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Prediction of the amount of waste cold from liquefied natural gas (LNG) regasification for gas-fuelled low-speed main engines

Piotr Kamil Korlak

https://orcid.org/0000-0002-4548-4796

Doctoral School of the Maritime University of Szczecin, Szczecin, Poland
e-mail: 27901@s.am.szczecin.pl

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Abstract

Existing and future regulations on ship energy efficiency and methods for their improvement are presented in this work. The design and operational features of gas-fuelled low-speed main engines, liquefied natural gas (LNG) regasification conditions, and amount of waste cold are compared. Using a simple linear regression model based on the least squares method, formulae were developed to predict the amount of waste cold as a function of the brake power of gas-fuelled low-speed main engines operating under ISO ambient conditions in Tier III-compliant mode. A sufficiently accurate prediction of the waste cold amount at the initial design stage is feasible due to the formulae developed as part of this work.

Introduction

Efforts have been taken around the world to reduce the environmental impact of shipping since the 2000s. One of these was the enlargement of MARPOL Annex VI to include a policy of reducing carbon dioxide (CO_2) emissions and minimum energy efficiency requirements. Chapter 4 of MARPOL Annex VI has been obliging shipowners to use technical solutions to reduce CO_2 emissions since July 2011. The Energy Efficiency Design Index (EEDI), used as a regulatory tool to control CO_2 emission, is mandatory for all newbuild ships over . Its value, calculated according to the procedure shown in Figure 1, must not be higher than required for a specified type and size of the ship (IMO, 2016; MAN Diesel & Turbo, 2017). It should also be noted that EEDI does not include CO_2 emissions from steam boilers. In addition, as agreed by the 76th session of the Marine Environment Protection Committee (MEPC), all in-service ships will be subject to the minimum energy efficiency standards defined by the EEDI-equivalent Energy Efficiency Existing Ship Index (EEXI) from 2023 (DNV, 2021).

The International Maritime Organization (IMO), as set out in MARPOL Annex VI, from 2025 will require a reduction in CO2 emissions per tonne mile of cargo transported (EEDI and EEXI indicators) of at least 30% from the 2013 baseline (MAN Diesel & Turbo, 2017). Progress in the reduction of the required EEDI value from 2013 to 2025 is shown in Figure 2.

Since Liquefied Natural Gas (LNG) is by far the least carbon-intensive available marine fuel (MAN Diesel & Turbo, 2012) by enabling the reduction of CO_2 emissions up to 14.2% per tonne of consumed fuel (i.e., Carbon Factor shown in Figure 3) and other methods have already been exhausted (e.g., hydrodynamic optimization of the hull and propeller form) or are not feasible for the structural reasons (e.g. use

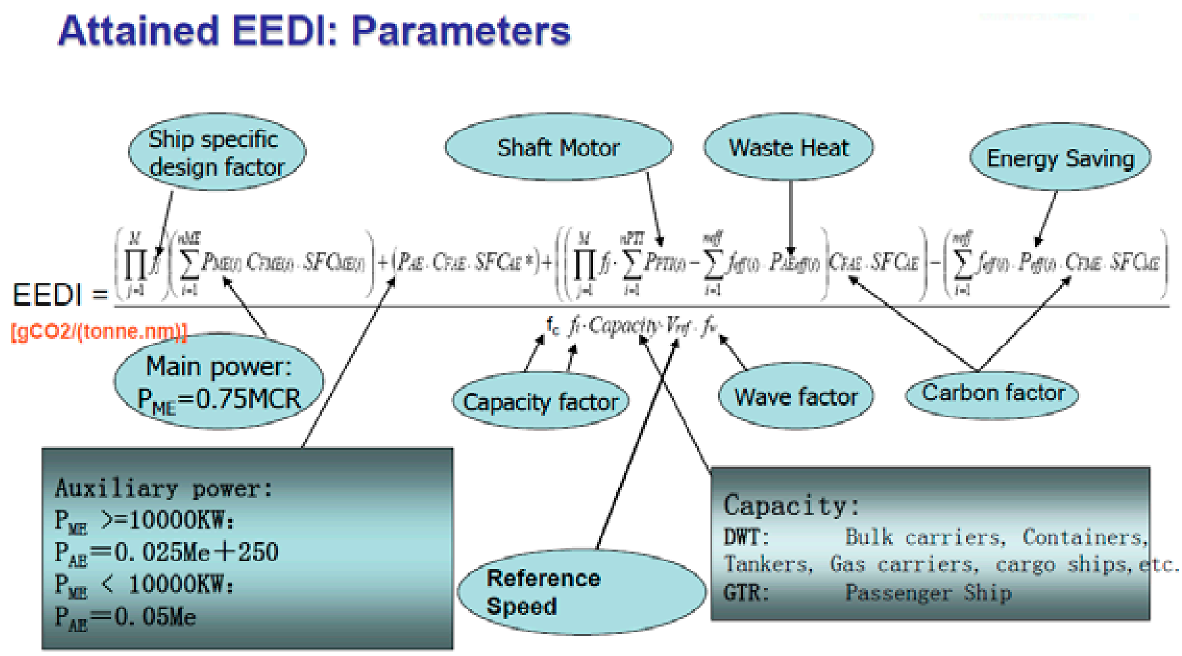

Figure 1. The procedure of the Energy Efficiency Design Index calculation (IMO, 2016)

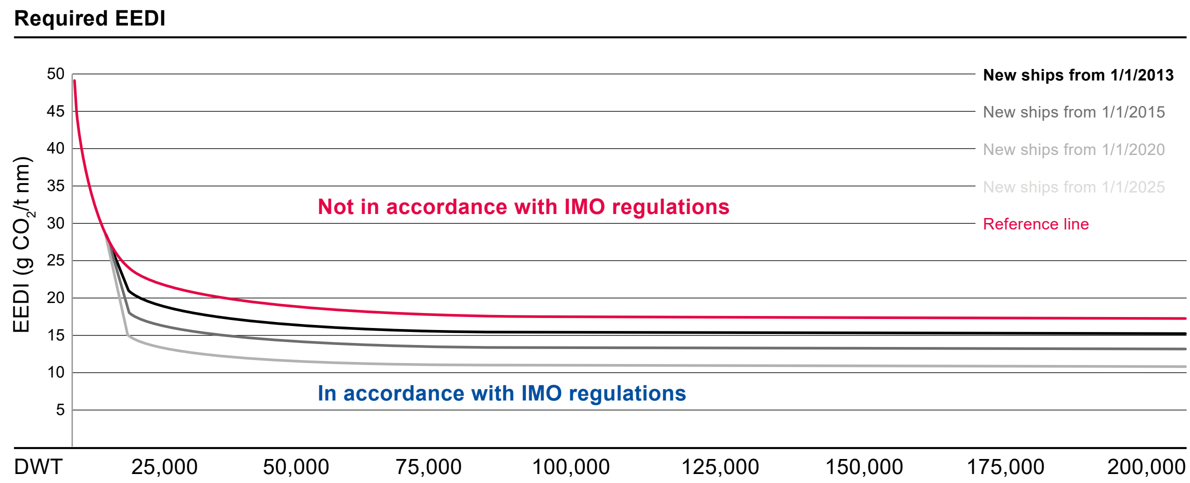

Figure 2. Progress in the reduction of the required EEDI value (MAN Diesel & Turbo, 2017)

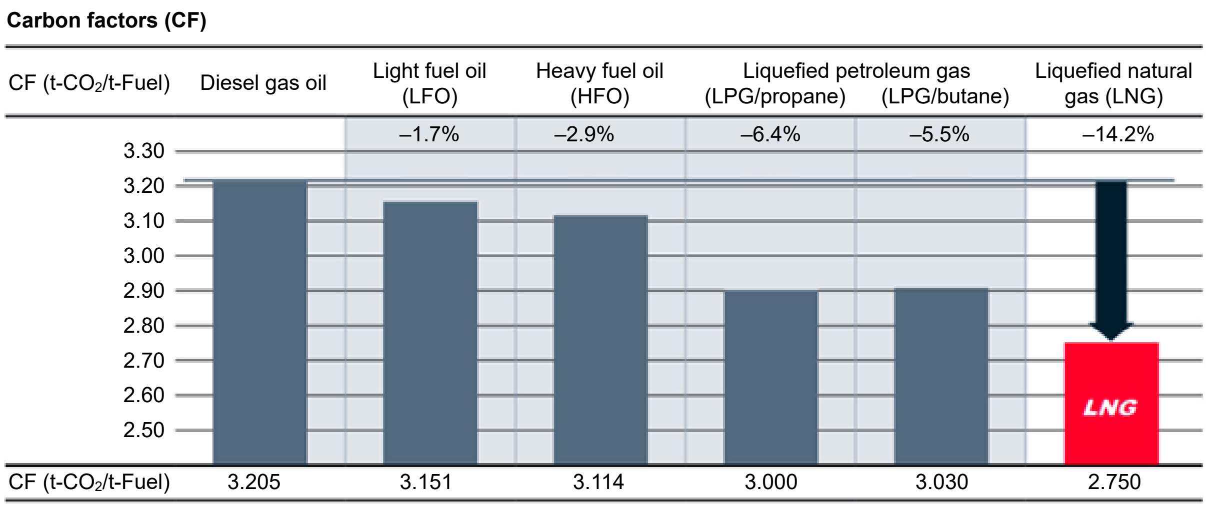

Figure 3. Carbon Factor of the most popular marine fuels (MAN Diesel & Turbo, 2012)

of the wind and solar energy on container ships), the further reduction in CO_2 emissions required by the IMO can only be achieved by reducing the fuel consumption of the main propulsion (main engines) and electric power station (auxiliary engines). This will be possible by using the available amount of both waste heat and cold to produce electricity (Giernalczyk, Górski & Krefft, 2015; Andreasen, Meroni & Haglind, 2017; Liberacki, 2019).

Exhaust gas waste heat recovery systems have already been implemented on many ships, especially those equipped with low-speed engines. The process of the waste cold recovery from LNG regasification has not yet been sufficiently researched and, consequently, no waste cold recovery system has been introduced into the ships' power plants. It is known, however, that the waste cold from LNG regasification and other low-temperature waste energy carriers, such as the heat loss of the scavenged air, cooling water and lube oil, could be used to produce electricity in a steam turbine utilization generator unit which operates according to the Organic Rankine Cycle (ORC) (Andreasen, Meroni & Haglind, 2017; Mondejar et al., 2018; Liberacki, 2019).

The ORC is a steam cycle where the difference is that the turbogenerator is not driven by steam water but by the steam of another medium that has a significantly lower boiling point. This allows the efficient use of low-temperature waste energy carriers. The working mediums are organic compounds, e.g. n-pentane, i-pentane, c-pentane, hence the circuit's name. Typically, refrigerants such as R134a, R245fa, or an organo-silicon compound (hexamethyldisiloxane) marked as MM or HMDSO are used. Depending on the medium chosen, boiling temperatures at normal pressure can be negative or at most a few tens of degrees Celsius.

It is, therefore, possible to bring them to boiling point and produce steam at pressures higher than atmospheric using waste energy carriers with temperatures lower than 373 K. The ORC systems

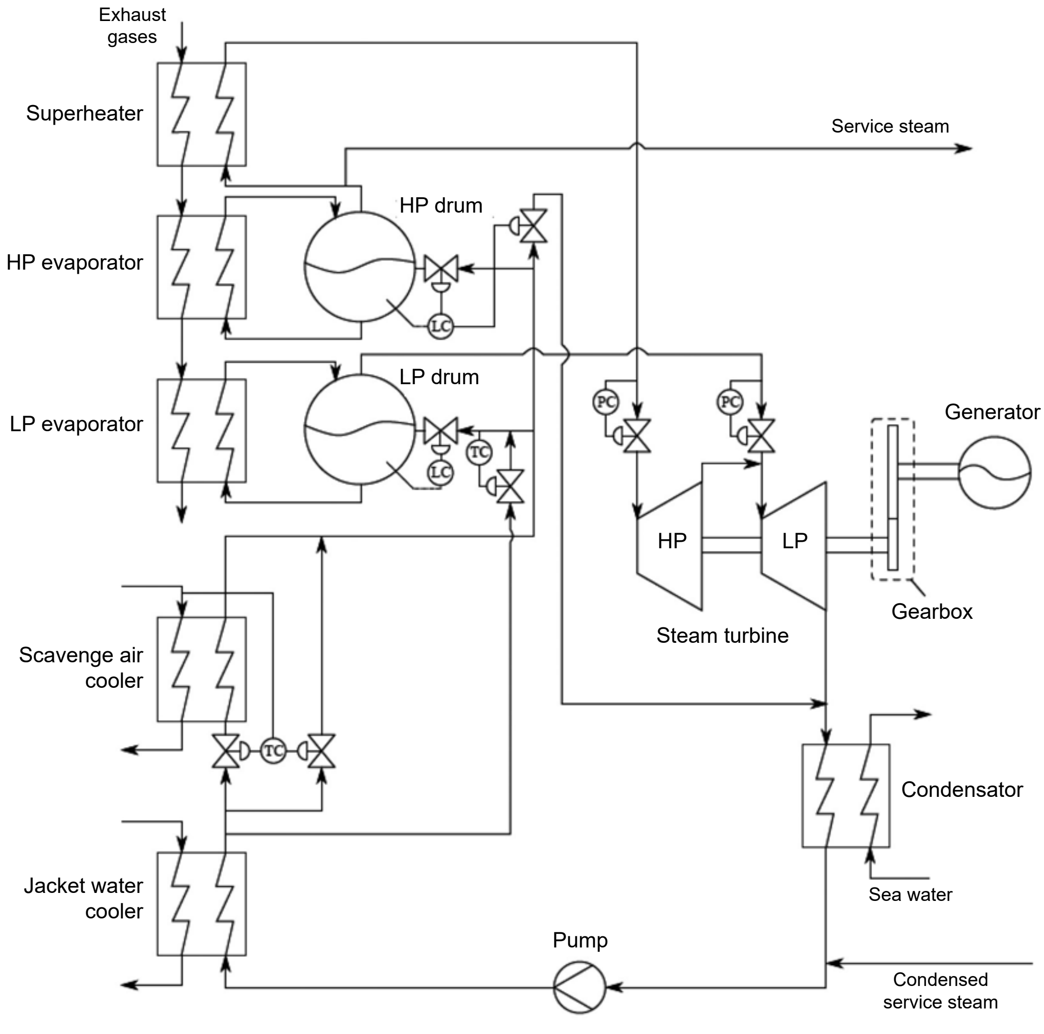

Figure 4. Schematic diagram of the ORC system with an indication of the control purpose of each valve: level control (LC), temperature control (TC) and pressure control (PC) (Andreasen, Meroni & Haglind, 2017)

can be built as single-pressure circuits, but because of the need to produce a certain amount of heating steam, they must also have part of a steam water circuit. A schematic diagram of the ORC system is shown in Figure 4. Moreover, there is a strong negative correlation between the amount of electricity obtained from the ORC system and the temperature of the condenser coolant. One interesting idea is to use the waste cold from LNG regasification to cool the steam in a condenser, which could increase the efficiency of the ORC (Liberacki, 2019).

Due to their low temperature and relatively small size, the use of the ORC technology is dedicated to the LNG-fuelled low-speed main engines (Andreasen, Meroni & Haglind, 2017; Mondejar et al., 2018). Moreover, ships equipped with low-speed main engines (e.g., container ships, bulk carriers, and tankers) find no other significant application for waste cold as, for example, the air conditioning of the numerous hotel rooms on RoPax and cruise ferries powered by the medium-speed engines (TT-Line, 2020).

This paper aims to determine the empirical mathematical relationships enabling the prediction of the amount of waste cold from LNG regasification for the full range of brake power of the currently available Tier III-compliant low-speed main engines operating under ISO ambient conditions. The least-squares method was applied to the calculations based on the performance parameters obtained from computer-aided engine selection software (MAN CEAS Engine Calculations v1.9.37 and WinGD General Technical Data 2.14.1.0) and the enthalpy values for specified temperatures and pressures of the LNG regasification process from MINI-REFPROP 10.0.

Table 1. Comparison of LNG-fuelled low-speed main engines (Christensen, 2017; WinGD, 2020; MAN Energy Solutions, 2021)

MAN ME-GI	WinGD X-DF
Diesel process maintained	Otto-process gas-air pre-mix
High-pressure gas injection means more expensive investments for fuel gas supply systems (compressors, pumps, components, etc.), higher electricity consumption and maintenance costs	Low-pressure gas injection means lower investments for fuel gas supply systems (compressors, pumps, components, etc.), lower electricity consumption and maintenance costs
Fuel in cylinder before gas	Gas in cylinder before fuel
Power remains the same	Power reduction required to pre-ignition/knocking risk, load ramp needed
Load response unchanged	
No pre-ignition & no knocking	
Insensitive to the gas mixture	Gas mixture important
Negligible methane slip	Methane slip significant
Particulate matter emissions are still significant	Particulate matter emissions are significantly reduced compared to diesel engine
NO_X reduction to IMO Tier III by SCR or EGR	Lower NO_X emission due to lower efficiency
MAN ME engine can be retrofit to ME-GI	WinGD X engine can only be retrofit if excess capacity is installed initially (20% larger engine, 20% greater fuel tanks, etc.)
Higher pilot fuel quantity, raging from 0,5 ÷ 8% of total energy consumption over engine power	Lower pilot fuel quantity, raging from 0,5 ÷ 1% of total energy consumption over engine power
Can only be operated when engine power is above 10% in gas mode	Can be operated on gas down to 5% power. Start/stop is requested in diesel mode.

Comparison of the LNG-fuelled low-speed main engines

The manufacturers of commercially available dual-fuel low-speed main engines (i.e., MAN and WinGD) offer engines that operate according to different thermodynamic power cycles (MAN Energy Solutions, 2021; WinGD, 2021). The major differences in design and operation between them are shown in Table 1.

The engines of the series mentioned in Table 1 require gas fuel supply at a temperature of 45±10°C and load-dependent pressure. The temperature of 45±10°C is specified mostly to reduce thermal loads on the gas piping itself and obtain a uniform gas density. Regardless of the required pressure of the gas fuel, the LNG regasification process is carried out in the same way. A cryogenic pump is used to generate the required pressure. After pressurization, the LNG is vaporized and burned in the engine. After running in the gas mode, the gas system is vented and depressurized to atmospheric pressure and the system can be purged with inert gas. Pipes in the gas supply system can be single- or double-walled and ventilated with negative pressure air. Due to the necessity of ensuring operational safety, suitable (e.g., control, safety, shut-off, and vent) valves are placed in the gas supply system (Christensen, 2017; WinGD, 2020; MAN Energy Solutions, 2021). A simplified diagram of the gas supply system for MAN ME-GI engines is shown in Figure 5.

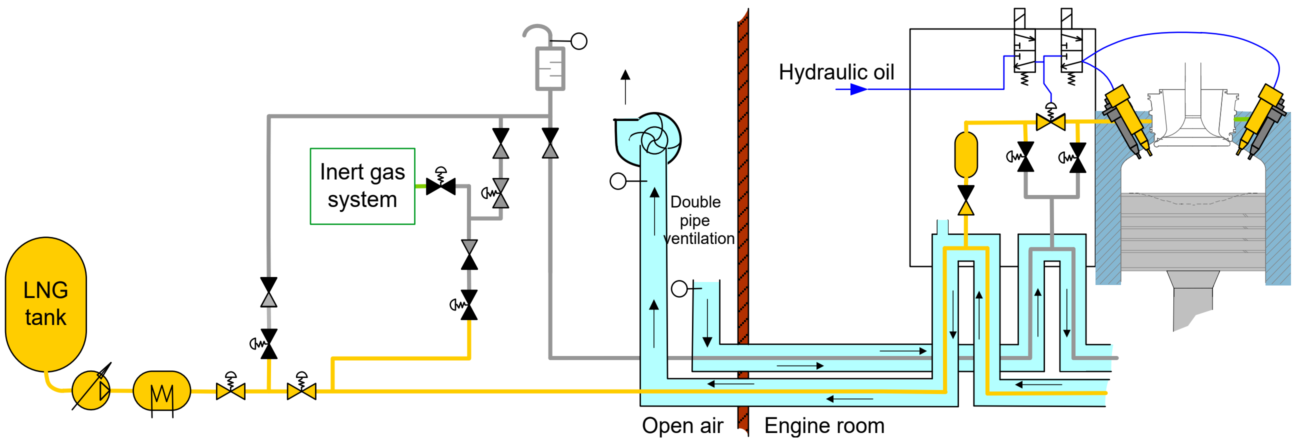

Figure 5. Gas supply system for MAN ME-GI engines (Christensen, 2017)

Results and discussion

The available amount of waste cold at the initial design stage can be predicted by comparing the actual performance parameters of the gas-fuelled low-speed main engines and LNG regasification. For this purpose, a list of selected real engine performance parameters (i.e., brake power and specific gas consumption in the load range of 25÷100% of maximum continuous rating under ISO ambient conditions) and LNG regasification parameters (initial and final pressure and temperature of the process) was prepared in the form of a database comprising all available engines equipped with suitable IMO Tier III technologies. Their availability is compared in Table 2.

The specified LNG regasification parameters, including the enthalpy values depending on the pressure and temperature set by the manufacturers (WinGD, 2020; MAN Energy Solutions, 2021), were obtained using MINI-REFPROP 10.0. A comparison of the LNG regasification parameters is presented in Table 3.

Table 2. Availability of the IMO Tier III-compliant technologies (Korlak, 2021)

<table>
<tr><th rowspan="2">IMO Tier III-compliant technology</th><th>MAN</th><th>WinGD</th></tr>
<tr><th colspan="2">Available for</th></tr>
<tr><td>LPSCR (Low-pressure selective catalytic reduction)</td><td colspan="2">All engines</td></tr>
<tr><td>HPSCR (High-pressure selective catalytic reduction)</td><td>Engines equipped with no more than 8 cylinders</td><td rowspan="4">Unavailable</td></tr>
<tr><td>EGRTC (Exhaust gas recirculation turbocharger cut-out)</td><td>Engines equipped with a piston bore ≥ 80 cm and at least 2 turbochargers</td></tr>
<tr><td>EcoEGR (Eco Exhaust gas recirculation)</td><td>All engines</td></tr>
<tr><td>EGRBP (Exhaust gas recirculation by-pass)</td><td>Engines equipped with piston bore ≤ 70 cm and only 1 turbocharger</td></tr>
</table>

Table 3. LNG regasification parameters

	MAN G-series ME-GI	MAN S-series ME-GI	WinGD X-DF (bore size ≤ 72 cm)	WinGD X-DF (bore size > 72 cm)
Pressure, MPa	30		1.33	1.5
Initial temperature, K	111.15			
Final temperature, K	318.15			
Initial enthalpy, kJ/kg	44.09		0.0037	0.0245
Final enthalpy, kJ/kg	761.41		944.33	943.54

The amount of waste cold from LNG regasification for each currently available gas-fuelled low-speed main engine, operating in the load range of 25÷100% of maximum continuous rating under ISO ambient conditions following the IMO Tier III emission standard, was determined using the following formula (Złoczowska & Adamczyk, 2017):

$$\dot{Q} = \frac{\dot{m}\cdot\Delta h}{3600} = \frac{\dot{m}\cdot\left(h_2 - h_1\right)}{3600} \qquad (1)$$

where:

- $\dot{Q}$ – amount of waste cold from the LNG regasification, kW;
- $\dot{m}$ – mass flow of the gas fuel, kg/h;
- Δh – enthalpy difference of the LNG regasification process, kg/h;
- h_1 – initial enthalpy of the LNG regasification process, kJ/kg;
- h_2 – final enthalpy of the LNG regasification process, kJ/kg.

Based on a comparison of the received results, it was found that, in the case of the application of the WinGD X-DF engines, it is possible to obtain a larger amount of waste cold for the same load in relation to both MAN ME-GI types due to a much larger enthalpy difference of the LNG regasification process Δh and increased gas fuel consumption at the cost of reducing pilot oil consumption. An example of calculation results for three engines of similar maximum continuous rating is shown in Table 4 and Figure 6. Engine performance parameters were taken from the computer-aided engine selection software (MAN CEAS Engine Calculations v1.9.37 and WinGD General Technical Data 2.14.1.0).

Another conclusion from comparing the performed calculations is that, unlike exhaust gas and scavenge air waste heat, applied IMO Tier III-compliant technology does not significantly affect the amount of waste cold from LNG regasification. Calculation results for MAN 8G95ME-C10.5-GI equipped with four different IMO Tier III-compliant technologies are presented in Table 5. Differences in the waste cold amount do not exceed 2% in relation to the LPSCR-equipped engine, which is within the tolerance of the specific gas consumption declared by the manufacturer, i.e. ±5% (MAN Energy Solutions, 2021; WinGD, 2021). Specific pilot oil consumption is identical in each of the analyzed variants, while the differences in the waste cold amount result only from slight differences in specific gas consumption. However, they represent the same percentage of the energy delivered in gas fuel.

Table 5. Comparison of the amount of waste cold from the LNG regasification

Engine	MAN 8G95-ME-C10.5-GI			
IMO Tier III-compliant technology	LPSCR	HPSCR	EGRTC	EcoEGR
Brake power = maximum continuous rating, kW	54,960			
Specific gas consumption, g/kWh	135.4	134.6	135.4	138
Specific pilot oil consumption, g/kWh	2.4			
Waste cold, kW	1482.78	1474.02	1482.78	1511.25
Waste cold as a % of energy delivered	1.41			
Waste cold as a % of value for LPSCR	100	99.41	100	101.92

Table 4. Comparison of the amount of waste cold from the LNG regasification

Engine	MAN 6G80ME-C10.5-GI	MAN 6S80ME-C9.5-GI	WinGD 6X82-DF-1.0
Fuel type	LNG + pilot oil		
IMO Tier III-compliant technology	Low-pressure selective catalytic reduction (LPSCR)		Not required in a gas mode
Brake power = maximum continuous rating, kW	29,120	27,440	25,920
Enthalpy difference, kJ/kg	717.32		943.52
Specific gas consumption, g/kWh	137.1	141.8	141.8
Specific pilot oil consumption, g/kWh	2.5	2.5	0.6
Waste cold, kW	772	764.57	963.29
Waste cold as a % of energy delivered	1.41	1.41	1.88

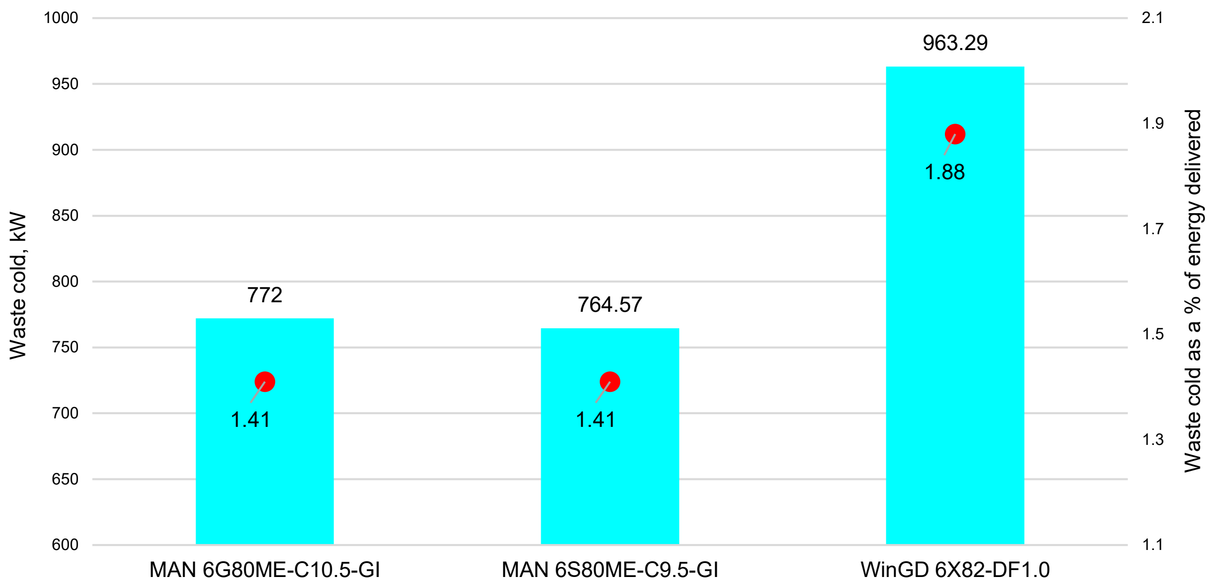

Figure 6. Comparison of the amount of waste cold from LNG regasification

Table 6. Waste cold as a % of energy delivered

	Engine load as % of maximum continuous rating	MAN G-type ME-GI	MAN S-type ME-GI	WinGD X-DF
Waste cold as a % of energy delivered	25	1.41		1.88
	100	1.38		1.86

It was also noticed that the waste cold amount as a percentage of the energy delivered in gas fuel slightly increases with engine load. Typical values for loads equal to 25% and 100% of maximum continuous rating are shown in Table 6.

A simple linear regression model based on the method of least squares was applied. For the statistical analysis, an intercept model was used, which includes parameters that may affect the final formula and were not included in the process of statistical analysis. The following formulae for MAN ME-GI and WinGD X-DF engines were obtained as a result of the calculations:

- **MAN ME-GI** (including all IMO Tier III-compliant technologies):

$$\dot{Q} = 0.0268 \cdot P_B + 2.7953 \tag{2}$$

where:

$\dot{Q}$ – amount of waste cold from the LNG regasification, kW;

P_B – brake power of the engine, kW;

and:

$N = 784$ – sample size (16 load points of 25÷100% of the maximum continuous rating for 14 available engine models in variants equipped with different SCR or EGR technologies),

$Z = 95\%$ – confidence level,

$R = 0.99915$ – Pearson correlation coefficient,

$R^2 = 0.9983$ – coefficient of determination,

$V = 0.0323$ – coefficient of variation (exemplary value for the fixed brake power of 25,000 kW),

$E = 19.17$ kW – standard error.

The amount of waste cold from the LNG regasification as a function of the gas-fuelled MAN ME-GI main engine brake power is shown in Figure 7.

- **WinGD X-DF**:

$$\dot{Q} = 0.0365 \cdot P_B + 12.596 \tag{3}$$

where:

$\dot{Q}$ – amount of waste cold amount from the LNG regasification, kW;

P_B – brake power of the engine, kW;

and:

$N = 256$ – sample size (16 load points of 25÷100% of the maximum continuous rating for 13 available engine models),

$Z = 95\%$ – confidence level,

$R = 0.9998$ – Pearson correlation coefficient,

$R^2 = 0.9996$ – coefficient of determination,

$V = 0.0159$ – coefficient of variation (exemplary value for the fixed brake power of 25,000 kW),

$E = 9.43$ kW – standard error.

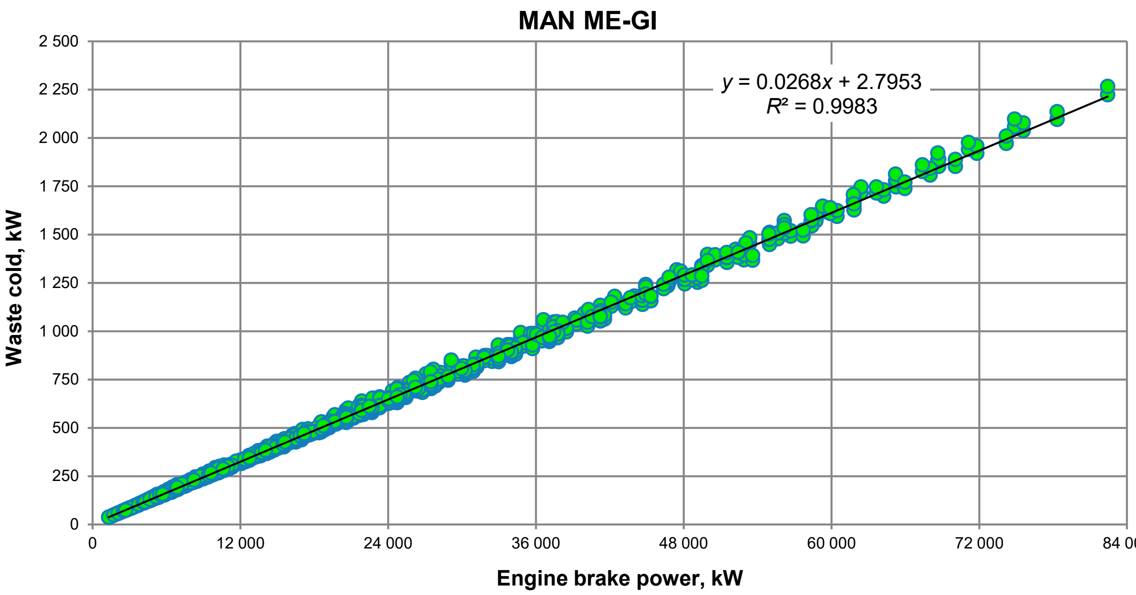

Figure 7. Amount of waste cold from the LNG regasification as a function of the gas-fuelled MAN ME-GI main engine brake power

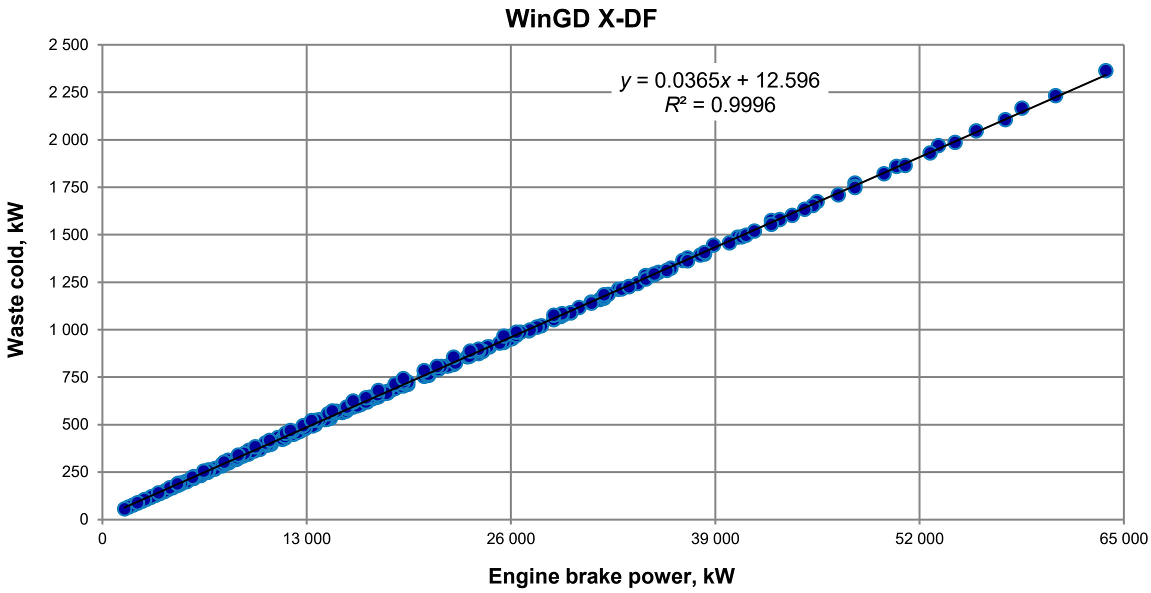

Figure 8. Amount of waste cold from the LNG regasification as a function of the gas-fuelled WinGD X-DF main engine brake power

The amount of waste cold from LNG regasification as a function of the gas-fuelled WinGD X-DF main engine brake power is shown in Figure 8.

Attention should be drawn to the very high coefficient of determination values ($R^2 \cong 1$) obtained in developing the formulae used to predict the amount of waste cold, which indicates a strong relationship between the studied dependencies and fit the appropriate regression model.

The obtained formulae (2÷3) allow a sufficiently accurate prediction of the waste cold amount at the initial design stage in a fast and simple way. The indicated value of the coefficient of determination R^2 allows the application of formulae (2÷3) with a high probability that the preliminary calculations will be satisfactorily close to the results of accurate verifying calculations at the technical design stage.

Conclusions

Based on a comparison of the received results, it was found that, in the case of the application of the WinGD X-DF engines, it is possible to obtain a larger amount of waste cold for the same load in relation to both MAN ME-GI types due to a much larger enthalpy difference of the LNG regasification process Δh and increased gas fuel consumption at the cost of reducing pilot oil consumption.

These differences result directly from their design and operational features, which determine the LNG regasification conditions and combustion process. WinGD X-DF engines realize the Otto cycle, in which the injection of gas fuel occurs under relatively low pressure, not exceeding 1.5 MPa. Under these conditions, while maintaining a fixed final temperature of the LNG regasification process, the enthalpy difference is significantly bigger (Δh = 943.52 kJ/kg) than for MAN ME-GI engines realizing the Diesel cycle and requiring the injection of gas fuel pressurized up to 30 MPa (Δh = 717.32 kJ/kg).

The very high values of the R^2 coefficient confirm that formulae determined by the least-squares method can be useful in the prediction of the waste cold amount at the initial design stage. It provides a basis for deciding how to utilize the available waste cold amount in an optimal way for a given type of ship equipped with a gas-fuelled low-speed main engine.

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