Ocean Engineering 264 (2022) 112493

Contents lists available at ScienceDirect

Ocean Engineering

journal homepage: www.elsevier.com/locate/oceaneng

Waste heat recovery from marine main medium speed engine block. Energy, exergy, economic and environmental (4E) assessment – Case study

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ARTICLE INFO

Keywords:
Energy
Exergy
Sustainability
CO_2 – NO_x emissions reduction
medium speed marine diesel engine

ABSTRACT

90% of traded goods are transported by ship, the vast majority of which are propelled by diesel engines with an energy efficiency of no more than 50%. Among the developments to increase it, waste heat recovery (WHR) technologies still have room to increase overall efficiency.

This study looks into the feasibility of recovering waste heat radiated by the block of a large marine diesel engine. An energy, exergy, economic and environmental (4E) assessment is presented. The analysis identifies the amount of heat from the block which is possible to recover and looks into the cost and CO_2 emissions reduction gained when waste energy is reused. A cement carrier vessel propelled by two four-stroke engines with 3000 kW nominal rating is used as case study vessel. From onboard measurements and fluids analysis, results unveil the overall heat radiated from engine block and coolers to be 14.93% of total. Scale factor makes available up to 369 kW of radiated heat. Fuel savings achieved by engine block heat recovery will lead to reductions of 3.31% of the total CO_2 and 8.33% of the specific NO_x gases emitted. Technological means of heat recovery can range from the temper of hot water to use in vessel's accommodation to the recharge of electric batteries by means of thermoelectric generators.

1. Introduction

Climate change due to global warming is a proved fact and can lead to high impact weather effects like floods, sea-level rise and desertification (Hanlon et al., 2021; Schleussner et al., 2016; United Nations, 2020). Because of this, United Nations approved the 2030 Agenda made up of 17 Sustainable Development Goals (SDG's) that seek to achieve prosperity that is respectful with the planet and its inhabitants (United Nations, 2015). Among these SDG's, the compliance with SDG 13: Climate Action, makes an active, determined contribution to a sustainable, low-carbon future to fight against climate change. By adopting the 2030 Agenda, the European Commission's Green Deal aims to make Europe the first climate-neutral continent by 2050 (Dr. Ing. Johan Breukelaar, 2019).

International Maritime Organization (IMO), following the 2030 Agenda, seeks to reduce CO_2 emissions by at least 40% by 2030, pursuing efforts towards 70% by 2050, compared to 2008 (International Maritime Organization, 2019). Latest IMO's Marine Environment Protection Committee (MEPC 77th) session has recognized the necessity of decarbonization. Consequently, the objective of zero-emission-vessels is being declared as a priority (International Maritime Organization, 2021b). One of the proposals, presented by the European Commission and United States, is the inclusion of information about vessel's energy efficiency of existing ships index (EEXI) and carbon intensity performance (CII) in the IMO Data Collection System (DCS) (International Maritime Organization, 2021a).

Worldwide commerce depends on maritime transport for 90% of traded goods. Despite being one of the most critical and massive means of transportation, the contribution to the anthropogenic CO_2 emissions of shipping is only 2.89% of global (International Maritime Organization, 2021). Additionally, large marine engines emit Nitrogen Oxides (NO_x) as a combustion byproduct. Environmentally, NO_x emissions are largely responsible of ozone layer destruction. In addition, NO_x are considered one of the main causes of respiratory diseases (Boningari and Smirniotis, 2016). Being aware of this, IMO established NO_x emissions regulation requirements for any engine over 130 kW (International Maritime Organization (IMO), 2016, 2013, 2008)

According to the United Nations Conference on Trade and Development report "*Review of maritime transport 2021*" world merchant fleet is, as an average, 21 year old (UNCTAD, 2021). These data reveal the necessity of adapting decarbonization measures to the already existing

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https://doi.org/10.1016/j.oceaneng.2022.112493
Received 13 April 2022; Received in revised form 24 May 2022; Accepted 3 September 2022
Available online 15 September 2022

Nomenclature

A	Exposed area of cylinder walls (m^2)
B	Cylinder bore
c	Carbon
c1 – c4	Experimental values for FMEP
Cos φ	Power Factor
Cp	Specific heat under constant pressure (J/kg K)
Cu	Circumferential velocity (m/s)
E	Energy (J)
Ex	Exergy (J)
FMEP	Friction Mean Effective Pressure (Pa)
g	Gravity (m/s^2)
H	Enthalpy (J)
h	Heat Transfer Coefficient (W/m^2 K)
h	Hydrogen
h	Specific enthalpy (J/kg)
HT	High Temperature (cooling water)
I	Current (Amp)
I	Irreversibility (J)
IMEP	Average Indicating Effective Pressure (Pa)
L	Losses (J)
LHV	Lower heating value (kJ/kg)
LT	Low Temperature (cooling water)
M	Torque (N/m)
$\dot{m}$	Mass flow rate (kg/s)
$\dot{n}$	Molar flow rate (mol/s)
n	Number of moles
o	Oxygen
ORC	Organic Rankine Cycle
P	Pressure (Pa)
Q	Heat (J)
R	Specific gas constant (J/kg K)
S	Entropy (J/K)
s	Sulphur
T	Temperature (K)
t	Time (s)
U	Internal energy (J)
Up	Piston speed (m/s)
V	Volume (m^3)
V	Voltage (Volts)
W	Work (J)
w	Local average gas velocity (m/s)
Y	Mass fraction
z	Height (m)

Greek Symbols

α	Thermal diffusivity
η	Efficiency
θ	Crankshaft angle (Degrees)
µi	Specific Gibs free enthalpy (J/kg)
Ϭcycle	Moran Shapiro irreversibility correlation
Ϭsystem	Entropy generation (irreversibilities) (J/K)

Subscripts

a	Absorbed
avoid	Avoidable
bb	Blow by
cham	Combustion chamber
cham.block	Heat transferred to the engine block
cham.comb	Heat due to combustion
cham.coolHT	Heat dissipated on HT water
cham.coolLT	Heat dissipated on LT water
cham.LO	Heat transferred to lube oil
charge.air	Charge Air
chem	Chemical
comb	Combustion
cool	Cooling fluid
cycle	Cycle
cylinder	Cylinder
EDP.fuel.pump	Fuel pump, engine driven
EDP.HT.pump	HT water pump, engine driven
EDP.LO.pump	LO pump, engine driven
EDP.LT.pump	LT water pump, engine driven
env	Environment
env.man	Manifold to environment
env.turbo	Turbo to environment
exh.gas	Exhaust gas
exhaust	Exhaust
fr.bear	Bearing losses due to friction
fr.pis	Piston losses due to friction
fuel	Fuel
fuel.inlet.engine	Fuel inlet at the engine
fuel.return	Fuel return to tank
gas	Mixture gas inside cylinder
Heat.trf	Heat transfer
ideal.gas	Ideal gases
inlet	Inlet
int.rev	Integrator on reversible function
intr	Intrinsic
IVC	Inlet Valve Closing
kin	Kinetic
leaks.clean	Clean fuel leaks
leaks.dirty	Dirty fuel leaks
LEDP	Engine Driven Pumps looses
Lfr.valv	Valve losses due to friction
lube.oil	Lube oil
mixture	Mixture
motor	Electric motor
outlet	Outlet
phys	Physical
pot	Potential
products	Products
pumping	Pumping work
reactives	Reactives
shaft	Shaft
startir.air	Starting Air
turbo	Turbo compressor
unb	Unburnt
uncert.ext	External uncertainty
uncert.int	Internal uncertainty
wall	Cylinder walls
Work	Work on engine shaft

Superscripts

sensible	Sensible

fleet.

Approximately 90% of merchant vessels in service are propelled by one or more diesel engines (Lamaris and Hountalas, 2010; Yao et al., 2019). Of those, in best large internal combustion engines only half of the energy supplied from fuel is transformed in mechanical energy at the propeller shaft (Wärtsilä, 2015; Winterthur Gas and Diesel, 2021). This means almost half of the energy supplied through the fuel is dissipated, its majority as waste heat (Ma et al., 2012; Wang et al., 2013; Wärtsilä, 2015). Several studies within the engine such as variable valve timing, dual-fuel, turbochargers and common rail injection are employed to improve performance (Ariani et al., 2019; Tadros et al., 2019; Zhou et al., 2017). After these, the room for an efficiency increment by tuning the engine is very little. This means almost half of the energy supplied through the fuel is dissipated, its majority as waste heat (Ma et al., 2012; Wang et al., 2013; Wärtsilä, 2015). Waste Heat Recovery technology is then the currently subject of ongoing research with potential to increase power output, and thus reducing specific fuel consumption and emissions. Also, it has been acknowledged as an effective technical solution for marine power plants (Singh and Pedersen, 2016).

Automotive industry has been the focus of many studies around energy and exergy efficiency and relevant literature has been developed (Alkidas, 1988; Benajes et al., 2015; Ding et al., 2011; Huber et al., 1990; Payri et al., 2014; Punov et al., 2016; Van Gerpen and Shapiro, 1990; Watson et al., 1980; Woschni, 1967). Payri et al. presented a Global Energy Blancae (Payri et al., 2014), later revised by Carreño (Carreño Arango, 2016). On the contrary, large diesel engines have received less attention. In general terms, manufacturers include a heat balance of their engines but always evaluating ideal, new cases (Wärtsilä, 2015, 2019) from the First Law of Thermodynamics point of view. Even in commercial engines, measurements are taken during Factory Acceptance Tests of the product, when the engine is at its best.

The study of the large diesel engine using the Second Law of Thermodynamics is still minor. Baldi et al. first studied a chemical tanker and later created an energy and exergy flow profile of a cruise ship operating in the Baltic Sea on measurements from one year (Baldi et al., 2014, 2018). Yao et al. proposed an analysis about the exergy losses inside the marine engine, concluding the combustion process is the biggest source of irreversibilities (Yao et al., 2019). Yao conclusions match with Heywood and Villalta studies about the phenomenology of heat release inside the combustion chamber as both determined peak gas temperature during the combustion process can reach 2500 K (Heywood, 1988; Villalta Lara, 2018). Li et al. studied the effect of the combustion by testing three different engine regimes and executing an energy - exergy analysis in a common rail, high speed engine (Li et al., 2016). Cavalcanti looked into a dual-fuel marine engine used for trigeneration system and analyzed energy, exergy and economic terms (Cavalcanti, 2021). Going further, Tsitsilonis et al. presented a methodology for ship propulsion energy monitoring and management (Tsitsilonis and Theotokatos, 2018) that can be applied to already sailing ships.

Over the years, the heat dissipated through exhaust gases have been reused in economizers to produce steam. Waste heat collected by the fresh water circuit is commonly used to produce technical water in evaporators. This study considered a third source of waste heat: dissipated heat radiated by the engine block and mounted coolers. Several authors have researched about the suitability of engine components for energy saving but no exhaustive studies on engine block low temperature heat have been carried (Butrymowicz et al., 2021; Tsitsilonis and Theotokatos, 2018). In percentage terms this energy source is lower than the other two, but for medium and large marine engines scale factor has to be in mind. Smith et al. isolated a smaller size engine and measured total heat losses to the environment to be 10–15% of the total energy supplied by the fuel (Smith et al., 2009).

A general analysis of the WHR of ship main engine was carried out by Zhemin et al. (Zhemin and Yuxin, 2020). Abdu Ahmed et al. analyzed different WHR systems and how suitable they are for the marine diesel engine, based on an exergy analysis (Abdu et al., 2016). Baldi et al. analyzed the feasibility of WHR systems for a chemical tanker concluding that 5–15% fuel savings can be expected (Baldi and Gabrielii, 2015). Dimopoulos et al. looked at the possibilities of WHR with an exergy analysis combined with a thermo-economic modeling in a marine combined cycle system which main machine was a 2-stroke diesel engine (Dimopoulos et al., 2012). Dere et al. proposed a model which controls cylinder liner temperatures in order to increase engine's efficiency (Dere and Deniz, 2020). Mito et al. used waste heat from scavenging to feed a steam power plant (Mito et al., 2018). Due to the presence of emission reduction systems like SCR and EGR, the amount of exhaust waste heat available has been limited so Kamil compared the waste heat availability among the different technologies in Tier III-compliant engines (Korlak, 2021). Several studies are developing theories around the combination of the diesel engine with an Organic Rankine Cycle (Durmusoglu et al., 2009; Mondal et al., 2020; Ouyang et al., 2020a; Su et al., 2020; Yang and Yeh, 2015). Ouyang et al. extended the studies on supercritical and subcritical Rankine cycles and combined them with the use of supercritical carbon dioxide Brayton and Kalina cycles (Ouyang et al., 2020b, 2021; Su et al., 2020). An extensive review of the WHR methods where also thermoelectric generators are proposed was published by Mohd Noor et al. (Mohd et al., 2015).

Residual heat was classified by Musharavati and Khanmohammadi into three temperature ranges: less than 230 °C, 230–650 °C and over 650 °C. They stated that, at all temperature ranges, energy recovery can be used to generate electric power but, in general, efficiencies are lower at low temperature (Musharavati and Khanmohammadi, 2021). Each WHR system works properly at a defined temperature range: Kalina cycle can work at different ranges if adapted (ammonia-water mixture) and same for the Rankine cycle (ORC) (Musharavati and Khanmohammadi, 2021; Qu et al., 2021) But for the case of more specific systems like the supercritical carbon dioxide Bratyon cycle, it works at its best with heat sources greater than 400 °C (Feng et al., 2020) Other WHR systems like thermoelectric generators show more advantages with low temperature heat, and will become damaged if exposed to temperatures over 230 °C (Marlow Industries, 2015). In order to maximize the recovery of energy at all temperature ranges, combinations of WHR systems like Power Turbine along with classical steam Rankine and Organic Rankine cycles were studied by Qu et al. (2021).

Methods to increase efficiency on marine engines are being studied in order to find three improvements: economic, as less consumption means lower cost per voyage; environmental, since CO2 emissions contribute largely to the greenhouse effect (International Maritime Organization, 2021) and sanitary, as other emissions like CO, SOx and NOx represent a hazard for human health (Lee et al., 2020). Chu Van et al. created a numerical model to investigate performance and exhaust emissions of a cargo vessel's marine engine (Chu Van et al., 2017).

In order to comply with the 2030 Agenda and its requirements, the holistic 4E (energy, exergy, economic and environmental) analysis of thermal engines becomes a priority. Shayesteh et al. determined the optimum parameters of an ORC, that recovers waste heat from ship's main engine and powers a reverse osmosis unit, based on a 4E analysis (Shayesteh et al., 2019). Xu et al. went further and used the 4E analysis to determine the size of the prime mover for its combined cooling, heating and power (CCHP) system (Xu et al., 2021). In general terms, 4E analysis will help enhancing heat recovery, as Musharavati and Khanmohammadi did in order to improve a system composed of a fuel cell and thermoelectric generators (Musharavati and Khanmohammadi, 2021).

This study takes in count the average age of the world merchant fleet and the lack of studies about the waste heat dissipated by the marine engine block and proposes an energy, exergy, economic and environmental method to examine the marine diesel engine, with emphasis on the WHR from the engine block of machines already in use. Real measurements and sampling of working fluids are taken in order to evaluate factual conditions of the engine studied. The analysis identifies and quantifies the amount of heat from the engine block and its attachments which is possible to recover and looks into the cost reduction gained when this energy is reused. Since waste heat released by the block of the case study engine is not greater than 100 °C, WHR systems implemented will need to work properly under conditions imposed by the heat source. A cut in specific fuel consumption is directly reflected on the amount of emissions released and so are calculated.

Moreover, the study provides a theoretical basis for the analysis of in use large marine engines by both First and Second Law of Thermodynamics methods. Results provide theoretical guidance for improving efficiency of marine diesel engines. By decreasing specific energy consumption with engine block waste heat recovery, operating costs and greenhouse gas emissions are reduced.

2. Material and methods

In this section, system performance is discussed in six separate subsections, including governing thermodynamic equations for energy and exergy balance along with economic and environmental analyses.

2.1. Case study vessel

The MV Cristina Masaveu (Fig. 1) has been used as case study vessel. The ship, with 133.50 m length and 8291 GT, was built in 2011 and is propelled by two Wärtsilä 6L32 engines; one of which was used for the research carried out for this paper. Characteristics of the engine were showed in Table 1.

Information related to output power, heat balance and measurements is available from manufacturer's Product Guide and Instruction Manual (Wärtsilä, 2015; Wärtsilä Finland Oy, 2010). Also, the vessel has an energy monitoring system installed which has proved useful in order to monitor data for the study. Energy monitoring system parameters are shown in Table 2:

2.2. Energy balance analysis

First principle of Thermodynamics is used on the analysis. Being the diesel engine an open system means that the law of conservation of energy and mass cannot be applied as there are several flows coming in and out of the device. In order to maximize the observation of the different effects the theory of the Global Energy Balance presented by Carreño (Carreño Arango, 2016), is used.

Energy Balance Analysis is divided into two parts: external and internal. In the first one only the external interactions the engine makes with the environment and its flows (air, fuel, exhaust gas, heat and power) were considered. The engine is seen as a blackbox where flows come in and out. In the second, only internal processes are studied so it can be observed how energy flows into the machine.

For the completion of the Energy Analysis the following assumptions are made:

Fig. 1. MV Cristina Masaveu (Cementos Tudela Veguin, 2011).

Table 1
Engine particulars of the W6L32 used during the research (Wärtsilä, 2015).

Engine particulars (Wärtsilä 6L32)	
Cylinder number	6
Strokes	4
Engine output	3000 kW
Speed	750 rpm
Crankshaft radius	356 mm
Bore	320 mm
Stroke	400 mm
Compression ratio	16.0
Firing order CW	1–5 – 3–6 – 2–4
Max peak pressure	192 bar
Fuel consumption at 100% load	206.093 g/kW h

Table 2
Parameters extracted from ship's energy monitoring system and additional measurements.

Parameter	Units	Device	Accuracy
Fuel flow	g/kW h	VAF PVE005/3001/16	±0.1%
Fuel rack	Mm	TWK IW253/40–0.25 KFN-KHN-A21	±0.25%
Propeller shaft torque	Nm	Coterena ARGOS COT02 Torsionmeter	±1%
Temperatures	°C	3 wire PT-100	±0.2%
Additional temperatures	°C	Fluke 62 MAX	±1.5%

- Pressure in combustion chamber is uniform. The speed of the internal fluids and the flame is lower than the speed of sound (Annand, 1963; Spellman, 2020).
- Gases inside combustion chamber are considered to behave as ideal gases. Lapuerta et al. studied the difference between ideal and real gas (including fuel and water vapour as well as unburnt particles) finding very slight difference (Lapuerta et al., 2006).
- The angle where combustion peak pressure occurs matches with TDC. Since the experimental engine peak pressure is below 200 bar, elastic deformations of the powertrain are not took into account (Martín, 2012). Piston, rings and liner are considered as a rigid solid body that is not affected by any other rotation, torsion or thermal internal deformation (Stanley et al., 1999).
- Heat from combustion process produces a diffuse heat radiation distributed uniformly along all directions (Lopez et al., 2012).
- Lube oil film around piston rings has constant thickness and this is considered as an incompressible fluid (Carreño Arango, 2016; Stanley et al., 1999).
- Exhaust gases deviation to the oil sump is considered to be zero. The blow by effect is a symptom usually found on worn engines. Since large marine diesel engines are monitored monthly trough the reading of peak combustion pressures the level of blow by is minimal. In case some blow by was present, the gas velocity and the short duration of the effect make the process adiabatic.
- Heat dissipated through the turbo-compressor casing is negligible. Energy entering the turbine splits in four forms: work, heat dissipated to the lubrication oil, and to the casing and finally heat still contained into turbine exhaust gas. On top of being a minuscule fraction, commercial turbo-compressor casings are heat insulated so no real heat recovery can be applied without major refits.
- Engine Driven Pumps (EDP's) specific consumptions declared by the manufacturer are considered valid due to the technical difficulties of the experimental measurement (Taraza et al., 2000). EDP's friction the circulating fluid, which slightly increases its enthalpy. This will be dissipated into the fluid's intercooler. The speed of the EDP's is proportional to engine speed (Carreño Arango, 2016).

2.2.1. External energy balance analysis

Fig. 2 shows the subsystems studied on the external energy balance: First Principle of Thermodynamics is applied to the system:

$$\dot{m}_{fuel}LHV = W_{shaft} + \dot{Q}_{cool} + \dot{Q}_{env.LO} + \dot{H}_{exh.gas} + \dot{Q}_{env.CA} + \dot{Q}_{fuel.return} + \dot{Q}_{leaks} + \dot{H}_{unb} + \dot{Q}_{block} \quad (1)$$

where first part of the equation represents the energy from the fuel. The second part contains work, heat and enthalpy flow. W_{shaft} and $\dot{m}_{fuel}$ were measured onboard with a torsion meter and a flow meter respectively. $\dot{Q}_{cool}$ represents heat transfer to the coolant. In large marine diesel engines, dissipated heat will be the sum of two circuits, High Temperature (HT) and Low Temperature (LT) water:

$$\dot{Q}_{cool} = \dot{Q}_{coolHT} + \dot{Q}_{coolLT} \quad (2)$$

$$\dot{Q}_{coolHT} = \dot{m}_{coolHT}Cp_{cool}(T_{coolHT.outlet} - T_{coolHT.inlet}) \quad (3)$$

$$\dot{Q}_{coolLT} = \dot{m}_{coolLT}Cp_{cool}(T_{coolLT.outlet} - T_{coolLT.inlet}) \quad (4)$$

HT and LT water fluids used for cooling the engine contain corrosion inhibitor chemicals. In order to know the specific heat of the mixture a sample is analyzed on the laboratory, by using a DSC calorimeter (Mettler-Toledo TGA/SDTA851).

$\dot{Q}_{env.LO}$ is the fraction of heat liberated into the lubrication oil not dissipated into the LT water circuit but radiated to the environment at the LO cooler:

$$\dot{Q}_{lube.oil} = \dot{Q}_{coolLT.LO} + \dot{Q}_{env.LO} \quad (5)$$

$$\dot{Q}_{lube.oil} = \dot{m}_{lube.oil}Cp_{lube.oil}(T_{lube.oil.outlet} - T_{lube.oil.inlet}) \quad (6)$$

$$\dot{Q}_{coolLT.LO} = \dot{m}_{coolLT.LO}Cp_{coolLT}(T_{coolLT.LO.outlet} - T_{coolLT.LO.inlet}) \quad (7)$$

Specific heat $Cp_{lube.oil}$ value is not normally declared by the oil suppliers. Even if so, this value will be for clean, non used oil. Due to the combustion processes, the oil becomes fouled so this value is obtained through the analysis of a sample took from the engine.

$\dot{H}_{exh.gas}$ represents the enthalpy flow of the outlet exhaust gas at the outlet of the combustion chamber, discounting the enthalpy of the charge air and fuel mixture at the inlet of the combustion chamber:

$$\dot{H}_{exh.gas} = \dot{m}_{exhaust}h^{sensible}_{exhaust} - \left(\dot{m}_{charge.air}h^{sensible}_{charge.air} + \dot{m}_{fuel}h^{sensible}_{fuel}\right) \quad (8)$$

$$h^{sensible}_{exhaust} = \int_{T_0}^{T_{exhaust}} Cp_{exhaust}dT$$
$$h^{sensible}_{charge.air} = \int_{T_0}^{T_{charge.air}} Cp_{charge.air}dT \quad (9)$$
$$h^{sensible}_{fuel} = \int_{T_0}^{T_{fuel}} Cp_{fuel}dT$$

Specific heat $Cp_{exhaust}$ value is obtained if the composition of the exhaust gas is known, as an addition of the specific heat of its species. In this study, as the composition is not known, the value determined by Koshy et al. $Cp_{exhaust} = 1.185$ kJ/kg K (Koshy, 2015) is used. The term dT represents the differential of the variable T (temperature) while the integration is carried out between ambient temperature ($T_0 = 298$ K) and the temperature of each fluid (exhaust gas, charge air and fuel).

Moreover $\dot{Q}_{env.CA}$ is the amount of heat liberated into the air cooler casing. Although much of the heat is dissipated into LT water circuit, there is a part radiated to the environment:

$$\dot{Q}_{charge.air} = \dot{Q}_{coolLT.CA} + \dot{Q}_{env.CA} \quad (10)$$

$$\dot{Q}_{charge.air} = \dot{m}_{charge.air}Cp_{charge.air}\left(T_{charge.air.outlet} - T_{charge.air.inlet}\right) \quad (11)$$

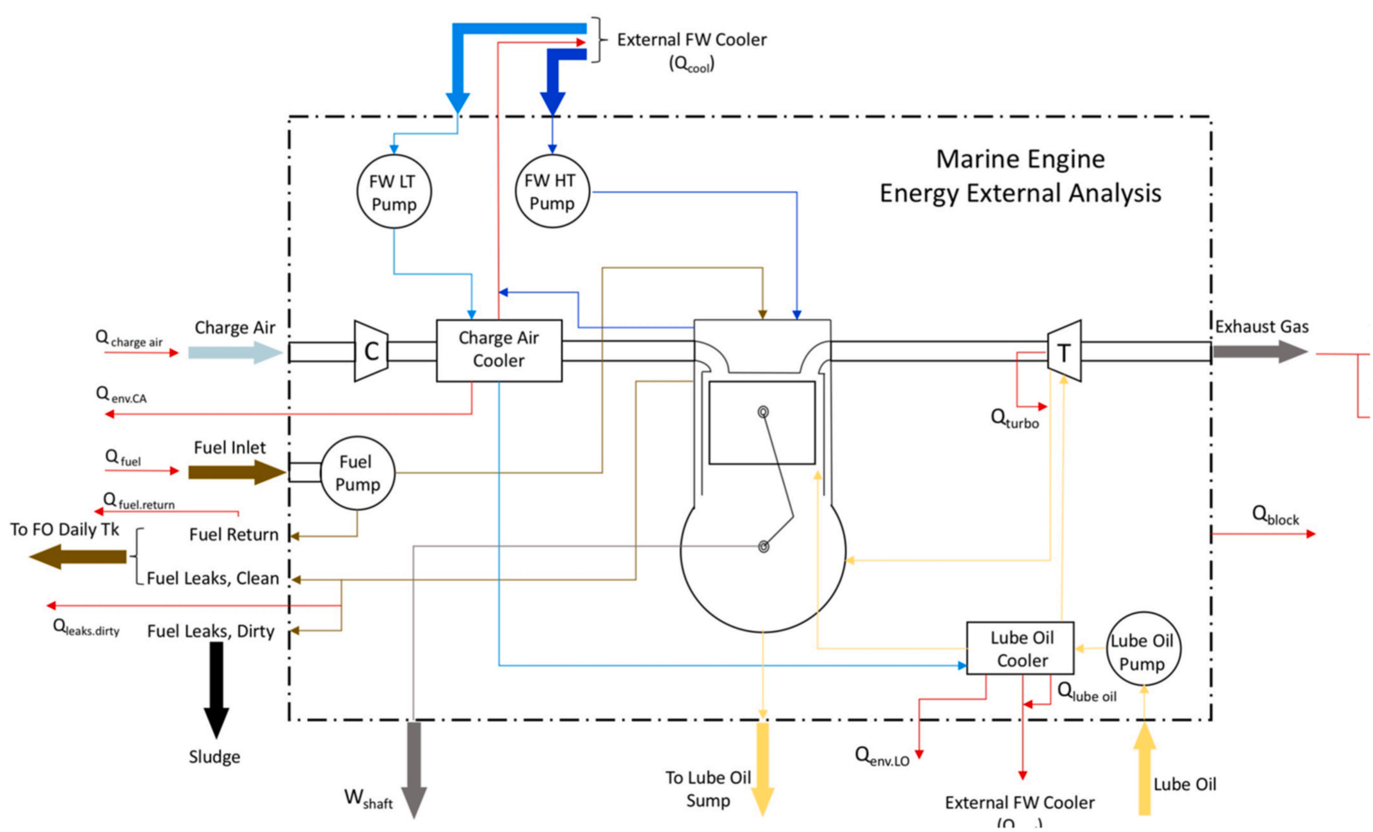

Fig. 2. Diesel engine schematics and the application of the External Energy Balance.

$$\dot{Q}_{coolLT.CA} = \dot{m}_{coolLT.CA} Cp_{cool} (T_{coolLT.CA.outlet} - T_{coolLT.CA.inlet}) \quad (12)$$

$\dot{Q}_{fuel.return}$ as a high fraction of the fuel supplied to the engine returns to the tank, the method accounts for the heat dissipated into the returning fuel. As this fuel passes through the engine it does take heat out of it, mainly from piping and injection pumps:

$$\dot{Q}_{fuel.return} = \dot{m}_{fuel.return} Cp_{fuel} \left(T_{fuel.return.} - T_{fuel.inlet.engine}\right) \quad (13)$$

Q_{leaks} large marine engines have fuel leaks that need to be considered. Clean Leaks are drained from injection pumps and injection valves and the fuel can be reused so they are sent back to the daily tanks. Dirty Leaks are unforeseen drains that cannot be reused due to possible contamination. These are treated as sludge so, in the end, there is energy wasted:

$$\dot{Q}_{leaks.dirty} = \dot{m}_{leaks.dirty} Cp_{fuel} \left(T_{leaks.dirty} - T_{fuel.inlet.engine}\right) \quad (14)$$

$\dot{H}_{unb}$ represents the energy losses due to incomplete combustion. The fraction of unburnt material varies during the operation of the engine and so does the lost energy (Carreño Arango, 2016; Rakopoulos et al., 2009):

$$\dot{H}_{unb} = (Y_{HC} LHV_{HC} + Y_{CO} LHV_{CO} + Y_C LHV_C) \dot{m}_{exhaust} \quad (15)$$

where heating values of HC, CO and C (soot) are 42900 kJ/kg, 10100 kJ/kg and 32800 kJ/kg respectively.

$\dot{Q}_{block}$ represents the heat transfer from the engine to the environment that happens out of the already studied sources (fluid coolers and fuel drains). After those, the engine block results to be the main source of waste heat:

$$\begin{aligned} \dot{Q}_{block} &= \dot{m}_{fuel} LHV - W_{shaft} - \dot{Q}_{cool} - \dot{Q}_{env.LO} \\ &- \dot{H}_{exh.gas} - \dot{Q}_{env.CA} - \dot{Q}_{fuel.return} - \dot{Q}_{leaks} - \dot{H}_{unb} \end{aligned} \quad (16)$$

2.2.2. Internal energy balance analysis

Fig. 3 shows the subsystems studied on the internal energy balance:

Internal processes inside the engine are observed and First Principle of Thermodynamics is applied:

$$\begin{aligned} \dot{m}_{fuel} LHV &= W_{shaft} + \left(L_{friction} + L_{EDP}\right) + \dot{Q}_{cham} \\ &+ \dot{Q}_{env} + \dot{H}_{exh.gas} + \dot{Q}_{env.CA} + \dot{H}_{unb} \end{aligned} \quad (17)$$

$L_{friction}$ represents energy lost by friction mechanisms. The considered ones are: piston rings – cylinder liner, bearings and camshaft – valves (Tormos et al., 2018):

$$L_{friction} = L_{fr.pis} + L_{fr.bear} + L_{fr.valv} \quad (18)$$

$L_{fr.pis}$ heat will be transferred to the cylinder liner and dissipated into the HT water circuit.$L_{fr.bear}$ and $L_{fr.valv}$ heat will be dissipated into the lubrication oil.

On this study, a torsion meter is used to measure the torque produced on the propeller shaft which is connected via a gearbox with the engine crankshaft. By knowing shaft radius, gearbox efficiency, starting air pressure used and the area where the air is applied, $L_{friction}$ can be estimated:

$$P_{starting.air} \cdot S_{cylinder} = \left(\frac{M_{shaft}}{radius_{shaft}}\right) + L_{friction} \quad (19)$$

$$L_{friction} = \left(P_{starting.air} \cdot S_{cylinder}\right) - \left(\frac{M_{shaft}}{radius_{shaft}}\right) \quad (20)$$

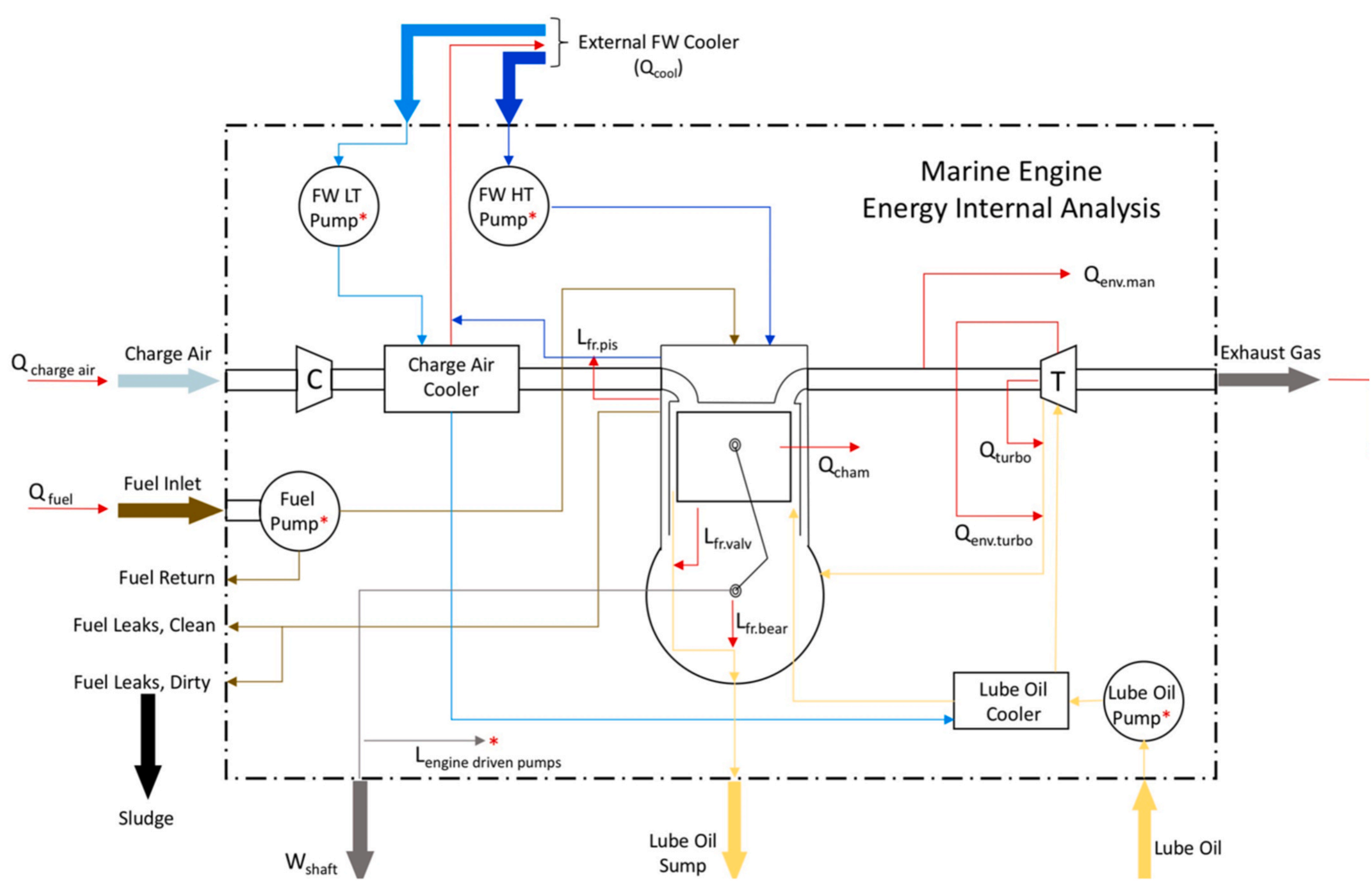

Fig. 3. Diesel engine schematics and the application of the Internal Energy Balance.

On this measurement,$L_{friction}$ contains not only friction energy but also the work dedicated to move the engine driven pumps.

In case $L_{friction}$ was estimated by any other method,L_{EDP} is accounted as follows:

$$L_{EDP} = L_{LO.pump} + L_{pumpHT} + L_{pumpLT} + L_{fuel.pump} \tag{21}$$

L_{EDP} is the energy needed to move the engine driven pumps (Lube Oil, HT and LT water and Fuel Oil). Engine's manufacturer provides the data related to the energy consumption of engine driven pumps on the Test Record document, in the form of g/kW h. By knowing the Lower Heating Value of the fuel used, the required work is calculated.

$\dot{Q}_{cham}$ represents the heat transfer between combustion chamber and cylinder liner walls. The heat transferred is ultimately dissipated into the HT water circuit. It is assumed that combustion, heat release and transfer from the gas to the cylinder walls occur simultaneously:

$$\dot{Q}_{cham} = Ah(T_{gas} - T_{wall}) \tag{22}$$

where A is the total area of exposed cylinder walls,h is the heat transfer coefficient formulated by Woschni (Dolz Ruíz, 2011; Heywood, 1988; Woschni, 1967):

$$h = 3.26B^{-0.2}p_{cham}^{0.8}w^{0.8}T_{gas}^{-0.53} \tag{23}$$

Being B cylinder bore,p_{cham} combustion chamber pressure,w average gas speed inside the cylinder andT_{gas} the average gas temperature inside the cylinder during the combustion process, which can be estimated if mixture is considered as an ideal gas (Lapuerta et al., 2006; Payri and Desantes, 2011). In this case, combustion pressure diagrams were taken onboard with a PREMET-C pressure indicator.

$\dot{Q}_{env}$ composed of$\dot{Q}_{env.manifold}$ and$\dot{Q}_{env.tc.casing}$, represents the heat transfer from the manifold and the turbo casing to the environment, respectively. In practice, the heat radiated by these surfaces is not really useable as SOLAS Reg. II-2/15.2.10 requires that all surfaces with temperatures above 220 °C, which may be impinged as a result of a fuel oil, lubricating oil and other flammable oil system failure be properly insulated (International Maritime Organization, 2009). This means no calculation will be carried out $\dot{Q}_{env} = 0$, since the dissipated heat to the environment is minimal (Saint-Gobain Marine Applications, 2018).

2.3. Exergy balance analysis

Quality of energy is defined as how capable is the energy to produce changes in a system (Kotas, 1985). Depending on its quality, the energy can be used up to 100% (totally reversible) or less (when contains irreversibilities). An irreversible process takes the non useable percentage of energy. Exergy is the standard used to compare the ratio of useable energy vs. maximum theoretical work.

2.3.1. Environment and dead state

For the exergy balance analysis it is needed to define the state of the studied system prior to the examination as environmental conditions not only depend on the engine but also on external factors as the location of the vessel and therefore the forecast. Ambient conditions according to ISO 15550 (Ambient temperature 25 °C [298.15 K], Pressure 100 kPa, LT water temperature 25 °C [298.15 K], Relative Humidity 30%) (ISO, 2016) have been used for this study.

2.3.2. Exergy balance analysis of the diesel engine

Exergy balance analysis was divided into two sections: first, interactions with the environment are considered. While the second studies the inner processes that occur between the different parts inside the engine. The following assumptions were made:

- Pipe head loss in piping and manifolds are negligible.
- No other heat losses apart from the ones studied occur so no other heat transfers are done.
- Gas mixture inside the combustion chamber behaves as an ideal gas.
- There is no exhaust gas recirculation.
- Non elastic deformations are inexistent.
- Magnetic and polarization does not occur or does not have an affect on the engine.

2.3.3. Global exergy balance analysis

The engine is studied as a black box which exchanges mass and energy with the environment. On top of the already related ambient conditions, final composition of the gas has to be as shown in Table 3:

The engine is fed by fuel and air in order to obtain power at the output but also thermal losses:

$$\dot{Ex}_{fuel} + \dot{Ex}_{charge.air} = \dot{W}_{shaft} + \dot{E}_{losses} \tag{24}$$

$$\dot{Ex}_{fuel} = \dot{Ex}_{phys} + \dot{Ex}_{chem} \tag{25}$$

Since the case study engine uses diesel fuel, Ex_{fuel} is represented by Gibbs free energy,μ_i:

$$\mu_i = Ex_{fuel} \tag{26}$$

In the case of marine power plants, fuel is usually preheated (from dead state temperature, T_0) before injection so there is some thermal availability in the form of physical exergy. But since temperature difference is minor and the increment on the fuel exergy results negligible (not greater than 0.2% of chemical exergy) this term is considered to be non-existent (Rakopoulos and Giakoumis, 2006).

$$Ex_{phys} = (h - h_0) - T_0(s - s_0) \simeq 0 \tag{27}$$

$$Ex_{chem} = \left(1.0401 + 0.1728\frac{h}{c} + 0.0432\frac{o}{c} + 0.2169\frac{s}{c}\left[1 - 2.0628\frac{h}{c}\right]\right)*LHV \tag{28}$$

being h, c, o y s the mass fraction of hydrogen, carbon, oxygen and sulphur. For diesel fuels with a formula of the type $C_{14.4}H_{24.9}$, the expression presented by Stepanov (1995) is used:

$$\frac{Ex_{chem}}{LHV} = 1.0699 \tag{29}$$

Exergy from compressed charge air is:

$$\dot{Ex}_{charge.air} = \left(1 - \frac{T_0}{T_{charge.air.inlet}}\right)\dot{Q}_{charge.air} \tag{30}$$

From Equation (24), $\dot{E}_{losses}$ are obtained:

$$\dot{E}_{losses} = \left(\dot{Ex}_{fuel} + \dot{Ex}_{charge.air}\right) - \dot{W}_{shaft} \tag{31}$$

$$\dot{E}_{losses} = \dot{Ex}_{outlet} + \dot{I} \tag{32}$$

$\dot{E}_{losses}$ is composed of$\dot{Ex}_{outlet}$ which represents the energy flow which is not work on the shaft but can be recovered (mainly heat from exhaust gas, water and radiation from the engine block) while$\dot{I}$ gathers the irreversible forms (combustion and mixture irreversibilities).

Table 3
Chemical composition of the reference ambient on Dead State (Rakopoulos and Giakoumis, 2006).

Species	Percentage
Nitrogen (N_2)	75.67
Oxygen (O_2)	20.35
Hydrogen (H_2)	3.03
Carbon dioxide (CO_2)	0.03
Others	0.92

2.3.4. Local exergy balance analysis

The exergy flows in a large marine diesel engine are:

Fig. 4 shows the subsystems studied on the local exergy balance:

The exergy balance composed of the previously listed flows can be expressed as:

$$\begin{aligned} E\dot{x}_{fuel} + E\dot{x}_{charge.air} = \dot{W}_{shaft} + E\dot{x}_{cool} + E\dot{x}_{env.LO} + E\dot{x}_{exh.gas} \\ + E\dot{x}_{env.CA} + E\dot{x}_{block} + \dot{I}_{comb} + \dot{I}_{friction} + \dot{I}_{mixture} \end{aligned} \tag{33}$$

$E\dot{x}_{cool}$ represents the exergy contained on HT and LT water circuits:

$$E\dot{x}_{cool} = E\dot{x}_{coolHT} + E\dot{x}_{coolLT} \tag{34}$$

$$E\dot{x}_{coolHT} = \left(1 - \frac{T_0}{T_{coolHT.inlet}}\right)\dot{Q}_{coolHT} \tag{35}$$

$$E\dot{x}_{coolLT} = \left(1 - \frac{T_0}{T_{coolLT.inlet}}\right)\dot{Q}_{coolLT} \tag{36}$$

$E\dot{x}_{env.LO}$. The majority of the heat dissipated from the engine internal processes into the lubrication oil is at the same time dissipated to the LT water in the LO cooler. Despite this, the intercooler radiates heat to the ambient so an extra heat fraction must be considered:

$$E\dot{x}_{env.LO} = \left(1 - \frac{T_0}{T_{LO.inlet}}\right)\dot{Q}_{env.LO} \tag{37}$$

$E\dot{x}_{exh.gas}$ is the dissipated heat from the engine to the environment. While the machine is operating in a stationary manner, thermodynamic properties can be measured from temperature and pressure sensors mounted on the engine:

$$E\dot{x}_{exh.gas} = \left(1 - \frac{T_0}{T_{exh.gas.afterTC}}\right)\dot{Q}_{exh.gas} \tag{38}$$

$$\dot{Q}_{exh.gas} = \dot{m}_{exh.gas} Cp_{exh.gas}\left(T_{exh.gas.afterTC} - T_0\right) \tag{39}$$

$E\dot{x}_{env.CA}$ heat dissipated from compressed charge air ends up mostly on the LT water. But a fraction of the waste heat is radiated to the ambient:

$$E\dot{x}_{env.CA} = \left(1 - \frac{T_0}{T_{charge.air.inlet}}\right)\dot{Q}_{env.CA} \tag{40}$$

$E\dot{x}_{block}$. Heat radiation from the block, previously considered as an irreversibility (Özkan et al., 2013), is studied as an exergy process where energy can be recovered:

$$E\dot{x}_{block} = \left(1 - \frac{T_0}{T_{block}}\right)\dot{Q}_{block} \tag{41}$$

$\dot{I}_{comb}$ is a process that involves significant temperature changes and therefore entropy generation. Rakopoulos, Baldi and other authors stated the major exergy destruction on a diesel engine is produced on the combustion process (Baldi et al., 2018; Paul et al., 2017; Rakopoulos and Giakoumis, 2006; Razmara et al., 2016):

$$\dot{I}_{comb} = \left(1 - \frac{T_0}{T_{comb}}\right)\dot{Q}_{cham} \tag{42}$$

$\dot{I}_{friction}$ represents the exergy destructed by friction processes. Chen and Flynn presented a correlation used to obtain FMEP (friction mean effective pressure) (Heywood, 1988):

$$FMEP = c_1 + c_2 P_{max} + c_3 U_p + c_4 U_p^2 \tag{43}$$

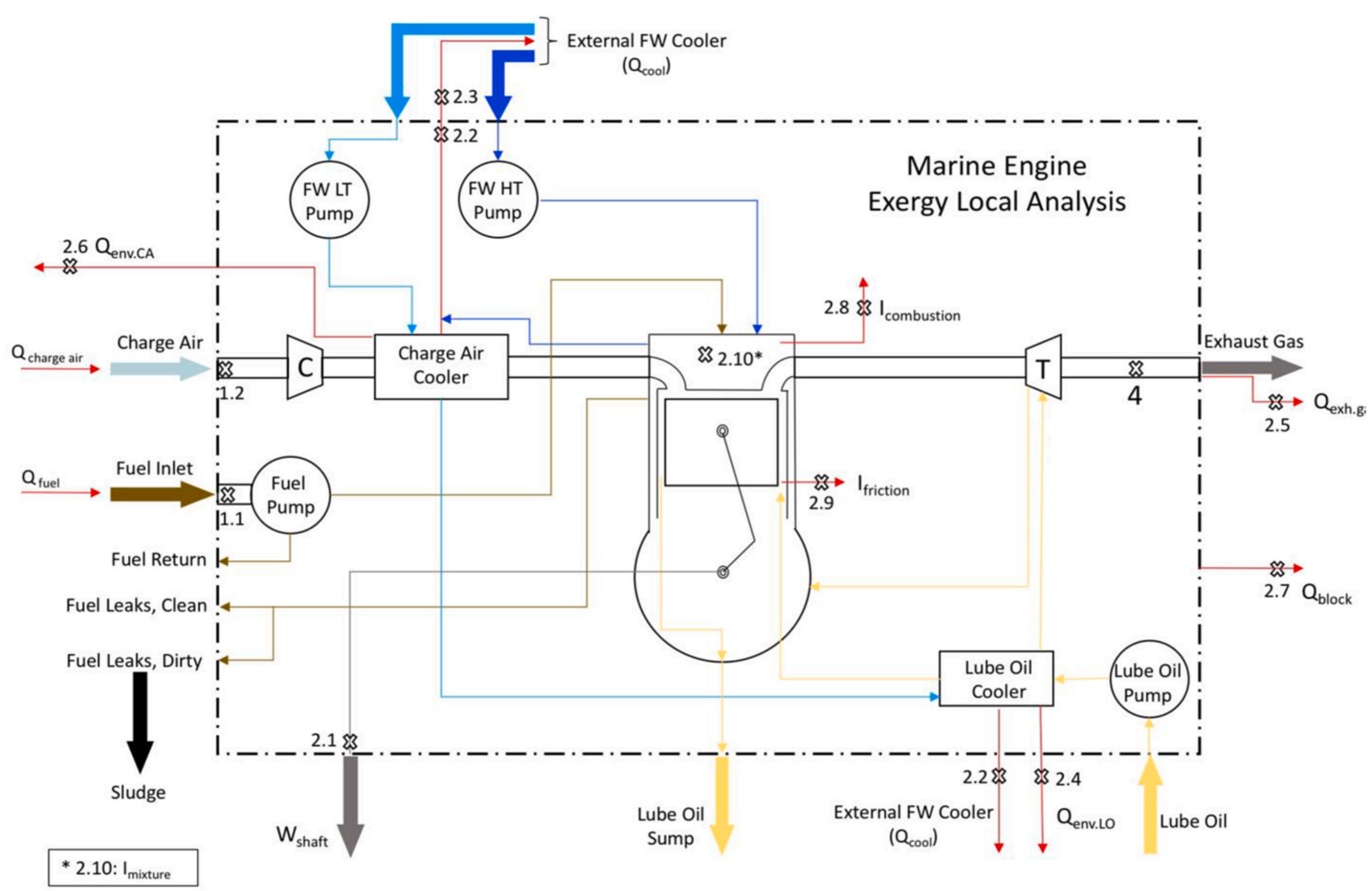

Fig. 4. Diesel engine schematics and the application of the Local Exergy Balance.

Being P_{max} the cylinder peak pressure and U_p^2 the mean piston speed. The terms c_1, c_2, c_3 and c_4 can be determined with experimental data. Lupul obtained the following values (Lupul, 2008):

$$c_1 = 0.20; c_2 = 0.004; c_3 = 0.007; c_4 = 0.0008 \tag{44}$$

Once FMEP is known, it is multiplied by the displaced volume:

$$\dot{I}_{friction} = FMEP \cdot V_d \tag{45}$$

$I_{mixture}$ during the mixture process both diesel and air enter in contact. This leads to two subprocesses, the chemical as they are heterogeneous substances and the physical one due to the difference in temperature and pressure of the substances. Assuming the chemical subprocess is negligible due to most of the chemistry is done in combustion and using the Gouy-Stodola method for the mixing with heat transfer (Kotas, 1985):

$$\dot{I}_{mixture} = T_0\left(\dot{m}_{charge.air}\left[s_{final} - s_{charge.air}\right] + \dot{m}_{fuel}\left[s_{final} - s_{fuel}\right] - \frac{\dot{Q}_{env}}{T_{env}}\right) \tag{46}$$

where T_{env} corresponds to the temperature of cylinder walls at the moment the mixture happens.

2.4. System performance

The following performance indicators are used to evaluate the efficiency of the engine:

- Exergy efficiency of the engine is shown as the ratio between the work extracted from the shaft as mechanical power and the total exergy provided by the fuel.

$$\varepsilon = \frac{\dot{W}_{shaft}}{\dot{Ex}_{fuel}} \tag{47}$$

$$\lambda = \frac{\dot{I}_{total}}{\dot{Q}_{fuel}} \tag{48}$$

2.5. Economic analysis

The economic analysis in this research is carried out using the specific exergy costing method, known as SEPCO (Moran and Shapiro, 2005; Seshadri, 1996; Valero et al., 2006). Once the exergy flows from and to the engine have been evaluated, the cost balance equation is applied:

$$\sum_0 \dot{C}_o + \dot{C}_{work} = \dot{C}_{fuel} + \sum_i \dot{C}_i + \dot{Z} \tag{49}$$

which in the case of the marine engine can be expressed as:

$$\dot{C}_{work} + \dot{C}_{exhaust} + \dot{C}_{heat} = \dot{C}_{fuel} + \dot{Z} \tag{50}$$

where $\dot{C}$ represents the cost flow rate in €/h and $\dot{Z}$ is the sum of capital investment and maintenance cost incurred.

$$\dot{Z} = \frac{PEC \cdot CRF \cdot \Phi}{N} \tag{51}$$

PEC is the purchase equipment cost in euro (which for the case study engine was 800000 euro) and *CRF* stands for Capital Recovery Factor, calculated as:

$$CRF = \frac{j \cdot (1+j)^n}{(1+j)^n - 1} \tag{52}$$

being j the interest rate (12%) and n the service life of the components (estimated in 20 years). N is the annual running hours (5000 running hours per year on the case study vessel) and Φ is the maintenance factor, considered to be 1.06 (Khanmohammadi and Azimian, 2015; Sayyaadi and Sabzaligol, 2009).

The cost flow $\dot{C}$ is obtained:

$$\dot{C} = c\dot{Ex} \tag{53}$$

$$c_w \dot{W}_{shaft} + c_{exh.gas}\dot{H}_{exh.gas} + c_{heat}\dot{Ex}_{heat} = c_{fuel}\dot{Ex}_{fuel} + \dot{Z} \tag{54}$$

where c is the cost per unit of exergy, in euro per kW h. $\dot{Ex}$ represents the exergy transfer rate.

Since the energy of the fuel is used in order to obtain work on the shaft and both non recovered heat and exhaust enthalpy are byproducts, it is assumed the cost unit is the same for the three of them: $c_{fuel} = c_{exh} = c_{heat}$ (55)where the energy of a liter of the fuel, in kW, can be obtained from its LHV and density. At the time of the test, the price of the fuel used was 0.5 euro per liter. Then, the cost per unit of exergy of the work on the shaft can be obtained:

$$c_w = \frac{c_{fuel}\dot{Ex}_{fuel} + \dot{Z} - c_{heat}\dot{Ex}_{heat} - c_{exh.gas}\dot{H}_{exh.gas}}{\dot{W}_{shaft}} \tag{56}$$

Also, heat recovered from engine block and coolers is a non specifically desired byproduct so its cost unit is the same as the work on the shaft.

$$c_{w.whr} = c_w = c_{ambient} \tag{57}$$

If heat radiated by engine block along with lube oil and charge air coolers is recovered, then Equation (53) becomes:

$$c_w \dot{W}_{shaft} + c_{ambient}\dot{C}_{ambient} = c_{fuel}\dot{Ex}_{fuel} + \dot{Z} - c_{exh.gas}\dot{H}_{exh.gas} - c_{heat}\dot{Ex}_{heat} \tag{58}$$

$$c_{w.whr}\left(\dot{W}_{shaft} + \dot{C}_{ambient}\right) = c_{fuel}\left(\dot{Ex}_{fuel} - \dot{H}_{exh.gas} - \dot{Ex}_{heat}\right) + \dot{Z} \tag{59}$$

$$c_{w.whr} = \frac{c_{fuel}\left(\dot{Ex}_{fuel} - \dot{H}_{exh.gas} - \dot{Ex}_{heat}\right) + \dot{Z}}{\left(\dot{W}_{shaft} + \dot{C}_{ambient}\right)} \tag{60}$$

2.6. Environmental analysis

One of the major techniques to impede the unwanted environmental effects of energy conversion processes, is to focus on fuel consumption decrement. The fact that internal combustion engines use a thermochemical reaction to produce work involves a transformation of the source. From the combustion of the diesel fuel, CO_2 emissions are released into the environment. Carbon dioxide emissions from the combustion of the diesel fuel are in the range of 73.3–75 g CO_2 per MJ (Edwards et al., 2011; IPCC, 2006).

Also, manufacturer of the case study engine states NO_x emissions are 10 g/kW h in engine's Test Report. An increment on the power, either thermal or mechanical, extracted from the engine will be of advantage to the specific emissions.

Waste heat dissipated from the engine block and accessories is in the end combusted diesel non used for any purpose. By recovering the useable heat, specific emissions can be reduced as more power is extracted from the same quantity of fuel.

3. - Results and discussion

Prior to the findings, study is contextualized with fuel rack and consumption measurements taken onboard the case study vessel. Fuel rack position was chosen as it is the clearest indicator of the engine load from those monitored onboard. Fig. 5 shows the distribution of the different positions of the fuel rack of the engine during running times in 2020.

From Fig. 5, it can be appreciated the range of loads the engine operates most of the time is 75–80% Maximum Continuous Rating (MCR). Table 4 summarizes the operating hours percentage at the different fuel racks and the time at that fuel rack range in a year period

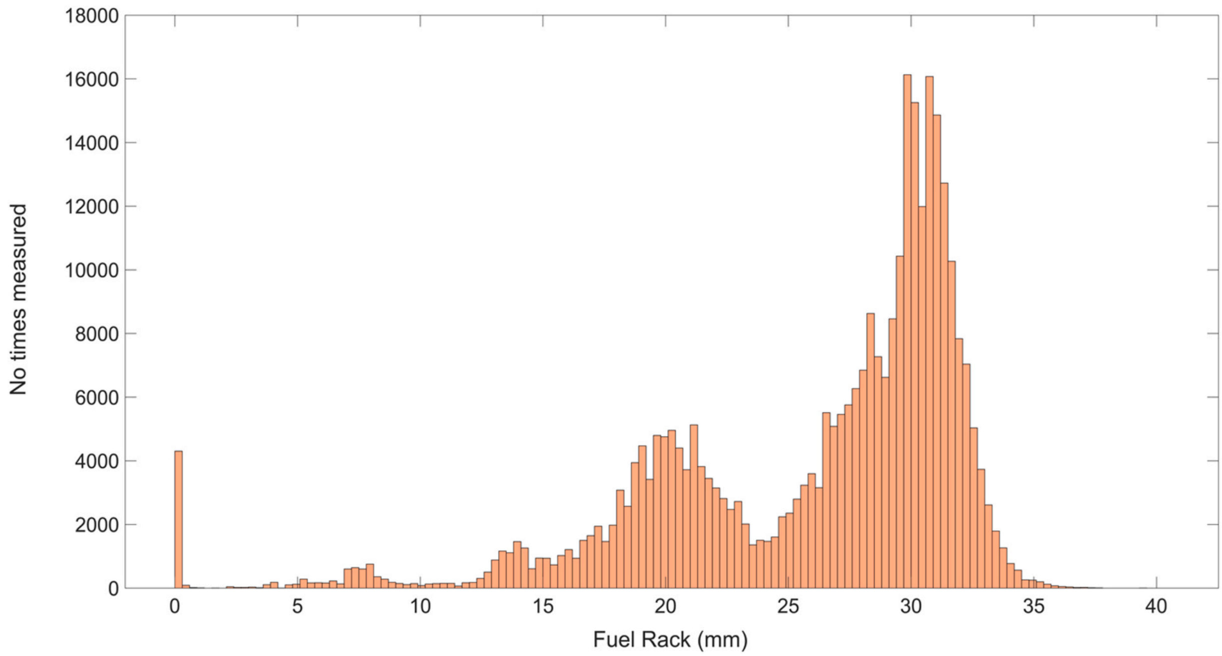

Fig. 5. Fuel rack distribution of the studied engine while running, from data collected in 2020.

Table 4
Results from the Energy analysis of the case vessel.

Fuel rack (mm)	Operating hours percentage	Operating hours
0–5	1.366	68.296
5–10	0.921	46.048
10–15	2.470	123.479
15–20	12.337	616.839
20–25	15.288	764.384
25–30	32.710	1635.486
30–35	34.909	1745.465

(5000 operating hours).

In Fig. 6, a relation of fuel rack position and specific consumption in grams per kilowatt-hour is presented.

3.1. Energy Analysis

Chemical energy from fuel injected into the engine contains energy is first subdivided in two terms: mechanical and thermal energies. The first one produces work on the shaft, commonly used for propulsion or electric generation purposes. Energy released as heat is dissipated, mainly into the cooling water circuit. Other sources of heat from combustion, compressed charge air or lube oil temperature increment are recovered in the cooling water as well. Contributions from other sources like fuel returning to the tank and dirty leaks are minor (1.5100–3.1565

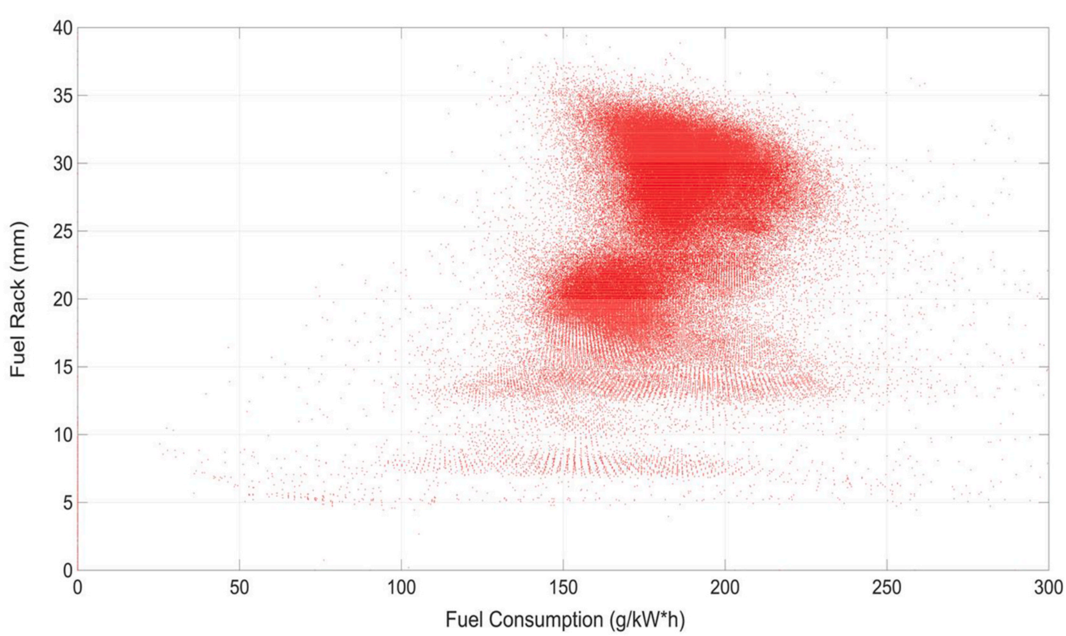

Fig. 6. Fuel rack and consumption distribution of the studied engine.

and 0.0047–0.0094 kW, respectively).

Heat radiated to the ambient comes from lube oil cooler casing, charge air cooler casing and engine block. The two first sources are cooled by the LT cooling water circuit but that does not remove the heat completely, a fraction is released into the ambient. Results show Lube Oil cooler dissipates 3.68–6.09% of the energy contained into the fuel. Charge Air cooler dissipates 6.16–6.35% of the energy contained into the fuel. The Engine Block takes a fraction of heat from several sources like combustion, friction, lube oil and fuel oil. Heat dissipated by the Block to the ambient is on the range of 4.99–5.95% of the energy supplied. If compared with the power extracted from the shaft the percentage of energy dissipated by the Engine Block is 12.30–15.28%. The sum of the three sources gives the total energy radiated to the ambient and represents 14.93–18.37% of the energy initially supplied by the diesel fuel. Despite being a small fraction, the scale factor plays an important role as in the test engine the radiated heat exceeds 290 kW in all the loads where the test was carried. Table 5 summarizes the results obtained of each parameter at the different output rates:

Fig. 7 shows the results of the energy balance analysis at different loads.

The largest energy loss occurs through the exhaust gases, which represent 32.25–34.86% of the total power, followed by the cooling water with 9.27–12.59%. The rest of the waste heat is dissipated through the lubrication oil and radiated heat.

Fig. 8 shows the amount of heat released to the ambient in comparison with the work generated on the shaft, at the loads measured. As it can be seen the main purpose of the system, the conversion of chemical into mechanical energy, is larger than the heat radiated to the ambient. Although the percentage of recoverable energy is low, large marine engines play with the scale factor as work on the shaft can vary from a thousand to tens of thousand kilowatts.

3.2. Exergy analysis

Exergy analysis has been performed at the same distribution of loads. Exergy is supplied to the engine from the diesel fuel in its majority, with a minor contribution of the charge air heat. Table 6 summarizes the results obtained of each parameter at the different output rates. It can be observed that a large part of the irreversibilities present are located in some specific parts of the system, like the case of the combustion, which represents 25.44–26.71% of the total exergy supplied. Exergy from the heat dissipated into the ambient accounts for 2.97–3.31% in total.

From the point of view of the exergy losses, exhaust gas represent the biggest source of exergy lost. A 38.36–42.89% of recoverable energy goes as exhaust gas waste heat. Part of this heat is already used but there is a fraction that cannot be recovered due to the cold corrosion issue as sulphur is still present in marine diesel fuels. The sum of exergies from engine block, lube oil and charge air coolers represent a 2.98–3.31% of the total exergy supplied by the fuel and a 3.90–4.21% of the recoverable energy. Results of the exergy balance are presented in Fig. 9.

Dissipated exergy is candidate to be reused in order to maximize the efficiency of the system. Fig. 10 shows the amount of exergy from waste heat released into the ambient in comparison with the work generated on the shaft, at the loads measured. It can be seen that the larger term corresponds to the exhaust gas waste heat, followed by the cooling water LT and HT circuits. Exergy from the heat radiated into the environment from the engine block represents a minor term but is still recoverable and so is the heat radiated by both lube oil and charge air coolers.

Table 5
Results from the Energy analysis of the case vessel.

Load (% MCR)	75	80	100
External Energy Balance (kJ)			
Fuel energy	5672.4126	6163.3000	7402.1750
Cooling	714.5437	571.6350	714.5437
Lube Oil	542.2766	709.1309	709.1309
Lube Oil cooler to ambient	208.8229	375.6772	280.4047
Exhaust Gas	1838.5817	1988.2508	2580.8821
Charge Air	681.1241	709.5042	821.2134
Charge Air cooler to ambient	361.3641	389.7442	455.7734
Fuel return to tank	3.1565	1.5589	1.5100
Fuel dirty leaks	0.0094	0.0047	0.0047
Unburnt matter	65.1416	69.6294	84.8609
Engine Block	295.9343	366.7998	369.0563
Internal Energy Balance (kJ)			
Friction	130.2485	123.4863	119.1763
Engine Driven Pumps	80.8125	74.7067	79.0167
Combustion chamber	1918.9724	1999.8842	2325.6322

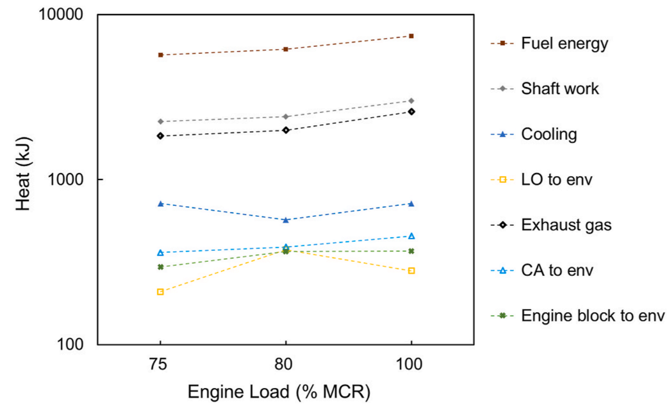

Fig. 7. Relationship between Energy distribution and load.

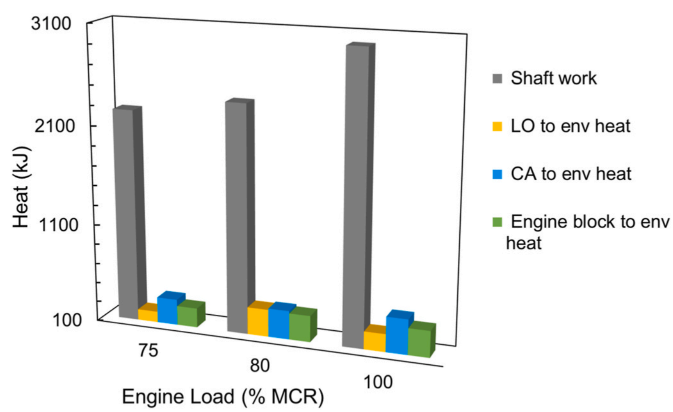

Fig. 8. Relationship between Work on the Shaft and Waste Heat released to the ambient.

Table 6
Results from the Exergy analysis of the case vessel.

Load (% MCR)	75	80	100
Global Exergy Balance (kJ)			
Fuel	6068.9142	6594.1147	7919.5871
Charge Air	243.5993	253.7492	293.7012
Losses	4062.5134	4447.8639	5213.2883
Local Exergy Balance (kJ)			
Cooling	67.9027	56.3185	67.4755
Lube Oil cooler to ambient	27.3846	47.3520	36.0596
Exhaust Gas	1099.7058	1170.6635	1259.0679
Charge Air cooler to ambient	129.2393	139.3893	163.0042
Engine Block	27.2186	31.4275	36.7775
Combustion	1608.1935	1761.3781	2015.3513
Friction	0.0020	0.0021	0.0025
Mixture	38.3342	44.8340	53.7633

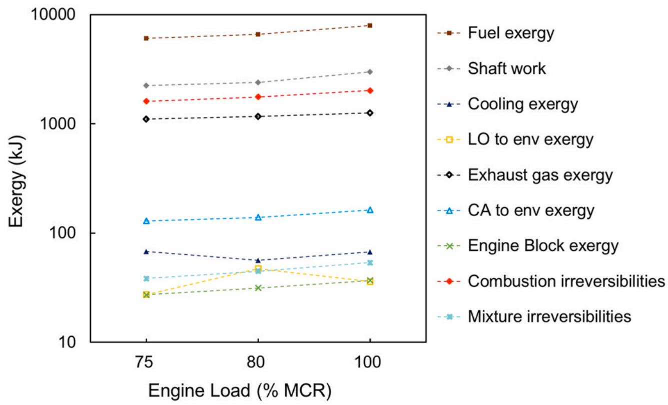

Fig. 9. Relationship between Exergy distribution and load.

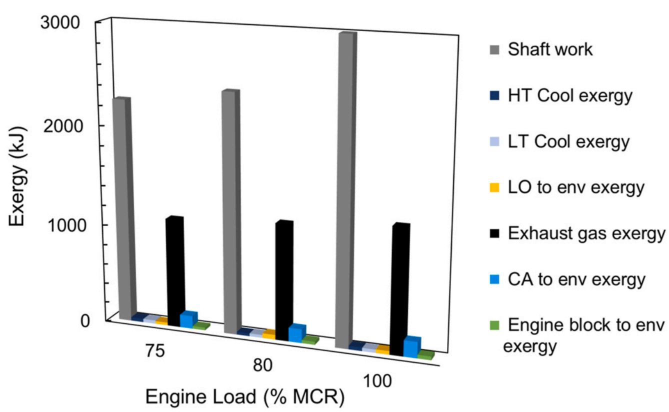

Fig. 10. Relationship between Work on the shaft and waste Exergy.

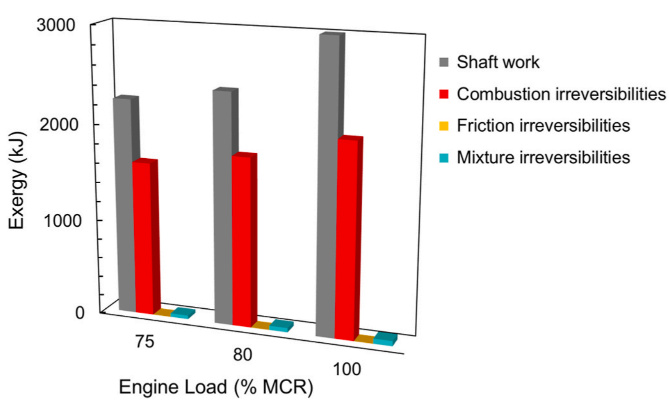

Fig. 11. Relationship between Work on the shaft and engine Irreversibilities.

On the contrary, Fig. 11 shows engine irreversibilities in comparison with the work generated on the shaft, at the loads measured. It can be observed that the largest irreversibility is due to the combustion process, contributing in a minor form friction and mixture processes.

If relationships between the work on the shaft, exergy and irreversibilities are considered, it can be appreciated, as shown in Fig. 12, that each one of the terms takes almost a third of the energy supplied by the fuel.

3.3. System performance

An engine system will be more efficient with high exergy efficiency and lower irreversibilities efficiency. Table 7 presents the values obtained at the different measured loads.

Results from the sum of the power extracted from the shaft and irreversibilities efficiencies range from 68.24 to 68.69% being the remainder the exergy present in the waste heat.

First, the relationship between work and fuel exergy is desired to be as high as possible since the higher the mechanical work extracted from the same amount of fuel the higher efficiency what will affect operation cost and CO_2 emissions. On the other hand, the irreversibilities and heat from the fuel ratio is wanted to be as low as possible as that is a fraction of energy that cannot be reused.

3.4. Economic analysis

The use of waste energy produces an increment of the total energy available for a specific amount of fuel. Therefore, the cost of using the machine drops accordingly to the recovery. Table 8 summarizes a comparison of the results when using engine block waste heat and the classical application. Case study vessel operates 5000 h per year, an average of 13.69 h per day at 80% MCR. In those conditions a cut in 8.33% of the daily cost is produced while recovering the radiated waste heat.

3.5. Environmental analysis

Along with a cost reduction, the use of the waste heat dissipated into the environment brings a reduction in specific emissions as more power is extracted from the same amount of fuel. Table 9 accounts for the CO_2 and NO_x specific emissions saved while recovering the exergy related to waste heat dissipated into the ambient at the different loads studied in section 2.3. From the results a CO_2 reduction of 2.97–3.31% and a NO_x specific emissions reduction of 7.28–8.33% are appreciated. For the operational profile of the case study vessel this reductions means 5306 kg of CO_2 and 11041 kg of NO_x per year can be cut down.

Operating costs and CO_2 emissions are directly linked to the system performance as the price of operating the machine descends if from the same amount of fuel more available energy is extracted, whether in the form of mechanical work or heat. As the fraction of energy increases for the same amount of fuel, total CO_2 emissions do not vary as the same quantity of carbon is being burnt. But in specific terms CO_2 emissions are lower per each mass unit of fuel used. Same happens with NO_x emissions as more power is used from the same amount of fuel.

4. - Conclusions

In this paper, a comprehensive methodology to analyze the energy, exergy, economic and environmental (4E) balance has been proposed for large marine diesel engines. By taking into account engine particularities, the results presented are a significant contribution to scientific literature in terms of the detail and extent of the available data. Below, major findings of this research are presented.

(1) As discussed, both energy and exergy analyses are examined from two points of view:
   - External – Global: the engine is treated as a blackbox where it exchanges energy with the environment.
   - Internal – Local: where only internal flows of energy are considered. Some of the terms can be measured or obtained through analysis but others are calculated or estimated.

(2) Along with the description of the methodologies, a determination of each term in the energy and exergy balances have been presented. While mechanical work on the shaft accounts for over 40% of the total energy, only a 25% of the total energy is available for recovery, exergy. Major source of irreversibility is found to be combustion, taking 26% of the total energy supplied. Inside the engine, an exothermic combustion produced in the combustion chamber releases heat to the cylinder walls which, at the

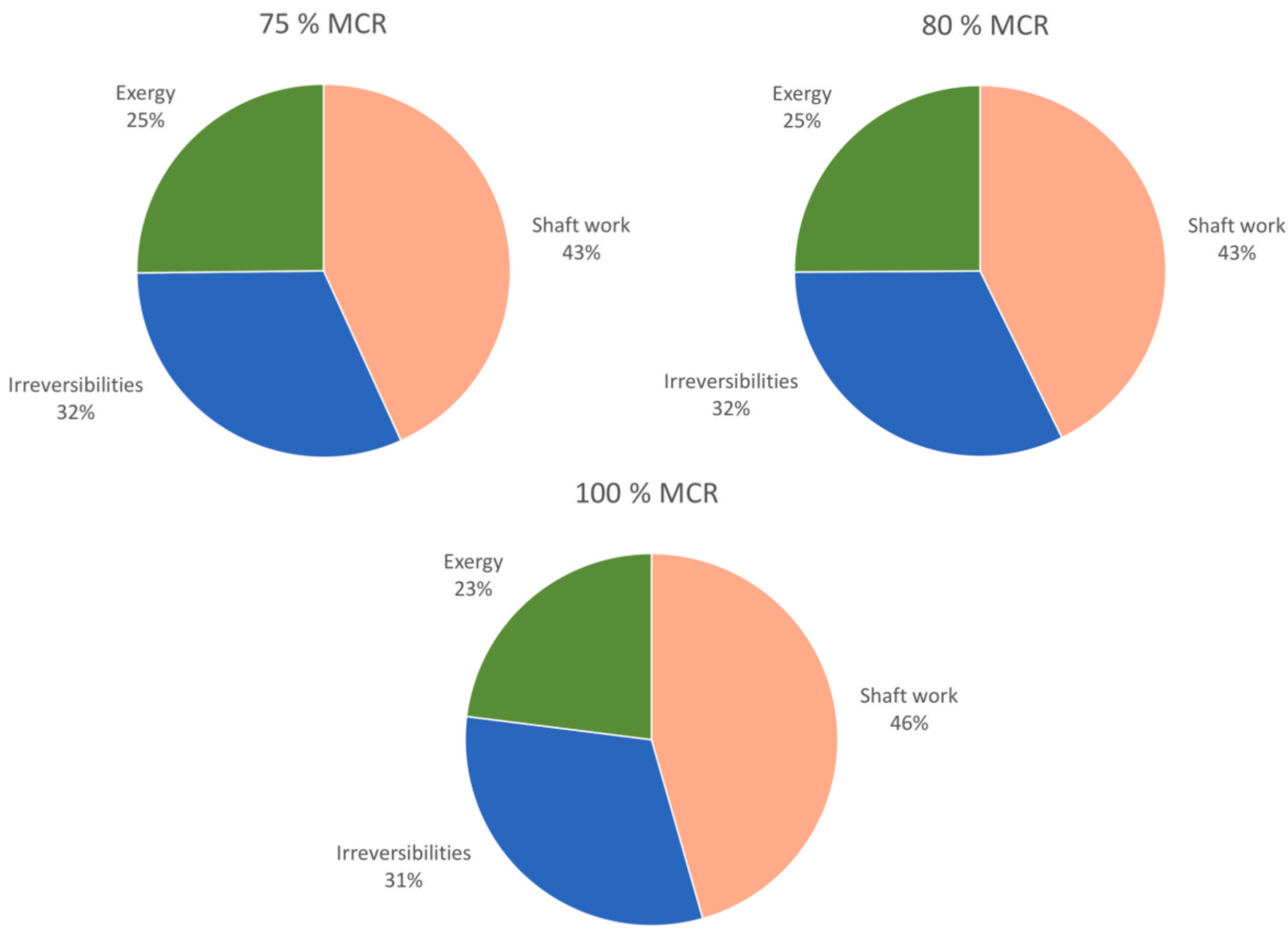

Fig. 12. Distribution of the energy supplied by the fuel at the loads measured.

Table 7
Results from the Energy analysis of the case vessel.

Load (% MCR)	75	80	100
System Performance			
Exergy Efficiency	0.3707	0.3640	0.3788
Irreversibilities Efficiency	0.2902	0.2931	0.2795

Table 8
Results from the Exergoeconomic analysis of the case vessel.

Load (% MCR)	75	80	100
Cost without WHR (Cw) (€/kW h)	0.0804	0.0812	0.0804
Cost with WHR (Cw.whr) (€/kW h)	0.0743	0.0745	0.0746
Savings (%)	7.5536	8.3329	7.2884

Table 9
Results from the Environmental analysis of the case vessel.

Load (% MCR)	75	80	100
CO_2 reduction (g/s)	13.7882	16.3627	17.6881
CO_2 reduction (%)	3.0292	3.3085	2.9779
NO_x specific emissions, only shaft work (g/kW h)	9.9000	10	10.1000
Total NO_x emissions, only shaft work (g/h)	22275	24000	30300
NO_x specific emissions, shaft work + WHR (g/kW h)	9.1522	9.1667	9.3639
NO_x reduction (%)	7.5536	8.3329	7.2884

same time, transfer the heat to the HT Cooling circuit. Any heat transfer between different elements produces entropy, which always destroys exergy.

(3) An exergoeconomic study has been done in order to asses savings that can be achieved if recovering available waste heat from engine block and coolers surface. Exergy is taken for the analysis in preference of energy as it does represent the real amount of energy available to be recovered. As expected, the recovery of waste heat results in a reduction of the cost flow (euro/kW h).

(4) An environmental assessment has been carried out in order to overview the reduction in carbon dioxide (CO_2) and nitrogen oxides (NO_X) emissions produced while the dissipated heat from engine block, LO and CA coolers is recovered. The more waste heat is recovered and used onboard, the less amount of fuel will be needed to produce the same amount of energy so the proposed method can help greatly to lessen air pollution.

Shipping industry is not seen as the cleaner mean of transportation due to press images of black smoke coming out of ship's funnels. As a matter of fact, commercial shipping is not very public due to port restrictions related to the ISPS Code but cruise and ferry industries are exceptionally exposed to the general opinion. For these last two sectors, air pollution trimming becomes relevant.

(5) From displayed results, it can be appreciated the system shows potential for optimization. Classical solutions like the use of ORC and recovery of exhaust and cooling water waste heat can be

combined with newer technologies like thermoelectric conversion.

Phenomenology of energy distribution inside a combustion engine is so complex that not all the parameters can be measured on a commercial, already working, engine. Due to that fact, measured and calculated terms have been combined in order to solve the proposed method.

The information presented in this study can provide a beneficial tool to examine those already installed marine engines in order to find the real energy balance. And thus, compare with initial test records and analyze the degradation of the system. Results about the recovery of the engine block waste heat provide a theoretical foundation for a better understanding of the machine, useful for maintenance operators and designers. Future studies based on this work will focus on implementing technologies able to recover the waste heat.

4.1. Limitations and future work

The estimation of waste heat available on the engine studied represents one of the major contributions of this work to the scientific literature. Nevertheless, it needs to be noted that estimation was based on some assumptions that could only be partly verified against the engine installed on the case study vessel, especially for some of the internal energy parameters which their measurement need of an on purpose system and/or virtual models.

Ideally, the study should continue by creating a virtual model with particular calibration for the studied engine where internal parameters could be refined from current values. Secondly, the develop of a feasible system for the recovery of the heat radiated by the engine should be studied and tested.

Funding

This research did not receive any specific grant from funding agencies in the public, commercial, or not-for-profit sectors.

CRediT authorship contribution statement

L.A. Díaz-Secades: Conceptualization, Methodology, Writing – original draft. **R. González:** Writing – review & editing, Supervision, Project administration. **N. Rivera:** Investigation, Writing – review & editing, Supervision.

Declaration of competing interest

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

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