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POLISH MARITIME RESEARCH 3 (115) 2022 Vol. 29; pp. 78-88
10.2478/pomr-2022-0028

FUTURE SUSTAINABLE MARITIME SECTOR: ENERGY EFFICIENCY IMPROVEMENT AND ENVIRONMENTAL IMPACT REDUCTION FOR FISHING CARRIERS OLDER THAN 20 YEARS IN THE FLEET PART II

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ABSTRACT

For the maritime sector to be sustainable and to have an intact blue economy, shipowners should be ready to implement Ship Energy Efficiency Management Plans alongside energy efficiency projects. The problem for organizations and shipowners having fishing carriers older than 20 years is highlighted and the following challenges arise for decision-making authorities. To keep such ships in the fleet for the next decade, shipowners should deploy energy efficiency projects for marine system retrofitting to improve energy efficiency and meet environmental regulations. An energy audit is performed and an energy efficiency program is proposed with guidelines for regulations that are currently coming into force. To improve energy efficiency, reduce the environmental impact, and cut fuel consumption costs, marine system retrofitting is done, in a particular case, with two options proposed. The first is a cascade refrigeration system with hydrocarbons and carbon dioxide, where the shipowner gains an energy efficiency improvement of about 20%. The second option is a two-stage refrigeration system with ammonium as the environmentally friendly refrigerant, which improves the energy efficiency by about 26%. Technical and economic issues have been discussed.

Keywords: Blue Economy, Energy Efficiency, Fishing Carrier, Vapor Compression Refrigeration Machine

INTRODUCTION

According to the final communication from the Commission to the European Parliament in May 2021 [1], a new approach to a sustainable blue economy has been discussed. The EU Directive on the Implementation of Maritime Spatial Planning in 2022 proposes cross-border cooperation as a key element of the EU policy. The cooperation is facilitated by the EU Commission to push the Member States closer to the integration of off-shore renewable energy development objectives in their national spatial plans, in order to promote the use of renewable energy systems.

Energy efficiency together with energy security regulations is a huge part of the issues concerning the environment within the maritime sector for ship manufacturers and owners.

There are still some reefers in the fleet (fish carriers) [2] that were manufactured in the year 2000 and the last decades of the 20th century.

How can they be adapted to the sustainable maritime blue economy?

Here the key challenges are as follows:
- to improve energy efficiency;
- to reduce fuel consumption;
- to reduce the environmental impact of onboard systems.

The Energy Efficiency Existing Ship Index (EEXI) together with the Carbon Intensity Indicator (CII) incorporates a valuation of potential operational changes considering optimal energy efficiency technologies for energy efficiency improvements that are best suited for the ship under investigation.

For potential operational changes, the following directions can be worked:

– Speed reduction across the ship fleet can reduce greenhouse gas (GHG) emissions.
– Ship sailing route assessments to define methods for *ship route* planning and *route* optimization as a part of *energy* efficiency [3].
– Onboard systems estimation to make technical modifications and reduce energy consumption.
– Cargo handling system estimation for seeking modification and optimization opportunities to reduce energy consumption.
– Sea water pump assessment to find the potential to reduce energy consumption.
– Variable frequency drives assessment to save energy and reduce GHG emissions in existing *ships based on the research of ABB marine [4].*
– Smart HVAC&R.

Performing energy quality analysis [5] contributes to the electric energy quality monitoring system to achieve green shipping for sustainable blue economy regulations.

Energy efficiency technologies [6-9] can be adapted to ship systems where there is high potential for cutting costs: air lubrication systems, wind-assisted propulsion [10] such as Flettner rotors, kite and sail propulsion, marine battery systems, or any energy storage systems [11], waste heat recovery systems [12-14] or energy recovery systems, exhaust gas cleaning systems [15] according to requirements set by maritime regulations, propeller efficiency retrofit, rudder efficiency retrofit, operated system or equipment auxiliary saving technologies to reduce energy consumption, shaft generators, engine tuning, trim optimization, coating or antifoul evaluation, cold ironing, HVAC&R to improve energy efficiency, and insulation improvements to reduce energy losses.

Refrigeration systems and refrigerants. In vapor refrigeration systems, natural refrigerants have been used in different industrial applications for some time. For marine refrigeration, R744 was used more often than R717 in marine applications. The key reason was the safety requirements, R744 being less toxic in comparison with R717. Despite the lower efficiency of R744, marine refrigeration systems were the most common solutions for onboard installations. Modern technology developments have made the R744 system economically competitive as well. Several of its thermodynamic properties are more valuable than the phasing-down of HFCs group refrigerants. As a result of the marine regulations in force, the demand for a transition to natural refrigerants is becoming more common in the maritime sector and other industries.

Addressing the first phase in global trends of refrigerants offered by the Montreal Protocol [16] and Kyoto Protocol [17], the next phases in the direction of environmentally friendly refrigerants have been performed. The global warming potential (GWP) and ozone depletion potential (ODP), which are regulated by the European Commission [18], confirm no bounds for natural refrigerants such as R717, and hydrocarbons (R290) as well as the carbon-dioxide based R744. Of these natural refrigerants, the last one has many additional advantages besides global environment safety. For applying carbon dioxide, local safety regulations for system maintenance and transport refrigeration are provided by [19] Bitzer and ASHRAE, as well as going through recommendations for the use of R744, being non-flammable as well as non-toxic and meeting the required safety class for refrigerants. Hence, the mass and compact sizing system characteristics, suitable for marine performance, are permitted for the development of refrigeration units intended for fishing vessels (fishing carriers) [20].

R290 hydrocarbons. These have a very low GWP=3. Systems with R290 have been in operation globally for many years. It is commonly used within compact systems with low charges – home refrigerators, self-contained commercial refrigeration equipment, and retrofitted automotive systems. The main hindrance to the use of R290 is its high flammability (A3 safety group). R290 is supplied to the automotive aftermarket sector and makes up about 15% of the Cold Food Chain (Australian research).

R717 (GWP=0) has been used as a common refrigerant throughout the 20th century, particularly for huge industrial plants intended for food processing. The European Commission (April 2022) proposed updating the EU F-Gas Regulation, opening up opportunities for the use of ammonia/ NH_3 (R717). For vapor compression refrigeration systems, R717 has a high latent heat as well as the highest refrigeration capacity per unit mass flow of all the refrigerants in current use. Since R717 has a low molar mass, it can have a higher particle velocity compared with other refrigerants, which enables small pipes to be used. While it is flammable and toxic, it is classified in the B2L safety group and additional safety measures are required.

The usage of R744 (GWP=1) in marine systems ceased in the 1950s, mostly due to technical complications that occurred and the reduced possibility to use synthetic working fluids that operate at lower working pressures. Nowadays, these problems have been solved, however, and there are applications for which R744 is the selected working fluid, such as for freezing applications, commercial refrigeration, etc.

These marine refrigeration systems contain 1000 kg of refrigerant. To make the engineering transition away from ODS refrigerant is one of the objectives that must be met. As for alternatives to ODS refrigerant, it has been proposed to develop an energy-efficient marine system with natural refrigerant in charge. A cascade refrigeration system (R290/R744) and a two-stage refrigeration system (R717) are proposed.

The study aims to adapt fishing carriers within a sustainable maritime sector according to the modern legislative base and IMO requirements.

The study objectives are as follows:

– To conduct an energy audit and offer a program to improve energy efficiency and reduce the environmental impact of the refrigeration system.
– To make an engineering transition away from the ODS refrigerant currently in use.

METHODS

As a result of the energy audit performed for the marine refrigeration system of the fish carrier of this study, the energy potential was identified and a program was proposed to improve its energy efficiency and reduce its environmental impact. Using natural refrigerants in both a cascade refrigeration system R290/R744 and two-stage R717 refrigeration system allows for meeting the requirements of maritime sector environmental regulations.

CASCADE SYSTEM R290 / R744

Technical data:

- Condensing temperature $t_{condens}$ = 30°C;
- Evaporating temperature t_{evap} = -40°C;
- High-temperature R290 circuit, low-temperature R744 circuit.
- Cooling capacity (refrigeration effect) Q_0 = 450 kW

It is essential to use a cascade refrigeration system when the difference between the temperature at which heat is rejected and the temperature at which refrigeration is required has a high value, in which case there is no possibility to find a single refrigerant with suitable properties.

The cascade refrigeration system works as two independent single-stage refrigeration machines, which are connected by an intermediate cascade heat exchanger. This system element connects the two refrigerant circuits thermally by working simultaneously as an evaporator and a condenser, where CO_2 condensation processes occur (as at the condenser) as the refrigerant of the R744 low-temperature circuit and evaporating processes occur when the refrigerant "boils off" (as at the evaporator) for the R290 high-temperature circuit.

In the high-temperature basin, R290 vapors are compressed in the high-temperature circuit (HTC) compressor and enter the condenser, where they are cooled, condensed, and supercooled down to the T_{13} temperature. Then the liquid refrigerant enters the regenerative heat exchanger, where it is supercooled due to the heat exchange process with the steam of low-temperature potential for the cascade heat exchanger. Refrigerant at the T_{14} temperature is throttled in the expansion valve, then the vapor–liquid mixture "boils off" in the cascade heat exchanger and through the regenerative heat exchanger, the refrigerant passes to the HTC compressor, and then the cycle repeats, as shown in Fig. 1.

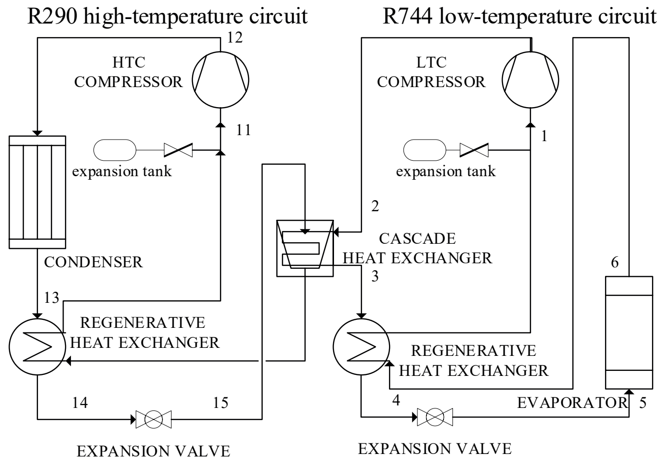


Fig. 1. Cascade refrigeration machine

In the low-temperature basin, R744 vapors are compressed in the LTC compressor and feed the cascade heat exchanger, where the R744 vapors are condensed. The liquid refrigerant is supercooled in the regenerative heat exchanger and enters the evaporator, where the refrigerant "boils off", absorbing the latent heat from the object being cooled. The low-temperature steam is heated in the regenerative heat exchanger to the suction temperature T_1. Refrigerant with temperature T_1 is fed to the LTC compressor. The cycle repeats.

When the machine is stopped for a long period, the temperature of the refrigerant becomes equal to the ambient temperature. The liquid refrigerant evaporates. Certainly, as the temperature rises, so does the pressure. Thus, a high saturation pressure is set in the machine, which corresponds to the ambient temperature. This pressure is quite high and can lead to serious problems when starting the compressor. Therefore, an expansion tank (balloon) is installed on the suction line of the compressor, which is switched on only when the machine is stopped.

The cascade refrigeration machine cycle is as follows:

– the intermediate temperature in the cascade heat exchanger is defined as:

$$T_{\text{int}ermediate} := \sqrt{T_{condens\_HTC} \times T_{evap\_\text{LTC}}}\,,\ K \tag{1}$$

where:

$T_{condens\_HTC}$ – absolute condensing temperature of the high-temperature circuit, $T_{condens\_HTC}$ =303 K,

T_{evap_LTC} – absolute evaporating temperature of the low-temperature circuit,

T_{evap_LTC} =233 K, $T_{\text{int}ermediate}$ =265 K;

– evaporating temperature of the high-temperature circuit:

$$T_{evap\_HTC} := \sqrt{T_{\text{int}ermediate} - \Delta T},\ K \tag{2}$$

where: ΔT=5 K, T_{evap_LTC} =260 K;

– condensing temperature of the low-temperature circuit:

$$T_{condens\_LTC} := T_{\text{intermediate}} \div \Delta T,\ K \tag{3}$$

$$T_{condens\_LTC} = 270\ \text{K};$$

– temperature overheating at the outlet of the compressor is 5 °C;
– the supercooling of the refrigerant in the condenser is 5 °C;
– the temperature of insufficient heat recovery is taken equal to 10 °C;
– regenerative heat exchanger efficiency ή=0.8.

THERMAL CALCULATION

The specific mass cooling capacity is calculated by:

$$q_{evap\_LTC} = h_6 - h_5,\ kJ/kg \tag{4}$$

$$q_{evap\_HTC} = h_{16} - h_{15},\ kJ/kg \tag{5}$$

where h – enthalpy of nodes (Table 1).

The specific heat of condensation is determined by:

$$q_{condens\_LTC} = h_2 - h_3,\ kJ/kg \tag{6}$$

$$q_{condens\_HTC} = h_{12} - h_{13},\ kJ/kg \tag{7}$$

The specific volumetric cooling capacity is calculated by:

$$q_v = q_{evap\_LTC} \div V_1,\ kJ/m^3 \tag{8}$$

where V_1 is the refrigerant volume at the corresponding nodes in the circuit, m^3/kg.

The refrigerant mass flow rate through the low-temperature circuit:

$$\dot{m}_{LTC} := Q_{0\_LTC} \div q_{0\_LTC},\ kg/s \tag{9}$$

where Q_{0_LTC} – system refrigeration capacity, kW.

The refrigerant mass flow rate through the high-temperature circuit is calculated using the heat balance equation for the cascade heat exchanger:

$$\dot{m}_{HTC} := \left(\dot{m}_{LTC} \times q_{0\_LTC}\right) \div q_{0\_HTC},\ kg/s \tag{10}$$

Adiabatic work in the compressor:

$$W_{comp\_LTC} := h_2 - h_1,\ kJ/kg \tag{11}$$

$$W_{comp\_HTC} := h_{12} - h_{11},\ kJ/kg \tag{12}$$

The actual displacement capacity (ADC) in the compressor:

$$V_{act.displ.capacity\_LTC} := \dot{m}_{LTC} \times V_1,\ m^3/s \tag{13}$$

$$V_{act.displ.capacity\_HTC} := \dot{m}_{HTC} \times V_1,\ m^3/s \tag{14}$$

External pressure ratio:

$$\pi_{external} = p_{condens} \div p_{evap} \tag{15}$$

In the standard range of refrigeration screw compressors, three values of the geometric compression ratio are adopted: for high-temperature and booster compressors $\varepsilon_{geometrical} = 2.6$ at $\pi_{external} \leq 4.0$; for the medium temperature $\varepsilon_{geometrical} = 4.0$ at $\pi_{external} \leq 8.0$; for low-temperature $\varepsilon_{geometrical} = 5.0$ at $\pi_{external} \rangle 8.0$. $\varepsilon_{geometrical} = 2.6$ is chosen as the geometric compression ratio.

Theoretical compressor displacement:

$$V_{theoretical} = V_{actual} \div \lambda\ ,\ m^3/s \tag{16}$$

where the feed rate of the screw compressor = 0.9 is found from [21] for the accepted brand of injected oil and the geometric compression ratio $\varepsilon_{geometrical} = 2{,}6$

To determine the power required to drive the compressor:
Adiabatic:

$$N_{a\_LCT} := \dot{m}_{LTC} \times W_{comp\_LTC},\ kW \tag{17}$$

$$N_{a\_HCT} := \dot{m}_{HTC} \times W_{comp\_HTC},\ kW \tag{18}$$

Indicator:

$$N_{i\_LTC} := N_{a\_LTC} \div \eta_{i\_LTC},\ kW \tag{19}$$

$$N_{i\_HTC} := N_{a\_HTC} \div \eta_{i\_HTC},\ kW \tag{20}$$

where $\eta_{i\_LTC}, \eta_{i\_HTC}$ – indicator coefficient:

$$\eta_{i\_LTC} := \lambda_{w\_LTC} + b \times t_{evap\_LTC} \tag{21}$$

$$\eta_{i\_HTC} := \lambda_{w\_HTC} + b \times t_{evap\_HTC} \tag{22}$$

where b – factor (b=0.001).

The indicator efficiency $\eta_{i\_LTC}, \eta_{i\_HTC}$ and effective efficiency $\eta_{effective}$ for a screw compressor operating with the injection of a small amount of liquid into the working cavities, mainly to reduce the temperature of the compressible medium, and a screw compressor with oil supplied to the working cavities, are found from the dependencies shown in [21] depending on the geometric compression ratio $\varepsilon_{geometrical} = 2.6$.

The dependence of the effective efficiency for screw compressors on the external degree of pressure increase $\pi_{external}$ at various $\varepsilon_{geometrical}$ can be derived by the graphical method as well. The dependences of the mechanical efficiency of screw compressors of the types described above, as well as for a dry screw compressor, can be taken by the same method.

Power consumed by friction:

$$N_{friction\_LTC} := V_{h\_LTC} \times P_{friction} \text{ , } kW \quad (23)$$

$$N_{friction\_HTC} := V_{h\_HTC} \times P_{friction} \text{ , } kW \quad (24)$$

Effective power:

$$N_{e\_LTC} := N_{friction\_LTC} + N_{i\_LTC} \text{ , } kW \quad (25)$$

$$N_{e\_HTC} := N_{friction\_HTC} + N_{i\_HTC} \text{ , } kW \quad (26)$$

Coefficient of performance (COP):

a) Carnot cycle:

$$COP_c := T_{evap\_LTC} \div \left(T_{condens\_HTC} - T_{evap\_LTC}\right) \quad (27)$$

b) theoretical:

$$COP_{theoretical} := q_{evap\_LTC} \div \left(W_{compressor\_HTC} - W_{compressor\_LTC}\right) \quad (28)$$

c) actual:

$$COP_{actual} := Q_{evap\_LTC} \div \left(N_{e\_LTC} + N_{e\_HTC}\right) \quad (29)$$

Thermodynamic perfection factor:

a) theoretical:

$$\eta_{theoretical} := COP_{theoretical} \div COP_c \quad (30)$$

b) actual:

$$\eta_{actual} := COP_{actual} \div COP_c \quad (31)$$

Theoretical refrigerant coefficient:

$$\varepsilon_{theoretical} := Q_{evap\_LTC} \div \left(N_{a\_LTC} + N_{a\_HTC}\right) \quad (32)$$

REFRIGERATION SYSTEM HCFC-22

Technical data:
Cooling capacity $Q_0 = 450$ kW
Condensing temperature $t_к = 30$ °C;
Evaporating temperature $t_o = -40$ °C;
Refrigerant – R22

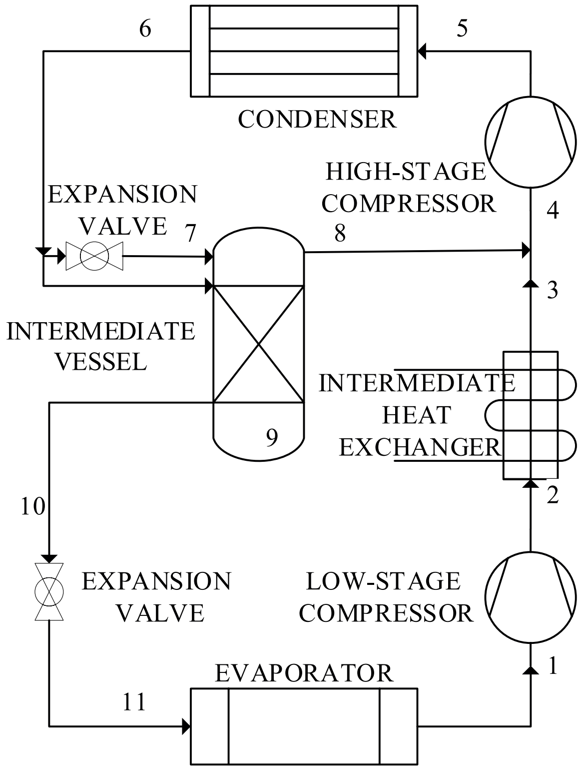


Fig. 2. Refrigeration system for two-stage refrigeration machine with one-time throttling and incomplete intermediate cooling

Tab. 1. Parameters in the corresponding nodes

№	1	2	3	4	5	6	7	8	9	10	11
p, bar	1.049	3.537	3.537	3.537	11.919	11.919	3.537	3.537	3.537	11.919	1.049
T, °C	-20	34.461	30	24	85	25	-10	-10	-10	-5	-40
h, kJ/kg	401	432	430	425	460	230	230	402	189	194	194
v, m³/kg	0.226	–	–	0.076	–	–	–	–	–	–	–

The average pressure of the high-stage compressor entering the suction:

$$P_{suction} \sqrt{11{,}9 \times 1{,}049} = 3{,}537 \text{ bar}$$

$$T_{suction} = -10^{\circ} C$$

The refrigeration cycle parameters are calculated:

Specific cooling capacity:

$$q_{evap} = h_1 - h_{11} = 401 - 194 = 207 \text{ kJ/kg}$$

The mass flow of the high-pressure stage compressor $\dot{m}_{2\_HPS}$ is greater than that of the low-pressure stage compressor $\dot{m}_{1\_LPS}$, since, in addition to the steam coming from the low-pressure stage compressor in the amount $\dot{m}_{1\_LPS}$, it also receives steam formed during the "boiling off" of the liquid in the economizer. The volumetric cooling capacity of the high-pressure compressor is about three times less due to the reduction in the volume of steam when compressed in the low-pressure compressor.

The mass supply of the compressor low-pressure stage, kg/s, is determined by the following:

$$\dot{m}_{1\_LPS} = Q_{evap} \div q_{evap}\text{, } kg/s \tag{33}$$

where Q_{evap} – cooling capacity, kW; q_{evap} – specific cooling capacity, kJ/kg:

The mass supply of the compressor high-pressure stage, kg/s, is determined by Eq. (34):

$$\dot{m}_{2\_LPS} = \dot{m}_{1\_LPS} \times \frac{h_8 - h_{10}}{h_8 - h_7}\text{, } kg/s \tag{34}$$

To determine the parameters of node 4, the enthalpy for suction in the compressor high-pressure stage is as follows:

$$h_4 = h_8 + \frac{\dot{m}_{1\_LPS}\left(h_3 - h_8\right)}{\dot{m}_{2\_LPS}}\text{, } kJ/kg \tag{35}$$

The specific work of the cycle, equal to the work of the low-pressure stage compressor:

$$w_1 = h_2 - h_1\text{, } kJ/kg \tag{36}$$

The specific work of the cycle, equal to the work of the high-pressure stage compressor:

$$w_2 = h_5 - h_4\text{, } kJ/kg \tag{37}$$

The specific load on the condenser:

$$h_{condenser} = h_5 - h_6\text{, } kJ/kg \tag{38}$$

The coefficient of performance (theoretical):

$$COP_{theoretical} = q_{evap} / \left(w_1 - w_2\right) \tag{39}$$

The energy conversion factor of the cascade refrigeration unit of the Carnot cycle:

$$COP_c = T_{evap} / \left(T_{cond} - T_{evap}\right) \tag{40}$$

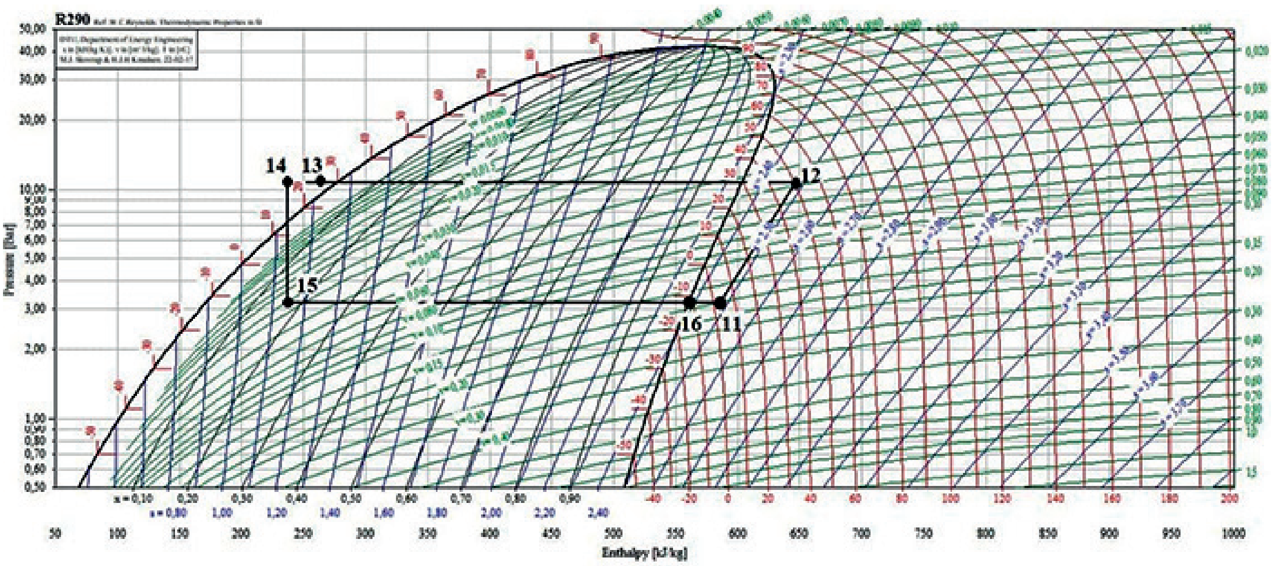
The thermodynamic perfection factor (theoretical) for the two-stage refrigeration system:

$$\eta_{theoretical} := COP_{theoretical} \div COP_c \tag{41}$$

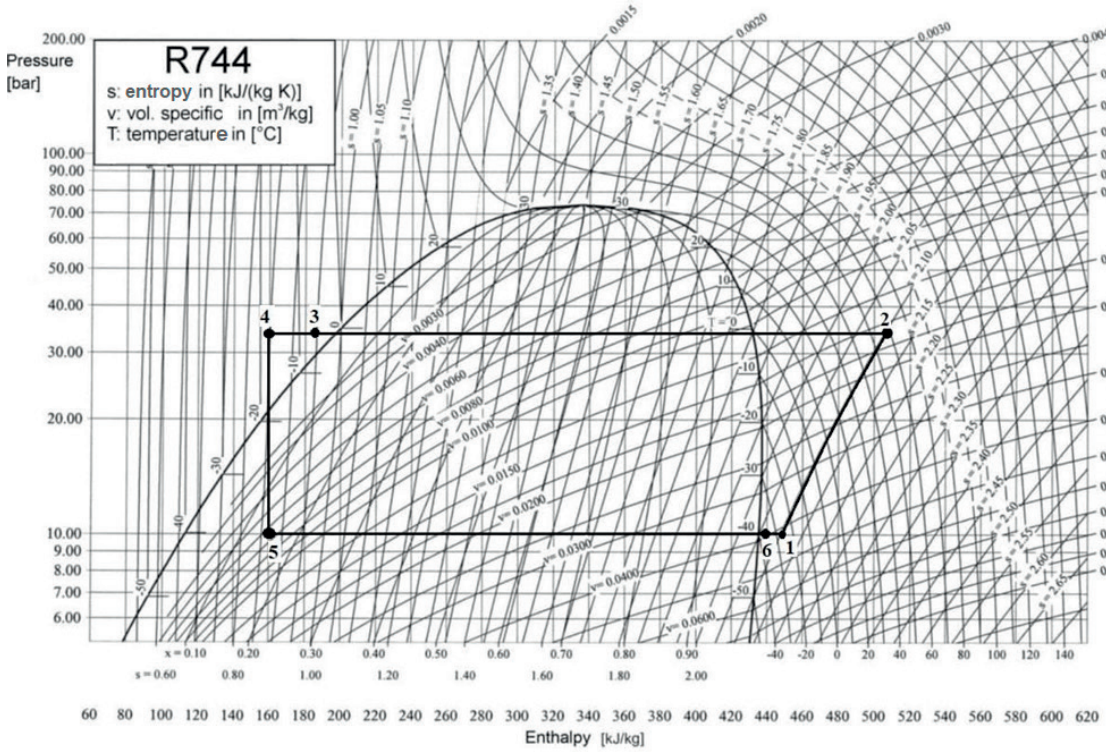
The full degree thermodynamic perfection factor (actual) for the cascade refrigeration system:

$$\eta_{actual} = COP^{n}_{actual} \div COP_c \tag{42}$$

The calculation for other refrigerants is done by analogy.



a) high-temperature cycle R290



b) low-temperature cycle R744

Fig. 3. Cascade refrigeration machine (a, b) R290 / R744 log P-h diagram.

RESULTS

The log P-h diagram is intended to determine the parameters of the nodes presented in Fig. 3, using CoolPack software for the simulation refrigeration system model.

The cascade refrigeration system thermodynamic processes:

- 11–12 adiabatic compression of refrigerant vapors by the LTC compressor;
- 12–13 isobaric condensation of refrigerant vapors in the condenser;
- 13–14 isobaric supercooling of liquid R290 in the "vapor-liquid" type regenerative heat exchanger;
- 14–15 isoenthalpy liquid throttling of refrigerant in the throttle valve;
- 15–16 isobaric evaporation of the vapor-liquid mixture into the cascade heat exchanger;
- 1–1 adiabatic vapor compression of the refrigerant vapor by the HTC compressor;
- 2–3 isobaric condensation of refrigerant vapors in the cascade heat exchanger;
- 3–4 isobaric supercooling of liquid R744 in the "vapor-liquid" type regenerative heat exchanger;
- 4–5 isoenthalpy throttling of liquid refrigerant in the throttle valve;
- 5–6 isobaric evaporation of vapor-liquid mixture in the cascade heat exchanger.

Tab. 2. Node parameters for cascade refrigeration system

High-temperature circuit						
Node number	11	12	13	14	15	16
p, bar	3.204	10.75	10.75	10.75	3.204	3.204
T, °C	3	50.4	25	15	-12	-12
h, kJ/kg	585.641	646.543	264	237.572	237.572	560
v, m3/kg	0.15					
Low-temperature circuit						
Node number	1	2	3	4	5	6
p, bar	10	33	33	33	10	10
T, °C	-25	58	-7	-17	-40	-35
h, kJ/kg	450	507	184	161	161	436
v, m^3/kg	0.041					

Tab. 3. Thermal calculation results

Parameter	Value	Parameter	Value	Parameter	Value
q_{evap_LTC} kJ/kg	275	$V_{act.displ.capacity\_HTC}$ m³/sec	0.246	$N_{friction\_HTC}$ kWt	15.1
q_{evap_HTC} . kJ/kg	322	λн	0.82	N_{e_LTC} kW	117.7
$q_{condens\_LTC}$ kJ/kg	323	λ_w	0.81	N_{e_HTC} kW	132.9
$q_{condens\_HTC}$ kJ/kg	382	V_{h_LTC} m²/sec	0.081	COP_c	3.33
q_v . kJ/m³	6707	V_{h_HTC} m²/sec	0.303	$COP_{theoretical}$	2.33
$\dot{m}_{LTC}$ kg/sec	1.63	N_{a_LCT} kW	93.3	COP_{actual}	1.79
$\dot{m}_{HTC}$ kg/sec	1.64	N_{a_HCT} . kW	99.8	$\eta_{theoretical}$	0.701
W_{comp_LTC} kJ/kg	57	N_{i_LTC} kW	113.6	η_{actual}	0.539
W_{comp_HTC} kJ/kg	61	N_{i_HTC} kW	117.7	$\varepsilon_{theoretical}$	2.23
$V_{act.displ.capacity\_LTC}$. m³/sec	0.067	$N_{friction\_LTC}$. kW	4.1		

REFRIGERATION SYSTEM WITH HCFC-R22

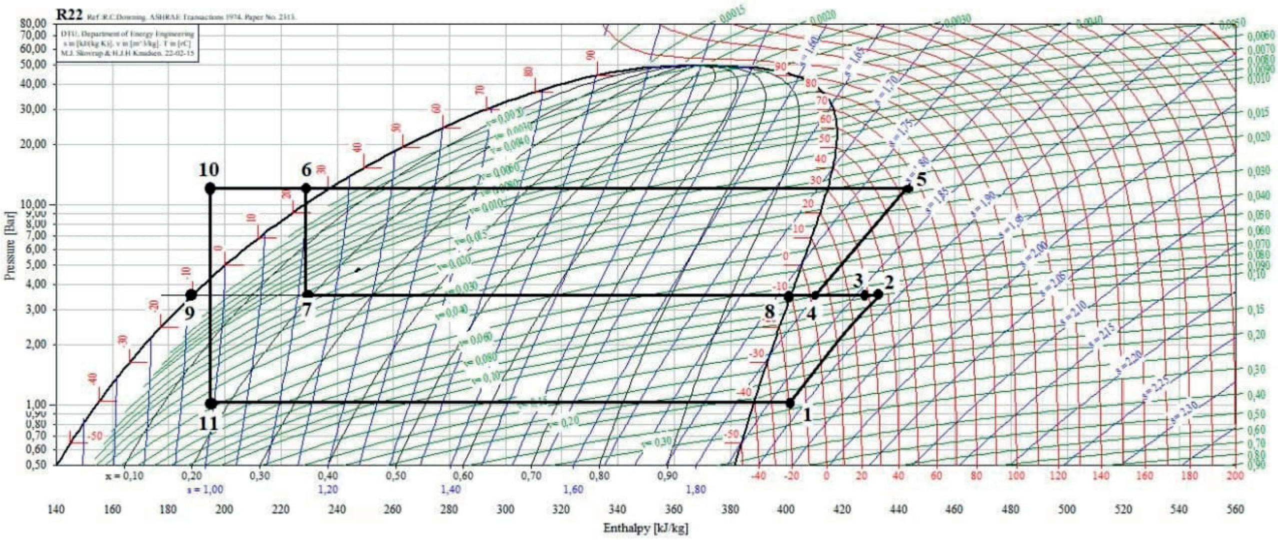


Fig. 4. Two-stage refrigeration machine R22 log P-h diagram using CoolPack software

Refrigeration system thermodynamic processes:
1 – 2 adiabatic compression in the high-pressure stage compressor from P_0 to $P_{superheat;}$
2 – 3 removing superheated steam in the intermediate heat exchanger, $P_{superheat}$ = const;
3 – 4 suction steam superheat in the high-pressure stage compressor, $P_{superheat}$ = const;
4 – 5 adiabatic compression in the high-pressure stage compressor from $P_{superheat}$ to $P_{condens}$;
5 – 6 condensing processes occurring in the condenser for $P_{condens}$ = const, $t_{condens}$ = const;
6 – 7 throttling expansion valve 1 from $P_{condens}$ to $P_{superheat}$ for h = const;
7 – 8 "boiling off"in the intermediate vessel for $P_{superheat}$ = const and superheat = const;
6 – 10 liquid subcooling in the coil in the intermediate vessel for $P_{condens}$ = const;
10 – 11 throttling in the expansion valve 2 from $P_{condens}$ to P_0 for h = const;
11 – 1 "boiling off " of refrigerant in the evaporator for P_0 = const, t_0 = const.

We summarize the results of the calculations for all the proposed alternative refrigerants in a table.

Tab. 4. Result for refrigeration systems

Parameter	R22 one-stage	R22 two-stage	Cascade R290/R744	R717 two-stage
q_{evap}, kJ/kg	170	207	275	1110
W_{comp}, kJ/kg	68	66	117	380
q_v, kJ/m³	754	914	6707	634
N_a, kW	180	160	193	156
N_i, kW	300	230	250	236
$COP_{theoretical}$	2.5	3.11	2.33	2.92
COP_{actual}	1.5	1.95	1.8	1.9
$\eta_{theoretical}$	0.75	0.935	0.7	0.877
η_{actual}	0.45	0.587	0.54	0.571

Capital costs for the cascade refrigeration system (R290/R744) are presented in Table 5.

Tab. 5. Equipment cost for cascade refrigeration system

#	Equipment and its characteristics	Unit	Quantity	Cost per single unit, euro	Overall cost, euro
1	Compressor	piece	4	60000-70000	130000
2	Condenser	piece	2	9250	18500
3	Heat exchanger	piece	2	2050	4100
4	Cascade heat exchanger	piece	1	3000	3000
Total cost of equipment	155600 euro				
Cost of other equipment 10%	44896 euro				
Estimated cost of equipment	138767 euro				
Packaging costs and transporting costs 15%	20815 euro				
Assembly costs 20%	27753 euro				
Total	201213 euro				

Tab. 6. Cost of equipment and refrigerants for the two-stage ammonia refrigeration system and the regular one-stage vapor compression refrigeration system

Equipment for two-stage marine system R717 and one-stage marine system HCFC-22	Cost for two-stage marine system R717	Cost for one-stage marine system HCFC-22
Shut-off, control, and automatic valves	1105 euro	18674 euro
Pipes	2306 euro	6841 euro
Brazed plate heat exchangers	224 euro	-
Lamellar heat-exchange collapsible devices	3401 euro	-
Shell and tube heat exchangers	-	3729 euro
Compressor	90780 euro	119629 euro
Receiver	210 euro	521 euro
Pumps	1041euro	1320 euro
Air coolers	8063 euro	13150 euro
Refrigerants		
R717- 0.9t	737 euro	-
Heat transfer medium - 9.28t	12067 euro	-
Oil	56 euro	130 euro
HCFC-22 - 2t	-	1450 euro
Technical maintenance per year		
Filters, spare parts of compressor and pump replacement, etc.	931 euro	1737 euro
Total	116694 euro	169397 euro

DISCUSSION

The one-stage vapor compression refrigeration system is the object of an energy audit investigation. The owner of the fishing carrier is trying to adapt its ship which has installed a refrigeration system with HCFC-R22 refrigerant still operating in compliance with the existing regulations in force. After 8–10 years, the owner is planning to withdraw the ship from service. In order to meet the expected new regulations, which impose strict requirements concerning the environmental impact of on-board systems, an energy audit has been performed for the marine system, and the energy potential was derived. A program is proposed to improve the marine system's energy efficiency and reduce its environmental impact. For the energy efficiency improvement program, two refrigeration systems have been chosen: a cascade refrigeration system R290/R744 which can improve the marine system's efficiency by about 20% and reduce its environmental impact because it uses natural refrigerants that do not harm the environment; and a two-stage refrigeration system R717 that can contribute still greater efficiency up to 26% and reduce the environmental impact due to its use of ammonium with GWP=0.

From the economic perspective, if we look at the capital costs, the two-stage vapor compression refrigeration system with R717 as the refrigerant requires 26% less investment than the cascade refrigeration system R290/R744.

Each of the proposed refrigeration systems can not only lead to energy savings and satisfy the environmental regulations requirements, but also cut the costs for fuel consumption and reduce the environmental impact.

Not only is an energy audit by professionals in the area of interest who have experience in the projects and competencies required for the development and deployment of a good energy efficiency program, but also energy management from the shipowner's side should be implemented to enable additional savings by the organization.

To look for extra steps that can be implemented for system performance improvements, to adopt an energy policy and ensure active commitment from the managerial side, to fully integrate energy management requirements and rules, to be open to new energy efficiency projects to make further gains, to train staff to define the needs and requirements, and finally, to enhance the efficient energy management with good communication concerning energy issues both within the organizational system and outside it – all these activities will enable the organization to become a leader in the market.

Guidelines are offered on how to develop a Ship Energy Efficiency Management Plan for future understanding and its subsequent deployment concerning the current regulations that each ship with a gross tonnage of more than 400 ton must comply with from 2021. By studying the development principles and purpose of such an SEEMP, the shipowner can be ready for its use, including an energy efficiency operational indicator, the existing energy efficiency ship index, and a carbon intensity indicator. These will be in use in the very near future and shipowners should have a clear understanding that their ship may be excluded from the fleet even before the desired term (after 8–10 years in the fleet) if the proposed program for improving energy efficiency is not implemented.

For a sustainable blue economy for the maritime sector, green ships programs, within the existing regulations, create new opportunities for on-board system retrofits or modernization and can prompt shipowners to move forward and to look for new savings opportunities through improved energy efficiency, at the same time as addressing new environmental restrictions.

CONCLUSIONS

Marine transport is a key player in the food supply cold chain. To improve communication within an organization concerning energy efficiency issues, not only to deploy energy efficiency projects by retrofitting or optimizing onboard systems to improve systems performance, to cut fuel costs, and reduce the environmental impact, it is crucial to have good energy management.

An energy audit has been performed. On the basis of the derived energy potential, a program for energy efficiency improvements and environmental impact reduction was offered for fishing carrier owners. This paper presents marine system retrofitting for a vapor compression refrigeration system with the HCFC-22 refrigerant in current use. An engineering transition away from ODS refrigerant is proposed for natural refrigerants (R290, R744, R717) which are approved by the Montreal and Kyoto protocols, the Kigali amendment, and other environmental regulations that are in force.

Shipowners can benefit from energy efficiency improvements of 20% to 26% from marine system retrofitting, as well as cutting the costs for fuel consumption.

By integrating energy efficiency projects, the organization may be able to retain fishing carriers in the fleet for 8–10 years longer as an effective part of the cold chain for African Union food security, and to contribute to maritime and security strategy problem-solving.

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