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Comparative analysis of the heat balance results of the selected Tier III-compliant gas-fuelled two-stroke main engines

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Two-stroke engines are distinguished by the highest overall efficiency among all main engines. This is not only due to the low speed, and large piston stroke, but also to the high combustion temperature, which results in an increase in nitrogen oxides (NO_x) emission. Technical solutions applied to bring main engines into compliance with current NO_x emission standards set by the Tier III limits include the use of SCR and EGR systems, the implementation of the Otto cycle, and the application of liquified natural gas (LNG) as the low-emission fuel. Impact of the available Tier III-compliant technologies on the heat balance results is analysed using the example of the currently most popular dual-fuel main engines, i.e. WinGD X92DF and MAN G95ME-C10.5-GI. The possibilities of waste heat recovery in the electricity generation process and thereby improving the ship energy efficiency are discussed.

Key words: *heat balance, two-stroke engine, LNG, SCR, EGR*

1. Introduction

Two-stroke engines, achieving an overall efficiency of more than 50% and not requiring the use of reduction gears, are unrivalled as the most efficient source of mechanical energy needed to move the ship at the set sailing speed. As such, they are widely used on all types of cargo ships (including container ships, bulk carriers, and tankers), whose geometry allows the main engine of considerable height to be installed inside the ship's engine room space in the plane of symmetry of the hull's skeg. The highest overall efficiency among all internal combustion engine types is due not only to the low speed and large piston stroke, but also to the relatively high combustion temperatures of the compressed fuel-air mixture. It promotes efficient utilisation of the chemical energy contained in the supplied fuel, but is associated with an undesirable significant increase in nitrogen oxides (NO_x) emission [7]. NO_x emission increasing with the combustion temperature of hydrocarbon fuels, especially intensively above 1300°C, are an inherent part of the operation of all internal combustion engines [3]. The negative impact of NO_x emission on the environment and human health has been recognised by the International Maritime Organisation (IMO) and MARPOL Annex VI has been expanded to include provisions obliging shipowners to implement technical solutions to reduce NO_x emission from ships. The requirements for the prevention of NO_x emission apply to engines with a rated brake power of more than 130 kW as defined in rules 2.12 and 2.14. The most stringent NO_x emission standards have been set by the Tier III limits since 2015 [13]. Despite the enduring popularity of direct-driven main propulsion systems with a two-stroke engines, publicly available analyses and studies of the heat balance results refer to an outdated state of the art. The main engine operation was then unaffected by the selective catalytic reduction (SCR) or exhaust gas recirculation (EGR), ensuring the exhaust gas composition in compliance with current NO_x emission standards, liquified natural gas (LNG) was not yet in widespread use as the low-emission fuel and there were no main engines implementing the Otto cycle instead of the Diesel cycle [10]. Thus, the purpose of this paper is to compare the percentage share of heat losses in the heat balances of the most popular dual-fuel two-stroke main engines, i.e. WinGD X92DF and MAN G95ME-C10.5-GI, operating in a Tier III-compliant gas mode under ISO ambient conditions [7]. The realisation of the chosen objective will facilitate the selection of a gas-fuelled two-stroke main engine in terms of waste heat recovery in the process of electricity generation based on the thermodynamic cycle appropriate for the parameters of available heat losses [1].

2. Heat balance of the internal combustion engine

The heat balance is a compilation of the distribution of the total heat supplied to the main engine with fuel into useful heat, equivalent to the brake power, and the sum of heat losses. To make the heat balance more readable, the percentage of all components in relation to the total heat supplied is given. The heat balance is carried out at a constant load for the contract parameters after the engine has reached thermal equilibrium, and the results obtained are presented analytically. Performing a heat balance with the presentation of results makes it possible to determine ways of increasing its efficiency, which is particularly important in terms of waste heat recovery. Equations (1)–(4), applicable to the heat balance of all internal combustion engines, are listed below [2].

$$\dot{Q}_T = \dot{Q}_u + \sum \dot{Q}_L \tag{1}$$

hence:

$$\sum \dot{Q}_L = \left(\dot{Q}_g + \sum \dot{Q}_{lt} + \dot{Q}_r\right) \tag{2}$$

hence:

$$\sum \dot{Q}_{lt} = \left(\dot{Q}_o + \dot{Q}_w + \dot{Q}_{sa} + \dot{Q}_{EGR}\right) \tag{3}$$

and:

$$\dot{Q}_T = \frac{\dot{Q}_u}{\eta_o} = \frac{P_B}{\eta_o} \tag{4}$$

thus for LNG-fuelled engines [12]:

$$\dot{Q}_T = \frac{P_B \cdot (LCV_{FO} \cdot SFOC + LCV_G \cdot SGC)}{3600} \tag{5}$$

where: $\dot{Q}_T$ – total heat supplied with fuel, kW; $\dot{Q}_u$ – useful heat, kW; $\sum \dot{Q}_L$ – sum of all heat losses, kW; $\dot{Q}_g$ – exhaust gas waste heat, kW; $\sum \dot{Q}_{lt}$ – sum of the low-temperature heat losses, kW; $\dot{Q}_r$ – radiation waste heat, kW; $\dot{Q}_o$ – cooling oil waste heat, kW; $\dot{Q}_w$ – jacket water waste heat, kW; $\dot{Q}_{sa}$ – scavenge air waste heat, kW; $\dot{Q}_{EGR}$ – EGR cooling waste heat, if applicable, kW; P_B – main engine brake power, kW; η_o – main engine overall efficiency; LCV_{FO} – lower calorific value of the pilot fuel oil, $\frac{kJ}{kg}$; SGC – specific LNG consumption, $\frac{g}{kWh}$; SFOC – specific pilot fuel oil consumption, $\frac{g}{kWh}$; LCV_G – lower calorific value of the LNG, $\frac{kJ}{kg}$.

$\dot{Q}_{EGR}$ is a total heat dissipated in plate heat exchanger according to Fig. 1. This heat loss is unique to engines equipped with each of the EGR system variants [15].

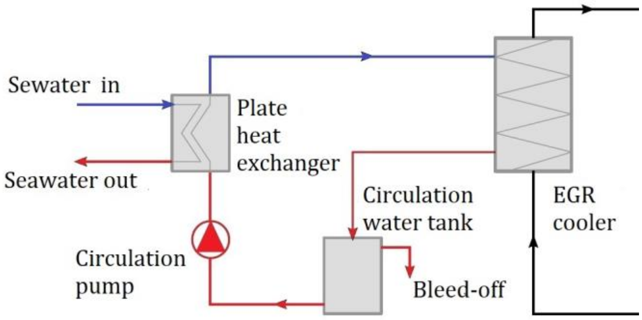

Fig. 1. Overview of the EGR cooling system [21]

The overall efficiency of the contemporary two-stroke dual-fuel main engines operating at maximum continuous load under ISO ambient conditions exceeds 50%, which means that such a proportion of the fuel-supplied energy is converted into brake power, with the rest being lost [11]. Potentially, therefore, very large amounts of heat could be recovered, which is severely hampered by the relatively low temperature of the waste heat carriers. Conventional waste heat recovery systems reach for exhaust gas characterised by sufficiently high inlet and outlet temperatures that allow the use of this waste heat carrier in electricity generation based on steam Rankine cycle (SRC), Brayton cycle (BC) or combined cycle (CC) [1]. The other waste heat carriers, i.e. scavenge air, jacket water, lubricating oil and EGR cooling water, are distinguished by significantly lower temperatures (Table 3) that preclude water steam production. Only the scavenge air gives off a portion of its heat to the feed water supplying an exhaust gas economiser in several maritime applications. Consequently, they have not yet been used in the electricity generation required to reduce the energy efficiency index for designed (EEDI) and existing (EEXI) ships [14]. The amount of the useful heat and the individual heat losses in the heat balance as well as the temperatures of waste heat carriers are subject by both the engine thermodynamic cycle and the operation of the SCR or EGR systems used [1]. The impact of the operation of the mentioned systems and thermodynamic cycle realised by the two-stroke dual-fuel main engine on the available waste heat amounts and temperatures are discussed and illustrated by the heat balance results later in the paper.

3. Methodology

The impact of technical solutions to adapt main engines to the Tier III emission standards on the heat balance results and the temperature of waste heat carriers is illustrated using the example of the currently most popular two-stroke dual-fuel main engines, i.e. WinGD X92DF and MAN G95ME-C10.5-GI. These have found the application, among others, on all the series of dual-fuel ultra-large container ships built in the last three years [7]. Table 1 contains the basic design and operating parameters of the analysed main engines.

Table 1. Basic design and operating parameters of the WinGD X92DF and MAN G95ME-C10.5-GI main engines [15]

Parameter	Unit	WinGD X92DF	MAN G95ME-10.5-GI
Piston stroke	m	3.468	3.46
Piston bore	m	0.92	0.95
Maximum continuous speed	min^{-1}	80	80
Maximum continuous power	$\frac{kW}{cyl.}$	5320	6870
Available number of cylinders	–	6–12	6–12
Available range of contract maximum continuous power	kW	23,520–63,840	27,120–82,440
Compression ratio	–	6.25	12
Gas supply pressure	MPa	0.75–1.5	20–30

Simulations of the operation of selected main engines were performed in the MAN Computer Engine Application System and General Technical Data for WinGD 2-Stroke Engines software for the boundary conditions indicated in Table 2. In this manner the full heat balance results of the analysed main engines were received, including the amount of all heat losses as well as the temperature and mass flow of the available waste heat carriers. The most important data are presented in graphical (Fig. 2–3) and tabular (Table 3) form later in the paper.

The reference engine load equal to 75% of the maximum continuous rating allows obtaining information useful, among other things, in calculating EEDI and EEXI and in developing how to recover available waste heat amount. In addition, it should be noted that the scope of EEDI and EEXI does not include the use of waste heat for heating services as well as in the process of domestic hot water and fresh water production. Improving the energy efficiency of a ship as defined by these technical measures, therefore requires the production of electricity in recovery turbine generator units. In this way, some of the diesel generating sets of the ship's electric power station can be set aside and the surplus power transferred to the main engine crankshaft by means of a shaft generator operating in a PTO (Power Take-Off) mode [4].

Table 2. Boundary conditions assumed for the simulation of the selected main engines operation [4]

<table>
<tr><th colspan="2">Boundary conditions</th><th>Remarks</th></tr>
<tr><td>Engine load</td><td>75% of maximum continuous power</td><td>Engine load is related to the calculation of energy efficiency design (EEDI) and existing ship (EEXI) index</td></tr>
<tr><td>Engine speed</td><td>91% of maximum continuous speed</td><td>Engine speed corresponds to the operation of the engine according to the propeller curve</td></tr>
<tr><td>Ambient conditions</td><td>ISO</td><td>Ambient air and scavenge air coolant temperature is 25°C</td></tr>
<tr><td rowspan="2">Fuels used</td><td>Regasified LNG</td><td rowspan="2">Gas mode in which the pilot oil dose is involved in the initiation of ignition</td></tr>
<tr><td>MGO as a pilot oil</td></tr>
<tr><td>Compliance with NO_x emission standards</td><td>Tier III and consequently less stringent Tier II</td><td>WinGD X92DF in a gas mode does not require the use of SCR or EGR systems</td></tr>
</table>

In turn, the presentation of heat losses as a percentage of the energy supplied with fuel makes it easier to relate these values to main engines with any available number of cylinders. On the other hand, the temperature of each of the waste heat carriers refers to the thermodynamic cycle whose realisation enables the production of steam to drive the steam turbine generator unit [11].

4. Results and discussion

The heat balance results of the WinGD X92DF and MAN G95ME-C10-GI operating in a Tier III-compliant gas mode under 75% maximum continuous rating and ISO ambient conditions received in the simulation carried out for the boundary conditions listed in Table 2 are shown in Fig. 2. All the symbols used are identical to those mentioned below equations (1)–(5).

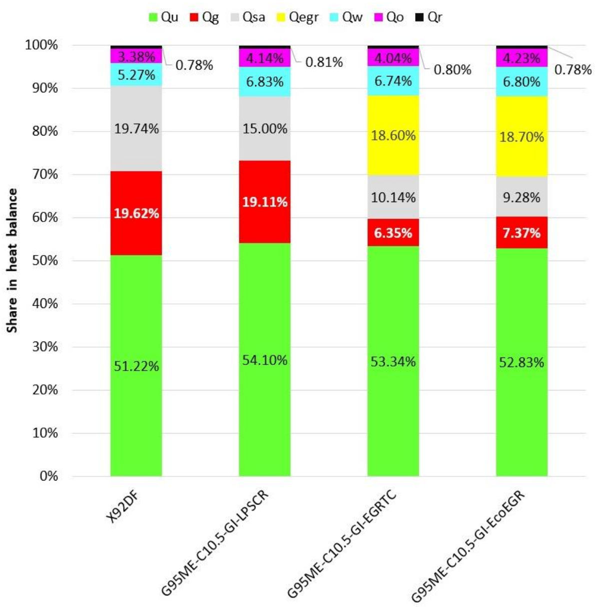

Fig. 2. Heat balance results of the gas-fuelled MAN G95ME-C10.5-GI and WinGD X92DF operating in a Tier III-compliant gas mode under 75% maximum continuous rating and ISO ambient conditions

On the basis of a comparison of the heat balance results shown in Fig. 2, it was found that the realisation of the Otto cycle by the WinGD X92DF was associated with up to 2.88 percentage points lower overall efficiency and thus much higher waste heat resources than the Diesel-cycle MAN G95ME-C10.5-GI. This was caused by a significantly lower compression ratio as well as the combustion temperature of the compressed fuel-air mixture (Table 2). Otto cycle is the most efficient thermodynamic cycle of the internal combustion engines assuming the same compression ratio. However, the almost twice lower compression ratio than in the case of the Diesel cycle determines the lower efficiency of Otto-cycle WinGD X92DF [10]. In addition, it is important to note the largest share of exhaust gas and scavenge air heat losses with the smallest share of jacket water and lube oil heat losses in the WinGD X92DF engines waste heat. This is due to the lack of any additional systems to ensure the exhaust gas composition complied with Tier III emission standards, and a slightly larger piston stroke (Table 1) than the competing MAN G95ME-C10.5-GI [21].

MAN G95ME-C10.5-GI equipped with the LPSCR system stands out for the highest overall efficiency and share of exhaust gas waste heat in the total heat loss. This was attributed to the lack of interference of low-pressure selective catalytic reduction in the combustion process and thus the highest exhaust gas temperature and mass flow values among the available main engine configurations from this manufacturer [23]. Those equipped with both EGR system variants, on the other hand, are characterised by an unique EGR cooling heat loss $(\dot{Q}_{EGR})$, which is greater the higher the percentage of exhaust gas is recirculated. The appearance of this heat loss in the heat balance is linked to the significant decrease in the contribution of exhaust gas and scavenge air heat loss to total waste heat (Fig. 2) [13].

The inlet T_i and outlet T_o temperatures of the available waste heat carriers obtained from the MAN Computer Engine Application System and General Technical Data for WinGD 2-Stroke Engines software are shown in Table 3.

Table 3. Inlet T_i and outlet T_o temperatures of the available waste heat carriers

<table>
<tr><th colspan="2">Waste heat carrier</th><th colspan="2">Exhaust gas</th><th colspan="2">Scavenge air</th><th colspan="2">Jacket water</th></tr>
<tr><td colspan="2">X92DF</td><td>351</td><td>206</td><td>185</td><td>31</td><td>84</td><td rowspan="4">75</td></tr>
<tr><td rowspan="3">G95ME-C10.5-GI</td><td>LPSCR</td><td>424</td><td>254</td><td>188</td><td>32</td><td rowspan="3">87</td></tr>
<tr><td>EGRTC</td><td>294</td><td>214</td><td>193</td><td>33</td></tr>
<tr><td>EcoEGR</td><td>290</td><td>213</td><td>192</td><td>35</td></tr>
<tr><td colspan="2">Symbol</td><td>T_i</td><td>T_o</td><td>T_i</td><td>T_o</td><td>T_i</td><td>T_o</td></tr>
<tr><td colspan="2">Unit</td><td colspan="6">°C</td></tr>
<tr><td colspan="2">Waste heat carrier</td><td colspan="2">Lube oil</td><td colspan="2">EGR cooling water</td></tr>
<tr><td colspan="2">X92DF</td><td>55</td><td rowspan="4">45</td><td colspan="2" rowspan="2">N/A</td></tr>
<tr><td rowspan="3">G95ME-C10.5-GI</td><td>LPSCR</td><td rowspan="3">54</td></tr>
<tr><td>EGRTC</td><td>44</td><td rowspan="2">25</td></tr>
<tr><td>EcoEGR</td><td>45</td></tr>
<tr><td colspan="2">Symbol</td><td>T_i</td><td>T_o</td><td>T_i</td><td>T_o</td></tr>
<tr><td colspan="2">Unit</td><td colspan="4">°C</td></tr>
</table>

The temperature values of the available waste heat carriers shown in Table 3 are not the highest achievable due to the reduction of the main engine load to 75% of the maximum continuous rating. The exhaust gas temperatures of all the analysed main engines are among the lowest range due to the location of the operating point in the area of highest overall efficiency, when the temperatures of the rest of waste heat carriers increase proportionally to the main engine load [11]. Exhaust gas is the only waste heat carrier whose sufficiently high temperature makes it possible to use them directly in the electricity generation process on the basis of both conventional thermodynamic cycles, i.e. the steam Rankine cycle (SRC), Brayton gas cycle (BC) or combined gas-steam cycle (CC), as well as the organic Rankine cycle (ORC). Each of the other waste heat carriers, due to their temperature precluding the steam production, can only be used to generate electricity in an ORC using an organic working fluid with a sufficiently low boiling point. In this way, superheated steam can be produced using waste heat carriers with a temperature below the water boiling point [1]. So far, only waste heat recovery based on conventional (SRC, BC and CC) thermodynamic cycles adapted to the main engine output has been implemented on some ships [8]. On the basis of comparison of the data contained in Figure 2, it is concluded that the use of a main engine equipped with any of the available EGR system variants can make waste heat recovery of the exhaust gas alone unprofitable in terms of the economy and improving ship energy efficiency defined by EEDI and EEXI values. This results from the relatively small share (6.35–7.37%) of exhaust gas waste heat from in the heat balance. This means that its amount, expressed in kilowatts, is almost 3 times smaller than that of main engines equipped with LPSCR or implementing the Otto cycle. The share of exhaust gas and low-temperature (3) waste heat in the sum of heat losses are shown in Fig. 3.

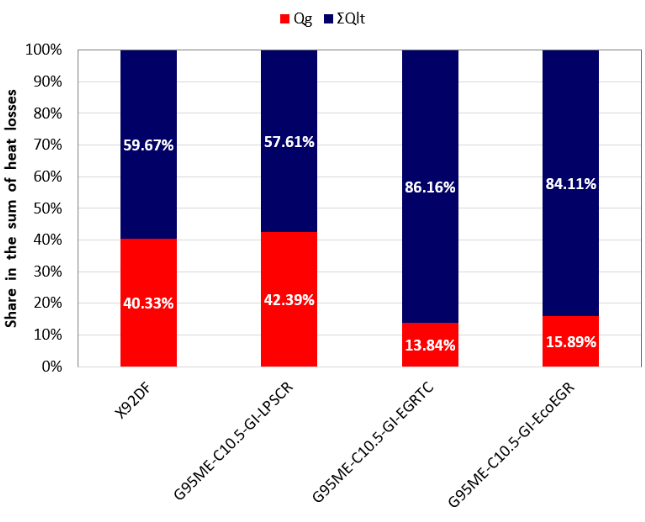

Fig. 3. Share of exhaust gas and low-temperature waste heat in the sum of heat losses

Analysis of the data contained in Fig. 3 has shown that waste heat recovery based on the ORC is particularly important for improving the energy efficiency of ships powered by main engines equipped with EGR systems. The waste heat of low-temperature carriers (3) in their case is equivalent to as much as 84.11–86.16% of the sum of heat losses, while for Diesel-cycle main engines equipped with LPSCR and Otto-cycle X92DF this share is within the limits of 57.61–59.67%. The indicated percentage of available waste heat can only be recovered in the power generation process in turbine generator units based on the ORC. It should also be emphasised that there is some waste cold amount from the LNG regasification process, which can be utilised to discharge heat from the steam in the condenser and in consequence significantly increase the ORC efficiency [1].

Conclusions

Performed analysis of the heat balance results of the two most popular two-stroke gas-fuelled main engines, i.e. WinGD X92DF and MAN G95ME-C10.5-GI, revealed that all available Tier III-compliant technologies have a significant impact on overall efficiency and the share of heat losses in the total waste heat.

Gas-fuelled Otto-cycle WinGD X92DF meets the most stringent Tier III emission standards without the use of any additional systems due to the lower combustion temperature of the less compressed fuel-air mixture. However, this is linked to the less efficient use of the chemical energy contained in the supplied fuel and therefore a decrease in overall efficiency and exhaust gas temperature (Table 3). Furthermore, it stands oud for the largest share of exhaust gas and scavenge air heat losses, while having the smallest share of cooling water and lube oil losses in total waste heat.

Gas-fuelled Diesel-cycle MAN G95ME-C10.5-GI requires the use of one of the available Tier III-compliant SCR or EGR systems due to the higher combustion temperature of the more compressed fuel-air mixture. LPSCR does not interfere with the combustion process, so main engines equipped with this system are characterised by the highest overall efficiency and exhaust gas temperature, as well as the largest share of exhaust gas waste heat in the sum of heat losses. Those equipped with both EGR system variants, on the other hand, are distinguished by their unique EGR cooling waste heat, which is greater the higher the percentage of exhaust gas is recirculated. The appearance of this EGR heat loss in the heat balance is associated with a significant decrease in the share of the exhaust gas temperature and waste heat as well as scavenge air to the total waste heat. In contrast, the temperature differences of the other waste heat carriers are negligible regardless of the Tier III-compliant technology used or lack thereof.

The reduction of EEDI and EEXI values requires the use of available waste heat in the electricity generation process. Exhaust gas is the only waste heat carrier with a sufficiently high temperature to enable its direct use in the electricity generation process on the basis of conventional thermodynamic cycles, i.e. the steam Rankine cycle (SRC), Brayton gas cycle (BC) or combined gas/steam cycle (CC), as well as the organic Rankine cycle (ORC). Any of the other low-temperature waste heat carriers, due to their temperatures precluding steam production, can only be used for power generation based on an ORC using an organic working fluid with a sufficiently low boiling point. Having regard to the dominant share of low-temperature waste heat carriers in the total heat loss of EGR-equipped main engines, it becomes ineffective to improve the energy efficiency of the ship driven by them without the use of waste heat recovery based on the ORC. Available waste cold amount from the LNG regasification process, which can be utilised to discharge heat from the steam in the condenser and consequently significantly increasing the ORC efficiency.

Nomenclature

EEDI	energy efficiency design index	EEXI	energy efficiency index for exisiting ships
EGRTC	exhaust gas recirculation turbocharger cut-out	EcoEGR	Eco-exhaust gas recirculation
LPSCR	low-pressure selective catalytic reduction	ORC	organic Rankine cycle

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