

ANALYSIS OF CONTAINER SHIP ENERGY SYSTEMS

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ABSTRACT

Nowadays energy efficiency is a global concern for the maritime organization and business. Rising operating costs, mainly fuel prices and stricter environmental regulations are forcing the shipping industry to find more effective ways of designing and operating the ships in an energy-efficient manner. An analysis of the energy system of ships and understanding the different energy flows can contribute to better energy management and efficiency in the ships. Keeping this objective in mind, the energy systems of typical medium-sized container ships are analysed in this paper based on the vessel's operating data and equipment parameters collected by the crew over three months. The study focuses on the energy and exergy balances of the main components. It has been shown that more than 80% of total energy consumption is caused by the propulsion plant, while the electric power generation accounts for 14–17%. Up to 60% of the energy supply is lost to the environment through the cooling system, radiation, friction and exhaust from diesel engines. Exergy losses caused by the exhaust gas and heat transfer are other contributors. Roughly 6% of the fuel exergy input is lost due to heat transfer. The exergy lost through the exhaust gas flue mainly from the main engine is estimated at 12% of the total exergy input.

There is considerable potential for waste heat recovery on container vessels. However, to improve the overall energy efficiency, a techno-economic study has to be carried out on waste heat recovery solutions for each category of vessels and trading mode.

Keywords: Container ship, energy efficiency, energy, exergy.

1 INTRODUCTION

1.1 Background

Ships have proved to be an efficient way to move cargo, transporting approximately 80% of the world's goods by volume [1], but they are also known to threaten human health, ecosystems, and the climate. As global trade increased and shipping industry developed, global fuel consumption by the shipping sector has been increasing year on year. According to the third greenhouse gas (GHG) study by the International Maritime Organization (IMO) [2] and the International Transport Forum (ITF) report [3], GHG emissions from the shipping currently represent around 2.6% of the total global emissions, but this proportion could more than triple by 2050 if suitable measures are not undertaken at present to reverse the trend. The CO₂ emission from the ship is directly related to the energy efficiency of the ship's power plants.

In 2011, the IMO adopted mandatory technical and operational energy efficiency measures, which were expected to significantly reduce the amount of CO₂ emissions from international shipping [4]. All ships built after 2013 with 400 gross tonnage and higher must meet the minimum standard of energy efficiency, and the ships must become 10% more efficient (equalling CO₂ emission reduction) from 2015, 20% more efficient by 2020 and 30% more efficient from 2025 [5]. Consequently, new energy and energy-saving technologies of ships need to be developed by ship designers and builders, using which newly built ships can meet higher energy efficiency standards in the future. To meet this requirement, the ship's energy consumption has to be continuously monitored, measured and analysed.

Considerable potential exists in the shipping sector for reducing emissions. Various solutions already exist and a host of technologies are available for retrofitting on existing vessels [6–7]. However, the large number of variables influencing the ship's energy efficiency makes it hard to assess the ship performance to a standard baseline; consequently, energy and exergy systems shall be analysed for each ship or category of the ship to define the adequate solution.

1.2 Related work

Many papers related to energy efficiency improvement in ships have been published [8–13] using energy criteria (first law) for performance assessment. Some others are addressing the performance of the ship's propulsion [14–15] considering both the energy and the exergy criteria. Few studies focusing on the analysis of the overall the ship's energy system can be found in the literature; in Ref. [16], the interaction and interrelations within a generalized ship energy system are analysed from the reference energy system perspective, but this work is more theoretical since it is founded on a generic ship. Baldi and co-workers [17–18] present a good example of energy and exergy analysis of a cruise ship system but that is not valid for other types of vessels. The energy needs and distributions depend on the ship's type, a trading area, and operation mode. In this paper, the focus will be on the container vessel energy system based on the previous case study [19] presented at the eighth International Conference on Energy and Sustainability 2019. That study is extended here by considering two sample vessels and excluding the boiler energy consumption, which is considered negligible.

1.3 Aim and approach

This study aims to understand how energy consumed by a container ship is used and where the potential for heat recovery exists. To this end, the fundamentals of energy and exergy analysis will be applied to the machinery system of a typical container vessel of a medium size. This category represents 20% of the total world container fleet and 30% of the total power installed and fuel consumed by the whole container vessel fleet. About 5,000 container ships, making up roughly 14% of deadweight in the global fleet, consume the most energy (26%) and fuel (25%) of any ship class and emit most amount of black carbon (26%) [1].

The study will be based on ship's records, mainly the daily report (noon report). This report is prepared by the ship's master and sent daily to the company staff ashore providing data on several parameters, namely fuel consumption, the speed of the vessel, the weight of the cargo, the speed and the wind sector, the engine power, the electric power, among others. These data are sent every day regardless of the vessel's position (at sea or in the port).

The ship machinery is continuously monitored, and all parameters are recorded and stored in the automation system; this includes sensors for temperature, pressure, flow, speed, electric current, voltage frequency power, among others. The technical documentation (instruction manual, test protocol, technical files) issued by the component manufacturers is also used.

After collecting all the information and data from the sample ships, the energy and exergy analysis fundamentals are applied. As the boiler energy consumption is estimated at 1% of the total energy flow in the container vessel category, it will not be considered in this study.

1.4 The ship's energy system

Figure 1 represents the typical energy flows of the container ship system. The ship is a large and complex system that must be self-sustaining in its environment for long periods. To

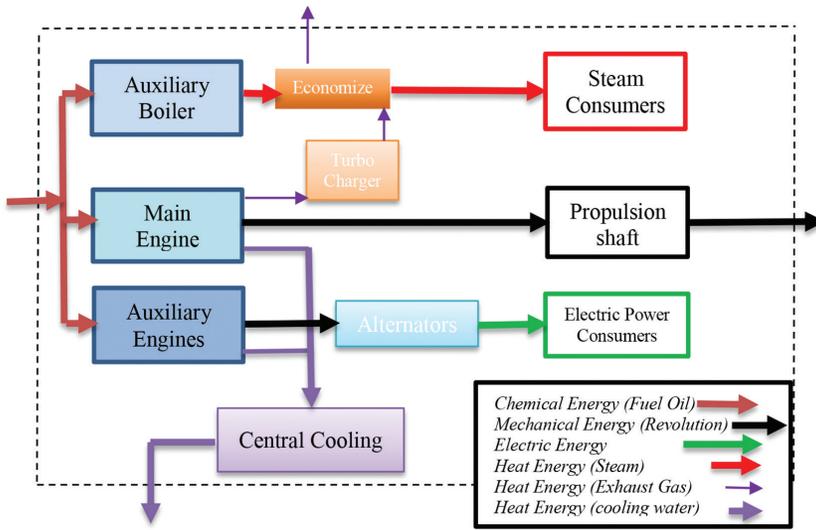


Figure 1: Ship energy system.

enable this operation, she must be fitted with the machinery necessary for propulsion and operations, which mainly consist of:

- The propulsion plant; the most commonly used is the diesel engine [20], which converts chemical energy to mechanical energy by burning fuel oil as depicted in Fig. 2.
- Electric power production: mostly by diesel generators.
- Auxiliary machines necessary for the safety operations of the propulsion system, navigation system and crew living conditions on board: boilers, cooling pumps, lubricating pumps, fuel oil treatment, firefighting system, air compressors, air conditioning, deck machinery, among others.
- Heating system: For heat production, a steam boiler or thermal oil heater is fitted on board as an auxiliary machine for heating systems of fuel oil, lubricating oil, cooling water, sanitary and air conditioning.

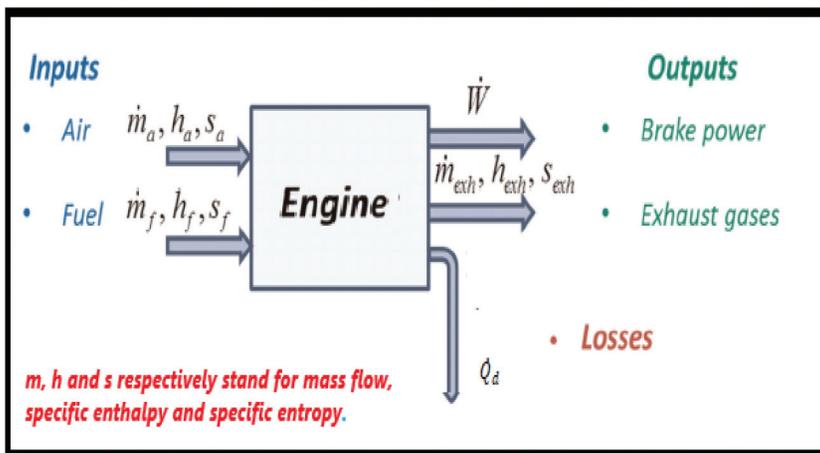


Figure 2: Diesel engine energy flows.

During the passage through the sea or ocean, the exhaust gas flow from the main engine is enough to produce steam or thermal oil without the need for firing the oil-fired boiler/thermal heater. At the port and during ship slowdown, the main engine exhaust gas flow is missing or reduced, and the oil-fired boiler is used to supply the heating steam or thermal oil.

2 ENERGY AND EXERGY PRINCIPLES

It is well known that applying only the first law of thermodynamics is inadequate for evaluating some features of energy resource utilization. For this reason, the use of the second law of thermodynamics has been intensified in internal combustion engines. An exergy-based performance analysis generally means the performance analysis of a system based on the second law of thermodynamics that overcomes the limit of an energy-based analysis. The equations for the second law analysis of the system are used to analyse the operation of ship-board engines [21].

Even though exergy analysis is similar to energy analysis in its characteristics, there are some fundamental differences between them [22]. By applying it to a thermal system in addition to the energy analysis, irreversibility could be detected and so the best ways to minimize energy destruction, losses and increase efficiency can be understood. Energy analysis provides data on the amount of energy variation as a function of energy transfer.

2.1 Energy flow

Energy analysis provides a calculation of the internal energy variation as a function of energy transfers across the boundaries as heat or work and the enthalpy, which is related to the mass flow passing through these boundaries [23].

For the purpose of simplifying the first law calculations of the engine, the following assumptions were made: the engine operates at steady-state; the whole engine, including propeller shaft, is selected as a control volume; the combustion air and the exhaust gas each form an ideal gas mixture and potential; and kinetic energy effects of the incoming and outgoing fluid streams are ignored.

After these assumptions, the fuel energy rate of the control volume is given by

$$W = \dot{m}(K_e + P_e) + W_i \quad (1)$$

$$Q = \dot{m}h + Q_i \quad (2)$$

where \dot{m} represents the mass flow rate carrying the kinetic (K_e) and potential (P_e) energy work or enthalpy (h) in or out of the system; W_i and Q_i are the direct input/output energy to/from the system.

The conservation of mass principle for a general steady flow system with multiple inlets and outlets can be expressed in a rate form as

$$\sum \dot{m}_i = \sum \dot{m}_o \quad (3)$$

where \dot{m}_i represents the inlet flow rate of the mass that consists of air and fuel and \dot{m}_o represents outlet mass consisting of exhaust gases. It is considered that the fuel enters the engine with a mass flow rate \dot{m}_f and is mixed with air \dot{m}_a to form \dot{m}_i . Subscripts (i) and (o) denote input and output respectively.

The energy balance for the control volume in steady-state conditions, with kinetic and potential energy being neglected, is, in the general sense, given by:

$$\dot{Q} + \sum \dot{m}_i h_i = \dot{W} + \sum \dot{m}_o h_o \quad (4)$$

where \dot{m} is the mass flow rate and h is the specific enthalpy. Besides, \dot{Q} indicates the net heat transfer rate and \dot{W} indicates the brake power (the network output).

The amount of energy entering a control volume in all forms (by heat, work, and mass) must be equal to the amount of energy leaving it.

Considering that the diesel engine generates brake power and some heat produced by combustion is transferred to the entropy environment (combustion air enthalpy ignored), Eqn (4) can be written as

$$\dot{Q}_f = \dot{W} + \dot{Q}_{ex} + \dot{Q}_d \quad (5)$$

where \dot{Q}_f is the fuel energy rate, \dot{Q}_{ex} is the exhaust energy rate, \dot{Q}_d is the lost energy rate, which is transferred to the environment by cooling, lubricating, that is, heat transfer except for the exhaust.

The brake power is given by

$$W = \omega \tau \quad (6)$$

where ω is the angular velocity; and τ is the engine torque.

Since the combustion air is in the same state as the standard reference state defined as $T_0 = 293$ K and $P_0 = 1$ atm, the energy of the combustion air can be neglected so the heat input rate to the control volume by mass is represented only by chemical energy of fuel \dot{E}_f which can be calculated using the mass flow rate of the fuel \dot{m}_f and its net calorific value (NCV) as expressed by

$$\dot{E}_f = \dot{m}_f (NCV) \quad (7)$$

The exhaust energy rate \dot{Q}_{ex} can be calculated as a function of the mass flow rate of each exhaust gas components \dot{m}_{ex} which is obtained using combustion equations and the enthalpy change (Δh) represented by the difference between the enthalpy of the exhaust temperature and that of the reference temperature of each exhaust gas species as given by

$$\dot{Q}_{ex} = \dot{m}_{ex} (\Delta h) \quad (8)$$

The engine characteristics are essentially expressed by the thermal efficiency (η) and brake-specific fuel oil consumption *Sfoc*. While the thermal efficiency is determined as the ratio of brake power to the fuel energy rate of the control volume, *Sfoc* is defined as a measure of how much fuel is consumed in 1 h to obtain 1 kW brake power and they are, respectively, denoted as in eqns (9) and (10):

$$\eta = \frac{\dot{W}}{\dot{E}_f} \quad (9)$$

$$Sofc = \frac{\dot{m}_f}{\dot{W}} \quad (10)$$

2.2 Exergy flow

Unlike energy, the value of exergy depends on the state of the environment as well as the state of the system. Therefore, exergy is a combination property. The exergy analysis of thermal systems is performed to improve energy source utilization by determining the order of exergy destructions and losses in the processes and components of the system and then by reducing them.

The assumptions made for energy analysis are also applicable to exergy analysis. Based on these assumptions, the exergy balance for the control volume can be given in the general sense by

$$\dot{B}_Q + \dot{B}_W + \sum \dot{m}_i \psi_i - \sum \dot{m}_o \psi_o - \dot{B}_d = 0 \quad (11)$$

where \dot{B}_Q is the exergy transfer rate related to heat transfer between the control volume and the environment; \dot{B}_W is the exergy transfer rate associated with work transfer; \dot{B}_d is the exergy destruction rate of the control volume and $(\dot{m}_i \psi_i, \dot{m}_o \psi_o)$ are the exergy transfer through the intake and exhaust process, respectively, where ψ_i, ψ_o are the specific exergies of the fuel and exhaust gas, respectively, and similarly \dot{m}_i, \dot{m}_o are the mass flow rates of the fuel and exhaust gas, respectively.

The exergy input rate with mass transfer to the engine is formed mainly by fuel exergy; the combustion air exergy can be neglected by assuming that intake air enters the engine at ambient conditions. The specific chemical exergy of fuel is obtained by multiplying the Net Calorific Value (NCV) of fuel and chemical exergy factor as

$$\psi_f = \varphi NCV \quad (12)$$

The chemical exergy factor φ is based on the composition of fuel oil (C, H, O, S) [16].

The exergy transfer rate associated with work is equal to the network for the engine, that is,

$$\dot{B}_W = \dot{W} \quad (13)$$

Exergy output rate with mass transfer from the engine consists of exhaust exergy, which can be expressed as

$$\dot{B}_{ex} = \dot{m}_{ex} \psi = \dot{m}_{ex} C_{pex} T_0 \left(\frac{T}{T_0} - 1 - \ln \frac{T}{T_0} \right) \quad (14)$$

Exhaust exergy per unit mass ψ can be expressed as the sum of the specific thermomechanical and chemical exergies of the exhaust gases with the assumption that the exhaust gas is an ideal gas mixture [24–25].

Exergy transfer rate associated with the heat transfer is obtained by

$$\dot{B}_Q = \sum \left(1 - \frac{T_0}{T} \right) \dot{Q}_d \quad (15)$$

where T is the cooling water temperature equal to the system boundary temperature; \dot{Q}_d is the lost (wasted) energy rate, which is transferred to the environment by cooling, lubricating, heat transfer except for the exhaust gas. When determining the rate of exergy transfer accompanying heat transfer, it was assumed that \dot{Q}_d is rejected into the ambient air from the boundary having the same temperature as the engine coolant exiting the engine block.

The exergy efficiency is expressed as the ratio of the brake power exergy to the fuel exergy of the control volume, that is,

$$\gamma = \frac{\dot{W}}{\dot{B}_f} \quad (16)$$

3 CASE STUDY

3.1 Container ships under study

As already stated in the introduction, two typical container vessels (A) and (B) are chosen for the present study, with the main details as shown in Table 1. Each vessel is fitted with machinery consisting of the propulsion system and auxiliary machines:

Table 1: Main particulars of ship under study.

	Vessel (A)	Vessel (B)		Vessel (A)	Vessel (B)
Length (m)	200	261	Propulsion Engine	21,770 kW	36,560 kW
Breadth (m)	29.8	32	Diesel generator	3 sets: 1,890 kW	4 sets: 1,840 kW
Draught (m)	11.4	12.6	Boiler	2.5 t/h_9 bars	3.5 t/h_11bars
Gross tonnage	26190	40542	Cargo capacity	2600 TEU	4250 TEU

- Both vessels are propelled with one main diesel engine (ME) turning a single propeller at a maximum speed of 25 Knots.
- Ship (A) and (B) are also fitted, respectively, with three and four diesel generators, each producing electric power needed for auxiliary machinery operations and cargo reefer container feed.
- Auxiliary machinery for the ship safe operation, including boiler, which is not covered by this study as stated before.

3.2 Measurements

Vessel (A) is operating in regular-line short trips between the Northwest European ports and West African Ports. Vessel (B) is trading in long voyages between China and the US East Coast.

The noon reports and ship's records for three-round voyages lasting approximately three months were collected. Sorting of these data was necessary, as without this, the calculations could be altered because of zero speed or consumption values corresponding to periods of stoppage or anchorage. The data were filtered, and the non-consistent values recorded by mistake or inaccurately by the crew were ignored.

In addition to the noon report, the engine and auxiliary engine performance records at different loads were provided by the crew in addition to equipment instruction books. Some parameter values were collected from engine logbook, which were mainly engine parameters and electric power consumption.

In Table 2, the average percentage of load versus running hours is summarized. The fuel oil consumption at each percentage of maximum load and the exhaust temperature is obtained from the ship's records. The fuel chemical energy is obtained from eqn (7) while the exhaust gas flue is calculated based on the engine specifications and tests as stated in the engine-approved technical files. The same approach was adopted for the auxiliary engine (AE) parameters as shown in Table 3.

The average power represents less than 60% of the engine nominal power according to the performance record provided by the crew. The diesel engines installed on both ships are from the different makers, but from the same engine generation; therefore, complying with the same requirements related to performances and emissions.

The exhaust gas temperature after the turbocharger is taken from machinery parameter records. The exhaust gas flow is calculated based on the engine intake pressure and temperature recorded by the crew.

Table 2: Main engine average parameters vs % of running time.

	% Load of MCR	% time running	Power (kW)	FO Cons. (kg/h)	E_f (MJ) Eqn. (7)	T_{ex}* after T/C (°C)	m_{ex} (t/h)
Vessel (A)	15	11	3100	750	8771	295	39
	25	15	5100	1120	13109	290	58
	35	24	7150	1416	16560	290	75
	55	15	11700	2192	25634	280	115
	65	32	14050	2625	30698	280	140
Vessel (B)	10	04	3600	805	9550	290	41
	20	44	7300	1510	17910	290	78
	30	21	11000	2145	25440	290	121
	40	12	14500	2810	33570	285	151
	50	19	18000	3380	40100	285	185

MCR: *Maximum continuous rate* (100% load), T/C: *Turbo charger*, FO cons: *Fuel coil consumption*, T_{ex}*: *Exhaust gas temperature*, m_{ex}: *Exhaust gas flow*.

Table 3: Auxiliary engine average load and parameters.

	Average Power (kW)	m_{ex} (kg/h)	T_{ex}. (°C)	FO cons. (kg/h)
Vessel (A)	1050	10850	355	232
Vessel (B)	1270	13120	340	266

The engine manufacturer provides diagrams and empirical formulas to calculate exhaust gas flow and temperature after the turbocharger at different engine loads, but it is based on bed test conditions [26].

For the engines with the same output, the exhaust gas flow and temperature are different depending on the maker's design specifications.

4 RESULTS AND DISCUSSION

4.1 Energy and exergy inputs

The total measured fuel oil consumption for ME, AE and AB are shown in Table 4. The energy and exergy values stated are calculated. The fuel used onboard is the heavy fuel oil IFO 380, which has a density @ 15°C, kg/m³. The net calorific value of this fuel oil category is NCV=42.7 MJ/kg. The fuel oil-specific exergy corresponds to the gross calorific value. The gas constant of air is $R = 0.287 \text{ kJ/kg}^\circ\text{K}$. The specific heat of air at the average temperature of exhaust gases (350°C) is $c_p = 1.065 \text{ kJ/kg}^\circ\text{K}$ [17].

As shown in Fig. 3, the propulsion is the main energy consumer on board, responsible of respectively 82% and 85% of the total fuel oil consumption of the vessels (A) and (B).

The auxiliary engines for electric power production account for 17% and 14% of the energy demand respectively for vessels (A) and (B).

Table 4: Energy and exergy inputs.

Consumer	Main Engine		Auxiliary Engine		Auxiliary Boiler	
	A	B	A	B	A	B
Vessel						
FO cons. (ton)	2313	3846	485	630	29	28
Running hours	1384	2090	1905	2177	412	87
\dot{E}_f (TJ)	98765	164224	20709.5	26901	1272.5	1196
\dot{B}_f (TJ)	105086	174735	22035	28623	1353.9	1272
\dot{E}_f (%)	81.8	85.4	17.15	13.98	1.05	0.6
\dot{B}_f (%)	81.8	85.4	17.15	13.98	1.05	0.6

Percentages of energy consumption

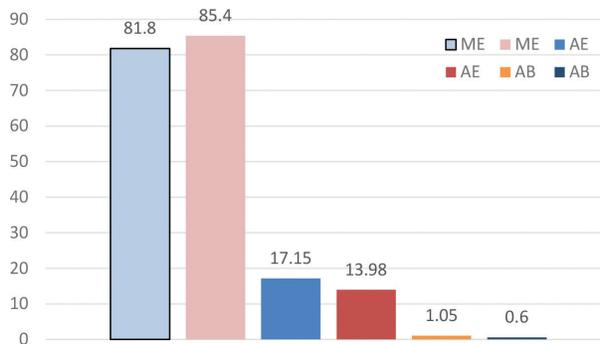


Figure 3: Energy consumption shares.

The auxiliary boiler consumption is approximately at 1% or less because the heating is mainly maintained by ME exhaust gas boiler where the exhaust gas wasted heat is recovered. For this reason, boiler energy consumption is neglected in this study. Since the specific fuel exergy is aligned with the lower heating value, the percentages of fuel exergy are similar to those for energy values.

4.2 Efficiencies

The parameter values shown in Table 5 are calculated by substitution in the eqns (4), (7), (8), (9), (13), (14) and (15) hence the different efficiencies are obtained. The reference state is defined as $T_0 = 293$ K and $P_0 = 1$ atm.

4.3 Energy distribution

The energy distribution for each vessel is depicted in Fig. 4. For both vessels, 38–40% of the energy flow consumed by the main engine is converted to work while only 36–38% of the fuel oil entering the auxiliary engines is converted to useful work. The differences between

Table 5: Diesel engine efficiencies.

	Vessel	Av. (KW)	\dot{Q}_{ex} (KW) (7)	\dot{Q}_d (KW) (4)	\dot{B}_{ex} (KW) (13)	\dot{B}_Q (KW) (14)	\dot{B}_d (KW) (10)	η (8) (%)	γ (15) (%)	Sfoc (9) g/ KwH
Main Engine (ME)										
ME	A	7530	5750	6542	2470	1188	9389	38.1	36.6	221
	B	10900	8560	7690	2993	1360	13736	40	37.6	210
% of \dot{E}_f	A	38	29	33.1						
	B	40	31	28.2						
% of \dot{B}_f	A	35.7			12	5.8	45.6			
	B	37.6			10.3	4.7	47.4			
Auxiliary Engine (AE)										
AE	A	1050	960	1009	328	176	1647	34.9	32.8	238
	B	1270	998	1211	403	260	1820	35.9	33.8	233
% of \dot{E}_f	A	34.9	32	33.5						
	B	36	29	34.3						
% of \dot{B}_f	A	32.7			10.2	5.5	51.4			
	B	33.9			10.7	6.9	48.5			
Vessels (A) & (B): Global efficiencies										
Glob- al	A	8580	6710	7551	2798	1364	11036	37.7	36	
	B	12170	9458	8901	3396	1620	15556	39.5	37.2	
% of \dot{E}_f	A	37.1	29.5	33.2						
	B	39.5	30.7	28.9						
% of \dot{B}_f	A	35.4			11.8	5.7	46.4			
	B	37.2			10.4	5	47.5			

the two vessels can be explained by the difference in maintenance conditions of the engines, the ship's hull, and the operation mode.

Also, approximately 29–31% of the energy, which is lost to exhaust gas, will be partly recovered in the exhaust gas boiler to produce heating steam. Besides, the rest of energy (respectively 33% and 29% of vessels A and B) is wasted to the environment by the cooling system, radiations, and mechanical frictions.

The global energy performance of vessels A and B is similar to that of the main engine; they are respectively 37.7% and 39.5%, as the main engine consumes fuel oil to the major extent. The percentage of energy contained in the exhaust gas is 29.5% and 30.7% for ship A and B, respectively, the difference between the two engines is due to the slight difference in the exhaust gas temperature and the difference in the air/fuel ratio.

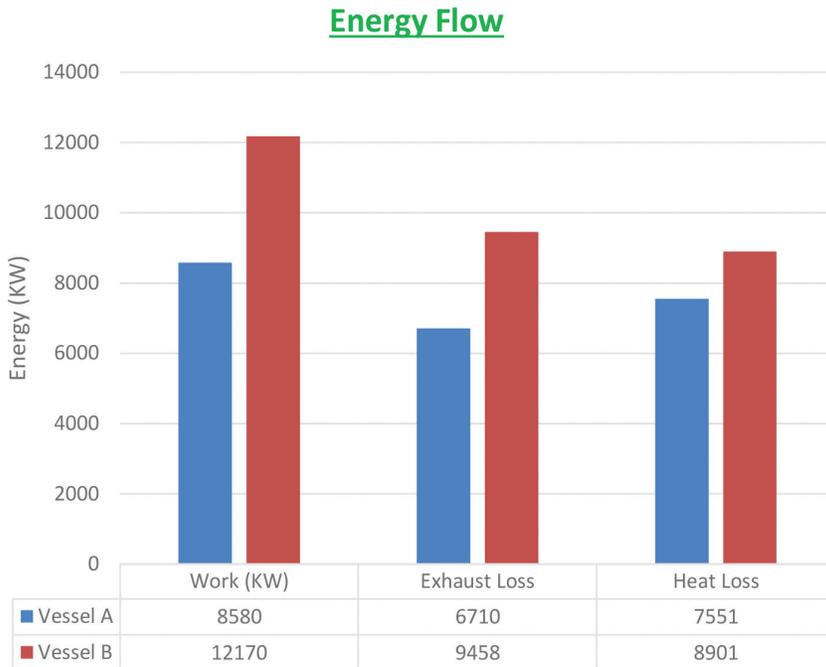


Figure 4: Ship energy values.

Up to 33.2%, 29% of the energy input is wasted to the environment by cooling systems, radiation and mechanical frictions. Globally, it can be noted that, for container vessels, less than 40% of energy is converted to useful work.

4.4 Exergy distribution

As shown in Fig. 5, the exergy balance is similar to that of energy, that is, 36–38% of the exergy input to the main engine is transformed on useful work (propulsion) while only 33–34% of fuel energy consumed is converted to electricity by auxiliary engines. Furthermore, 5–6% of the fuel exergy input is lost due to heat transfer. The exergy lost through the exhaust gas flue is estimated at 11–12% of the total exergy input; also, 46–47% of exergy is destructed.

4.5 Break specific fuel consumption (bsfc)

The specific fuel oil consumption measured for the main engines is 210–221 g/kWh, much higher than the manufacturer's value, which is less than 180. The same is true for the auxiliary engines; it increases to 233–238 g/KWh instead of 195–200 g/KWh specified by the manufacturers. The increasing of the Bsfc can be explained by the operating mode of the engines, which are mostly used at less than 60% of the nominal output and by the maintenance conditions of the engines.

The difference in Bsfc between the main engine and the auxiliary engine is due to the cycle type. The main engines are two-stroke engine type while the auxiliary engines are medium-speed four-stroke engine type. It is known that the two-stroke engine has better thermal efficiency.

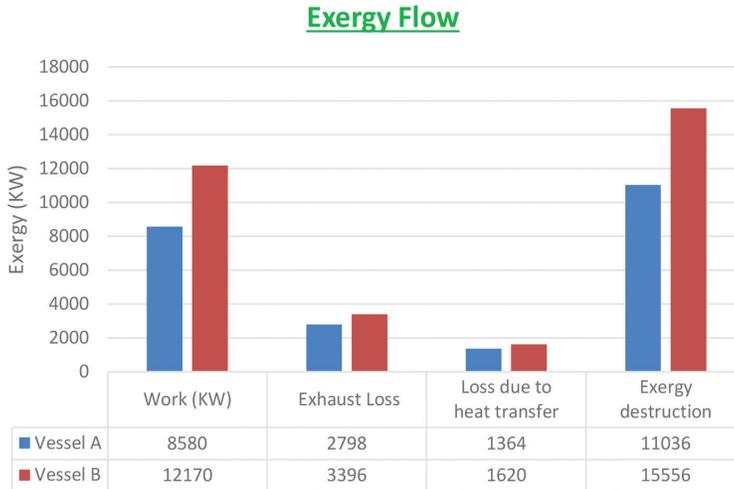


Figure 5: Ship exergy values.

It should be noted that as is normal practice in shipbuilding, part of the exhaust gas energy is recovered by an economizer fitted on the main engine exhaust manifold, but still, the gas is expelled to the atmosphere at 190°C. In fact, at gas temperatures below 180°C, the vaporized un-burned oil particles in the gas condense to an adhesive mixture of soot and oil, resulting in troublesome deposits on the tubes and the heating surface of the boilers. Also, there is the risk of low-temperature corrosion on the gas side of the tubes (the sulfuric acid dew point temperature is assumed to be at 130–140°C).

It was demonstrated by other works in the literature [27] that for such auxiliary engine size, the global energy efficiency can be enhanced by installing an economizer on the exhaust system similar to the one fitted to the main engine. Hence the steam produced will partly compensate for the lack of steam produced by main engine economizer in the port.

This result can be extended to a similar vessel category; however, it should be noted that the container vessels are designed for a speed of 20–25 knots corresponding to engine nominal output, but during the last decade, due to market shrinkage and fuel oil price, the container vessels are operated at low streaming mode (less than 50% of engine power). In normal operations, the share of propulsion energy consumption would be much higher with better energy efficiency.

The diesel generators are designed for the normal safe operation of the ship and for supplying the reefer containers. Due to the low market, the number of reefer containers loaded is less than the ship's capacity; therefore, the auxiliary engine is running under load. Otherwise, it could be operated at higher loads within the design load range with a better efficiency range.

5 CONCLUSION

In this paper, the energy and exergy balance of two container vessels operating at a different zones and different voyage patterns are studied. The analysis was based on the data and measurements recorded by the crew and transmitted to the ship management.

It has been established that 82–85% of the energy consumed by the container vessel is for propulsion use while the electric power generation accounts for 14–17%. Steam production is responsible for less than 1% of energy consumption and therefore it is not considered in this study. Heating needs are mainly satisfied by the steam produced in the exhaust gas economizer fitted on the main engine.

The global energy performance of both vessels is similar to that of the main engine as the latter consumes fuel oil to a major extent. Approximately 40% of fuel energy is converted to useful work propulsion and electricity.

The percentage of energy contained in the exhaust gas is approximately 30%. The remaining 30% of energy is wasted to the environment by cooling systems, radiations and mechanical frictions.

The exergy balance is similar to energy balance, that is, 36–38% of the exergy input to the main engine is transformed on useful work (propulsion) while only 33–34% of fuel energy consumed is converted to electricity by auxiliary engines. Moreover, 5–6% of the fuel exergy input is lost due to heat transfer. The exergy lost thorough the exhaust gas flue is estimated at 11–12% of the total exergy input mainly in the main engine.

The efficiencies may be improved by operating the engines at an optimal point 75–85% of the nominal load. The best diesel engine efficiency is 50%, and therefore, there is still a huge amount of energy loss from the cooling system and the exhaust gas.

There is considerable potential for waste heat recovery on container vessels. Various solutions already exist and a host of technologies available for retrofitting on existing vessels; however, to improve the overall energy efficiency, a techno-economic study has to be performed on waste heat recovery solutions for each category of the vessels and trading mode.

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