Exergetic performance analysis of high pressure air systems on ships

Asim Sinan Karakurt*, Ibrahim Ozsari and
Veysi Bashan

Department of Naval Architecture and Marine Engineering,
Yildiz Technical University,
Istanbul, Turkey
Email: asinan@yildiz.edu.tr
Email: ozsari@yildiz.edu.tr
Email: veysibashan@gmail.com
*Corresponding author

Abstract: Energy consumption, efficiencies, and eco-friendly issues are among the most researched topics. One of the high-energy consumption systems on ships is high-pressure air-start systems. This study performs energy and exergy analyses to optimise the use of high pressure air system with respect to performance outputs. The study calculates the exergy destructions resulting from the power consumed by the main compressors and the improper use of air stored at high pressures. Additionally, the ECOP is used to compare different models with respect to the First and Second Laws of Thermodynamics. The results from the analyses are as follows. The power consumptions are 1,530 kW, 1,283 kW, and 1,276 kW; the exergy destructions are 250 kW, 471 kW, and 450 kW; the ECOPs are 6.15, 2.72, and 2.84, respectively. The study presents the effects of sea water and ambient temperatures, and compressors' isentropic efficiencies on the power consumption, exergy destruction, and ECOP.

Keywords: ecological coefficient of performance; ECOP; air storage; exergy analysis; compressed air system.

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Biographical notes: Asim Sinan Karakurt is working as an Assistant Professor in the Naval Architecture and Marine Engineering Department at Yildiz Technical University, Istanbul, Turkey. He is currently working in the area of design, analysis, and optimisation of thermal systems.

Ibrahim Ozsari is a PhD student in the Naval Architecture and Marine Engineering Department at Yildiz Technical University, Istanbul. He is presently a research assistant at the Yildiz Technical University. His main research areas are energy, exergy, gas turbine, oxy-combustion, and thermodynamics.

Veysi Bashan is a PhD student in the Naval Architecture and Marine Engineering Department at Yildiz Technical University, Istanbul. He is presently a research assistant at the Yildiz Technical University. His main research areas are marine diesel engines, catalytic combustion, and thermodynamics.

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1 Introduction

Homo sapiens neanderthalensis, one of the first ancestors of humanity, used their lungs, which served as a natural low-pressure compressor (LPC), to light fire and later used the lungs for their blow darts while hunting or protecting themselves from wild animals. Since the transition to the metal age, the use of compressors alongside many technological developments has become widespread over the 5,000 years since the first primitive compressors, the blowers that had been developed against the need for greater air power than the human lungs (Elliott, 2006). Compressors, which in the past had been mostly used in the field of metallurgy, are now widely used in various fields, from transportation vehicles and refrigeration systems to industrial facilities and energy storage systems in various capacities and sizes. One of the places where compressors are frequently used is ships, where pressurised air, (i.e., compressors) is needed to operate many systems on ships such as heating, ventilation, and air conditioning (HVAC) systems; refrigeration systems; and the main engine compressed air system. This situation has given importance to designing compressors under optimum conditions and using them efficiently (i.e., to optimise both the initial investment costs and maintenance/operating costs). The importance of improvements in compressors can be better understood when considering that Europe consumes more than 80 TWh of energy and, correspondingly, more than 55 million tons of CO_2 for compressed air (Radgen, 2004; Radgen and Blaustein, 2000). In this context, many field studies have been done, analysing thermodynamics' first and second laws for compressors and compressed air systems according to different criteria and parameters. Mascarenhas et al. (2019) performed comprehensive energy, exergy, and sustainability analyses for an industrial air compression system with respect to different outputs such as the factors of depletion number, sustainability index, waste exergy ratio, and environmental effects that carried out comprehensive economic and ecological analyses alongside these in the study. Liu and Wang (2016) and Zhou et al. (2018) carried out studies on the comparative analysis of different compressed air energy storage systems (adiabatic and advanced adiabatic) using energy and exergy analyses with respect to different parameters. Venkataramani et al. (2016) studied the various aspects of a compressed air energy storage system they presented, including a thermodynamic analysis, a model, and a simulation analysis. Budt et al. (2016) studied various approaches for applying compressed air-energy storage systems applying the concept of exergy. They also aimed to reduce energy losses and increase the efficiency of compressed air energy storage systems while making their energy and exergy analyses. Szablowski et al. (2017) determined the exergy damages of all system components by simulating the compressed-air loading and unloading processes. Peng et al. (2016) proposed an energy storage system based on adiabatic compressed air energy storage (A-CAES), demonstrating a 5% efficiency increase to have been achieved using a heat recuperator A-CAES system, as heat energy remains in the packed bed until the discharge process is completed. Silva et al. (2017) aimed to reduce energy consumption by eliminating waste points in compressed air distribution using their determinations and improvements such as minimising leakage points, replacing line filters, and implementing satisfactory maintenance of pipe fittings (primarily cylinders, valves, and filters). They compared the before-and-after results in terms of energy and economy. Kaya et al. (2002) showed producers the energy losses associated with compressed air systems, their costs, and also methods to make savings. Similarly, Dindorf (2012) investigated the effects of specific power, annual energy costs, compressed air costs, compressed air leakages, and pressure drops in compressed air systems. Meanwhile, other studies have presented an overview of present compressor technology with a focus on preserving the directions for screw and sliding vane machines, considering each sub-term that contributes to the overall efficiency; analyses have shown thermodynamic improvement during compression, which is achievable by splitting compression into two stages at lower compression ratios, to open the way to significantly reducing energy consumption (Saidur et al., 2010; Vittorini and Cipollone, 2016). Mozayeni et al. (2019) developed a thermodynamic exergy model for analysing pumped-hydro and compressed air system performance for key parameters such as storage pressure, pre-set pressure, air-compression mode, and pump/hydro turbine efficiency. Guo et al. (2017) examined detailed exergy and sensitivity analyses of compressed air energy storage system performance with static and dynamic air reservoir models under different working parameters. Saputro and Farid (2018) studied a theoretical model that show the change of temperature and pressure in a high-pressure tank at the charging and discharging process in a compressed air energy storage system. Musharavati et al. (2021) assessed a comparative analysis of the energy and exergy analyses of a compressed air energy storage system integrated with a thermoelectric generator. Khanmohammadi et al. (2021) conducted a parametric analysis under different operating parameters to introduce a current compressed air energy storage system combined with a Rankine cycle, proton exchange membrane fuel cell, and thermoelectric unit.

Ust (2005) defined the ecological coefficient of performance (ECOP) criterion as the ratio of the desired output per total entropy production, which is used for different power cycles (Ust et al., 2005, 2006; Yeginer et al., 2013), jet propulsion systems (Sohret, 2018), refrigeration systems (Wouagfack and Tchinda, 2011, 2014), and heat pump systems (Chen et al., 2007). The criterion also can be effectively applied to ships' auxiliary machinery systems such as compressed air systems. Using the ECOP criterion helps in obtaining information on what kind of impacts the systems have on the environment while achieving their goals (power production/consumption, cooling load, or heating load).

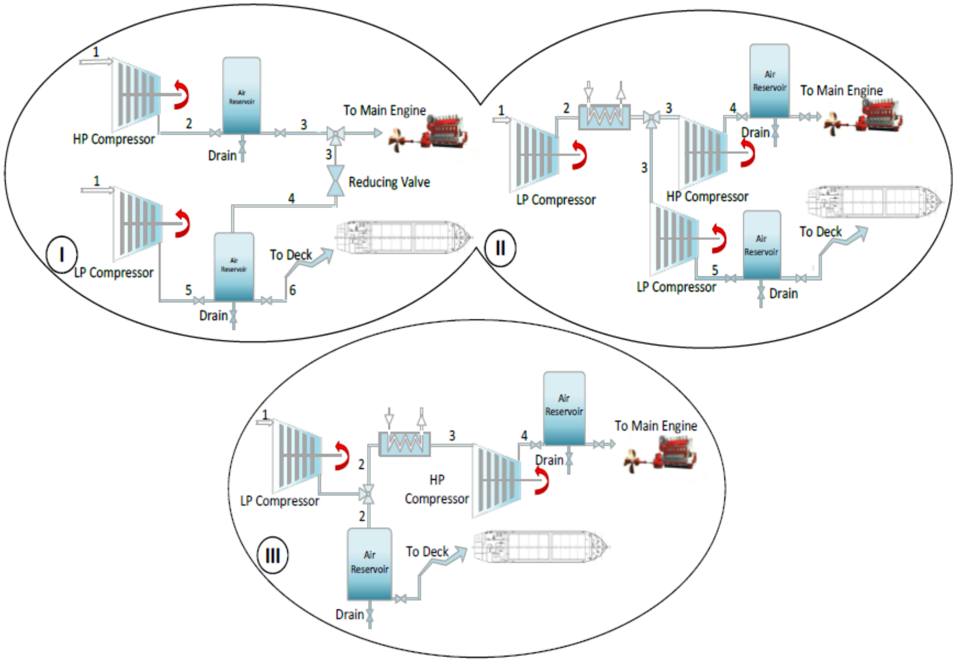
Energy efficiency is very important in ships as well as in terrestrial systems. One of the high power consuming auxiliary machines in ships is air compressors. Air compressors must reach a compression pressure of approximately 30 bar to start the main engine. Reducing the pressure of the air again with the seven bar pressure reducing valve, which is the service, system and safety air, causes huge destruction of exergy. The main purpose of this study is the modification of ship compressed air systems by considering both the energy and exergy efficiency. The use of valves to obtain lower pressures from a high-pressure source is a situation on ships that needs improvement, as it increases exergy destruction and reduces efficiency. In this context, the current study has developed from Karakurt et al.'s (2019) study and makes energy and exergy analyses by taking into account the effects various operating conditions (seawater temperature, compressor inlet temperature) have on ECOP, the total exergy destruction (exdTot), and the total power consumption (WTot) of high pressure air systems on ships.

2 The theoretical model

Ships have systems that require large volumes of compressed air. According to the international rules imposed by the International Maritime Organisation (IMO), a ship's compressed air tanks must be able to operate/start the main engine at least 12 times in a row. Ship's have two compressors that fill the air reservoir, one of which must operate independently of the main engine. In addition, this compressor must produce at least half of the total required capacity. The first air compressor on board should be capable of filling the air reservoirs within one hour. Therefore, these compressors are very powerful with high capacities. Both compressors fill the air reservoirs through a check valve. The filling circuit is usually provided with an oil/water separator. Filled with compressed air, the chambers are equipped with air valves to start the main engine and diesel generators. Ships also have an auxiliary air cylinder to run diesel generators that can be filled by an emergency compressor as needed. In addition to the main engine and diesel generator air valves, ships have multiple valves for needs such as control air, service air, and ship horn air that are separated by valves after the pressure-reducing valves. For high pressure compressed air, compressor outlets have high pressures and the compressor increases its power consumption.

A system flowsheet diagram is shown in Figure 1 and was modelled in the program engineering equation solver (EES) (Klein and Alvarda, 2007). In the compressor, compressed air up to 3,000 kPa is sent to the tube (Wankhede, 2019). The compressed air stored here is sent to the main engine as needed. The first system (I) under consideration consists of one LPC, one high-pressure compressor (HPC), and one independently operating pressure-reducing valve. This system takes the air from the surrounding environment (100 kPa) and compresses it up to 3,000 kPa (at 2 kg/s) in the HPC and 700 kPa (at 0.5 kg/h) in the LPC. The second system (II) under consideration has a LPC, HPC, and intermediate pressure compressor (IPC) in addition to an intercooler (cooling using seawater). The LPC compresses air up to a pressure ratio (547 kPa at 2.5 kg/s) corresponding to the optimum compression ratio, which is equal to the square root of the min and max pressure. After this, compression is carried out up to 700 kPa (at 0.5 kg/h) in the LPC and up to 3,000 kPa (at 2 kg/s) in the HPC. The third system (III) under consideration consists of two compressors and an intercooler. The air is compressed up to 700 kPa (at 2.5 kg/s) with the LPC located here; a 0.5 kg/s flow rate is sent to deck use, which is then compressed up to 3,000 kPa (at 2 kg/s) in the HPC for starting the main engine. The following assumptions have been made in the analyses: The isentropic efficiency for each turbine is 0.8 (Siddiqui and Dincer, 2018); the temperatures of the ambient environment, seawater, and compressor inlet environment are 298 K, 273 K, and 298 K, respectively; pressure losses at the heat exchanger or intercooler are 3% of the inlet pressure (Olabi et al., 2021).

Figure 1 Flowsheet diagram of the compressed air systems (see online version for colours)



The specific exergy and exergy destruction of the system components are found using equations (1) and (2).

$$ex_j = (h_j - h_0) - T_0 (s_j - s_0) \tag{1}$$

where h is the specific enthalpy, s is the specific entropy, T is the temperature, j is the state points of the related components, and 0 is the dead state conditions.

$$\begin{aligned} exd_j = &\left\{ \sum\left[q\left(1 - \frac{T_0}{T}\right)\right]_{in} - \sum\left[q\left(1 - \frac{T_0}{T}\right)\right]_{out} \right\} \\ &+ \sum\left[(w)_{in} - (w)_{out}\right] + \sum\left[(ex)_{in} - (ex)_{out}\right] \end{aligned} \tag{2}$$

where q, w and ex refer to specific heat transfer, specific work and specific exergy, respectively. On the right side of the equation, the exergy destruction describes that which occurs through heat, work, and exergy. Equation (2) can be specialised for a compressor in equation (3) with the assumption of heat transfer from/to the compressor being neglected.

$$exd_C = ex_{in,C} - ex_{out,C} + w_C \tag{3}$$

$$exd_{Tot} = \sum exd_C + \left(\sum exd_{HE} + \sum exd_{RV}\right) \tag{4}$$

where sub-indices Tot, C, HE and RV refer to the total, compressor, heat exchanger/intercooler, and reducing valve, respectively. The exergy destruction in kJ/kg units can be obtained using equation (4), and exergy destruction rates in kW units can be obtained by multiplying the related flow rates of components.

Power consumption in compressors and energy balances for the heat exchanger and reducing valve are created with inlet and outlet conditions of the components in equation (5). The ECOP is defined as the ratio of power consumption to total exergy destruction rate (i.e., loss rate of availability) in equation (6).

$$\dot{W}_{Tot} = \sum \dot{m}\left(\mathrm{h}_{out} - \mathrm{h}_{in}\right) \tag{5}$$

$$ECOP = \frac{\dot{W}_{Tot}}{e\dot{x}d_{Tot}} \tag{6}$$

3 Results and discussion

Thermophysical properties of system points (j) obtained from the above equations and the assumptions for the three different models (I, II, III) using the program EES are given in Table 1. Pressure (P), temperature (T), specific enthalpy (h), specific entropy (s), and specific exergy (ex) values can be found in Table 1.

Table 1 Thermophysical properties of system points

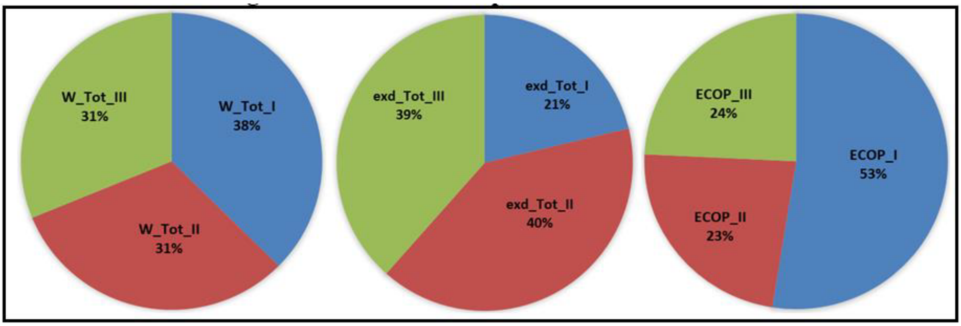
j	P_I	P_{II}	P_{III}	T_I	T_{II}	T_{III}	h_I	h_{II}	h_{III}
	[kPa]			*[K]*			*[kJ/kg]*		
1	100	100	100	298	298	298	298.3	298.3	298.3
2	3000	547.7	700	877.7	528.1	570.5	910.3	532.4	576.5
3	3000	531.3	679	877.7	373.8	377.6	910.3	422	438.2
4	700	3000	3000	570.5	741.7	712.8	576.5	760	728.5
5	700	700		570.5	463.3		576.5	465.4	

j	s_I	s_{II}	s_{III}	ex_I	ex_{II}	ex_{III}
	[kJ/(kg K)]			*[kJ/kg]*		
1	6.864	6.864	6.864	0	0	0
2	7.013	6.956	6.966	567.6	206.5	247.7
3	7.013	6.731	6.699	567.6	163.1	189
4	6.966	6.827	6.783	247.7	472.7	454.1
5	6.966	6.75		247.7	200.8	

The first and second laws analyses for all models were conducted at design flow rate conditions and off-design conditions were not into considerations. However, it is clearly observed that there is a linear relationship between flow rates and performance outputs. If the flow rates increase, the values of performance output also increase, vice versa.

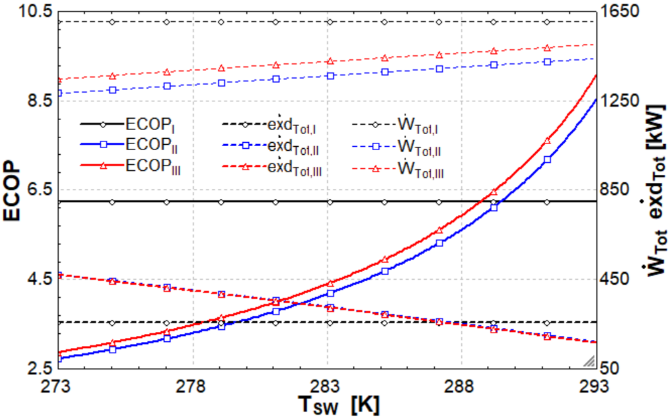
According to the assumptions in the thermodynamic model, performance outputs (consumed power, exergy destruction, and ECOP) from the first law and second law analyses are given in Figure 2. The effects of using an intercooler on the performance outputs can be seen in Figure 2, where models II and III have nearly the same ratios for all criteria While power consumption is 1,530 kW (38%) and ECOP is 6.151 (53%), where the values from Model I are higher than the others, the exergy destruction ratio is 248.7 kW (21%), which is lower because it lacks an intercooler (heat exchanger) and an additional compressor. Higher power consumption in the system/model means higher operating costs as well as a higher ECOP value, which means a higher power consumption per exergy destruction ratio.

Figure 2 Performance outputs of different models (see online version for colours)



The effects seawater temperatures (between 273 K and 293 K) have on the performance outputs are given in Figure 3 for constant isentropic efficiency and constant compressor inlet temperature. Because no cooling occurs with seawater in model I, the ECOP value remains constant; while the ECOP value increases exponentially with the increasing temperatures in models II and III (the rate of increase is higher for higher-temperature values). While the ECOP value is higher in Model I up to T = 290 K, it is higher in models II and III for $T > 290$ K. ECOP values at any temperature T up to this temperature value are highest for models I, III, and II in descending order and then models III, II, and I in descending order for $T > 290$ K.

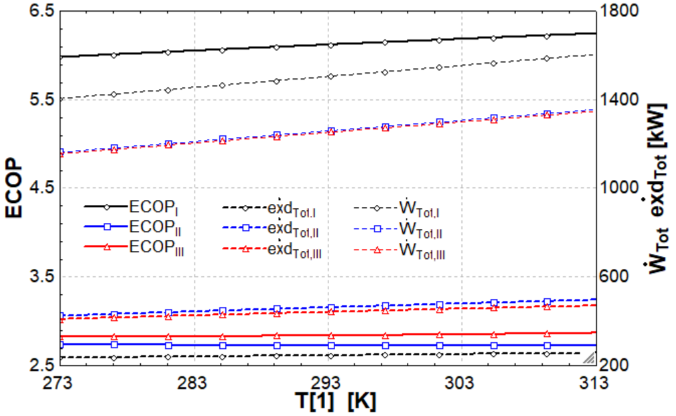
Figure 3 The effects of seawater temperature on the performance outputs (see online version for colours)



Exergy destruction and consumed power values remain constant with changing seawater temperatures in model I, while they increase linearly in models II and III. While the exergy destruction in model I is lower until the seawater temperature reaches 288 K, the values from models II and III are lower for saltwater temperatures above 288 K. This is because the increase in seawater temperature adversely affects the cooling in the intercooling unit. Additionally, the amount of power required in the HPC and LPC also increases linearly. According to the analysis, lower ECOP and higher exergy destruction values are obtained for different sea water temperatures, both outputs are seen to be suitable at temperatures around 288–289 K.

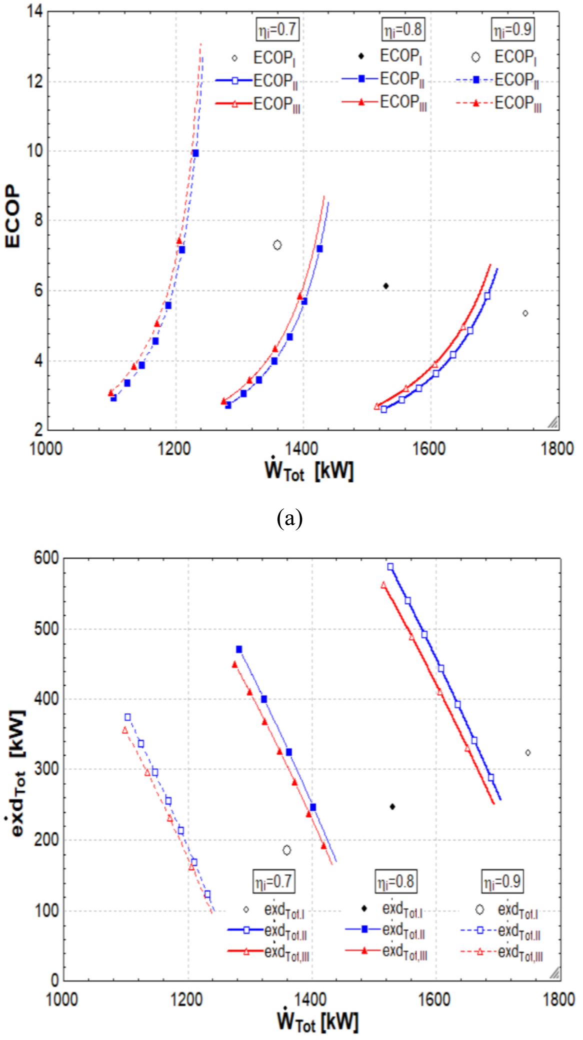
The effects of compressor inlet temperature (273 K > *T* > 313 K) on the performance outputs are given in Figure 4 for constant isentropic efficiency and constant seawater temperature. The ECOP values for each model increase linearly with increases in the compressor inlet temperature; the ECOP values corresponding to the same temperature value, from the most to the least, are respectively models I, III, and II. While the increase in ECOP value for model I is approximately 5.5%, the increase rates in models II and III are nearly 3.2% and 4.7%, respectively.

Figure 4 The effects of compressor inlet temperature on the performance outputs (see online version for colours)



The effects of isentropic efficiency on the performance outputs are given in Figures 5(a) and 5(b) for variable seawater temperatures and a constant compressor inlet temperature. The relationship between ECOP and exergy destruction values, (i.e., outputs of the 2nd law) with respect to power consumption for variable seawater temperature and variable isentropic efficiency values are given in Figures 5(a) and 5(b). With the increase in isentropic efficiency, both the exergy destruction and power consumption in model I decreased while the ECOP value increased (due to the change in seawater temperature not affecting model I, only one value was obtained). The ECOP value corresponding to the same seawater temperature decrease with increases in isentropic efficiency in models II and III while the exergy destruction and power consumption values decrease. In addition, an exponential relationship exists between ECOP and power consumption while a linear relationship exists between exergy destruction and power consumption in models II and III.

Figure 5 (a) and (b) The effects of isentropic efficiency on the performance outputs for variable seawater temperature (see online version for colours)

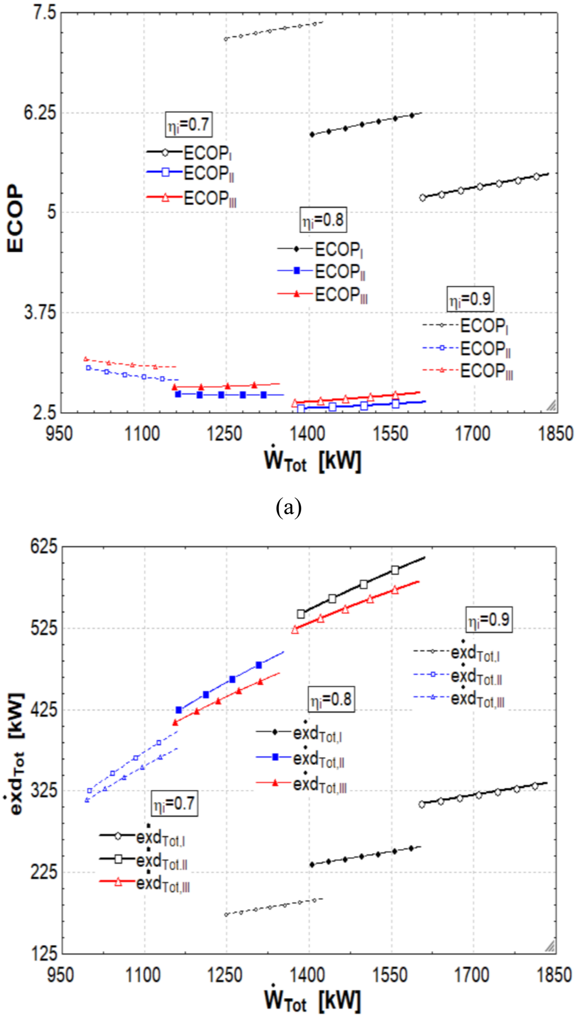


(a)

(b)

The effects of isentropic efficiency on the performance outputs are given in Figures 6(a) and 6(b) for variable compressor inlet temperatures and constant seawater temperature. In all three models, as the isentropic efficiency increases, the exergy destruction and power consumption values corresponding to the same compressor inlet temperature value decrease, while the ECOP value also increases. In addition, these figures show a linear relationship to exist between exergy destruction and power consumption, as well as between ECOP and power consumption in all models.

Figure 6 (a) and (b) The effects of isentropic efficiency on the performance outputs for variable compressor inlet temperature (see online version for colours)



(a)

(b)

4 Conclusions

This study has conducted energetic and exergetic performance analyses of a ship's high-pressure air system under various conditions, analysing the effects seawater temperature, compressor inlet temperature, and isentropic efficiency have on the criteria of power consumption, exergy destruction, and ECOP criteria. The analyses investigated both the relationships between the variable parameters and performance criteria as well as among performance criteria. These analyses show that using an intercooler reduces total power consumption and the ecological performance (ECOP) value but also increases total exergy destruction of the system. Within the scope of the 1st and 2nd law analysis, the effects the intercooler and additional compressor(s) used in models II and III have on the total cost (initial, operation, and maintenance costs) of the system will be the subject of future studies. In this way, a multi-objective optimisation will be realised by determining which system will be economically more suitable.

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Nomenclature

ECOP	Ecological coefficient of performance
(ex) Ex	(Specific) Exergy, kJ(/kg)
Exd	Exergy destruction, kJ/kg
h	Specific enthalpy, kJ/kg
HPC	High pressure compressor
HVAC	Heating, ventilation, and air conditioning
IPC	Intermediate pressure compressor
LPC	Low pressure compressor
$\dot{m}$	Flow rate (kg/s)
P	Pressure, kPa
q	Heat transfer, kJ/kg
s	Specific entropy, kJ/(kg K)
T	Temperature, K
w	Specific work, kJ/kg
Subscripts	
0	Dead state conditions
I, II, III	Different models
C	Compressor
HE	Heat exchanger
i	Isentropic
in	Inlet
j	State point
out	Outlet
RV	Reducing valve
SW	Seawater
Tot	Total
Greek letters	
η	Efficiency