Design and simulations of Refrigerated Sea Water Chillers with CO_2 ejector pumps for marine applications in hot climates

Jakub Bodys[(a)], Armin Hafner[(b)], Krzysztof Banasiak[(c)], Jacek Smolka[(a)], Yves Ladam[(d)]

[(a)] Institute of Thermal Technology (ITT), Silesian University of Technology (SUT), Gliwice, 44-100, Poland, jakub.bodys@polsl.pl

[(b)] Norwegian University of Science and Technology, Trondheim, 7465, Norway, armin.hafner@ntnu.no

[(c)] SINTEF Energy Research, Trondheim, 7465, Norway, krzysztof.banasiak@sintef.no

[(d)] Kuldeteknisk AS, Tromsø, 9010, Norway, yves@kuldeteknisk.no

Abstract

Various system configurations have been developed to improve the R744 systems under hot ambient conditions. However, stationary land applications are characterised by negligible limits on space for system equipment, unlike the marine industry, i.e. on-board fishing vessels. The baseline CO_2 refrigeration system for fishing vessels was developed by a cooperating industrial company, namely the Refrigerated Sea Water Chillers operation on the Norwegian coast, which confirmed the successful application of this approach. In this study, modified layouts are evaluated for operation in warmer climates without the need for an additional compressor unit, thus maintaining the compactness of the unit. Flash gas valve-, parallel compression- and multi-ejector systems were numerically investigated including ejectors section and flooded evaporator. Sea water temperatures as occurring in Mediterranean and East-Asian waters were investigated. Both the optimal high-pressure as well as the pressure level in an intermediate pressure receiver were controlled to achieve low energy consumptions. Finally, an up to 70% performance improvement was obtained in the case of the most advanced installation working in warm East-Asian waters. The obtained results showed that the proper design of the system should ensure no necessity for an additional compressor in warmer climates while still maintaining the designed cooling capacity.

Keywords

R744, CO_2, multi-ejector system, marine application, efficiency improvement

Nomenclature

Acronyms and abbreviations

GWP	Global Warming Potential
ODP	Ozone Depletion Potential
R717	Ammonia
HC	Hydrocarbons
R744	Carbon-dioxide
TFA	Trifluoroacetic acids

HF	Hydrogen fluoride
R1234yf	Tetrafluoropropene
HFC	Hydrofluorocarbons
HFO	Hydrofluoroolefins
R134a	Tetrafluoroethane
VRC	Volumetric Refrigeration Capacity
COP	Coefficient of Performance
IHX	Internal Heat Exchanger
LPR	Low Pressure Receiver
MER	Mass Entrainment Ratio
SN	Suction Nozzle
MN	Motive Nozzle
IPR	Intermediate Pressure Receiver
SST	Sea Surface Temperature
NEO	NASA Earth Observation
EES	Engineering Equation Solver
IP	Intermediate Pressure

Roman Letters

p	pressure, *bar*
h	specific enthalpy, $kJ\ kg^{-1}$
s	specific entropy, $kJ\ kg^{-1}\ K^{-1}$
$\dot{m}$	mass flow rate, $kg\ s^{-1}$

Greek Letters

χ	Mass Entrainment Ratio, -
η	Efficiency, %

Subscripts

in	Ejector inlet
out	Ejector outlet
is	Isentropic
mn	Motive nozzle
sn	Suction nozzle
COMP	Compressor
EVAP	Evaporator
DIF	Diffuser outlet
VALVE	Expansion valve
MOT	Motive nozzle port
FGAS	Flash gas
LPR	Lower Pressure Receiver
PAR	Parallel compressor
BASE	Base compressor

1. Introduction

According to the first turn in global trends of refrigerants presented by the Montreal [1] and Kyoto [2] protocols, the next steps toward the direction of environmentally friendly working fluids have already been undertaken. According to Global Warming Potential (GWP) and Ozone Depletion Potential (ODP), regulations presented by European Commission [3] ensure no limits for natural working fluids such as ammonia (NH_3, R717), hydrocarbons (HC) or carbon-dioxide (CO_2, R744). According to the listed natural refrigerants, the last ensures many additional advantages besides global environment safety. When applying R744, local safety during exploitation and transport is provided by its non-toxic, non-flammable characteristics and, as a consequence, the least stringent safety class, A1, is achieved [4]. It is worth noting that both safety ranges should be satisfied - global and local. Meanwhile, produced synthetic refrigerants characterised by very low GWP values might simultaneously have serious disadvantages. Namely, the decomposition processes (with or without fires) of these ultra-low GWP synthetic refrigerants result in toxic products such as trifluoroacetic acids (TFA) or hydrogen fluoride (HF), which pose real dangers to human health in closed spaces such as garages and ships [5]. On the other hand, refrigerants from the R1234 family are characterised by safety class A2/L, for which the potential for safe servicing and maintenance have been confirmed [6]. An analysis of alternative mixtures based on hydrofluorocarbons (HFC) and unsaturated HFCs to substitute for high GWP refrigerants has been provided as well [7]. Nevertheless, this study presents a comparison of economic benefits that shows that R744 is a more efficient solution than systems applying the mixtures mentioned.

Economic and technical aspects of R744 application provide the same positive perspective as the aforementioned environmental factors and legal regulations. This is due to the thermodynamic properties of R744, which result in high performance operation in real cycles [8]. First, the levels of high- and low-pressure sides provide lower pressure ratios than traditional halocarbons. Consequently, a higher efficiency of compressor operation is provided [9] [10]. In addition to lower pressure ratios, the pressure values in R744 systems are higher than in classical units using tetrafluoroethane (R134a). This provides for a lower specific volume and smaller size compressors - and further lowers investment costs [9], [11]. Moreover, smaller sizes of heat exchangers can be obtained according to relatively high volumetric refrigeration capacity (VRC) and high heat transfer coefficients in CO_2 flows. Next, very low temperature drops with corresponding pressure drops in installations allow designing smaller piping systems with higher velocities of flowing working fluids. These features can be summarized by the compact sizes of R744 installations and their high performance in operation [11].

The described thermodynamic and ecologic features find application in fishing vessel refrigeration units, where cooling of a catch during transportation is one of the crucial factors of final fish quality and achievable prices. Nevertheless, the quantity of catch is important for economic balance as well. Due to this, the refrigeration unit and its equipment should concern machinery space limitations and maximum refrigerated storage space. Hence, the aforementioned compact sizing and satisfactory performance have allowed the development of refrigeration units for fishing vessels applications. Such installations have been developed by Kuldeteknisk AS for new marine applications applying R744 refrigeration units. The catch is cooled by Refrigerated Sea Water (RSW) Chillers, in which storage tank water temperature is maintained at a level of -1 °C. In Scandinavian ambient conditions, where heat rejection is ensured by relatively cold sea water (5-12 °C), such operations result in high performances of the refrigeration units without sacrificing large amounts of space for the installation of the main components. Regarding performance and the ecological aspects related to the green label of R744, many of these installations are currently found in Norwegian fishing vessels.

Nevertheless, besides the mentioned advantages, some challenging areas have to be taken into account for the process of further development. One such challenge is operation under high ambient conditions such as off the southern Mediterranean coast or in Indonesian climates, the reasons for which are related to the thermodynamic properties of R744. Namely, the relatively low temperature of the critical point (30.98 °C) [12] enforces the cycle to operate in transcritical mode. In addition, the transcritical mode results in high expansion losses, which affect system Coefficient of Performance (COP) in a negative way [9] [11]. Hence, more advanced solutions have to be utilised in the case of R744 refrigeration units.

To maintain the applicability of the RSW system and its advantages in hot climates such as in south Europe or Asia, some improvements could be introduced to the CO_2 refrigeration cycle. The literature reports several studies in which the positive influences of various components configurations were described. These solutions were developed on the basis of other CO_2 applications such as supermarket heating and cooling systems [13] [14] [15], mobile refrigeration units [16] and residential heat pumps [17] [18].

The fundamental modification of the R744 system is based on the introduction of an intermediate pressure receiver, which is sometimes called a liquid receiver. The potential energy savings of this solution were described in the work of Gullo [19]. The author theoretically analysed a refrigeration system for supermarket applications in three cities characterised by high year-averaged temperatures - Rome (Italy), Valencia (Spain) and Seville (Spain). The investigation showed up to a 9.6 % COP improvement in a combined case with evaporator overfeeding and a parallel compression mode in comparison with a cycle based on refrigerant R404A. In the work of Carvalho [20], the investment cost of liquid receiver and additional equipment was evaluated to be high with regard to the obtained performance improvement. On the other hand, the compact sizing for CO_2 showed potential for application with small units of 1 kW power. Similar challenges in the investment cost area are related to the mentioned HFO working fluids, thus most initial applications are focused on Mobile Air Conditioning and small domestic refrigerators [21]. The higher performance of an R744 system was presented by Sarkar [22], but a larger installation was analysed. The authors investigated various configurations based on the parallel compression idea. In the case of the most promising parallel compression with economiser, the COP increment was equal to 47.3 %. Cases of smaller temperature differences resulted in COP improvements on the level of 15 %. Further possibilities for system improvement are related to proper integration of heating and cooling functions. A fully integrated building design process becomes a standard indicator of a well-planned state-of-the-art investment [23]. An energy savings based on an integration of transcritical CO_2 and desalination systems was reported in the work of Farsi et al. [24]. In the work of Manjunath et al. [25], waste heat from shipboard gas turbines was utilised for heating purposes as well as to provide a power supply for a transcritical CO_2 refrigeration unit. Another cogeneration approach based on carbon dioxide was reported in the paper presented by Akbari and Mahmoudi [26]. Those authors presented promising results of a supercritical Brayton and transcritical refrigeration cycle integration. The analysis showed benefits in the form of energy savings and optimised unit-cost production.

In addition to the heat recovery approach, work recovery of expansion losses is a perspective way to improve unit COP. The aforementioned expansion losses could be described as having a large potential for work recovery in the R744 system [11]. Direct and indirect work recovery for the expansion process was described as having yielded satisfactory results. However, direct solutions in the form of gear expanders or turbines could be described as having less demand in mobile units according to reliability. The mentioned reliability can be provided by devices with no moving parts and simple construction. Such features are delivered by introducing ejectors into transcritical CO_2 refrigeration systems [27]. The recovered work could be received in two ways regarding actual needs. The ejector operation can be focused on the pressure increment before the suction ports of compressors, resulting in lower energy demand. On the other hand, the ejector provides a pumping effect and recirculation of liquid CO_2, resulting in a lower mass flow rate through the compressor section. In consequence, it provides lower compressor work. Potential for highly-efficient operation was indicated in the work of Bai et al. [28], where an advanced exergy analysis on a transcritical R744 ejector system was presented. A developed decomposition of exergy destruction sources has shown that up to 43.44 % of exergy destruction could be avoided. The most significant component was the compressor and next ejector. Hence, a substantial improvement buffer can still be developed. A similar system configuration was studied by Zhu et al. [29]; nevertheless, those authors used experimental methods and were concerned with the influence of ejector performance on overall system COP. Moreover, developed coefficients allowed for an analysis of other system components' states, i.e. that of the liquid separator. Interesting results were provided by Zheng et al. [30], who utilized a dynamic simulation of a transcritical R744 ejector system. Those authors introduced a two-stage evaporator integrated with the ejector, obtaining increased functionality and better performance in the transient states of the system.

An experimental comparison provided by Lucas showed a 17 % COP improvement due to the ejector implementation [31]. The authors investigated the influence of the high pressure side on ejector and overall system performances. The range of investigated gas cooler temperatures was constrained from 30 °C to 40 °C, whereas the evaporation temperatures were between -10 °C and -1 °C. The COP improvement showed good potential for R744 transcritical system operation under relatively high ambient conditions. According to the described ejector solution, fully developed solutions were presented for applications such as in supermarket refrigeration systems [13]. The authors described the idea of parallel working ejectors to cover various system loads with simultaneously high efficiency for these devices. Several authors investigated this solution based on a multi-ejector block. A performance mapping of a multi-ejector block was delivered on the basis of laboratory tests and described in the work of Banasiak [32]. The presented results of the block performance throughout the wide range of operating conditions characteristic of supermarket operations delivered a range of efficiencies that were a function of pressure ratio (the outlet to the suction pressure) and motive pressure. Depending on the mentioned parameters, the efficiency ranged from 12 % to 33 % for a pressure ratio of 1.1 and 75 bar and a pressure ratio of 1.3 and 95 bar, respectively. The mentioned multi-ejector block efficiency can be described by the same function as that for a single ejector, according to the definition used [33]. Further analysis of a global multi-ejector system was provided by Haida [34]. The authors described the comparison of PC and multi-ejector system performances in a laboratory test rig based on high ambient temperatures. The obtained results showed up to 8 % system COP improvement when operating in the multi-ejector mode. Numerical analyses of multi-ejector block performance were performed in cooperation with the authors of the mentioned experimental tests [35]. According to those results, an even higher efficiency of 38 % could be obtained when pressure drops in collectors are reduced. Moreover, the first studies on multi-ejector implementation to a heat pump system were provided as well [36]. Having regarded that the concept of this device was planned for refrigeration applications [13], it could be said that constant development of this technology is visible.

In this study, an investigation of a modified RSW installation for fishing vessels operating under high ambient conditions is provided. To the best of the authors' knowledge, a study of the R744 installation for fishing vessels with constrained machinery room space is not provided in the literature. The baseline case with a liquid ejector designed for Scandinavian conditions was simulated on the basis of a developed mathematical model and measurement data from an actual working RSW installation (Kuldeteknisk AS, Tromsø). Highly efficient operation of the actually operating unit on the northern Norwegian coast was confirmed. To investigate system performance under high ambient conditions, the developed baseline model was modified by introducing an intermediate pressure receiver and parallel compression of the flash gas. Moreover, an additional model of a multi-ejector system was developed and simulated as well. On the basis of satellite data, Mediterranean and East-Asian water temperatures were chosen as representative of high-temperature climates. Parameterisation of the operating conditions delivered data on the most efficient system operation. Simulated configurations were compared in the light of the system COP and space requirements. Additional equipment was analysed and is discussed to propose the best solution with regard to performance and necessary modifications for each of the analysed climates. Finally, the relation between multi-ejector module efficiency and system performance is discussed. The overall conclusions on the most promising modification of RSW installation are stated.

2. Refrigerated Sea Water installation

2.1. Scandinavian operation - Baseline System

The Baseline System of the analysed RSW installation is presented in Fig. 1. Similar installations are used on fishing vessels in the region of northern Norway. This CO_2 cycle is built on the basis of the cycle proposed by Gustav Lorentzen [11]; nevertheless, a liquid ejector was implemented as an additional component. Additional control and measurement equipment is marked by frames with proper letters, where T is temperature measurement, P is pressure measurement, and V is flow measurement. Moreover, in Fig. 1, state points used in further calculations are marked. Operation of the installation is focused on cooling the water from a storage tank loop, where the set-point temperature of the water is approximately -1 °C. Heat rejection is ensured by a sea water supplied condenser. Scandinavian conditions ensure water inlet temperatures usually below 10 °C. The analysed installation is equipped with two compressors with a maximum electrical power consumption equal to 44 kW each at 34.85 bar of evaporation pressure and 10 K superheating [37]. The suction gas is supplied from internal heat exchangers (IHX) separately for each compressor. Evaporator load varies depending on water storage tank load and share of fresh water. From the refrigerant side, the evaporator is supplied by a stream expanded in a throttling valve and the ejector. The aforementioned ejector ensures liquid circulation between a low pressure receiver (LPR) and the evaporator. Finally, according to the collaboration with the Kuldeteknisk AS, some data of the system components used in the study had confidential character. Due to the mentioned collaboration, the comprehensive analysis of the considered refrigeration system was available. In general, classic oil recovery from the low-pressure side was adjusted in order to meet the pressure in low-pressure receiver. Next, the oil was pumped back into the oil separators nearby the compressors section. Generally, the auxiliary oil-receiving loop is built by high-pressure side separator and the receivers installed together with the CO_2 tanks. A system using this approach was described by Haida et al. 2016 [34]. Moreover, the literature reports that in the case of heat transfer, integrated lubricant-R744 tanks allows for improved heat transfer. In consequence there is a possibility to minimize the lubricant leakage [38].

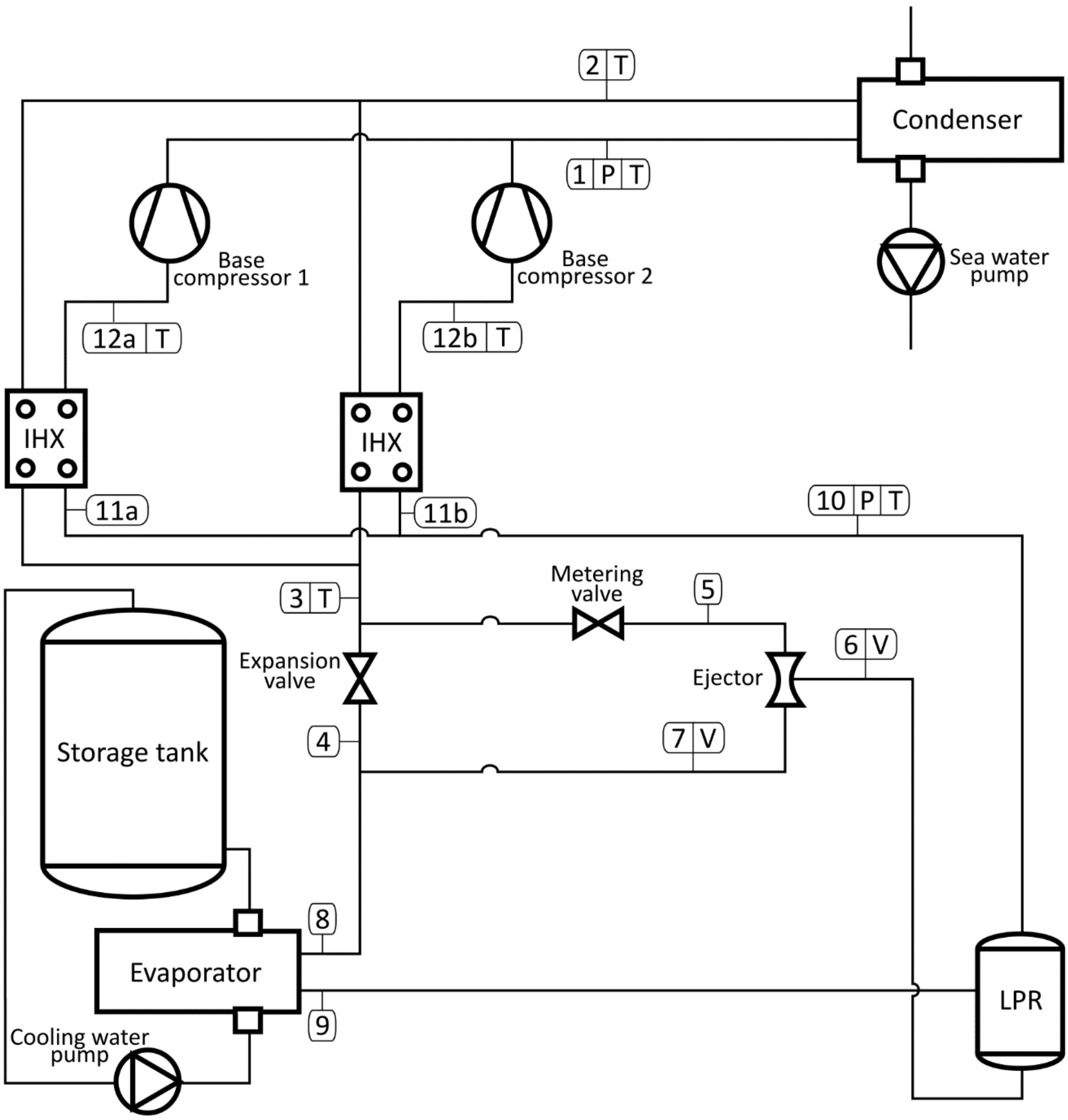

Figure 1. Baseline RSW chillers - R744 refrigeration unit installed in a fishing vessel operating under Scandinavian conditions.

Operation of the mentioned liquid ejector in the analysed RSW installation is focused on the internal circulation of liquid. Energy required for this circulation is recovered from expansion losses on the basis of the ejector work principle. Namely, a flow of subcooled R744 from the IHX is divided into two streams at point 3 (see Fig. 1). One stream is directly expanded in the throttling valve, and the second stream flows through the ejector. The basic scheme of the ejector geometry is presented in Fig. 2, where a motive nozzle, suction nozzle, pre-mixing chamber, mixer and diffuser are schematically shown. The mentioned high pressure subcooled motive stream is expanded in the motive nozzle and converted to a high velocity flow in the premixing chamber. The expansion process in the motive nozzle reaches pressures below that of the suction nozzle port, hence a suction phenomenon occurs. Next, the pressure of the mixed motive and suction streams is increased in the diffuser. Nevertheless, phenomena of the suction and pressure lift are related to each other. Moreover, ejector operation results in only one of the mentioned phenomena being characterised by high intensity, and in the second becomes simultaneously minor. Thus, obtaining high values of suction stream mass flow rate are related to low values of pressure difference (pressure lifts) between the suction and the outlet ports. In the case of the presented RSW installation, the ejector ensures circulation of the liquid, where the goal of its operation is given by the high mass flow rate of the suction stream. Such an operation results in smaller mass flows through the compressors. On this basis, system COP is improved in comparison with that of the traditional cycle without the ejector.

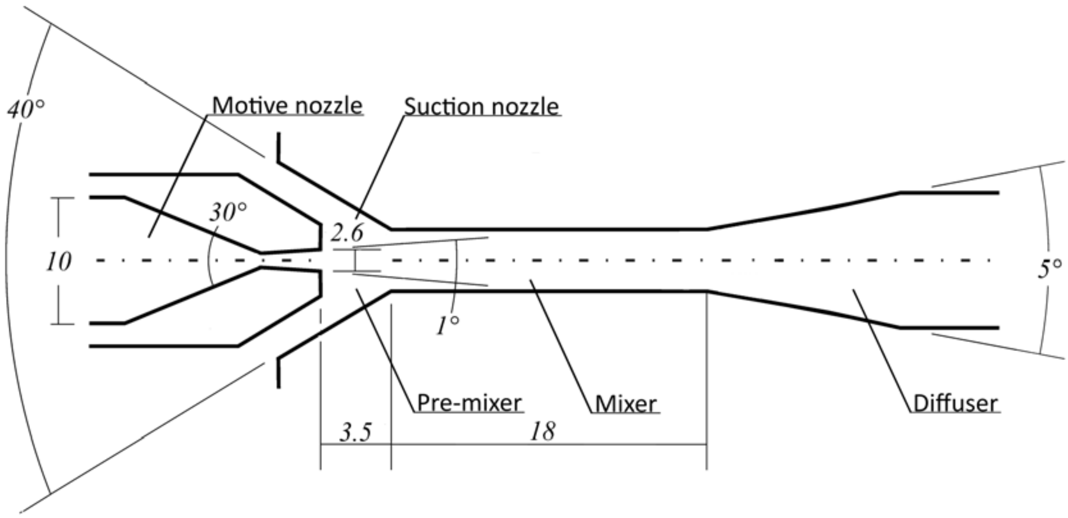

Figure 2. Liquid ejector geometry scheme with the marked flow sections.

2.2. Efficiency of ejector operation

For this study, the ejector efficiency definition (Eq. 1) presented by Elbel and Hrnjak [33] was used. The efficiency of the ejector is given as a ratio between recovered work and maximum available work delivered in the motive nozzle. Namely, the numerator is defined as a difference of enthalpies obtained from an isentropic and isenthalpic compression process from the suction nozzle pressure to the ejector outlet pressure. In the second part, the numerator is defined similarly but considers the expansion process in the motive nozzle:

$$\eta_{EJ} = \chi \cdot \frac{h|_{s=SN,in\ p=p_{out}} - h_{SN,in}}{h_{MN,in} - h|_{s=MN,in\ p=p_{out}}}, \tag{1}$$

where h is the specific enthalpy, subscript s represents the specific entropy in the suction nozzle (SN) and the motive nozzle (MN), p is the pressure, and *in* and *out* are the ejector inlets and outlet, respectively. In this definition, parameter χ, which is called mass entrainment ratio (MER), is used (Eq. 2):

$$\chi = \frac{\dot{m}_{SN}}{\dot{m}_{MN}}, \tag{2}$$

where $\dot{m}$ is the mass flow rate.

3. RSW system at high ambient temperatures

3.1. Warm waters of the Mediterranean and East-Asian regions

The challenging matter of higher heat rejection temperatures should be solved to maintain aspects of high performance and economy. It is worth noting that even seas located in northern conditions report rising temperature levels. An example is given on the basis of satellite data and analysis focused on the basin of Gulf of Finland (Baltic Sea) [39]. In this region, the average annual SST in 1982 was 6.8 °C. Due to the significantly visible warming of approximately 0.04 K per year, the mentioned value increased to 8.2 °C in 2014. However, the temperature change was not constant, i.e., in the middle of the 1980s, the temperature dropped to 5.0 °C, and noting a significant increase up to 7.3 °C in 1989. In the more global case of the Mediterranean Sea, a similar increasing trend has been described [40]. An interesting fact of the same kind as in the Gulf of Finland, an increasing temperature rate of 0.4 K per decade in the last 30 years was observed. Moreover, simulation predictions based on data from the period 1986-2015 showed an approximately 5.8 K increment in the average SST at the end of XXI century.

3.2. Constraints according to fishing vessel construction

Higher temperature differences between ambient and cooled media usually require increased power consumption and larger refrigeration unit sizes. The R744 RSW unit provides a solution in the form of the overall compact size of the installation. However, analysis of power consumption increases and compressor size should allow further economic analysis of such an implementation to fishing vessels according to available space in the machinery room. Constrained space for system modifications and enlargement could be described as a one of the challenges in such an application.

3.3. Analysed modifications to the Baseline System

The motivation for introducing the RSW for hot climate waters is concentrated on the compact size of the system and the ecological label assigned to the natural refrigerant. However, the challenge of heat rejection at higher sea water temperatures has to be solved to maintain the performance and economic aspects. Meanwhile, the temperatures on the Mediterranean coast and in the south-east region of Asia vary from 18 °C to 21 °C and from 30 °C to 33 °C, respectively. According to Sea Surface Temperature (SST) data available in NASA Earth Observation (NEO) databases, the waters of the mentioned East-Asian regions can even reach 35 °C [41]. In the region of the Mediterranean Sea, the temperature differences in comparison to the baseline north conditions are smaller. Nevertheless, water temperatures reach up to 23 °C [41]. The R744 RSW unit is a compact installation. However, analysis of power consumption increases and compressor size should allow further economic analyses of such an implementation on fishing vessels according to the available spaces in their machinery rooms. The constrained spaces available for system modifications and enlargement could be described as a one of the challenges in such applications.

According to the above described space constraints and simultaneous higher power demands, the configurations of ejector-, flash gas- and parallel compression- units were analysed without modification to the rest of the Baseline System installation (black lines). In Fig. 3, the scheme of the modified Baseline System model (red dotted and blue dashed lines) is presented. An Intermediate Pressure Receiver (IPR) was introduced with a second low-pressure expansion valve for liquid expansion. Flash gas (red dotted line) is expanded via the flash gas valve and then mixed with the refrigerant stream from Lower Pressure Receiver (LPR). The parallel compressor line (blue dashed line) was separated from the flash gas line and directed to main line leading to the gas cooler. To simulate hot climate conditions, higher heat rejection temperatures were assumed. To analyse the energy demands of the RSW unit at various fishing vessel locations, two temperature levels were taken into consideration. Hence, temperatures of 21 °C and 33 °C characteristic of the Mediterranean Sea and the waters of east Asia, respectively, were assumed [41]. To investigate the influence of each modification, two systems were simulated separately. The first system was based on flash gas expansion (FGV), for which the entire amount of flash gas was directed to the flash gas valve. Therefore, when the FGV mode was tested, the parallel compression line was turned off. The second system was based on parallel compressor utilisation (PC). In that mode, the flash gas valve was closed, and the entire flash gas stream was draw in by the parallel compressor.

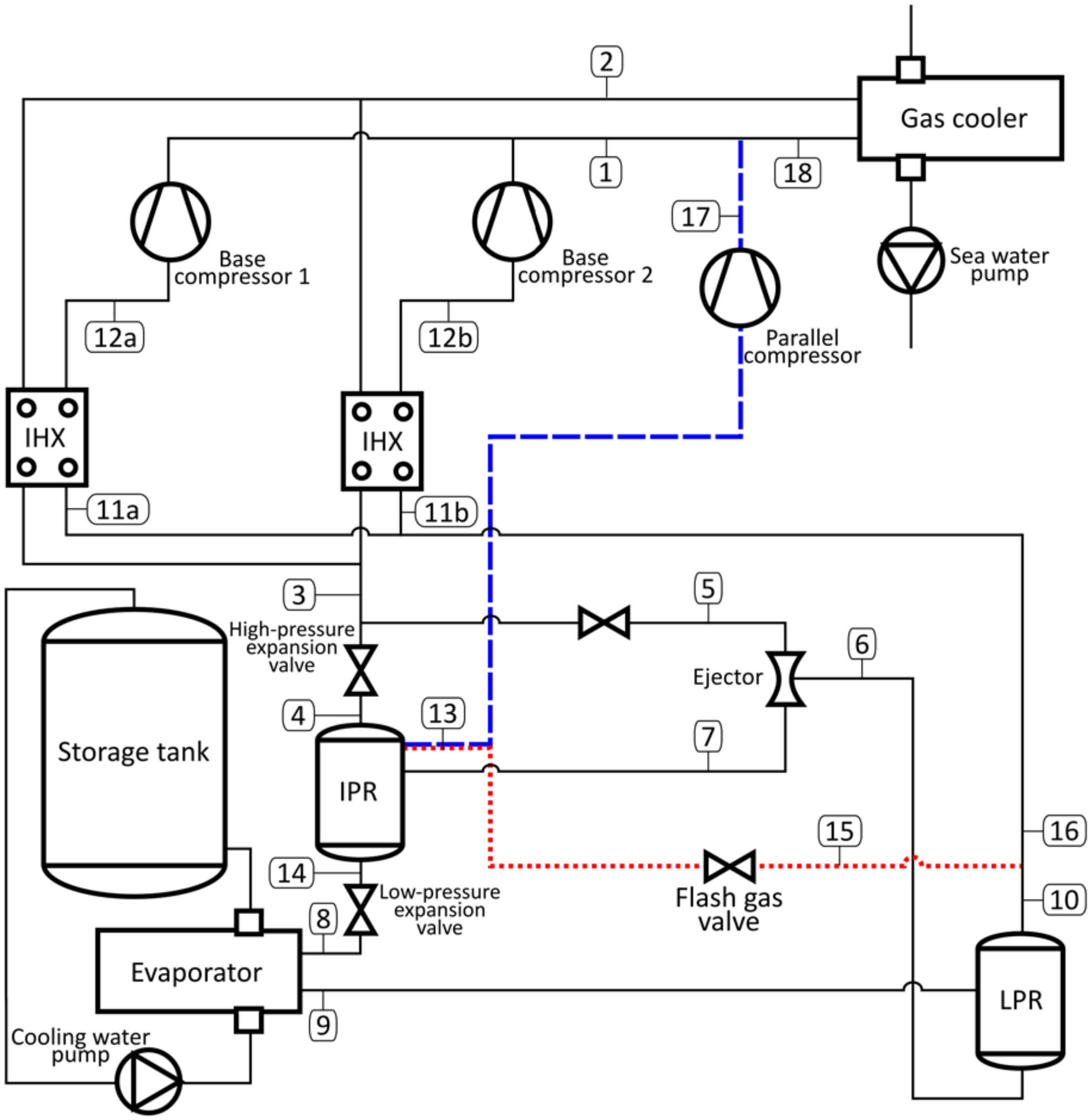

Figure 3. Modified RSW installation with introduced additional equipment: IPR, flash gas line (red dotted) and parallel compression line (blue dashed).

On the basis of the presented FGV and PC installations, the next generation of R744 was developed and described in the literature [13]. Namely, the throttling valve was exchanged with an ejector device, which served as a basis for further cycle improvement, and such an installation is presented in Fig. 4. The basis of this modification is related to the fact that the ejector motive nozzle provides similar mass flow rates as during expansion in a throttling valve. Moreover, to maintain the compact sizing and improve system reliability, ejectors were connected in a multi-ejector module to form one compressed device (green frame in Fig. 4). Each ejector is controlled by individual valves. Due to this, overall regulation is based on the binary idea of opened and closed fixed geometry ejectors working in a parallel mode. The concept of such an approach was delivered in the work of Hafner [13]. The same idea was investigated in this study through simulation of a separated multi-ejector system (ME). The module work is utilised to pump working fluid from the LPR to the IPR. The operation of the vapour ejectors in the multi-ejector module provides unloading of the base compressors by sucking vapour produced in the evaporator to the higher pressure of the IPR, and high enough ejector performance and sufficient motive mass flow rates allow drawing the entire evaporator stream. Eliminating the base compressor and operating with parallel compressors only would be a potential solution for RSW implementation.

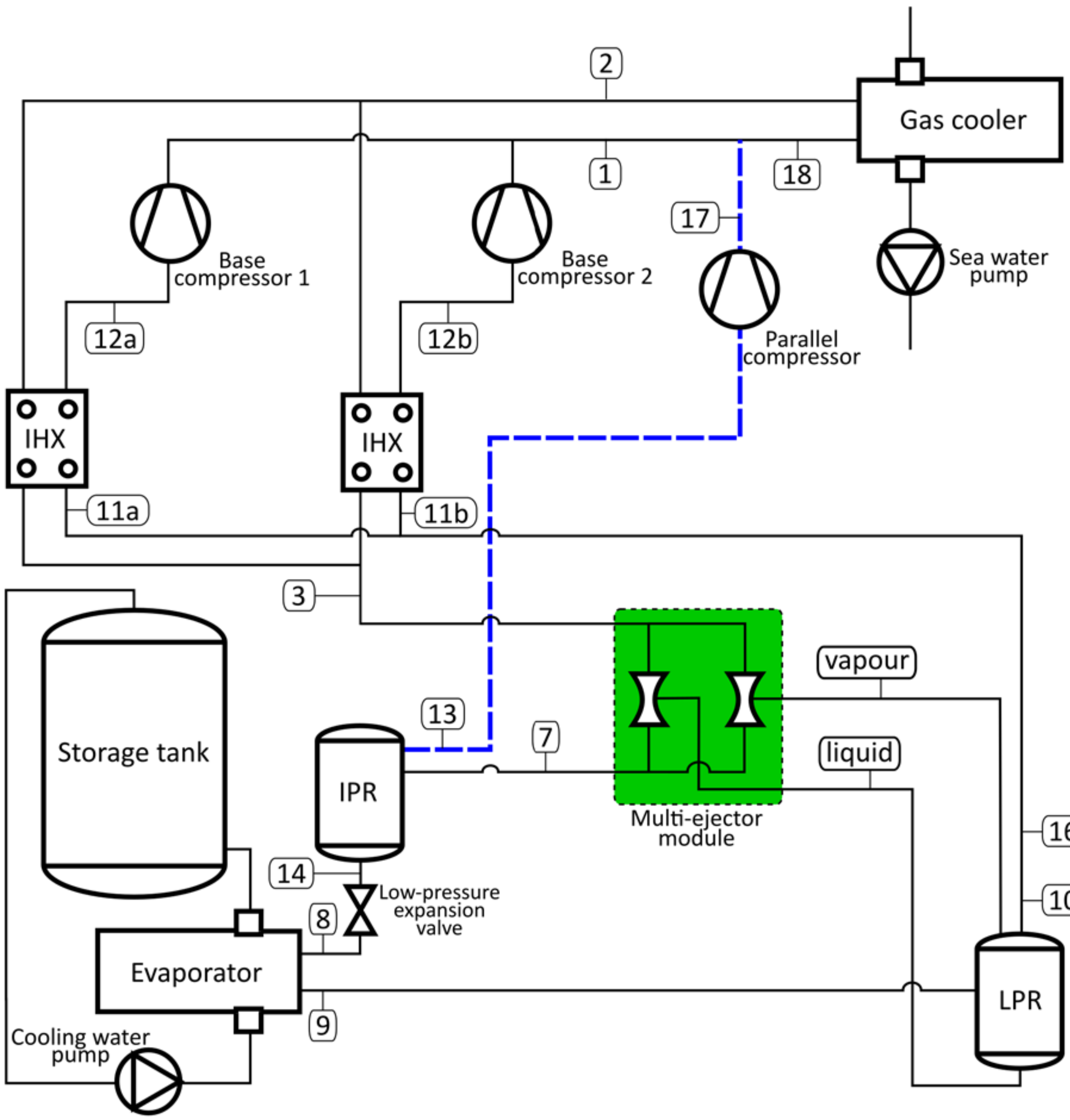

Figure 4. Concept of a new RSW installation based on parallel working ejectors contained in a multi-ejector module.

The main components of the layouts mentioned were presented on the pressure-enthalpy diagram of R744 in Fig. 5. In order to maintain a clarity, the colours used for the representation of each modification correspond to the colours used in Fig. 3 and Fig. 4 – red is FGV, blue is PC and green is ME. Moreover, the processes of each ejector section were marked by green dashed lines. In the ME system, the throttling of the high-pressure refrigerant is exchanged to expansion in the motive nozzle, the expansion ends below the evaporator pressure what results in the entrainment via the suction nozzle (vapour suction illustrated in Fig. 5). After mixing of the primary and secondary streams in the mixer, the pressure is lifted in the diffuser up to the IPR level. From the point of view of the system performance, an introduction of the ejectors benefits in the pressure lift between the evaporator and IPR. In a consequence, the parallel compressor operates with higher suction pressure and the lower pressure ratio what results in the reduced input power. Next, the advantageous approach of the evaporator flooded operation and proper adjustment of the intermediate pressure allows for increased cooling capacity. Finally in Fig. 5, the clearly visible technical advantageous of R744 as the working fluid can be discussed as well. Firstly, the low pressure ratio in the range from 1.5 to 4 could be characterised as substantially lower than that for the synthetic refrigerants. The result of such a value is obtained at the higher efficiency of the R744 compressors. Next, consideration of high operational pressures more than 30 bar leads to other advantageous properties such as low specific volume and more compact sizes of heat exchangers and compressors. Moreover, small pressure drops (and consequently very low temperature drops) in CO_2 installations allow for the selection of the smaller piping systems what again leads to the compact sizing – very demanded from the marine industry.

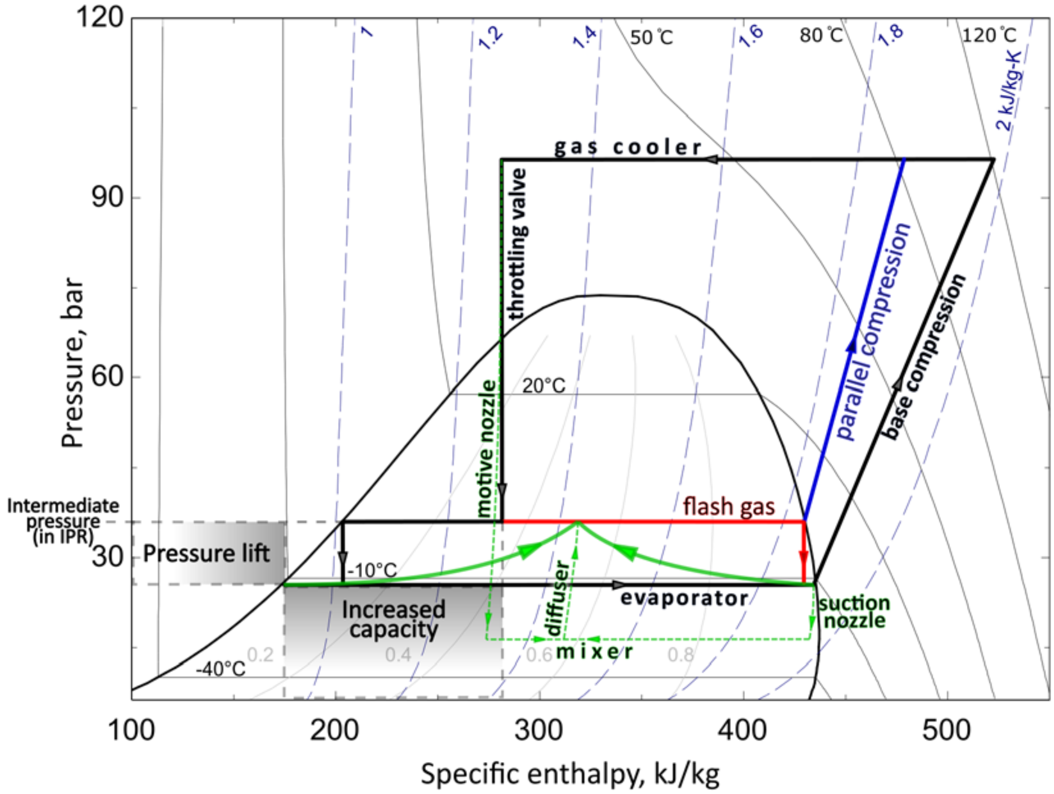

Figure 5. Representation of the modified system layouts (red is FGV, blue is PC and green is ME).

4. R744 cycle modelling - Baseline and modified configurations

The utilised computational approach was presented in the form of flowchart in Fig. 6. The system layout and corresponding mathematical model constituted the first step. The points used in the stream formulations are presented in Figs. 1, 3 and 4. A real fluid property library available in the employed software was used for the determination of the thermodynamic parameters at a given system point. Next, the convergence criteria and range of simulation were established for the given system configuration. The convergence criteria were set to 10^{-5} for both relative residuals and maximum variable change. The authors found that the precise initialisation point is challenging but crucial for the final convergence of the obtained solution. A system of equations was implemented in the Engineering Equation Solver (EES) to iteratively solve each model [42]. This tool offers the Newton-Raphson method as a built-in default solving algorithm for obtaining solutions of sets of non-linear equations. The solution obtained for the given operating point was utilised for the next point computations what substantially improved the convergence time. A series of calculations of firstly established range was finalised by data collection prepared for the further analysis.

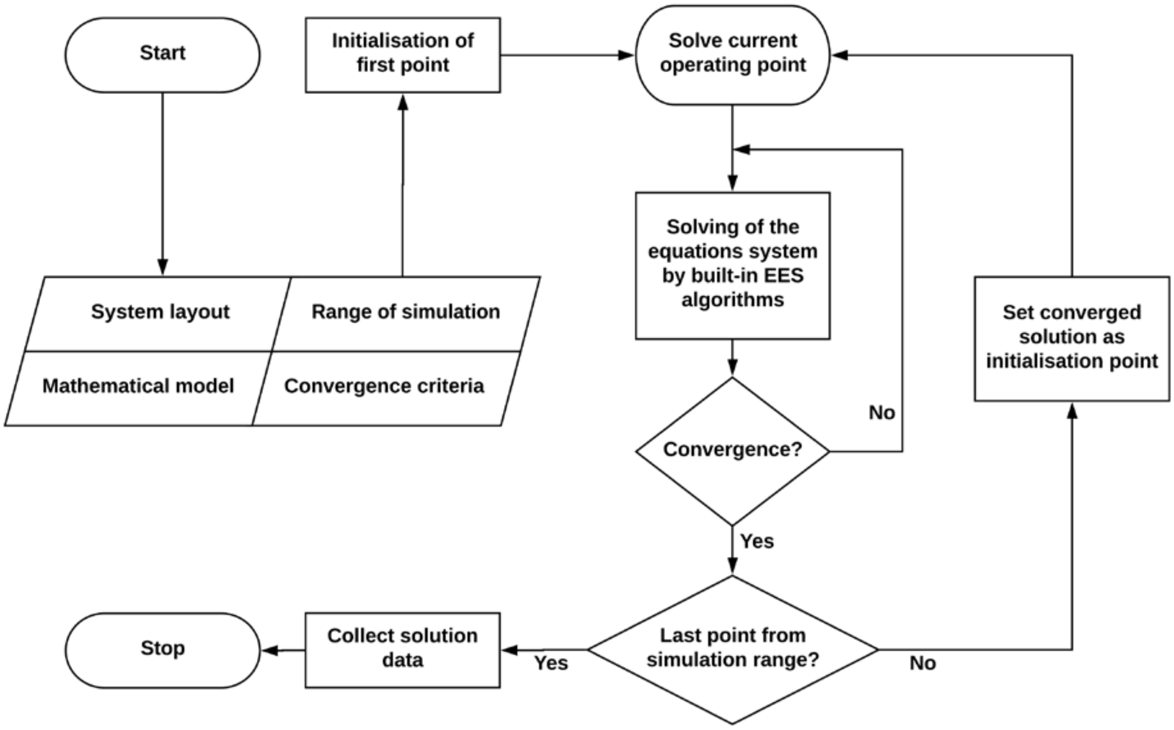

Figure 6. Solving procedure presented in the form of flowchart.

4.1. Fundamentals of the mathematical modelling approach implemented in the Engineering Equation Solver

The developed Baseline model was based on the measurement data delivered by the fishing vessel operator. The model was used to evaluate the performance of the actual installation. To simulate higher operating conditions, the modified models were prepared. However, the models were developed on the basis of the Baseline model used for actual cycle evaluation. Namely, the analyses of the Baseline and modified RSW installations were executed on the basis of energy and mass balance equations. In the following formulations, point identification is based on the Baseline System (Fig. 1) and the modified system scheme (Figs. 3 and 4).

The compressors consume energy delivered to the systems. Hence, a total compressor power equation was formulated:

$$\dot{W} = \dot{m}_{COMP1} \cdot \left(h_1 - h_{12a}\right) + \dot{m}_{COMP2} \cdot \left(h_1 - h_{12b}\right) \cdot \qquad (3)$$

To calculate the energy distribution between the evaporator, ejector diffuser and expansion valve, the energy balance of this section is formulated in Eq. 4:

$$\dot{m}_{EVAP} \cdot h_8 = \dot{m}_{DIF} \cdot h_7 + \dot{m}_{VALVE} \cdot h_4 . \qquad (4)$$

Mass stream balances from Eq. 3 and Eq. 4 were formulated to obtain distribution of mass flow through the evaporator, expansion valve and ejector motive nozzle:

$$\dot{m}_{COMP} = \dot{m}_{VALVE} + \dot{m}_{MOT} \text{, and} \tag{5}$$

$$\dot{m}_{EVAP} = \dot{m}_{VALVE} + \dot{m}_{DIF} \text{,} \tag{6}$$

where $\dot{W}$ is power, $\dot{m}$ is mass flow rate, and h is specific enthalpy. The subscript COMP denotes the compressor, EVAP denotes the evaporator, DIF denotes the ejector outlet port, VALVE denotes the expansion valve, and MOT denotes the ejector motive port. In addition to the compressor power consumption, the evaluation of the compressor work was based on the equation for the compressors' isentropic efficiency (Eq. 7):

$$\eta_{is} = \left(h_{1,s} - h_{12}\right)/\left(h_1 - h_{12}\right), \tag{7}$$

where the subscript *is* denotes isentropic, and *s* is specific entropy. As a simulation result, the process mass flow rates of the system were calculated. Hence, calculation of the system power demand at given operating conditions and evaporator load was possible. Further, the system performance was presented in the form of the COP factor, which is defined as follows in Eq. 8:

$$COP = \frac{\dot{Q}_{EVAP}}{\dot{W}}, \tag{8}$$

where $\dot{Q}_{EVAP}$ is the heat transferred in the evaporator. The equation of evaporator energy balance (Eq. 9) is formulated as

$$\dot{Q}_{EVAP} = \dot{m}_{EVAP} \cdot \left(h_9 - h_8\right). \tag{9}$$

The analysis of the flash gas valve and the parallel compression systems were based on the modified Baseline model. In the case of the flash gas valve, the introduction of the IPR, two expansion valves and an additional flash gas line was necessary. To model these modifications, the additional energy balance related to the IPR was formulated, as presented in Eq. 10:

$$\dot{m}_{VALVE} \cdot h_3 + \dot{m}_{DIF} \cdot h_7 = \left(\dot{m}_{EVAP} + \dot{m}_{FGAS}\right) \cdot h_{IPR}, \tag{10}$$

where subscript FGAS represents the flash gas, and IPR represents the intermediate pressure receiver. The vapour quality value in the IPR was estimated on the basis of the obtained IPR enthalpy and the assumed pressure in the tank. Moreover, the mixing of the flash gas stream and saturated vapour from LPR was modelled on the basis of mass (Eq. 11) and energy balance (Eq. 12):

$$\dot{m}_{FGAS} + \dot{m}_{LPR} = \dot{m}_{COMP} \text{, and} \tag{11}$$

$$\dot{m}_{FGAS} \cdot h_{15} + \dot{m}_{LPR} \cdot h_{10} = \dot{m}_{COMP} \cdot h_{16}. \tag{12}$$

The parallel compression was related to the additional equation for compressor work (Eq. 13). The mixing of the base compressor stream and auxiliary compressor stream was modelled by the mass (Eq. 14) and energy (Eq. 15) balances:

$$\dot{W}_{PAR} = \dot{m}_{COMP,PAR} \cdot \left(h_{17} - h_{13}\right), \tag{13}$$

$$\dot{m}_{COMP,PAR} + \dot{m}_{COMP,BASE} = \dot{m}_{COMP} \text{, and} \tag{14}$$

$$\dot{m}_{COMP,PAR} \cdot h_{17} + \dot{m}_{COMP,BASE} \cdot h_1 = \dot{m}_{COMP} \cdot h_{18}, \tag{15}$$

where the subscript $COMP_{PAR}$ represents the parallel compressor, and $COMP_{BASE}$ represents the base compressor. Moreover, the separated isentropic efficiency equation (Eq. 16) for parallel compression was added:

$$\eta_{is,PAR} = \left(h_{17,s} - h_{13}\right)/\left(h_{17} - h_{13}\right), \quad (16)$$

where $\eta_{is,PAR}$ is the isentropic efficiency of the parallel compressor. Similarly, as in the baseline simulations, the system COP was used as the evaluation factor. However, in the case of the parallel compression, the work of the auxiliary compressor was included in the COP factor defined in Eq. 17:

$$COP = \frac{\dot{Q}_{EVAP}}{\dot{W} + \dot{W}_{PAR}}. \quad (17)$$

4.2. Assumptions for the simulations of the modified systems

According to the introduced devices, the following assumptions were provided for the flash gas, parallel compression and multi-ejector systems.

Ejector operation was modelled on the basis of a 1-D homogeneous equilibrium model, in which each section's efficiency and the pressure in the mixing section were assumed. The efficiencies of the motive nozzle, suction nozzle and diffuser were assumed to be equal to 85 %, 80 % and 80 %, respectively, for both the vapour and liquid ejectors. Similar modelling approach was presented by Liu and Groll [43] when slightly higher motive efficiency and slightly lower diffuser efficiency were assumed. Moreover, similar results were obtained in the other papers as well [44], [45]. Especially in the work of Liu and Groll [45] as well as the work of Zhang et al. [46], the wide literature survey provided data of the efficiencies. In the work of Ahammed [47], some conclusions listed by Liu and Groll [45] were used. The authors assumed the constant mass entrainment ratio on the level of 0.85 [47]. Additional assumptions of chocked flow in the motive nozzle and constant pressure mixing section were introduced as in this study [47]. Moreover, these results were validated with the experimental data presented by Nakagawa [48]. The comparison resulted in some similarities between the simulated and experimentally tested ejector performance simultaneously showing substantial discrepancies between global factors of the system performance. After exergy analysis of the system components, the exergy destruction of ejector components was substantially lower than in the case of heat exchangers and compressor. Finally, the authors showed that the assumption of ejector efficiency could be characterised as crucial having regard comparison of the ejectors. On the other hand, it might have relatively larger margin in the case of whole system comparison. Some additional examples of assumed ejectors efficiency can be found in more recent paper of Zheng and Deng [49]. In that paper, the authors confirmed the most common approach of the assumed isentropic efficiency value of 80 %. Moreover, the efficiency of the motive nozzle was mostly higher than that of the mixer and diffuser and took values on the level higher than 85 %. On the other hand, as presented in comprehensive review about ejector refrigeration system modelling [50], the case of R744 is very specific because only few studies linked the ejector model with the system modelling. In a consequence, the choice of the proper model assumptions such as the ejector efficiency is still a challenging matter. Hence, the assumptions of sections' efficiency in this study in range 80 % - 85 % could be characterised as typical but not the highest from the reports available in the literature. The assumed pressure drop between the suction nozzle outlet and the mixer section was equal to 100 kPa on the basis of the authors' previous experience [44]. In the Baseline System, pressure lift is utilised only for the pressure drop between LPR tank and the evaporator, and thus the estimated ejector efficiency was 1.15 %. Furthermore, the liquid ejectors in the case of the modified systems were described by a constant overall efficiency equal to 15 %. This assumption was made for the single liquid ejectors as well as the liquid ejectors section in the multi-ejector module. According to the various pressure levels in the evaporator and IPR tank, the necessary motive stream was calculated.

In the case of the ME system, two different approaches were used in the computational procedure. According to the large amount of potential recovery work in the case of East-Asian conditions, it was assumed that the ejector work would be enough to intake the entire evaporator stream. This means zero power consumption by the base compressors. Due to that, the necessary efficiency was calculated and is further analysed in the discussion of the results. The evaluation of efficiency allows for the statement that this assumption was reasonable. Another approach was provided in the case of the Mediterranean climate, for which potential recovered work was lower. In this case, ejector efficiency was assumed to be a function ranging from 20 % to 35 % on the basis of performance maps presented in the work of Banasiak [32]. This assumption provided results in the form of evaporator stream distributions for the ejector and base compressor suction port.

According to the liquid circulation ensured by the liquid ejectors, the vapour quality at the evaporator outlet was assumed to be 0.95. The liquid phase of this stream was drawn by the implemented liquid ejectors from the LPR to the IPR.

The same compressor manufacturer was used for the Baseline System and modified installations. However, different types of compressors were utilised for parallel compression purposes due to the higher values of suction pressure. Moreover, on the basis of the auxiliary compressor operating limits, a simulated intermediate pressure (IP) range was assumed. Namely, a simulated 35 bar in the IPR tank was the lowest, and the highest suction pressure was 45 bar.

The isentropic efficiency of the base compressors and the parallel compressor was calculated for each simulation on the basis of the data provided from most of the manufacturers. Namely, the power input and heat released are given in certain operating conditions of the condenser and evaporator. According to these data, simple calculations based on the thermodynamic relations for one-stage refrigeration system resulted in the map of the compressors performance. Every of manufacturer's point was used, while the operation between these points were approximated linearly. The efficiency function involved two previously mentioned pressure arguments and was obtained on the basis of the data on the semi-hermetic transcritical CO_2 compressors delivered by the compressor manufacturer [37]. In Fig. 7 and Fig. 8, the isentropic efficiency maps are presented in a function of the high pressure and saturation temperature which correspond to the given pressure level in the evaporator or IPR. The maps ranges were limited to the conditions analysed in the study. Hence, it is not a full range from the manufacturer's website, but only an area needed for the calculations.

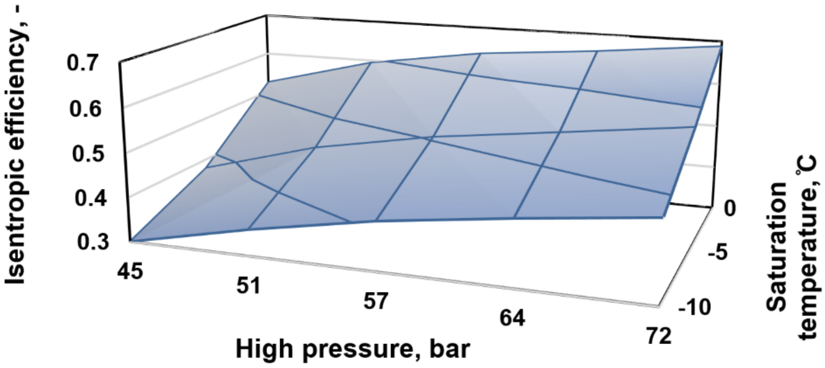

Figure 7. Isentropic efficiency mapped on the basis of the manufacturer's data for subcritical operation.

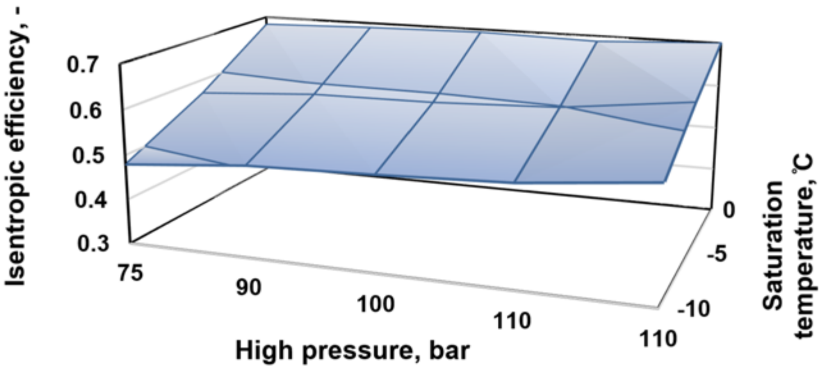

Figure 8. Isentropic efficiency mapped on the basis of the manufacturer's data for supercritical operation.

As mentioned, the performance map available from the manufacturer data was utilised in order to calculate the compressors efficiency. In this way, the whole range of the assumed pressure and temperature conditions was covered. Unfortunately, the heat loss data in a full range of the simulated parameters was not available. Hence, the heat loss would be assumed without any basis. Moreover, it would be hard to estimate conditions in the fishing vessel machinery due to still developing R744 technology in the case of such a marine application. Due to that, the heat loss was neglected in this study. According to this assumption, the temperature at the compressor outlet was obtained on the basis of the enthalpy calculated from the isentropic efficiency equation, where values of the isentropic efficiencies were delivered from the performance maps of the manufacturer used in this study. Having regard small marine applications, the heat loss from the compressors would be relatively lower when comparing to the stationary applications with large input powers. Finally, the influence of the isentropic efficiency onto the specific enthalpy before the gas cooler could be estimated as much more significant than compressor heat loss [51].

On the basis of the compressor manufacturer's data, a superheat of gas at the base compressor suction port was assumed to have a temperature of 10 K [37]. Equations for the IHX energy balances were added as well, for which the intermediate heat exchanger efficiency was assumed to be 100 %.

In general, the pressure drops in the R744 system could be evaluated as a relatively low in comparison to those for the synthetic refrigerants as well as for most of the hydrocarbons. Moreover, in the Baseline installation, the lines dimensions were adjusted in order to minimise the pressure drops. Hence, the pressure drops in the Baseline installation are negligible and were not taken into account in this study. Moreover, an evaluation of the other possible pressure drops in the filters, complicated arrangement (due to limited space) of pipelines and in the valves of the whole system (in the form auxiliary equipment of the separators, compressors and ejectors) would be very challenging having regard various conditions of the system work. On the other hand, most of the components mentioned are as compact as possible in order to fit into the restricted areas in the considered marine applications Consequently, in the cases of the modified installations, the same space limitation could be assumed. Due to that, the specific values of the pressure drops could be evaluated only after a complete design of the installation and its fitting to the fishing boat machinery room. At this stage of the analysis, it would be challenging and simultaneously would influence the results in the minor way due to very advantageous properties of the carbon dioxide.

Moreover, some literature reports that the highest pressure drops occur in the evaporator and they are even lower than 1 bar [52]. In this study, the evaporator pressure was iteratively calculated on the basis of the temperature difference between the refrigerant and cooled water. According to a cooled water temperature equal to -1 °C, the required temperature of the refrigerant was calculated as a function of the vapour quality at the evaporator inlet. On the basis of the heat transfer coefficient correlation presented in the work of Cheng [53], a proper function was approximated for vapour quality in the range of .0 to 0.6. The function described the deterioration in the heat transfer conditions with the reduced amount of liquid delivered to the evaporator. Finally, according to the constant evaporator load, the necessary temperature differences were calculated.

Cooling capacity was assumed on the basis of control terminal data delivered by the fishing vessel operator. On the basis of the obtained data, the representative evaporator load was estimated at a level of 250 kW, and that value was assumed for all the simulations of heat rejection high temperatures. Moreover, to evaluate possible implementations and amounts of corresponding compressors, the range of evaporator loads was additionally simulated for the case of the most promising solution. That range was assumed to be from 250 kW to 455 kW.

To investigate the two mentioned hot climate conditions, two heat rejection temperatures in the gas cooler were assumed. Moreover, the temperature difference between the refrigerant and sea water at the gas cooler outlet was assumed to be 5 K as in the Baseline installation. Hence, 26 °C and 38 °C refrigerant temperatures at the outlet of the gas cooler were tested.

4.3. Simulation range for high temperature heat rejection

The input data ranges for the simulations of the modified systems were based on the studies presented by Gullo that focused on the R744 booster system with a parallel compressor [54] [19]. That analysis was based on the optimisation of the high pressure, the temperature at the gas cooler outlet, the parallel compressor mass flow rate and the pressure level in the IPR. In this study, the pressure and temperature ranges presented in Table 1 were investigated. Thus, Mediterranean and East-Asian waters were simulated on the basis of two temperature levels after the gas cooler, and three pressure levels in the IPR were tested. The range of pressure levels in the IPR tank was assumed on the basis of operating limits delivered by the compressor manufacturer [37]. In the work of Gullo, a level of 35 bar was assumed as well. However, those authors assumed that value to be a constant [19]. In this study, three different values were simulated to investigate the influence of this parameter. Finally, for the Mediterranean climate, the high pressure level was tested in the range from 66 bar to 115 bar, where 66 bar is a limit for the subcritical mode. In the case of the East-Asian climate, the pressure at the compressor outlet was simulated in the range from 75 bar to 115 bar. The described parameters were introduced to the model as a set of boundary conditions for the Baseline System and two modified cases (FGV and PC). According to the works of Banasiak [32], Haida [34] and Bodys [35], the intermediate pressures (IP) for the ME systems operating with vapour ejectors were assumed with regard to the multi-ejector module operating range. Hence, the IP were different than for FGV and PC, namely 34 bar, 36 bar and 38 bar. Having regard the configuration based on the combined FGV with PC, the authors defined the scope of the paper in accordance to the state-of-the-art ejector technology. Consequently, the vast majority of the analysis was focused on the multi-ejector system. Hence, the direct analysis of the most perspective solution was chosen by the authors.

Table 1. Set of input data for simulations of high temperature heat rejection in Mediterranean and East-Asian climates.

Climate	Mediterranean	East-Asian
t_2, ℃	26	38
p_{ipr}, bar	35, 40, 45 (34,36, 38 for ME)	
p_1, bar	66, …, 115	75, …, 115

5. Discussion of the results for the Baseline and modified RSW systems

The Baseline System was simulated according to the measurement points of the actual system under operation. The set of input data and obtained system COP for Baseline System simulations in Scandinavian conditions are presented in Table 2. The system points are described in Fig. 1, C1 and C2 denote the frequency settings of compressors 1 and 2, respectively. $\dot{m}_{SUC}$ denotes the amount of liquid sucked by the ejector in a given system state. Evaporator load is determined by Q_{EVAP}. The high and low pressure sides of the cycle are given by p_C and p_O, respectively. According to the state-of-the-art technology discussed in the literature survey, the positive influence of the liquid ejectors were assumed. Hence, the experimental data of classic R744 refrigeration unit operating in the marine conditions is not available. Nevertheless, the reason for the liquid ejector implementation could be found in the expected performance improvement of the classic R744 system [11]. Five different system states were simulated on the basis of introduced ejector suction mass flow rate. The first two states represent the full load state, whereas the other three were related to partial load operation. According to the lack of motive nozzle measurements, an assumption of a constant MER value was made to calculate the motive stream. The MER value was assumed to be 1.5 on the basis of the ejector design process data delivered by SINTEF Energy Research. Moreover, a similar approach was utilised in the other studies [32], [33], [55]. The level of the obtained system COP ranged from 4.71 to 9.25. The increments were related to the declining condensation pressure p_C and increasing suction mass flow rate $\dot{m}_{SUC}$. Moreover, the relation between condensing pressure p_C and the declining temperature after condenser t_2 was maintained. The high COP values provided a wide perspective on further implementation areas. According to the relatively low values of the condensing pressure and simultaneously lower potential of the recovery work, the performance of the liquid ejector might be underestimated. On the other hand, in the case of restricted operating area in the light of the rejection temperatures, i.e. at the Norwegian Coast, more specialised design of the ejector would be valuable. In this case, a sensitivity study of the liquid ejector geometry influence on the R744 system operating in low ambient temperatures could be found as a very useful analysis. However, the actual development of this technology denotes that the COP improvement is ensured when comparing the classic R744 system layout and the system equipped in the ejector.

Table 2. Input data, data for the Baseline System and obtained COP for an actual RSW installation operating on the northern Norwegian coast.

C1 Hz	C2 Hz	p_c bar	t_1 ˚C	t_2 ˚C	t_3 ˚C	Q_{EVAP} kW	$\dot{m}_{SUC}$ kg/min	p_o bar	t_{11} ˚C	t_{12} ˚C	COP -
70	70	55.3	77.3	15.9	1.8	234	0.3	28.3	-6.1	14	4.71
70	70	56.3	70.9	16.5	6.9	353	4.9	30.9	-3.2	14.9	5.34
0	60	50	58.9	12.2	4.8	139	5.7	28.2	-6.2	1.9	5.75
35	35	50.3	64.7	12.5	1.1	144	8.3	28.3	-6.2	12.6	7.65
0	50	48.4	46.9	10.7	4.4	106	6.7	28.4	-6	1.8	9.25

5.1. Proposed modifications for hot climate applications

The Baseline System as well as the modified flash gas valve, parallel compression and multi-ejector systems were tested for the mentioned Mediterranean and East-Asian climates. The results from the first group of simulations (Mediterranean) are presented in Fig. 9, in which the relationship between COP and the high pressure is given. The Baseline System is described by a black curve. The results from FGV, PC and ME are described by the group of red, blue and green curves, respectively. In addition, the value of IP (bar) is indicated by a number after the system determination. The same manner of data presentation was used in further analysis. Some of the parameters were obtained on the basis of modelling assumptions. The evaporator load Q_{EVAP} was set to the constant value of 250 kW. The evaporator pressure p_O was quite constant and between 30.08 bar (-5.5℃) and 30.28 (-5.1℃) depending on the vapour/liquid conditions calculated at the evaporator inlet. On the basis of constant efficiency assumed for liquid ejector, the mass entrainment ratio was computed. In the best cases of Mediterranean climate, the values were of 1.8, 1.9 and 0.3 for the FGV, PC and ME system respectively. These relatively high values in the case of FGV and PC are related with low potential of work recovery corresponding to the motive nozzle pressure conditions. In the East-Asian climate, these values become lower due to higher motive nozzle pressures. Namely, they are 0.8, 0.9 and 0.7 for the FGV, PC and ME system, respectively. Finally, the temperature t_2 was 26℃ and 38℃ for the Mediterranean and East-Asian climate, respectively. According to the temperature assumed for the Mediterranean climate, the R744 systems should work in the subcritical mode. Hence, the obtained character of the curves shows that the optimum pressure is located at the lowest possible value. Further increment of the high-side pressure results in the additional compressor work with insufficiently enough increment of the cooling capacity. This subcritical mode can be compared to the others refrigerant with high temperatures of the critical point. According to the results presented in Fig. 9, the highest COP, 3.22, is related to the ME system and 34 bar in the IPR, namely for case ME-34. Increasing pressure in the IPR deteriorated the COP of the ME system, which is related to the too small amount of recovered work in the ejector. Having considered the higher motive pressure (gas cooler pressure), the ratio of the COP decrement decreased. An explanation is found in the higher potential for work recovery available in the ejector in the region of higher pressures. Nevertheless, in the case of the highest COP, the efficiency for ME was only on a slightly lower level than that of the Baseline System under favourable Scandinavian conditions. A similar situation is related to the FGV and PC systems.

Namely, the lowest pressure in the IPR provided the highest performance based on the increased cooling capacity. Regarding the FGV and PC systems, the obtained COPs were at a lower level than that for the ME systems. For the lowest pressure related to the operating limits of the subcritical mode, the PC and FGV systems ensured COP of 2.98 and 2.66, respectively. Due to this, the lowest possible level of high pressure should be ensured for optimal performance.

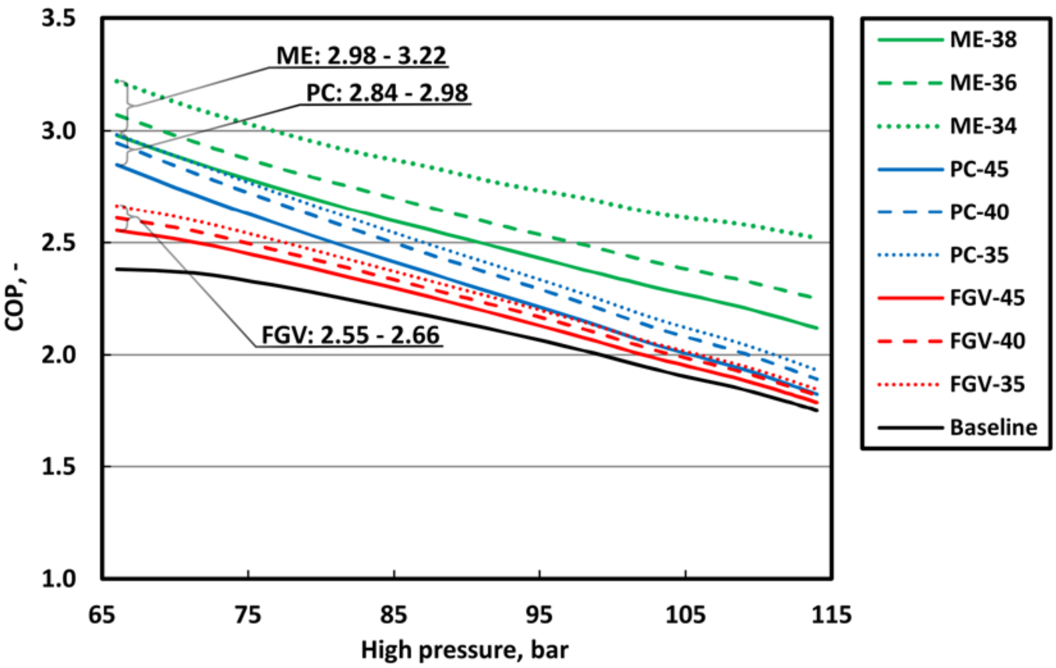

Figure 9. COP of the Baseline System and the modified cases presented as a function of high pressure from simulations performed for the Mediterranean climate.

Simulation cases of the heat rejection temperature characteristic for East-Asian waters provided the results presented in Fig. 10 in the same manner case identification was used for Fig. 9. However, in Fig. 10, the optimum pressure of each system is marked by a vertical line with the corresponding pressure value. Having regard higher temperature of the heat rejection which is above the CO_2 critical point, the character of the curves is substantially different comparing to Fig. 9. The reasons are located in the supercritical state of operation. Consequently, the condenser should be exchanged into a gas cooler because the heat rejection takes place above the critical point. This change is characteristic for R744 and brings several aspects to consider.

First of all, a wide range of optimum operations should be seen due to small changes in performance near optimum pressure. However, significant differences are visible between the performances of each system. Namely, the best prospective solution is related to the ME system, for which the maximum COP is equal to 2.58 at 94.6 bar of high side pressure. The optimum pressure in the IPR is different than for the lower heat rejection temperatures, and a higher ME system performance was obtained in the case of the highest pressure in the IPR tank. This relation is directly connected with the increasingly efficient operation of the multi-ejector module delivering the vapour to higher pressure levels. Simultaneously, higher pressure levels at the parallel compressor suction port directly result in lower power consumptions. However, the difference between the investigated IP is on the level of 5 % - the highest COP of ME-36 is equal to 2.46, whereas that for the mentioned ME-38 is 2.58. Moreover, the differences between system COP are more visible for various IP of the ME system than for the PC and FGV systems. The latter ones seem to be much less dependent on pressure in the IPR. Increasing high pressure delivers a quite visible better performance of mentioned 40 bar of the IP, which is related to the balance between recovered work and the favourable conditions of parallel compressor work. In the FGV case, even the smaller difference is visible, and the other trend is obtained as well. The FGV system should operate in the lowest IP throughout the range to obtain the highest performance, with a COP of 1.65. Regarding an optimum high side pressure, slightly different values were obtained for each system, which is related to other factors of crucial impact on system COP. In the case of ME, it is the ejector efficiency and the performance of the parallel compressor. In the case of the PC and FGV systems, the final results are created at an optimum point between parallel compressor efficiency and amount of gas in the IPR tank. The computations for the PC system resulted in the highest COP for a high side pressure of 92.5 bar in the PC configuration. Concerning the series of curves related to FGV, the optimal COP value was obtained at system high side pressure levels of approximately 97 bar. The global trends of every simulated system can be described as being similar. However, the modified systems are characterised by significant COP differences one to another, especially in the range of optimal pressures after the compressors. Moreover, different pressures for optimal operation should be ensured for each case, and the PC optimum should be reached for the lowest pressure in comparison to the other systems.

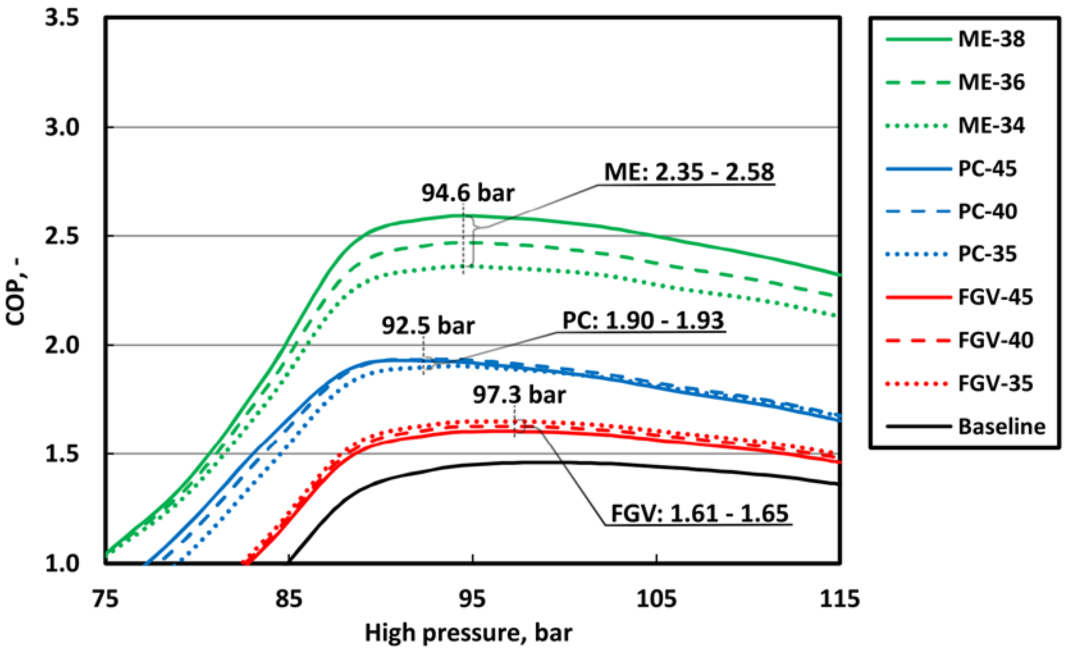

Figure 10. COP of the Baseline System and the modified cases presented as a function of high pressure from simulations performed for the East-Asian climate.

5.2. System modifications according to power demand increment

According to the goals, possible modifications were analysed in the light of the restricted spatial volumes in the machinery rooms of fishing vessels. From this perspective, the most demanding system would be a ME or PC unit due to the additional compressor implementation. However, under the higher ambient conditions (East-Asian), in the ME unit only the parallel compressors could be applied to circulate the refrigerant; this will be further explained below. Moreover, regarding the obtained COP improvements of the modified systems, the ME was assigned as the most promising solution. Nevertheless, due to the higher ambient (sea water) temperatures compared with those for Scandinavian conditions, higher power consumptions are expected. In the Baseline System configuration, two base compressors characterised by a maximum power of 44 kW are used [37]. The maximum power of the transcritical CO_2 compressors available from the manufacturer used in this study is approximately 58.3 kW [37]. To avoid introducing a third compressor into the fishing vessel machinery room, two compressors with a maximum power demand of 58.3 kW should be used for the purposes of PC. To evaluate these possibilities, an analysis of compressor power and corresponding system efficiency is provided.

Requirements on compressor power for the Mediterranean and East-Asian climates are presented in Fig. 11 and Fig. 12, respectively. The total compressor power for FGV is presented as a function of the high side pressure (red line). The ME (green) and the PC (blue) systems are represented by solid, dashed and dotted lines for their total power consumptions, base load compressors and parallel compressors, respectively. The same manner of case determination was used for the graph presented in Fig. 12. According to Fig. 9, the point of optimal operation is related to the lowest pressure, where the power consumption is at the lowest level. Operation of the FGV system can be obtained by two compressors of similar maximum power to obtain approximately 94 kW of total power at the optimal (lowest) high pressure. Moreover, the total work of each compressor could be on a level from below 58.3 kW up to approximately 90 bar. In the case of the more advanced solutions, lower power consumption is presented. However, despite that, one additional compressor should be delivered for the PC solution to cover parallel compression purposes. This is caused by the very low consumption of the parallel compressor and the high, increasing power of the base compressor section, which excides the assumed maximum power of one compressor - 58.3 kW. Hence, in the Mediterranean climate, the same set of compressors as in the Baseline system can be used only in the case of the FGV and ME configurations. Nevertheless, the low COP of the FGV system should disqualify such an implementation. The difference in required powers of FGV and ME is significant, and only the ME system should be considered. In the case of the ME system, similar compressors for base and parallel purposes should be used with a proper modification of the piping system. Finally, the required maximum power of compressors installed in the Baseline System would be enough for the Mediterranean climate, whereas the ME system would ensure the most efficient operation.

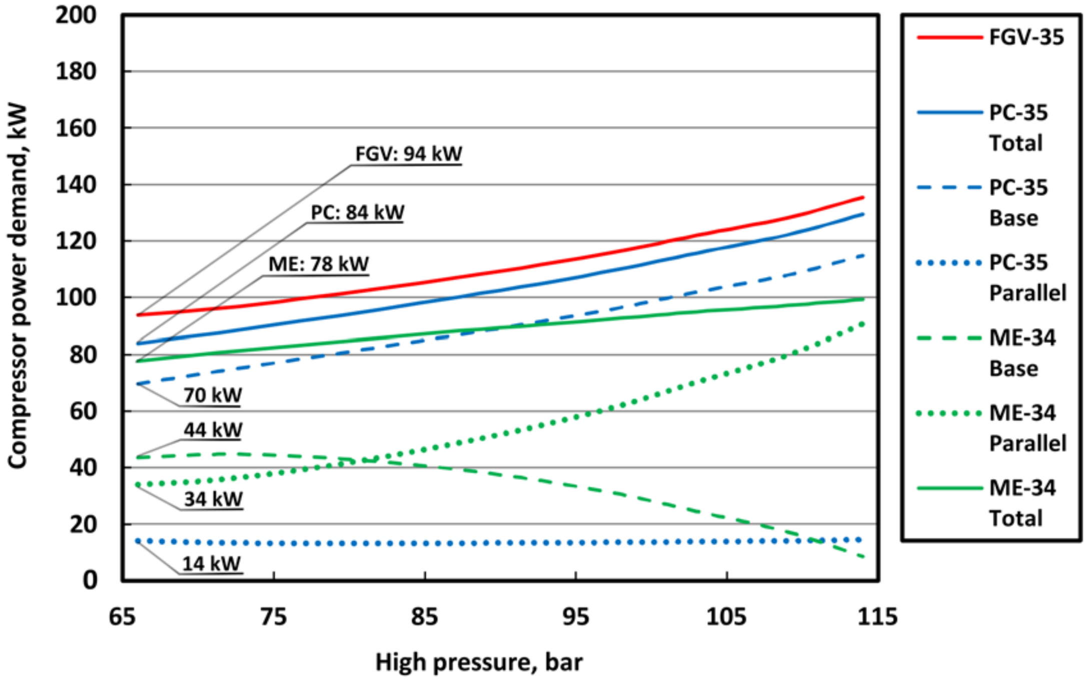

Figure 11. Compressor power demand of the modified systems characterised by the highest COP improvement in Mediterranean climate.

The power consumption for the East-Asian climate is characterised by definitely higher values than for the Mediterranean climate. Moreover, a non-linear trend is presented in Fig. 12, in which the power consumption results are described in the same manner as in Fig. 11. First, significantly more power is needed in the case of a FGV or PC unit. In comparison with the power consumption for the Mediterranean climate, approximately 60 % and 53 % more energy should be delivered for FGV and PC operating in an East-Asian climate. In the same comparison, the ME system operating under East-Asian conditions needs only 24 % more power. Due to the mentioned necessary power, a FGV system could not operate without an additional third compressor because the FGV total power (151 kW) is significantly higher than the 115 kW corresponding to two compressors. The base compressor section of the PC system would be covered by two compressors. However, only a very narrow buffer of additional cooling capacity would be ensured. Namely, the mentioned maximum power for two compressors is approximately 115 kW, whereas the base compressors of the PC system would operate at a level of 106 kW. On the other hand, any significant additional cooling load would be impossible to obtain in such an installation. Moreover, parallel compression could be employed by an additional third compressor of 23 kW. Thus, under these conditions, FGV and PC systems would require one additional compressor. Finally, the ME unit provides the only solution without the necessity of an additional third compressor. In this approach, the base load compressor is totally unloaded by the multi-ejector module. Hence, all of the refrigerant vapour from the evaporator could be delivered to higher pressures, applying the ejectors as a booster for the parallel compressors. The required power of the ME-unit would be at a level of 97 kW. The obtained reduced power consumption is based on the large amount of potential work recovery and the elevated pressure level at the parallel compressors' suction port and its high isentropic efficiency based on the low pressure ratio. Moreover, partially increased cooling capacity is ensured by liquid ejectors. As a final result, only two compressors of 58.3 kW would be enough to ensure operation of the R744 refrigeration unit in an East-Asian climate.

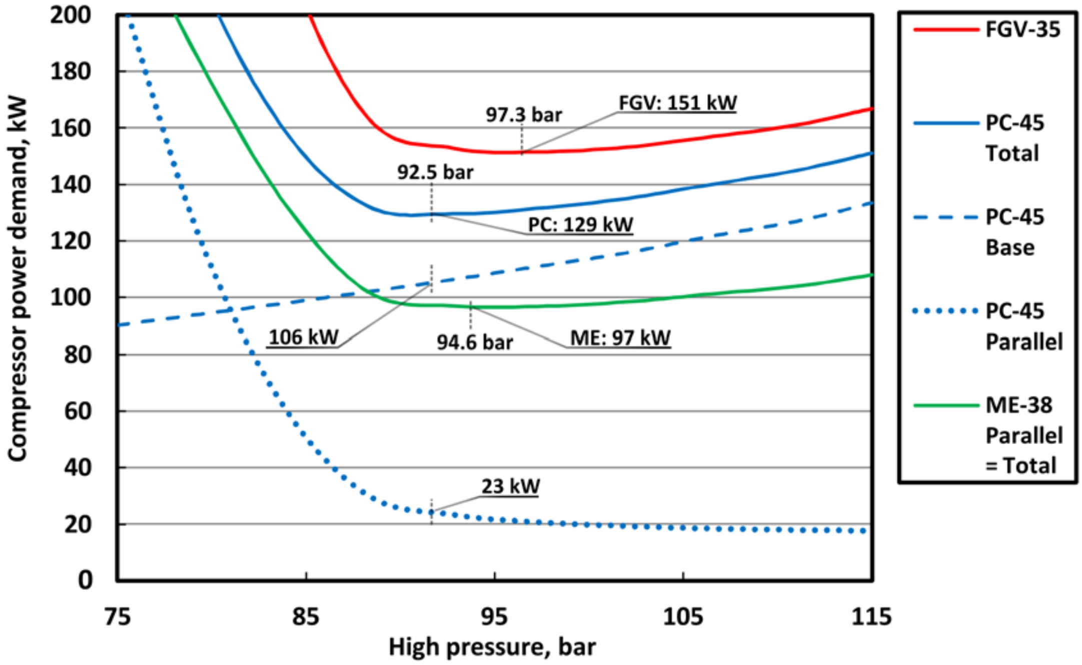

Figure 12. Compressor power demands of the modified systems characterised by the highest COP improvement for the East-Asian climate.

5.3. Ejector efficiency requirements in an East-Asian climate

In the case of the East-Asian climate, the system was modelled as utilizing parallel compressors only. Consequently, the entire evaporator stream was ingested by the vapour ejectors of a multi-ejector module. Due to that, the necessary efficiencies of the vapour ejector section to operate under the given conditions were calculated on the basis of the definition proposed by Elbel and Hrnjak [33]. In Fig. 13, the ejector efficiency obtained for each pressure in the IPR is presented. According to the COP data presented in Fig. 10, optimum pressure is marked to point to the necessary efficiency of the vapour ejector section. First, a significant difference between efficiencies is visible for higher pressures, including the range of optimum pressures. Having considered the data presented in the literature, efficiencies of vapour ejectors on the level of 35 % could be estimated as the highest reported [32]. Therefore, operation at a pressure level of 36 bar in the IPR tank could be stated as possible at this moment in vapour ejector development. Nevertheless, in the case of the highest IP, a significant improvement in ejector technology should be provided to reach the level of 52 % efficiency. The potential for significant improvements in efficiency up to a level of 45 % was described in previous work by the authors of this study [35]. In the case of the efficiency mentioned, the IP should be maintained on the level of 37.17 bar. On the other hand, as discussed, the COP difference between ME-36 and ME-38 is on the level of 5 %. Hence, the ME-36 performance remains at a high level comparable to that for ME-38.

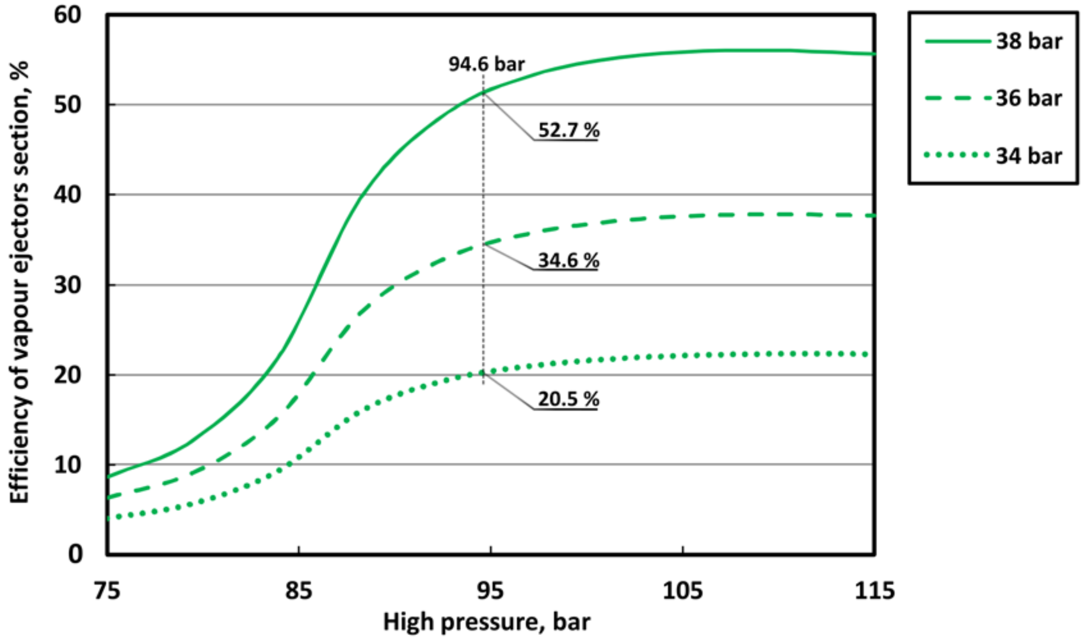

Figure 13. The required efficiency of the vapour ejector section in the multi-ejector module for each of the investigated pressures in the IPR.

5.4. Available cooling load in the function of installed compressor power

The assumed cooling load of 250 kW could be described as being representative for Scandinavian operation. Having regarded the various amounts and kinds of catches, different evaporator loads might be obtained. The necessary powers for the ME system as a function of evaporator load are presented in Fig. 14. The simulations were performed for optimum operation under East-Asian conditions. The maximum power for a single compressor was assumed on the basis of the manufacturer's data [37]. According to the limit of utilizing two compressors, evaporator loads ranging from 272 kW to 298 kW could be obtained depending on IP, which is directly related to ejector efficiency. This range is wider for higher capacities when differences between operations with higher or lower IPRs are more significant. On the other hand, three compressors could deliver a significantly higher evaporator load than two compressors. Namely, in the least efficient scenario of 34 bar, available cooling capacity would be on the level of approximately 412 kW. Hence, in an installation equipped with a third compressor, approximately 52 % of additional available evaporator load would be ensured. Moreover, in the case of the mentioned high efficiency multi-ejector module and 38 bar of IP, the increment of available evaporator load can even reach 67 %.

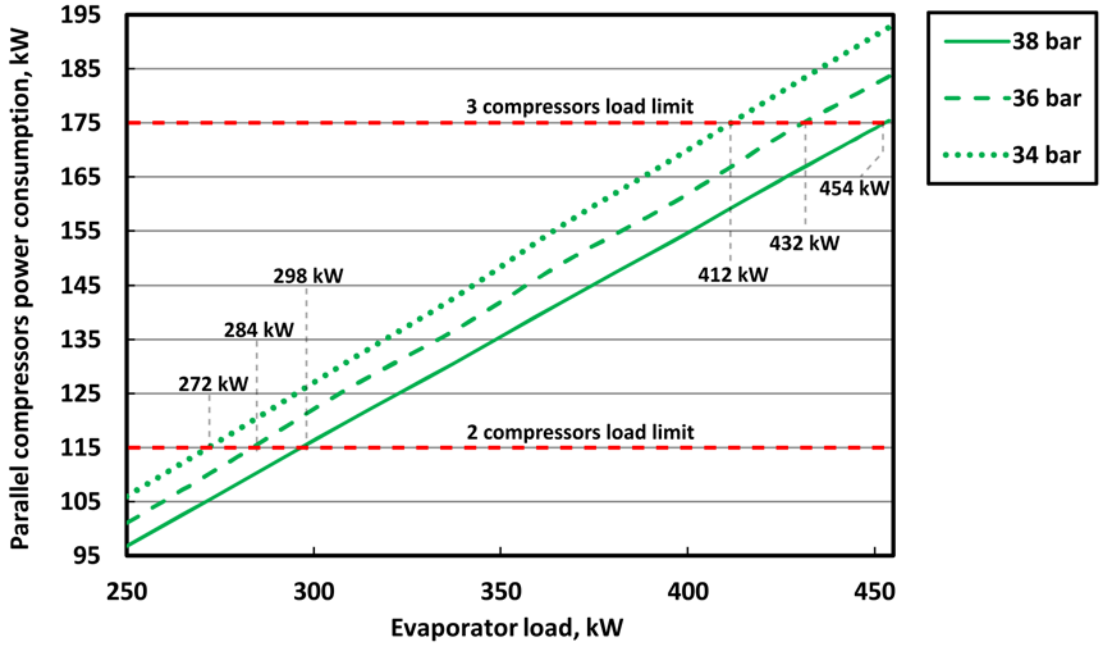

Figure 14. Necessary power for parallel compressors in the ME system operating at optimum pressure as a function of evaporator load.

6. Conclusions and further work

A mathematical model of the classic R744 refrigeration unit equipped with the liquid ejector was developed and utilised to evaluate performance of the installation operating in a fishing vessel in Scandinavian conditions.

On the basis of data obtained from NASA Earth Observatory online databases [41], Mediterranean and East-Asian sea waters were simulated for heat rejection temperatures of 21 °C and 33 °C, respectively. According to the thermodynamic characteristics of R744, system modifications were necessary to maintain high system performance. Nevertheless, spatial limitations of the fishing vessel machinery room required the additional consideration of introduced equipment related to the modified systems.

Simulations of East-Asian conditions resulted in the other trend of the COP distribution. First, significant differences were presented between each of the tested configurations. Again, the most prospective solution was ME, where the maximum obtained COP was 2.58. In the investigated configurations, the optimal high pressure was approximately 95 bar. Moreover, the range of high performance could be described as being relatively wide according to the high pressure value. That characteristic should provide advantages in regulation and automation areas. Substantial benefits could be found in the field due to the minimised requirements on equipment space.

East-Asian, which offered more demanding conditions, showed only one proper solution in the form of ME. In this case, as for the Mediterranean climate, a third compressor is not necessary. However, two compressors of higher power input should be delivered for parallel compression.

In the case of the analysed highest heat rejection temperatures, results in the form of necessary vapour ejector section efficiency were obtained. According to the operation with parallel compressors only, the entire stream from the evaporator was consumed by the vapour ejectors. According to literature data, ejectors of the demanded efficiency can be designed for two of the three investigated IP pressures - 34 bar and 36 bar. To obtain a higher system efficiency on the basis of an IP of 38 bar, improved vapour ejectors would be necessary. However, this efficiency increment could be described as relatively high for next generation ejectors. Finally, the results showed good prospects for higher cooling capacities according to the implementation of the third compressor.

According to the main goal of the paper, the analysis for the marine application with refrigeration system equipped in the liquid ejector was presented and more efficient solution for hot ambient conditions was proposed for such a system. Hence, the system analysis was performed having regard overall performance of the cycle. Nevertheless, the presented investigation was focused on the possibilities related with the multi-ejector system. A strong relation between the ejector technology and refrigeration system was discussed in Section 5.2 and especially in Section 5.3. Despite the global character of the analysis, the presented range of the promising operation for ejectors in a certain configuration and the specified efficiency seems to describe this solution as a very universal from the considered application point of view. Moreover, consequence of the ejector technology is illustrated as a substantial reduction of space requirements in the limited area of machinery room at the fishing vessel. Finally, the maintaining of the required cooling capacity without additional equipment as well as possible additional buffer related with only one additional compressor was obtained.

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