that can better accommodate solar power generation. Through some regions offer incentives specifically aimed at the modernization and greening of older vessels. Financial support or tax incentives may encourage shipowners to invest in solar panel installations. With respect of Integrating solar panels into a hybrid system alongside existing power sources can provide a balanced approach, optimizing energy use and extending the operational life of older vessels. For modifying older vessels with solar panels aligns with evolving environmental regulations and can enhance compliance with emissions reduction requirements. It is possible despite the initial retrofitting, costs, the long-term savings from reduced fuel consumption and maintenance costs can make solar panel installations economically viable. Goals can be reached by shipping companies with older fleets may adopt solar power as part of their sustainability initiatives, enhancing their corporate image and meeting the expectations of environmentally conscious stakeholders.

During the research studies, observation of real-world effects a set of challenges has been discovered for solar panel use in older reefer vessels and fishing carriers. First of all, it is the structural adaptation where older vessels may not have been designed with solar panel installation in mind. Retrofitting the structure to accommodate solar panels could be challenging and may require extensive modifications. Other problem is the limited deck space where older vessels might have smaller deck spaces, making it difficult to find suitable areas for solar panel installation. The limited space may restrict the capacity for solar energy generation. The outdated electrical systems can be the problem for older vessels which may have outdated or less efficient electrical systems that are not well-suited for integrating solar power. Upgrading these systems to work seamlessly with solar installations can be complex. Next rising problem is weight constraints where older vessels may have stricter weight limitations, and adding the weight of solar panels, along with necessary equipment, could impact the vessel’s stability and performance. The corrosion and wear can make problem for solar panel use where aging vessels may have more corrosion and wear, and exposure to a harsh ma-
The marine environment could accelerate the degradation of solar panels, reducing their effectiveness over time.

While challenges exist in retrofitting older vessels with solar panels, there are substantial opportunities to improve their efficiency, reduce environmental impact, and align with modern sustainability standards. The key lies in careful planning, technological upgrades, and leveraging available incentives to make such installations economically and environmentally beneficial [2] but it is not acceptable for older reefers, fishing carriers. Based on the analysis of the issues conducted, it can be stated that the feasibility of using solar panels on the older reefer vessel, fishing carrier is not justified, neither from an economic perspective nor in terms of system efficiency after adapting solar panel technology.

3. Ejector refrigeration: technologies

We have developed new thermodynamic cycles and configurations for combined compression-ejector refrigeration machines (CCERM), aiming to enhance the energy efficiency of the refrigeration cycle.

Figure 1 illustrates the configuration of the CCERM system, which can find widespread application in modern ship air conditioning and refrigeration systems that utilize waste heat from the main engine exhaust gases.

![Figure 1 - CCERM layout with internal combustion engine heat recovery](image)

In the system presented in Fig. 1, the ERM generator is heated directly by the exhaust gases, and the ERM itself operates with an environmentally friendly and safe refrigerant such as carbon dioxide (R744), ammonia (R717), and hydrocarbons. Cold water circulates in a closed loop, removing heat from the VCRM condenser, thereby increasing the energy efficiency of the refrigeration cycle by operating at lower condensation temperatures well below the temperature of the onboard seawater intake [1-10].

In the investigated single-stage VCRM, the working substance vapor is compressed in the compressor from the pressure $p_e$ to the pressure $p_c$. The compressed superheated vapor, with a high temperature ($t_2$) ranging from 80 to 100°C and correspondingly high specific exergy, then enters the condenser, where, through irreversible heat exchange with the cooling medium, the vapor is cooled and condensed. The exergy of the rejected heat is transferred to the surrounding environment and irreversibly lost.

A thermodynamic analysis of the VCRM cycles was conducted for the combined cold and heat generation to determine the possibility of transforming this heat into cold using the ERM. It was established that a rational method to increase the efficiency of VCRM with a high value of $t_2$ is the transformation of a portion of the superheated vapor's heat into cold, which is then used to cool the liquid after the condenser.

Fig. 2 shows one of the schematic solutions for a combined CCCERM, representing a non-traditional cascade refrigeration machine consisting of a single-stage VCRM and a single-stage ERM. The VCRM, which forms the lower stage of the cascade, operates as follows: Vapor generated in the evaporator 1 is compressed in the compressor 2 and enters the generator-cooler 3, where the main part of the superheating heat is removed.

The cooled vapor then enters the condenser 4, where further cooling and condensation occur at temperature $t_c$. The liquid refrigerant after condenser 4 is subcooled in the evaporator-subcooler 5, throttled in the control valve 6, and enters the evaporator 1 to produce cold.

The ERM is the upper stage of the cascade, and its operation is as follows: Saturated working vapor of the refrigerant with a temperature $t_G$, formed in the generator-cooler 3 due to the heat input from the superheated vapor of the compression cycle, enters the nozzle of the ejector 7, expands in it, and draws in the cold vapor from the evaporator-subcooler 5. The compressed vapor mixture in the diffuser of the ejector 7 enters the condenser 8, where it is liquefied at temperature $t_c$. The liquid leaving the condenser 8 is divided into two streams, one of which reduces its pressure in the control valve 9 and enters the evapor-
tor-subcooler 5, while the second stream, through the feed thermo-pump 10 of the float-lever principle of action, returns to the generator-cooler 3.

![Diagram of CCERM layout with condensation heat recovery](image)

Figure 2 – Combined CCERM layout with condensation heat recovery

Unlike a conventional cascade machine, the proposed CCERM has two separate condensers for each stage, from which heat is dissipated to the environment at the same temperatures \( t_c \). It also has two combined heat exchangers – the generator-cooler and the evaporator-subcooler – in which heat is dissipated from the lower stage to the upper stage at different temperature levels, respectively, above and below the temperature \( t_c \). Thus, in the CCERM, the heat from the compressed superheated vapor is usefully employed for the operation of the upper stage, and the cold obtained in the ERM is used to subcool the liquid refrigerant after the condenser of the lower stage. This leads to an increase in the coefficient of performance of the vapor compression cycle.

We proposed and developed a combined compression-ejector refrigeration system operating on a continuous basis, schematically presented in Fig. 2.

4. System analysis of the combined compression-ejector refrigeration machine

A system analysis of the combined compression-ejector refrigeration machine is carried out in the present study. Governing equations based on the conservation of energy and mass are derived for every component of the CCERM.

4.1. Governing equations of the components in CCERM

4.1.1. Generator

In the generator, the refrigerant in the ERM undergoes a phase-change process (evaporation) and the refrigerant in the VCRM undergoes a cooling process at vapor state. The generator is thus a heat exchanger like an evaporator. The neck-point temperature difference of the generator is \( \Delta T_{NG} \) which is a given design parameter for the generator. Denoting the state at the neck point as state 02 for the SSRM refrigerant vapor and as state 01’ for the ERM, we obtain the following governing equations:

\[
T_{02} = T_g = T_{NG} \quad (1)
\]

\[
m_r (h_{02} - h_g) = m'_r (h'_{01} - h'_g) \quad (2)
\]

\[
h_{02} = h_g (T_{02}, P_r); h_2 = h_g (T_2, P); h'_{01} = h_j (T_{01'}) \quad (3)
\]

\[
T_{01'} = T_g \quad (4)
\]

\[
T'_3 = T'_g \quad (5)
\]

\[
h'_3 = h'_{01} - c'_p (T_{01'} - T'_3) = h'_1 - c'_p (T'_{g} - T'_3) \quad (6)
\]

\[
m_r (h_2 - h_{02}) = m'_r (h'_3 - h'_{01}) \quad (7)
\]

\[
Q_g = Q'_g = m_r (h_2 - h_g) = m'_r (h'_3 - h'_g) \quad (8)
\]

4.1.2. ERM ejector

The primary component of the ERM is the ejector – a jet device designed to suction refrigerant vapor from the evaporator, compress them, and discharge them into the condenser (Fig. 3). Similar functions in a traditional VCRM are performed by a compressor, driven by an electric motor.

The key efficiency indicator of the ejector is the ejector efficiency coefficient \( U \), calculated by the equation:

\[
U = \frac{G_0}{G_e} \quad (9)
\]

where, \( G_0 \) and \( G_e \) represent the flows of the low-boiling working fluid through the evaporator-air cooler (ejected low-boiling working fluid at low pressure) and the generator (low-boiling working fluid at high pressure), respectively.
The efficiency of conversion of waste heat into refrigeration capacity is characterized by coefficient of performance \( \zeta = Q_0 / Q_s \) as the ratio of the refrigeration capacity \( Q_0 \) (heat extracted from the air at the diesel exhaust outlet to the low-boiling working fluid boiling in the evaporator-air cooler) to the consumed heat \( Q_s \), supplied to the boiling low-temperature working fluid at high pressure extracted from the engine exhaust gases and the supercharging air. It is calculated without considering the work of the feed pump and is shown as:

\[
\zeta = \frac{Q_0}{Q_s} = U \cdot \frac{q_0}{q_s} \quad (10)
\]

To justify the choice of rational methods for improving ERM efficiency at the initial stage of the study, let’s conduct an analysis that relates the cycle’s operating parameters and the thermodynamic properties of the working substance to the energy and operational performance of the ERM.

From equation (10), it follows that to increase \( \zeta \) one must strive to increase \( U \) and \( q_0 \) while reducing \( q_s \). In turn, to increase \( U \), it is necessary to increase the specific work of the working vapor expansion in the nozzle \( a = h_2 - h_1 \) and reduce the specific work of compressing the vapor mixture in the diffuser \( l_2 = h_3 - h_4 \).

An analysis of the influence of cycle operating parameters and thermodynamic properties of the refrigerant on the ejector efficiency coefficient shows that \( U \) increases with an increase in the value \( E = p_v - p_0 \) and the product \( p_v v_1 \) and a decrease in the value \( e = p_v / p_0 \) and the product \( p_v v_2 \). The adiabatic indices of the expansion process \( k_e \) and compression \( k_c \) processes differ slightly and do not have a significant impact on the value of \( U \).

In addition to thermodynamic properties, when choosing the working substance for the ERM, it is subjected to a series of traditional requirements: it must be environmentally harmless, inexpensive, non-aggressive to structural materials, thermally stable, non-toxic, non-flammable, explosion-safe, etc.

The new, non-flammable, environmentally safe refrigerant R245fa (CHF₂CH₂CF₃), with ODP = 0, GWP = 0.082, \( \mu = 134.05 \) kg/mol, \( t_{CR} = 154 \) °C, and \( t_5 = 14.9 \) °C, meets these requirements to the greatest extent and has been chosen as a promising low-boiling working substance for the ship ERM.

Thus, the thermal coefficient of the ERM depends on the thermodynamic properties of the working substance and the operating parameters of the cycle. The value of \( \zeta \) is higher for refrigerants with large values of \( t_{CR} \) and \( p_0 \) and small values of \( c_p \) [7-15].

![Figure 3 – Scheme of the flow part of the ejector with a conical-cylindrical mixing chamber: I – nozzle; II – suction chamber; III – mixing chamber; IV – diffuser](image-url)

In deriving the calculation equations to determine the maximally achievable \( U \), a one-dimensional model was used as a basis, with refinements and corrections obtained in the work [12]. It was established that the limit mode, corresponding to the maximally achievable ejector efficiency coefficient under given cycle operating parameters, is characterized by the ejective flow reaching the critical velocity at the inlet section of the mixing chamber.

For the simplification of the derivation of calculation equations in the proposed methodology, the following assumptions were made:

- The working and ejector flows have the same adiabatic index \( k \) and gas constants \( R \);
- Before entering the mixing chamber, on the section between the outlet section of the nozzle 1-1 and the inlet section of the mixing chamber 2-2, the working flow does not expand and does not mix with the ejector flow;
- The thickness of the edge of the outlet section of the working nozzle is assumed to be zero;
- Initial velocities of the working and ejector flows are considered to be zero due to their small values compared to the velocities of these flows in the mixing chamber.

The equation for calculating the ejector coefficient of the ejector with a conical-cylindrical mixing chamber (CCMC), operating on substances with the same physical properties, is given by [12].

\[
U \cdot \sqrt{\Theta} = \frac{K_1 \cdot \lambda_{e,0} - K_3 \cdot \lambda_{e,3}}{K_1 \cdot \lambda_{e,3} - K_2 \cdot \lambda_{e,0}} \quad (11)
\]

where

\[
K_i = \varphi_1 \cdot \varphi_2 \cdot \varphi_3 \quad (12)
\]
The calculation task involves finding such gas dynamic functions for these flows that the achievable ejector coefficient has the maximum value. In the model proposed by [12], this problem is solved using the method of successive approximations. It should be noted that this method is labor-intensive and, moreover, has low computational accuracy. Therefore, the authors propose another approach to solving this problem. It consists of determining the maximally achievable value of $U$ reduces to a nonlinear programming problem for the objective function (eq.11) when the ejector operates in the first or third limit mode.

4.1.3. ERM Condenser
Assuming that the condensate at the exit of the condenser (point 3') in the ERM is at a saturated-liquid state. The governing equations of the ERM condenser are

$$T'_v = T'_e$$  (22)
$$P'_v = P(T'_v)$$  (23)
$$h'_v = h_v(T'_v)$$  (24)
$$Q'_v = (m'_v - m'_e) \times (h'_v - h'_e)$$  (26)

4.1.4. ERM precooler
The precooler in the ERM is a simple heat exchanger that can be characterized by the effectiveness:

$$\eta_{pc} = \frac{T'_e - T'_v}{T'_v - T'_e}$$  (27)

where $\eta_{pc}$ is a given design parameter in the precooler design. Eq. 27 can be used to determine a temperature if the rest two temperatures are known.

4.1.5. Subcooler
The subcooler is used to subcool the liquid condensate in the VCRM by using the evaporation heat of the ERM. The subcooler is basically a heat exchanger like an evaporator with the refrigerant in the ERM undergoing an evaporating process. We assume that the thermodynamic state at the exit of the subcooler (state 8') for the ERM is a saturated-vapor state and the neck point (state 7') temperature difference of the subcooler is $\Delta T_{NS}$. $\Delta T_{NS}$ is given as the heat exchanger design parameter. Therefore, we obtain the following governing equations:

$$K_2 = \phi_2 \cdot \phi_3 \cdot \phi_4;$$  (13)

$$K_4 = 1 + \phi_4 \cdot \frac{p_c}{p_0} \cdot \Pi_\alpha - \frac{p_0}{p_c} \times \left[ (\beta - 0.5 \cdot (\beta - 1) \cdot \Pi_\alpha \cdot \left[ 1 + \left( \frac{p_c}{p_0} \right)^{l-1} \cdot \left( \frac{\Pi_\alpha}{\Pi_\beta} \right)^{l-1} \right] \right]$$  (14)

$$P_\alpha = \Pi_{gP} = \Pi_{0x}$$  (16)

The calculation task involves finding such gas dynamic functions for these flows that the achievable ejector coefficient has the maximum value. In the model proposed by [12], this problem is solved using the method of successive approximations. It should be noted that this method is labor-intensive and, moreover, has low computational accuracy. Therefore, the authors propose another approach to solving this problem. It consists of determining the maximally achievable value of $U$ reduces to a nonlinear programming problem for the objective function (eq.11) when the ejector operates in the first or third limit mode.

4.1.3. ERM Condenser
Assuming that the condensate at the exit of the condenser (point 3') in the ERM is at a saturated-liquid state. The governing equations of the ERM condenser are

$$T'_v = T'_e$$  (22)
$$P'_v = P(T'_v)$$  (23)
$$h'_v = h_v(T'_v)$$  (24)
$$Q'_v = (m'_v - m'_e) \times (h'_v - h'_e)$$  (26)

4.1.4. ERM precooler
The precooler in the ERM is a simple heat exchanger that can be characterized by the effectiveness:

$$\eta_{pc} = \frac{T'_e - T'_v}{T'_v - T'_e}$$  (27)

where $\eta_{pc}$ is a given design parameter in the precooler design. Eq. 27 can be used to determine a temperature if the rest two temperatures are known.

4.1.5. Subcooler
The subcooler is used to subcool the liquid condensate in the VCRM by using the evaporation heat of the ERM. The subcooler is basically a heat exchanger like an evaporator with the refrigerant in the ERM undergoing an evaporating process. We assume that the thermodynamic state at the exit of the subcooler (state 8') for the ERM is a saturated-vapor state and the neck point (state 7') temperature difference of the subcooler is $\Delta T_{NS}$. $\Delta T_{NS}$ is given as the heat exchanger design parameter. Therefore, we obtain the following governing equations:

$$K_2 = \phi_2 \cdot \phi_3 \cdot \phi_4;$$  (13)

$$K_4 = 1 + \phi_4 \cdot \frac{p_c}{p_0} \cdot \Pi_\alpha - \frac{p_0}{p_c} \times \left[ (\beta - 0.5 \cdot (\beta - 1) \cdot \Pi_\alpha \cdot \left[ 1 + \left( \frac{p_c}{p_0} \right)^{l-1} \cdot \left( \frac{\Pi_\alpha}{\Pi_\beta} \right)^{l-1} \right] \right]$$  (14)

$$P_\alpha = \Pi_{gP} = \Pi_{0x}$$  (16)

The calculation task involves finding such gas dynamic functions for these flows that the achievable ejector coefficient has the maximum value. In the model proposed by [12], this problem is solved using the method of successive approximations. It should be noted that this method is labor-intensive and, moreover, has low computational accuracy. Therefore, the authors propose another approach to solving this problem. It consists of determining the maximally achievable value of $U$ reduces to a nonlinear programming problem for the objective function (eq.11) when the ejector operates in the first or third limit mode.

4.1.3. ERM Condenser
Assuming that the condensate at the exit of the condenser (point 3') in the ERM is at a saturated-liquid state. The governing equations of the ERM condenser are

$$T'_v = T'_e$$  (22)
$$P'_v = P(T'_v)$$  (23)
$$h'_v = h_v(T'_v)$$  (24)
$$Q'_v = (m'_v - m'_e) \times (h'_v - h'_e)$$  (26)

4.1.4. ERM precooler
The precooler in the ERM is a simple heat exchanger that can be characterized by the effectiveness:

$$\eta_{pc} = \frac{T'_e - T'_v}{T'_v - T'_e}$$  (27)

where $\eta_{pc}$ is a given design parameter in the precooler design. Eq. 27 can be used to determine a temperature if the rest two temperatures are known.

4.1.5. Subcooler
The subcooler is used to subcool the liquid condensate in the VCRM by using the evaporation heat of the ERM. The subcooler is basically a heat exchanger like an evaporator with the refrigerant in the ERM undergoing an evaporating process. We assume that the thermodynamic state at the exit of the subcooler (state 8') for the ERM is a saturated-vapor state and the neck point (state 7') temperature difference of the subcooler is $\Delta T_{NS}$. $\Delta T_{NS}$ is given as the heat exchanger design parameter. Therefore, we obtain the following governing equations:
\[ T' = T'' = T''' \] \hspace{1cm} (28)
\[ T = T'' + T''_{NS} \] \hspace{1cm} (29)
\[ h'_s = h_s (T') \] \hspace{1cm} (30)
\[ h'_f = h'_f \] \hspace{1cm} (31)
\[ Q'_s = m'_s (h'_s - h'_f) \] \hspace{1cm} (32)
\[ Q_c = m_c (h_c - h_c) \] \hspace{1cm} (33)
\[ Q'_c = Q_c = m'_c (h'_c - h'_f) = m_c (h_c - h_c) \] \hspace{1cm} (34)

### 4.1.6. VCRM condenser

Assuming that the condenser of the VCRM has a degree of subcooling, \( \Delta T_{SC} \), the governing equations are:

\[ h_s = h_s (P, T_s) \] \hspace{1cm} (35)
\[ T_s = T_c - \Delta T_{SC} \] \hspace{1cm} (36)
\[ h_c = h_f (T_e) - c_p \Delta T_{SC} \] \hspace{1cm} (37)
\[ Q_c = m_c (h_c - h_c) \] \hspace{1cm} (38)

### 4.1.7. VCRM compressor

The compressor undergoes a non-isentropic process for vapor compression. The power input to the compressor can be represented by the following equation:

\[ W_c = m_c (h_c - h_c) / \eta_c \] \hspace{1cm} (39)

where \( \eta_c \) is the compression efficiency, including the motor loss. The outlet temperature \( T_c \) of the compressor can be determined by thermodynamic equation of state:

\[ T_c = f (h_c, P_c) \] \hspace{1cm} (40)

where

\[ h_c = h_c + (h_c - h_c) / \eta_s \] \hspace{1cm} (41)

where \( \eta_s \) is the isentropic efficiency of the compression process.

\[ h_{c2} = h_c (T_{c2}, P_c) \] \hspace{1cm} (42)
\[ T_{c2} = h_c (P_c, s2 = s1) \] \hspace{1cm} (43)

### 5. Discussion and results

Economic viability of solar panel application to the vessel under research becomes a critical consideration. Justifying the overall cost of retrofitting older vessels with solar panels must align with potential energy savings. This economic feasibility is particularly challenging for reefer, fishing carrier approaching the end of its operational life. Older reefer, fishing carrier might necessitate structural modifications to integrate solar panels, posing a challenge due to the original design not accommodating such additions. Retrofitting older reefer involve updating existing electrical systems to align with solar panels, presenting technological challenges. Limited deck space on older reefer, make it challenging to identify suitable areas for solar panel installations.

In order to have any profit for vessel under research, we need 150 kW installation at least. Solar modules are presently roughly 0.57 Euro per Watt of installed capacity. Taking into account the efficiency to be rated under standard conditions, and the capacity factor, representing the actual energy delivered over time, ranges from 10% to 30%, solar panels produce power both at sea and in port, but only during daylight, and are set to produce power 50% of the time. The cost breakdown includes the price of solar modules, additional equipment (cables, inverters, mounting structure), and estimates the installation cost for a vessel at 2.66 Euro to 3.22 Euro per watt. So, 150 kW installations will cost from 398269.20 Euro to 426717.00 Euro. The research indicates that the implementation of solar panels on the vessel under research lacks justification, considering both economic and operational efficiency aspects.

Using the above governing equations, a system performance calculation based on the concept of information-flow diagram [8] can be carried out. The information-flow diagram shows that there are three independent design variables for a CCERM, namely, the condensing temperature \( T_c \) and the evaporating temperature \( T_e \) of the single-stage refrigeration machine, and the evaporating temperature \( T_e \) of the ERM. Given \( T_c, T_e, T_e \) and the performance maps of the ejector and the compressor, the system performance of a CCERM can be carried out.

In the present study, we use R22 as the working fluid in the vapor compression refrigeration machine and R245fa as the working fluid in the ERM. Eq. (11) is used for ejector performance calculation in system analysis since R245fa is used in the single-stage refrigeration machine uses a screw-type compressor Stal R5 M which is made by ABB.

The coefficient of performance of the combine compression-ejector refrigeration machine, COPc, is
determined by the following definition:

\[ \text{COP}_2 = \frac{Q_c}{W_e + W_{pump}} \]  

(44)

where \( W_{pump} \) is the pumping power consumed by the circulation pump in the ERM. For comparison, the coefficient of performance of the VCRM, \( \text{COP}_1 \), is determined. \( \text{COP}_1 \) is defined as

\[ \text{COP}_1 = \frac{Q_c}{W_e} \]  

(45)

The analytical results presented in Table show that the COP of a CCERM is superior to that of a single-stage refrigeration machine. It can be seen that the improvement of COP is more significant at higher condensing temperature of the VCRM \( T_c \). For \( T_e = -40 °C \), the improvement in COP by using a CCERM can be as high as 19.3% at \( T_c = 45°C \) and \( T_e = 20 °C \) (evaporating temperature of the ERM).

When the ejector refrigeration machine is combined with the VCRM, the performance of the refrigeration system could be evaluated by calculating the COP improvement ratio as follow:

\[ \text{COP}_{\text{improv}} = \left( \frac{\text{COP}_2 - \text{COP}_1}{\text{COP}_1} \right) \times 100\% \]  

(46)

The higher the condensing temperature \( T_c \), the better the improvement in COP by using a CCERM. This indicates that a CCERM may be more significant for an ice-storage air-conditioning system using a condenser with air cooling device.

Research on this ejector was conducted in calculated operating modes with a conical-cylindrical mixing chamber in the temperature ranges of generation \( T_g = 80-100°C \), condensation \( T_c = 30-40°C \), and \( T_e = 20°C \). The calculation results for the values of \( U \) for the ejector are presented in Fig. 4

From Fig. 4, it can be observed that the values of \( U \) for the ejector increase with the increase in \( T_g \) and the decrease in \( T_c \), which can be explained by the growth of the value \( E \) and the decrease in \( e \). As a result of applying new methods to improve the efficiency of the ERM, values of ejector coefficients \( U \) within the range of 0.65 – 0.85 were obtained for real operating conditions. Currently, these are the highest values in the world for ERMs operating on low-boiling refrigerants. The obtained data will be used for the design and manufacture of semi-industrial prototypes of ERMs for various purposes.

<table>
<thead>
<tr>
<th>R22 condensing temperature ( T_e (°C) )</th>
<th>One-stage machine ( \text{COP}_1 )</th>
<th>( T_e = 20°C )</th>
<th>( T_e = 22°C )</th>
<th>( T_e = 24°C )</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>( \text{COP}_2 )</td>
<td>( \text{COP}_{\text{improv}} ) (%)</td>
<td>( \text{COP}_2 )</td>
<td>( \text{COP}_{\text{improv}} ) (%)</td>
</tr>
<tr>
<td>30</td>
<td>1.49</td>
<td>1.64</td>
<td>9.8</td>
<td>1.62</td>
</tr>
<tr>
<td>35</td>
<td>1.36</td>
<td>1.52</td>
<td>12.1</td>
<td>1.51</td>
</tr>
<tr>
<td>38</td>
<td>1.30</td>
<td>1.48</td>
<td>13.9</td>
<td>1.47</td>
</tr>
<tr>
<td>40</td>
<td>1.23</td>
<td>1.44</td>
<td>16.7</td>
<td>1.41</td>
</tr>
<tr>
<td>42</td>
<td>1.18</td>
<td>1.38</td>
<td>17.3</td>
<td>1.37</td>
</tr>
<tr>
<td>45</td>
<td>1.12</td>
<td>1.34</td>
<td>19.3</td>
<td>1.32</td>
</tr>
</tbody>
</table>

Table – Analytical results for COP of single-stage and combined compression-ejector refrigeration machine at \( T_e = -40 °C \).

\( \text{COP}_{\text{improv}} = \left( \frac{\text{COP}_2 - \text{COP}_1}{\text{COP}_1} \right) \times 100\% \)
6. Conclusions

The research reveals that the opportunities for solar panel use in reefer vessels and fishing carriers constructed 20-40 years ago encompass efficiency improvements, technological modifications, policy initiatives, hybrid system integration, environmental compliance, cost savings over time, and alignment with sustainability goals. However, several challenges emerge during the research, including structural adaptations, limited deck space, outdated electrical systems, weight constraints, and concerns about corrosion and wear. Economic viability becomes a critical consideration, particularly for vessels approaching the end of their operational life, where retrofitting costs may not align with potential energy savings. The analysis of cost estimates for a 150 kW installation indicates a considerable investment, further questioning the feasibility of solar panel implementation on the researched vessel. Consequently, the decision is made to refrain from adapting solar panel technology on the vessel, emphasizing the need for careful consideration of economic and operational aspects in such retrofitting endeavors.

The use of the ejector refrigeration machine as the bottom cycle of an inverse Rankine cycle using vapor compression refrigeration machine to lead to a combined compression-ejector refrigeration machine is a new concept. Since the ejector refrigeration machine is driven using the waste heat from the single-stage refrigeration machine, no additional energy is required except the negligible pumping power in the ERM. The improvement in COP for a combined compression-ejector refrigeration machine is thus expected. Both the system simulation results obtained in the present study verify this concept. The improvement of COP for the present study is about 19%, and are mostly greater than 10% depending upon the operating conditions.

References

Покращення енергоефективності старого рефрижераторного судна за допомогою комбінованої компресійно-ежекторної холодильної машини

В. В. Яглана¹, М. Г. Хмельнюк²

¹²Одеський національний технологічний університет, вул. Дворянська, 1/3, Одеса, 65082, Україна

e-mail: hmel_m@ukr.net

ORCID: ¹https://orcid.org/0000-0002-8827-4537; ²https://orcid.org/0000-0002-9310-1286

Запропоновано та досліджено суднову комбіновану компресійно-ежекторну холодильною машину, що складається з традиційної системи охолодження та кондиціювання повітря з використанням одноступеневої холодильної машин та ежекторної холодильної машини. Ежекторна холодильна машина приводиться в дію відпрацьованим теплом від одноступеневої холодильної машини та діє як нижній цикл останньої. Системний аналіз показує, що СОР компресійно-ежекторної холодильної машини значно вище, ніж одноступеневої холодильної системи. Покращення СОР може досягати 19,3% для температури випаровування одноступеневої холодильної машини при \( T_v = -40 \, ^\circ C \). Показано вплив проточної частини профілю ежектора, робочих параметрів, схеми та циклу суднової комбінованої компресійно-ежекторної холодильної машини на її експлуатаційні характеристики. Запропоновано та досліджено раціональні методи підвищення ефективності
суднової комбінованої компресійно-ејкекторної холодильної машини, що працює на екологічно чистому низькокиплячому холодоагенті R245fa. Дане дослідження показує, що комбінована компресійно-ејкекторна холодильна машина з використанням ейкекторної холодильної машини як нижнього циклу одноступеневої холодильної машини є життєздатною. Möglichості впровадження сонячних панелей, висунуті викили для старих рефрижераторних суден (20-40 років) змушують проаналізувати ситуацію. Незважаючи на те, що існують можливості для підвищення ефективності та екологічного вирівнювання, проблеми з модернізацією старих суден, особливо тих, що наближаються до кінця свого експлуатаційного терміну, роблять економічну життєздатність критичним питанням. Аналіз можливої застосування сонячних панелей оцінює діапазон витрат від 398 269,20 до 426 717,00 євро, наголошуючи на необхідності ретельної оцінки. Висновок виступає проти впровадження сонячних панелей на досліджуваному судні, підкреслюючи відсутність обґрунтування як з точки зору економічної, так і з точки зору операційної ефективності.

Ключові слова: Старе рефрижераторне судно; Енергоефективність; Ежекторна холодильна машина; Комбіновані компресійно-ејкекторні холодильної машини; Екологічно чистий холодоагент

Література


Отримана в редакції 12.08.2023, прийнята до друку 18.09.2023