Energy and exergy analysis of a cruise ship

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Abstract:

The shipping sector is today facing numerous challenges. Fuel prices are expected to increase in the medium-long term, and a sharp turn in environmental regulations will require several companies to switch to more expensive distillate fuels. In this context, passenger ships represent a small but increasing share of the industry. The complexity of the energy system of a ship where the energy required by propulsion is no longer the trivial main contributor to the whole energy use thus makes this kind of ship of particular interest for the analysis of how energy is converted from its original form to its final use on board.
To illustrate this, we performed an analysis of the energy and exergy flow rates of a cruise ship sailing in the Baltic Sea based on a combination of available measurements from ship operations and of mechanistic knowledge of the system. The energy analysis allows identifying propulsion as the main energy user (41% of the total) followed by heat (34%) and electric power (25%) generation; the exergy analysis allowed instead identifying the main inefficiencies of the system: exergy is primarily destroyed in all processes involving combustion (88% of the exergy destruction is generated in the Diesel engines and in the oil-fired boilers) and in the sea water cooler (5.4%); the main exergy losses happen instead in the exhaust gas, mostly from the main engines (67% of total losses) and particularly from those not equipped with heat recovery devices.
The improved understanding which derives from the results of the energy and exergy analysis can be used as a guidance to identify where improvements of the systems should be directed.

Keywords:

Energy analysis; exergy analysis; low carbon shipping

1. Introduction

1.1 Background

According to the third IMO GHG Study 2013, in 2012 shipping contributed to global anthropogenic CO_2 emissions with a total of 949 million tonnes, which represents roughly the 2.7% of the total [1]. Although such contribution appears relatively low, the trend is that shipping will play an even greater role in the CO_2 emissions in a near future due to the increased transport demand according to all IMO future scenarios. As an example, global transport demand has increased by 3.8 % in 2013, compared to a global GDP growth of 2.3 % the same year, which shows how shipping tends to rise even faster than global economy [2].

International Energy Agency data from marine bunker show that the OECD countries in fact have reduced the CO_2 impact from shipping, but a larger amount has been moved to the non-OECD countries [3]. The fact that shipping needs to even further reduce its CO_2 emissions in the near future is essential for being able to achieve the goals of maintaining global average temperature increase below 2^{o}C by 2050 [4]. Finally, in the Baltic Sea an emission control area is enforced by the International Maritime Organisation since January 2015 which stipulates that the fuel used must not contain more than 0.1 % sulphur, therefore requiring the use of more expensive distillate fuels.

Altogether, these conditions present a challenge to shipping companies, which are attempting to reduce their fuel use in an attempt to reduce both environmental impact and operative costs. A wide range of fuel saving solutions for shipping are available and partially implemented in the existing fleet, both from the design and operational perspective; several specific studies have been conducted on these technologies, and a more detailed treatise would be out of the scope of this work. In this context, it has been acknowledged that the world fleet is heterogeneous, and measures need to be evaluated on a ship-to-ship basis [5,6]. In this process, a deeper understanding of energy use on board of the specific ship is vital.

1.2 Previous work

A number of studies concerning ship energy systems can be found in literature. Thomas et al. [7] and Basurko et al. [8] worked on energy auditing fishing vessels; Shi et al. [9] proposed models for predicting ship fuel consumption for some specific vessel types; Balaji and Yaakob [10] analysed ship heat availability for use in ballast water treatment technologies. These studies have been of particular interest in their relative fields, but a more comprehensive approach of the totality of the ship energy system is missing. In addition, an analysis purely based on the First law of thermodynamics does not account for the irreversibilities of the systems and for the different quality of heat flows [11]. Exergy analysis, which is based on both the First and the Second laws of thermodynamics, can help addressing this shortcoming. Widely used in other industrial sectors, exergy analysis in not commonly employed in maritime technology studies, and is mostly related to waste heat recovery systems [12,13] and refrigeration plants [14,15]. The application of exergy analysis in shipping is still limited; Zaili and Zhaofeng [16] proposed the energy and exergy analysis of the propulsion system of an existing vessel showing that there is potential in improving ship power plant efficiency by recovering the exergy in the exhaust gas and by improving operations of the main engines.

In a previous study of the energy and exergy analysis of a product tanker [17] the dominance of propulsion as main energy user on board was highlighted, together with the substantial availability of waste heat for recover. On cruise vessels, the number of different uses of energy is larger and a complex system of different energy carriers (chemical, thermal, electrical or mechanical) is present in order to fulfil the needs for transport combined with passenger services and comfort, such as cooking and cooling in restaurants, air conditioning, and passenger entertainment facilities.

The complexity of the energy system of a ship where the energy required by propulsion is no longer the main contributor to the whole energy use thus makes this kind of system of particular interest for the analysis of how energy is transformed used on board. The complexity of such systems was modelled and investigated previously by Marty et al. [18,19], but to the best of our knowledge there is no study in literature describing cruise ships' energy and exergy analyses based on actual measurements.

1.3. Aim

The aim of this paper is to provide a better understanding of how energy and exergy are used on board of a cruise ship and where the largest potential for improvement is located by applying energy and exergy analysis to the a case study. The combination of a method rarely applied in the shipping sector to a ship type featuring a complex energy system is considered as the main contribution of this work to the existing literature in the field.

2. Methodology

This paper proposes the application of energy and exergy analysis (further described in Section 2.1) as a mean for improving understanding of energy conversion on board of a cruise ship. This application is shown for a specific case study vessel (see Section 2.2) for which extensive measurements from on board logging systems were available (see Section 2.3 for details on data gathering and processing). The results from the energy and exergy analyses are then discussed in order to propose possible improvements for ship operations and design.

2.1 Energy and exergy analysis

Energy and exergy analysis were performed on the system of study, taking the ship energy system, as presented above, as control volume. The energy flow rates are calculated by assuming that the chemical energy of the fuel flows is equal to its lower heating value, and the physical energy is taken as its relative enthalpy. The reference state, both for energy and exergy analysis, is assumed to be at atmospheric pressure and at the measured sea water temperature. Such an analysis allows tracking all the energy streams flowing through the ship and depicts the main heat and power users.

Energy may be transformed from one form to another, but it can neither be created nor destroyed; this results in the fact that a conventional energy analysis provides limited information on the system inefficiencies. *Exergy* is defined as the `*maximum theoretical useful work as the system is brought into complete thermodynamic equilibrium with the thermodynamic environment while the system interacts with it only*'. At the difference of energy, exergy is *not* conserved in real processes, and the exergy destroyed, or irreversibility rate, quantifies the system irreversibilities. The general exergy balance can be written as:

$$\dot{I} = \sum \dot{B}_{in} - \sum \dot{B}_{out} \,, \qquad (1)$$

where:

- $\dot{I}$ denotes the irreversibility rate, also called exergy destruction, which can also be calculated from the Gouy-Stodola theorem;
- $\dot{B}_{in}$ is the exergy flow rate entering the component/system under investigation;
- $\dot{B}_{out}$ represents the corresponding exergy outflow. This term is normally further subdivided in two parts: products($\dot{EX}_{prod}$) and losses ($\dot{EX}_{loss}$)

The exergy of a material flow is divisible into its physical, chemical, kinetic and potential components, in the absence of nuclear and magnetic interactions. The physical exergy represents the maximum amount of work obtainable from bringing the material stream from its initial state to the environmental state, defined by p_0 and T_0, taken here as the ambient pressure and the seawater temperature. The chemical exergy represents the maximum amount of work obtainable as the stream under consideration is brought to the dead state, by chemical reaction and transfer processes. The fuel chemical exergy is assumed equal to its higher heating value, which is derived based on the fuel H/C ratio according to the equation proposed by Szargut et al. [20]. The potential and kinetic exergies are neglected. The exergy transferred with power has the same value as its energy, while the exergy transferred with heat is lower and its value depends on the temperature at which heat transfer takes place. For more details, the reader is referred to the reference books of Szargut et al. [20], Kotas [21] and Moran [22].

The system performance is measured using several performance indicators:

- the *exergy efficiency (ε_t)*, defined as the ratio between the product $\dot{B}_p$ and the input $\dot{B}_{in}$ exergies:

$$\varepsilon_t = \frac{\dot{B}_p}{\dot{B}_{in}} \qquad (2)$$

  where the product exergy represents the desired output of the component or system, and the input exergy denotes the resources required to drive this process.

- the *exergy loss ratio (λ)*, proposed in [23], which illustrates how much of the exergy input to the system is actually destroyed through irreversibilities:

$$\lambda = \frac{\dot{I}}{\dot{B}_{in}} \qquad (3)$$

- the *irreversibility share* (δ), proposed in the works of Kotas [21] and Tsatsaronis [24], which is defined as the ratio between the exergy destroyed in the *i*-th component $\dot{I}_i$ in relation to the total system irreversibilities $\dot{I}_{tot}$.

$$\delta_i = \frac{\dot{I}_i}{\dot{I}_{tot}} \tag{4}$$

2.2 Case study vessel

The ship under study is a cruise ship operating on a daily basis in the Baltic Sea between Stockholm (Sweden) and Mariehamn (the Åland islands). The ship is 176.9 m long and has a beam of 28.6 m, and has a design speed of 21 knots. The ship was built in Aker Finnyards, Raumo Finland in 2004. The ship has a capacity of 1800 passengers and has several restaurants, night clubs and bars, as well as saunas and pools. This means that the energy system regarding the heat and electricity demand is more complex than a regular cargo vessel in the same size. Typical ship operations, although they can vary slightly between different days, are represented in Figure 1. It should be noted that the ship stops and drifts in open sea during night hours before mooring at its destination in the morning, if allowed by weather conditions.

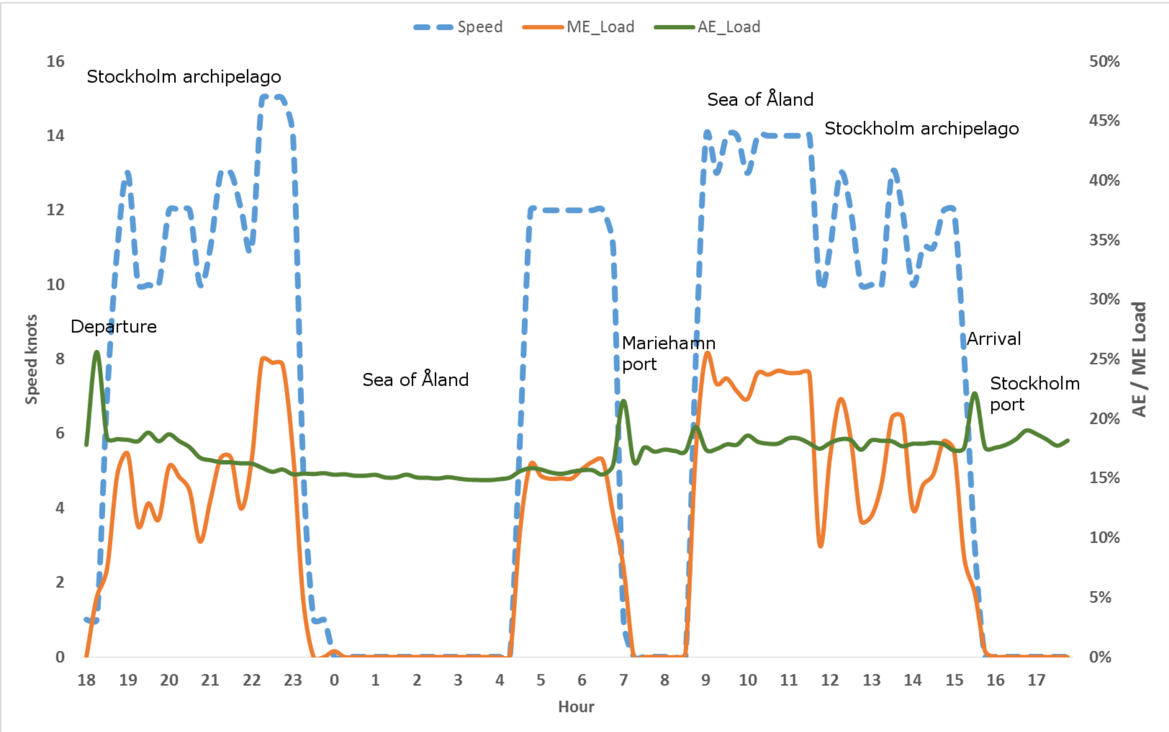


Fig. 1. Typical operational profile (ship speed, main engines load and auxiliary engines load) for the selected ship.

The ship systems are summarized in Figure 2. The propulsion system is composed of two equal propulsion lines, each made of two engines, a gearbox, and a propeller. The main engines are four Wärtsilä 4-stroke Diesel engines (ME) rated 5850 kW each. All engines are equipped with selective catalytic reactors (SCR) for NO_X emissions abatement. Propulsion power is needed whenever the ship is sailing; however, it should be noted that the ship rarely sails at full speed, and most of the time it only needs one or two engines operated simultaneously. When only one engine is used, power can be delivered to only one propeller. This requires the use of a significant rudder angle for keeping a straight course, which in turn increases ship resistance and, consequently, the amount of power to be delivered to the propeller.

Auxiliary power is provided by four auxiliary engines (AE) rated 2760 kW each. Auxiliary power is needed on board for a number of alternative functions, from pumps in the engine room to lights, restaurants, ventilation and entertainment for the passengers.

Auxiliary heat needs are fulfilled by the heat recovery steam generators (HRSG) located on the exhaust pipes of all four AEs and on two of the four MEs, and by the heat recovery on the HT cooling water systems (HRHT). When the available waste heat from the engines is not sufficient for fulfilling total heat demand, two oil-fired auxiliary boilers (AB) can be used to provide the remaining amount of heat. This situation mainly occurs when the ship is berthed in port, or during winter. The heat is needed for passenger and crew accommodation, as well as for the heating of the highly viscous heavy fuel oil used for engines and boilers. This last part, however, is drastically reduced since the 1st of January 2015, as new regulations entering into force require the use of low-sulphur fuels, which require a much more limited heating.

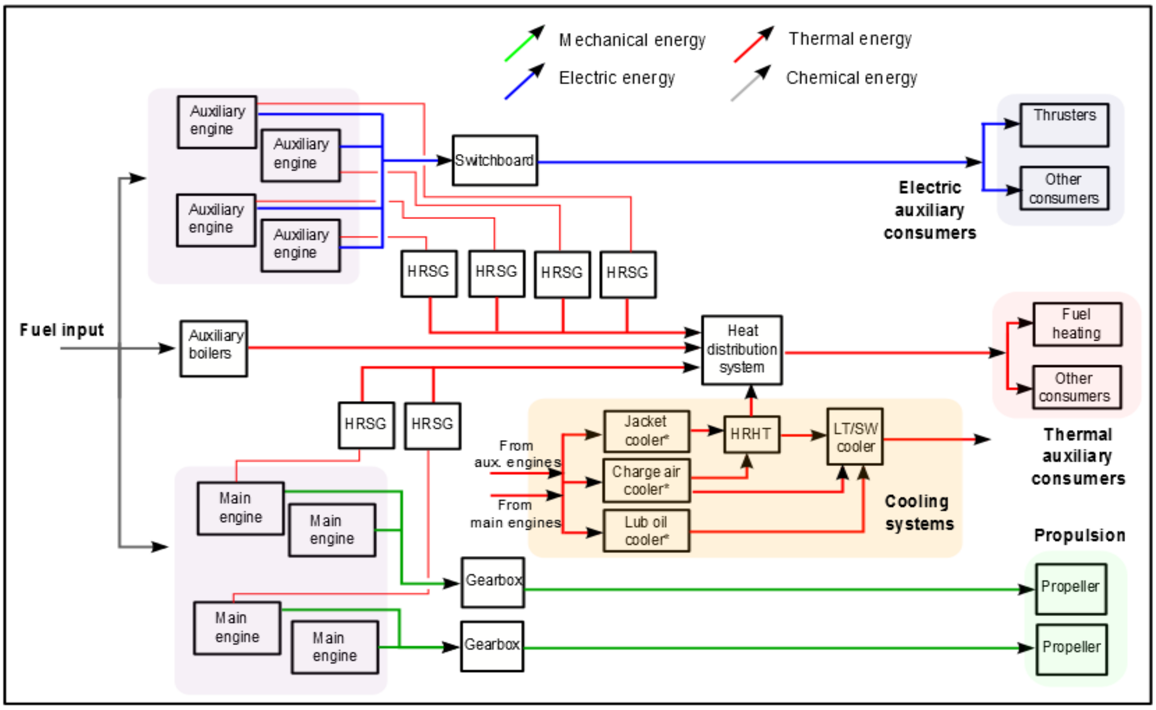


Fig. 2. Schematic representation of ship energy systems

2.3 Data gathering and processing

The ship under study is equipped with an extensive system for measuring and logging of operational variables, which logs the data with a 60 second interval. For this study an averaged 15 minute interval was chosen in order to cover a total of approximately one year of ship operations under the constraints related to the maximum number of data points in the database export tool.

A detailed accounting of all relationships and assumptions employed in this study in order to process the raw measured data are shown in Table A1 in Appendix A. Hereafter only the most relevant assumptions are discussed.

Main engines power and fuel mass flow ($\dot{m}_{fuel,ME}$) were not directly measured. In this study it was assumed that measures of the normalized fuel rack position (frp_{norm}) can be used as a predictor for the amount of fuel injected per cycle. The fuel flow to the main engines is consequently calculated according to the following equation:

$$\dot{m}_{fuel,ME} = \dot{m}_{fuel,ME,des} \left(a_0 + a_1 \, frp_{norm} \right) \left(\frac{n_{ME}}{n_{ME,des}} \right), \quad (5)$$

where the subscript *des* refers to design conditions of the engine at 100% of the maximum continuous rating (MCR) and n_{ME} represents the main engine speed. The regression coefficients a_0 and a_1 where determined based on the engine shop trial tests documents. Estimations calculated using (5) were validated against fuel flow measurements obtained from a recently installed mass flow meter. MEs delivered power ($\dot{W}_{ME}$) was then calculated according to (6):

$$\dot{W}_{ME} = 3.6^6 \frac{\dot{m}_{fuel,ME}}{bsfc_{ME}} \quad (6)$$

where the break specific fuel consumption (*bsfc*) of the MEs was calculated using a 2nd degree polynomial function calibrated on shop trial data. This method for the estimation of the required propulsion power involves certain accuracies, mostly related to engine bsfc (a margin of 5% is generally considered in related ISO standards) and to the relationship between fuel mass flow, engine speed and fuel rack position. However, given that no direct measurement of propeller power was available, this method is believed to be far more accurate than numerical estimations of the required propulsive power based on ship particulars and vessel speed.

The AEs power was available from on board measurements. The fuel mass flow rate to the AEs was calculated solving (6) for the fuel mass flow rate, where AEs' bsfc was estimated according to the same principle as described for the MEs.

The heat demand was not directly measured, and therefore needed to be estimated. As previously mentioned, on board heat demand is fulfilled by three different systems: the HRSGs, the HRHT and the ABs. The heat recovered in the HRSGs was estimated based on calculated engine exhaust flow and measured temperatures before and after the HRSGs; no information was available regarding the heat recovered in the HRHT; finally, the AB daily fuel consumption was available from a second logging system. In order to provide a reasonable assumption for the contribution of each of the above mentioned systems to the total amount of heat generated on board, it was assumed that heat demand is constant during each day, as suggested by Marty [19] . In addition, discussions with the crew allowed making the assumption that the ABs are only used when the main engines are not running. Based on these considerations, the following approximation was employed in this study:

$$\dot{Q}_{tot}(t) = \frac{m_{fuel,AB}}{\Delta t_{port}} LHV_{fuel}\eta_{AB} + \frac{\sum_i \dot{Q}_{HRSG,port}(t_i)\Delta t_i}{\Delta t_{port}} + 500\,[\mathrm{kW}], \quad (7)$$

where the first term represents the contribution from the oil fired boilers ($m_{fuel,AB}$, Δt_{port}, and η_{AB} represent the mass of fuel consumed by the ABs during the port stay, the time spent by the ship in port and the energy efficiency of the ABs respectively), the second term that of the HRSGs operated in port ($\dot{Q}_{HRSG,port}$ and Δt_i represent the heat flow recovered during port stays and the i-th time interval in which the port stay is divided respectively), and the third that of the HT cooling systems based on design calculations provided by the shipyard. When the ship is sailing/manoeuvring the ABs are turned off, and the required heat on board is generated by the HRSGs and the HRHT:

$$\dot{Q}_{rec,HT}(t) = \dot{Q}_{tot}(t) - \dot{Q}_{HRSG}(t) \quad (8)$$

where the total heat demand $\dot{Q}_{tot}(t)$ was calculated using (7) and the heat flow available from the HRSG $\dot{Q}_{HRSG}(t)$ is calculated based on available measurements of exhaust temperature and on the calculated exhaust gas flow rate from the engines.

3. Results

3.1 Operational profile

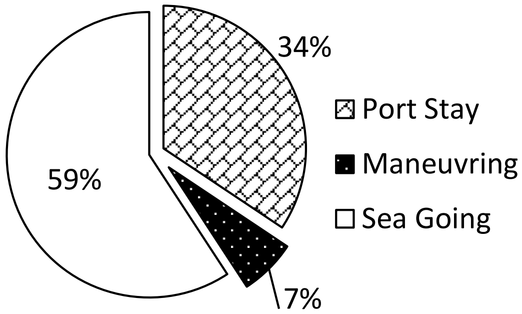
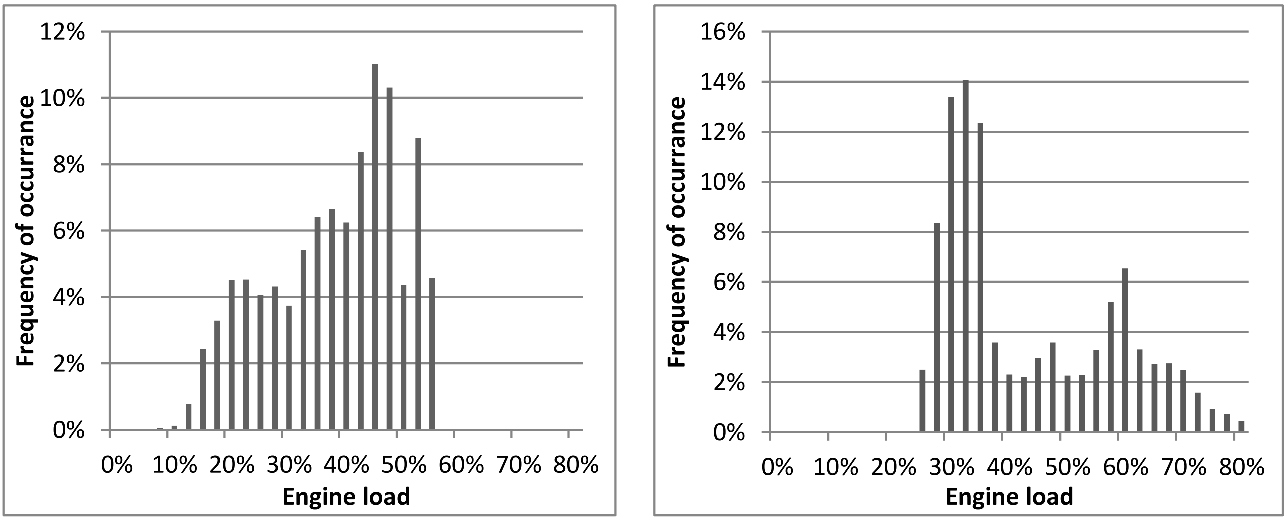


Fig. 3. Time spent by the ship in different operational modes



(a) **(b)**

Fig. 4. Load distribution for a) main engines and b) auxiliary engines

As shown in Figure 3 the ship spends most of the time sailing, while a significant amount of time is also spent in port. This is not surprising, in relation with the typical operations of this type of ship where loading and unloading of passengers is an operation that require a significant amount of time.

Figure 4 shows the load distribution for the main engines (a) and auxiliary engines (b), respectively. As it is observable from the figure, the main engines are most often operated at very low load, which leads to sub-optimal conditions in terms of efficiency and wear. This is a result of two concurring factors, as discussions with the crew revealed:

- The ship is operated most of the time at a speed which is much lower than the design value. This leads consequently to a strong reduction in propulsion power demand
- The engines are divided in two groups, each driving one propeller. This means that, even at very low load, it is not possible to operate on only one engine at medium-high load.

3.2 Energy analysis

Figure 4 shows the Sankey diagram for ship prime movers, converters, and users, where values are presented numerically in Table A2 in Appendix A. From the users' side, it can be seen that the energy demand for auxiliary power is comparable in size to that for propulsion (see Table 1). This is situation is expectable in the case of cruise ships / passenger ferries, but not common in other ship types. Thrusters, although they represent a high punctual demand, do not significantly contribute to the overall energy use. Auxiliary heat demand is also particularly large, but is mostly fulfilled by heat recovery boilers. It should be noted, however, that the yearly fuel consumption from the auxiliary boilers in the case under study is significant (5.2 %).

The contribution from the HT cooling water systems is also significant, comparable to that of the boilers and the HRSGs. Although this value was not directly measured and is therefore subject to a larger uncertainty, this observation suggests that heat integration has been carefully and successfully taken into account in the design of this particular vessel.

The energy analysis also shows that a large amount of heat is rejected to the environment, mainly with the exhaust gases exiting the heat recovery steam generators installed after the main engines, and heat from the low-temperature and seawater cooling systems. The amount of energy dispersed to the environment is in the same order of magnitude as the heat demand of the whole ship energy system. This situation suggests that additional heat could be harvested for other uses on-board, e.g. for use in heat recovery and heat-to-power systems, which would result in a smaller fuel consumption of the boilers or/and of the engines. This aspect will be however further investigated using the exergy analysis, which gives a better picture of energy quality and a better estimation of the amount of energy that could be actually recovered and converted into electricity. Large amount of energy is also dispersed via the LO cooling. This waste heat flow is by no means recovered on ship systems, differently from the heat to the HT systems.

Figure 5 shows the repartition of the energy use among different users and for the different operational modes. Propulsion represents the main part of energy use, but is only present when at sea or manoeuvring. Electric energy demand is instead rather constant over time and therefore it scales proportionally to the time in each phase. It should be noted that, both in Figure 1 and in Tables 2 and 3 the category "port stays" also includes the time spent by the ship drifting at sea.

The absence of any dedicated measurement made it impossible to identify the individual users; however, the use of bow thrusters during manoeuvring constitute a clear spike in the total auxiliary power demand and are therefore possible to separate from the total.

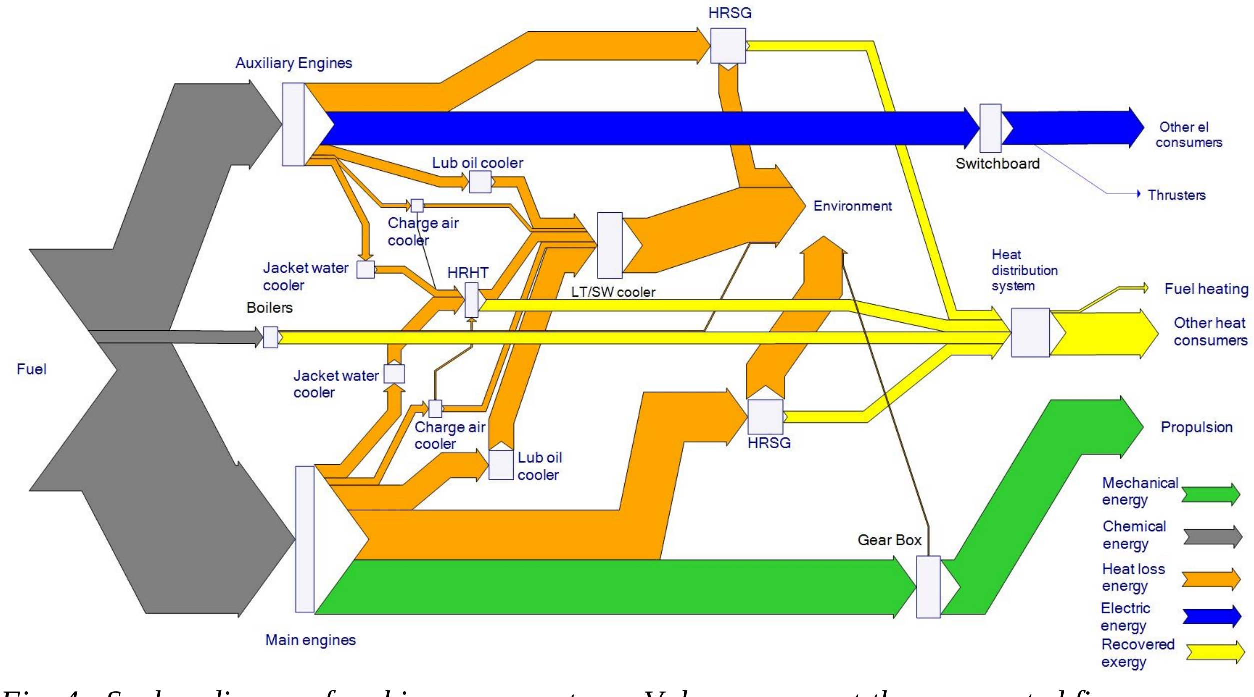


Fig. 4. Sankey diagram for ship energy systems. Values represent the aggregated figures over one year of operations. All main engines and auxiliary engines are grouped together.

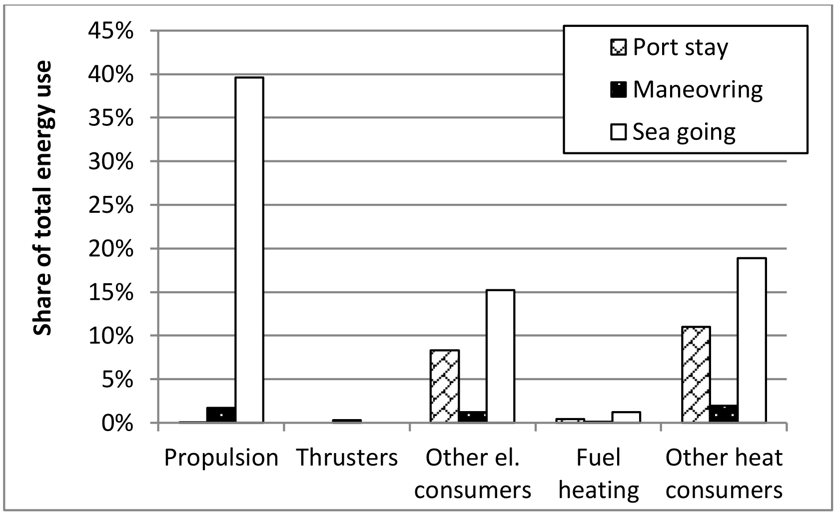


Fig. 5. Yearly energy demand for different users, separated per operational mode.

Table 1: Yearly shares for the different energy users on board, divided by operational mode

User	Port stay	Manoeuvring	Sea going	**All modes**
Propulsion	0,0%	1,7%	39,6%	**41,3%**
Thrusters	0,0%	0,3%	0,0%	**0,3%**
Other electric power users	8,3%	1,2%	15,3%	**24,8%**
Fuel heating	0,4%	0,1%	1,2%	**1,7%**
Other heat users	11,0%	2,0%	18,9%	**31,9%**
All users	**19,7%**	**5,3%**	**75,0%**	**100,0%**

3.3 Exergy analysis

The observation of the Grassmann diagram (Figure 6) allows for identifying, locating and quantifying the main sources of exergy losses and destruction. Numerical results are presented in Tables 2 and 3 in the text and Table 2A in the appendix. Most exergy destruction takes place in the main and auxiliary engines (respectively 53.1% and 27.8% of the total) followed by the boilers (6.9%), where the high rate of exergy destruction is strongly connected to the process of conversion of chemical to thermal energy, as well as to mixing and friction phenomena and heat transfer. A significant part of the exergy destruction takes place in the cooling systems (9.1%), which highlights potential for improvement in the design of the heat exchanger network. Finally the exergy flow rate lost to the environment in the exhaust gas after the HRSGs (8.0% of the total exergy input, 20.3% if compared to the total exergy output of the system) also represents a significant potential for improvement. It should be noted, as shown in Table 3, that most exergy losses take place during the seagoing phase, when on board heat demand is fulfilled through the use of waste heat available on board. The recoverable exergy theoretically available during port stays constitutes however more than half (52.5%) of the total exergy flow rate produced by the boilers.

The analysis of exergy performance indicators further highlights a number of observations about the system. Main engines show a lower efficiency compared to the auxiliary engines (34.7% compared to 38.3%) despite their higher efficiency at design conditions (46.8% compared to 44.2%), which provide further evidence to the fact that the main engines are often operated in non-optimal conditions. The high irreversibility ratio of the boilers (70.8%) suggests that reducing their use should be a priority in view of the exergy optimization of the system. Among the heat exchangers, the highest potential for improvements appears to lie in the lubricating oil coolers and in the jacket water coolers, where the high temperature difference between hot and cold flows suggests that the heat exchange process could be improved.

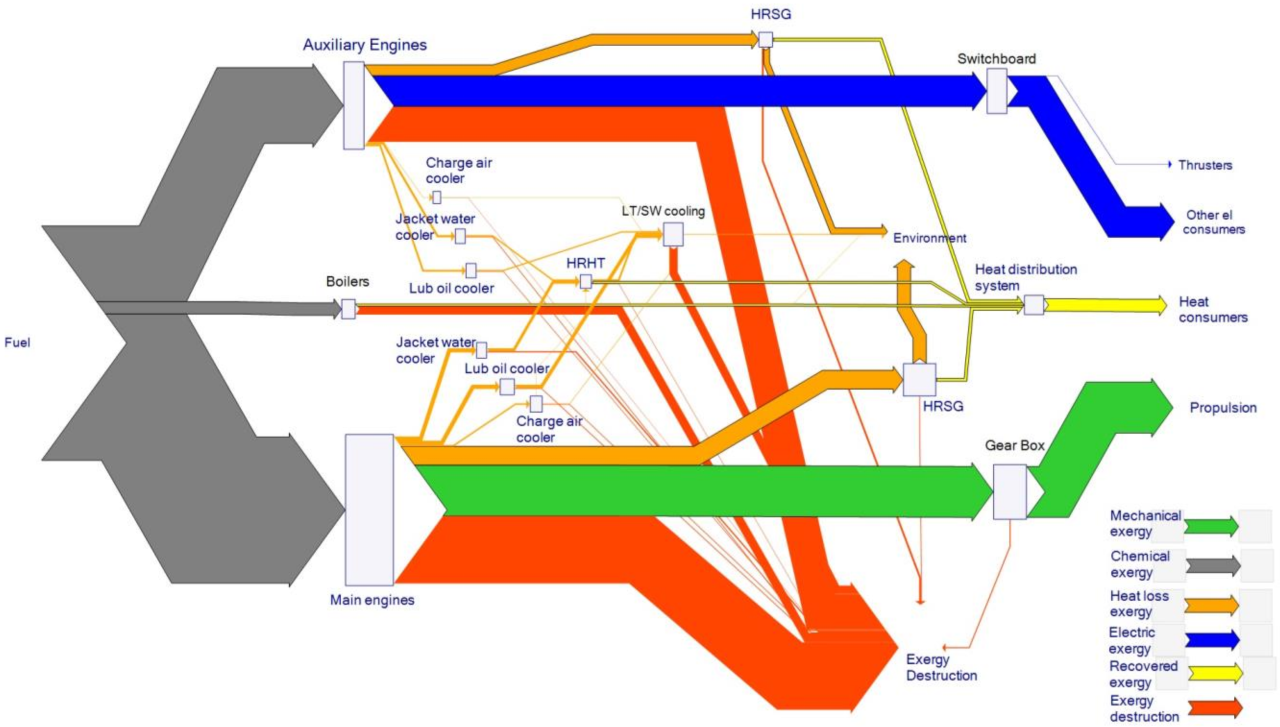


Fig. 6: Grassmann diagram for ship energy systems. Values represent the aggregated figure over one year of operations. All main engines and auxiliary engines are grouped together.

Table 2: Exergy efficiency, irreversibility ratio and irreversibility share for ship energy system thermal components

Component	Exergy efficiency	Exergy loss ratio	Irreversibility share
Main engines	34,7%	46,2%	53,1%
Auxiliary engines	38,3%	44,0%	27,8%
Auxiliary Boilers	29,2%	70,8%	6,9%
Charge air cooler (ME)	-	4,7%	0,5%
Charge air cooler (AE)	-	3,2%	0,2%
Lub oil cooler (ME)	-	6,6%	0,6%
Lub oil cooler (AE)	-	10,5%	0,5%
Jacket water cooler (ME)	-	7,8%	1,0%
Jacket water cooler (AE)	-	7,4%	0,5%
HT - LT mixer	-	1,7%	0,4%
LT - SW cooler	-	65,7%	5,4%
HRSG (ME)	37,0%	18,8%	1,5%
HRSG (AE)	28,5%	15,9%	1,4%

Table 3: Yearly shares for the different exergy losses on board, divided by operational mode

Flow	Port stay	Manoeuvring	Sea going	**All modes**
Exhaust, AE	9,1%	1,7%	19,2%	30,0%
Exhaust, ME	0,0%	2,6%	64,5%	67,1%
LT - SW cooler	0,6%	0,2%	2,2%	2,9%
All exergy losses	**9,7%**	**4,5%**	**85,9%**	**100,0%**

4. Discussion

The energy and exergy analysis of the selected ship reveal a rather well thought design, where there has been a significant attempt into the reduction of energy use, especially from the heat demand perspective. The amount of energy recovered from the exhaust gas and the engine cooling systems amounts to the most significant fraction of the overall head demand on board, and the boiler is used only in those situations when on board generated waste heat would not be sufficient to fulfil the totality of the heat demand. However, the system shows possibilities of improvement.

4.1 Suggested improvements

From an operational point of view, the main engines are often operated at very limited load, which significantly reduces the efficiency of the energy conversion. This situation is mostly due to high installed power (the vessel was designed for 21 knots but is normally operated at a maximum of 16 knots and, most often, at even lower speeds). In some parts of the journey operations on one engine only are prevented by regulations that impose to use both propeller lines in order to ensure the require capability to manoeuvre; it should be noted, however, that the achieved higher engine efficiency might be compensated by higher hull rudder resistance.

From a retrofitting/design perspective, even if engine substitution might not be possible due to the high related investment cost, engine de-rating could be a viable option for improving the off-design efficiency of the main engines. This could be achieved through cylinder disconnection, which would allow operating fewer cylinders closer to design conditions, and substitution of the existing turbocharger with one designed for lower power. An alternative worth investigating could also be that of an hybridisation of the whole system, through the installation of shaft motors/generators, which would allow both main and auxiliary engines to contribute to both propulsive and auxiliary electric power demand and, therefore, increase the flexibility of the system.

Efforts for improving the performance of a ship energy system should however not only focus on the main engines but on avoidable irreversibilities in the rest of the system, such as those caused in the HRSGs and cooling systems. These may be reduced by decreasing the temperature differences between the heat source (e.g. exhaust gases, lubricating oil) and the receiver streams.

From a thermal perspective, the existence of energy and exergy flow rates potentially available for recovery suggests that there is potential for improving the system’s efficiency and therefore reducing fuel consumption. However, the fact that most of the waste heat is available during sea passages, when on board heat demand is already fully fulfilled by the use of waste heat from the main and auxiliary engines, suggests that improvements would require more complex technical arrangements.

The utilisation of heat-to-power technologies represents one possible solution for making use of the waste heat available during sea passages. This possibility was explored by Ahlgren et al. [25] and showed significant potential for improving vessel performance. The use of WHR systems on board for heat-to-power conversion could also justify efforts in the improvement of the heat exchanger network in order to minimize exergy destruction and, therefore, allowing additional exergy to be recovered to useful power for on board use. This additional effort would be particularly justified in the case of the lubricating oil cooler and the jacket water cooler, where the exergy destruction happens at a higher rate therefore suggesting that the highest potential for improvement is located.

The use of thermal energy storage devices could constitute an alternative solution for reducing fuel consumption by providing a buffer between the excess energy available during sea passages and the unfulfilled demand during port stays. A dedicated study, as proposed by the authors in the case of a product tanker [26] is required for providing an estimate of the potential for recovery and of the required thermal storage capacity.

4.2 Limitations and further work

The limited amount of data, both from measurement and design perspective, limits parts of the analysis and therefore prevents to dig further into certain parts of the ship energy systems. The absence of measurements of the temperature levels of the heat demand in different parts of the ship prevents further considerations on heat i

ntegration. It is likely that a number of users on board require low-grade it, as in the case of HVAC pre-heaters and re-heaters, which could be provided by recovering heat from low-grade heat sources such as the lubricating oil cooler. In addition, strong assumptions were required for the calculation of the auxiliary heat demand on board and a significant improvement in the reliability of the results could be achieved if additional measurements were available, in particular for instantaneous fuel consumption for the ABs and for the heat flow recovered in the HT cooling systems.

A similar discussion can be presented for on board electrical users. The absence of measurements makes it impossible to draw conclusion on possible design and operational savings related to a minimized consumption. This influenced the possibility to analyse the operative efficiency of a number of systems and components, particularly HVAC, cooling systems and engine room ventilation, which not only are expected to contribute extensively to on board energy demand, but that are also often related to important improvement potential. Most of the cooling pumps on board are in fact equipped with frequency controllers, whose efficiency in the reduction of pump power demand was however impossible to determine.

5. Conclusion

The results of this study showed that the system under analysis was designed with significant efforts for improving energy efficiency; however, many parts of the system could be improved in order to reduce fuel consumption.

- The main potential proved to come from the main engines, which are most often operated at low load and, therefore, at low efficiency. This situation could be improved by engine de-rating or by a hybridization of the system.
- Exergy losses, mostly in the exhaust gas, also provide potential for improvement. Waste heat recovery through heat-to-power technologies is a viable option for the system under study, although a large part of the heat is already recovered for on board heat demand.
- Alternatively, the unbalance between heat availability and demand during sea passages and port stays could be solved through the use of a thermal energy storage system, which would lead to a reduction in the amount of fuel needed by the auxiliary boilers which today accounts for 7% of total the yearly fuel consumption.

The results generated by the energy and exergy analysis applied to the case study constitute a starting point for future work related to the improvement of the existing systems, as well as for the design of new similar ships. This paper partly fills the existing gap in literature concerning the analysis of ship operation in terms of energy use, with particular reference to cruise ships.

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Appendix A

Table A1: Summary of the assumptions employed in the processing of measured values for ship energy and exergy systems analysis.

Variable	Equation
$T_{air,comp,out}$	$T_{air,comp,in} + \frac{T_{air,comp,in}(\beta^{\left(\frac{k-1}{k}\right)} - 1)}{\eta_{TC,is}}$
$\eta_{is,comp}$	$P_2(load)$
$\dot{m}_{air}$	$\eta_{vol} \frac{V_{cyl,\max}\rho_{air} n N_{cyl}}{60*2}$
$\dot{m}_{fuel,ME}$	$\dot{m}_{fuel,ME,des}\left(a_0 + a_1 frp_{norm}\right)\left(\frac{n_{ME}}{n_{ME,des}}\right)$
$bsfc_{\mathrm{ME}}$	$P_2(load_{ME})$
$bsfc_{\mathrm{AE}}$	$P_2(load_{AE})$
$\dot{W}_{ME}$	$3.6^6 \frac{\dot{m}_{fuel,ME}}{bsfc_{ME}}$
$\dot{W}_{prop}$	$0.98\dot{W}_{ME}$ [27]
$\dot{m}_{eg}$	$\dot{m}_{air} + \dot{m}_{fuel}$
η_{vol}	$\frac{r_c}{r_c - 1} \frac{T_{air,in}[\mathrm{K}]}{313 + \frac{5}{6} T_{air,in}[^o C]}$ [28]
ρ_{air}	$\frac{p_{air,CAC,out}}{R_{air} T_{air,\mathrm{CAC,out}}}$
$\dot{m}_{eg,turb}$	$\dot{m}_{air} \frac{c_{p,air}(T_{air,comp,out} - T_{air,comp,in})}{\eta_{\mathrm{mech},TC} c_{p,eg}(T_{eg,turb,in} - T_{eg,turb,out})}$
$\dot{m}_{eg,bypass}$	$\dot{m}_{eg} - \dot{m}_{eg,turb}$
$\dot{Q}_{cooling}$	$\dot{Q}_{fuel} + \dot{Q}_{air,in} - \dot{Q}_{eg} - \dot{W}_{out}$
$\dot{Q}_{LT}$	$\frac{P_{2,LT}(load_{ME})}{P_{2,LT}(load_{ME}) + P_{2,HT}(load_{ME})} \dot{Q}_{cooling}$
$\dot{Q}_{HT}$	$\dot{Q}_{cooling} - \dot{Q}_{LT}$
$\dot{m}_{w,HT}$	$\dot{m}_{w,HT,des} load_{ME}$
$\dot{m}_{w,LT}$	$\dot{m}_{w,\mathrm{L}T,des} load_{ME}$
$\dot{m}_{w,\mathrm{LO}}$	$\dot{m}_{w,LO,des} load_{ME}$
$\dot{Q}_{CAC,HT}$	$\dot{Q}_{CAC} P_2(load_{ME})$
$\dot{Q}_{CAC,LT}$	$\dot{Q}_{CAC} - \dot{Q}_{CAC,HT}$
$\dot{Q}_{JW}$	$\dot{Q}_{HT} - \dot{Q}_{CAC,HT}$
$\dot{Q}_{LO}$	$\dot{Q}_{LT} - \dot{Q}_{CAC,LT}$

Table A2. Summary of the energy and exergy flow rates represented in the Sankey and Grassmann diagrams, referred to 11 months of ship operations. Values are provided in TJ

Type	From	To	Energy flow rate	Exergy flow rate
CH	Fuel	Main engines	203,5	216,6
CH	Fuel	Auxiliary engines	111,8	119,2
CH	Fuel	Auxiliary boilers	17,3	18,4
M	Main engines	Gearbox	75,2	75,2
H	Main engines	Charge air cooler (ME)	10,1	1,1
H	Main engines	HRSG (ME)	68,7	27,9
H	Main engines	Lubricating oil cooler (ME)	34,4	6,3
H	Main engines	Jacket water cooler (ME)	18,5	6,1
H	Charge air cooler (ME)	HT cooling systems	2,0	0,4
H	Charge air cooler (ME)	LT cooling systems	8,2	0,7
H	Jacket water cooler (ME)	HT cooling systems	18,5	4,2
H	Lubricating oil cooler (ME)	LT cooling systems	34,4	5,2
H	HRSG (ME)	Heat distribution system	15,8	5,4
H	HRSG (ME)	Environment	52,9	19,6
EL	Auxiliary engines	Switchboard	45,7	45,7
H	Auxiliary engines	Charge air cooler (AE)	5,0	1,0
H	Auxiliary engines	HRSG (AE)	39,5	16,0
H	Auxiliary engines	Lubricating oil cooler (AE)	14,2	2,7
H	Auxiliary engines	Jacket water cooler (AE)	9,5	3,1
H	Charge air cooler (AE)	HT cooling systems	0,3	0,1
H	Charge air cooler (AE)	LT cooling systems	4,7	0,5
H	Jacket water cooler (AE)	HT cooling systems	9,5	2,1
H	Lubricating oil cooler (AE)	LT cooling systems	14,2	1,9
H	HRSG (AE)	Heat distribution system	13,4	7,3
H	HRSG (AE)	Environment	26,1	8,7
M	Gearbox	Propeller	73,7	73,7
EL	Switchboard	Thrusters	0,6	0,6
EL	Switchboard	Other el. users	45,1	45,1
H	Heat distribution system	Fuel heating	3,2	1,0
H	Heat distribution system	Other heat users	58	17,6
H	HT cooling systems	LT cooling systems	14,0	2.8
H	HT cooling systems	Heat distribution system	16,3	3.3
H	LT cooling systems	Environment	75,6	0,9
H	Gearbox	Environment	1,5	1,5

Nomenclature

Letter symbols

B	exergy, J
$\dot{B}$	exergy flow, W
bsfc	break specific fuel consumption, g/kWh
c	specific heat, J/kg K
E	energy, J
$\dot{E}$	energy flow, W
frp	fuel rack position

h	specific enthalpy, J/kg
$\dot{I}$	irreversibility rate, W
k	specific heat ratio
m	mass, kg
$\dot{m}$	mass flow, kg/s
n	rotational speed, rpm
N_{cyl}	number of cylinders
p	pressure
P_n	polynomial of order n
$\dot{Q}$	heat flow, W
s	specific entropy, J/(kg K)
$\dot{S}_{gen}$	entropy generation rate, W/K
T	temperature, K or ^oC
V	volume, m^3
$\dot{V}$	volume flow, m^3/s

Acronymes

AB	auxiliary boiler
AE	auxiliary engine
CAC	charge air cooler
GDP	gross domestic product
HRSG	heat recovery steam generator
HRHT	heat recovery from the high temperature cooling systems
HT	high temperature cooling systems
HVAC	heat, ventilation and air conditioning
IMO	international maritime organization
JW	jacket water
LHV	lower heating value, MJ/kg
LO	lubricating oil
LT	low temperature cooling systems
ME	main engine
OECD	organisation for economic co-operation and development (OECD)
SCR	selective catalytic reactor
SG	shaft generator
SW	sea water
WHR	waste heat recovery

Greek letters

β	compression ratio
λ	irreversibility ratio
δ	irreversibility share
ε_t	exergy efficiency
η	energy efficiency
ρ	density, kg/m^3
Δ	finite difference

Subscripts

c cold
comp compressor
des design
eg exhaust gas
h hot
i component
in inlet flow
is isentropic
out output flow
prop propeller
tot total
turb turbine
0 reference state

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