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A CONCEPT OF A MARINE POWER PLANT SUPPLIED WITH NATURAL GAS WITH A REDUCED CO_2 EMISSION INDEX

Key words: marine power plant, CO_2, Waste Heat Recovery System, EEDI, Ro-Pax vessel.

Abstract: This paper shows the background for the development of the Ro-Pax type ships in the Baltic Sea area as the Emission Control Area (ECA) that led the authors of the paper to use the Energy Efficiency Design Index (EEDI) as a measurement of meeting the requirements of the International Maritime Organization (IMO). For the studied concept of the marine power plant, dual fuel main engines manufactured by MAN B&W and adjusted for the requirements of ECA have been chosen. The mathematical model for the power plant that was used to perform calculations for various ranges of engine load was described. The value of the EEDI has been determined for the three studied possible kinds of fuel.

Koncepcja okrętowego układu energetycznego zasilanego gazem ziemnym o zredukowanym wskaźniku emisji CO_2

Słowa kluczowe: napędy morskie, CO_2, system odzyskiwania ciepła z odpadów, EEDI, statki typu Ro-Pax.

Streszczenie: W artykule pokazano tło rozwoju statków typu Ro-Pax w akwenie morza Bałtyckiego jako strefy objętej Emission Control Areas – ECA. Fakt ten skłonił autorów do wykorzystania Energy Efficiency Design Index – EEDI jako miary spełnienia wymagań International Maritime Organization – IMO. Przyczynił się on do opracowania niniejszego artykułu. Dla opracowywanej koncepcji układu energetycznego dobrano dwupaliwowe silniki główne firmy MAN B&W, dostosowując je do wymagań strefy ECA. Opisano model matematyczny układu energetycznego, w oparciu o który wykonano obliczenia w funkcji różnych zakresów obciążeń silników. Wartości współczynnika EEDI wyznaczono dla rozpatrywanych trzech możliwych rodzajów paliwa.

Introduction

Economic development of the countries from the Baltic Sea region generates the need to build new ships that will meet the requirements of the increasing transportation needs while abiding by environmental protection regulations. As a consequence of intensive development, the construction of the first in a series of ferries of the Ro-Pax type ordered by a Polish ship owner was started and one of a similar type has been purchased and is in operation. Such ships are destined to operate in the ECA region with limits on the emission of sulphur and nitrogen oxides (SO_x and NO_x). A significant contribution of marine transport into the global emission of carbon dioxide made IMO introduce EEDI in 2013. According to the definition, the index is a quotient of the amount of the emitted carbon dioxide to the amount of cargo transported by the ship per nautical mile. The expected value of EEDI for a given type of ship may be reached in a number of ways, the most effective of which is by employing waste heat recovery from the ship engine operation.

The complexity of processes of chemical treatment of exhaust and waste heat recovery as well as the chance to use LNG, apart from marine fuels, creates a decision making problem at the designing stage. Those facts convinced the authors to make an attempt to work out a concept of a power plant for a passenger-vehicle vessel enabling the performance of a transportation task in the ECA region.

1. Analysed vessel and power plant

Dimensions and main parameters of the passenger-vehicle vessel described in paper [1] and listed in Table 1 were used for initial calculations of the ship power plant. A decomposition diagram of the designed power plant together with marked state parameters of the working media of the waste heat recovery system is shown in Fig. 3.

For the propulsion system of the vessel, two driving units with medium-speed diesel engines were proposed. Using a reduction gear and shafts, two propellers are driven with a regulated pitch. The ship power plant also comprises machines and equipment servicing the main power, enabling the realization of cycles, and the controlled transfer of power. Holtrop's method [1, 5] was used for initial calculations of power in the conceptual design, as well as characteristics of free propellers of Wageningen-C models developed by Maritime Research Institute Netherlands – MARIN (Maritime Research Institute Netherlands) [4]. A system of non-linear equations in the range of [120–145 min^{-1}] rotational speeds of the propeller was solved using an iteration method in order to determine the non-dimensional coefficients of advance, moment, and thrust of the propeller. Optimum rotational speed was determined at the moment when the maximum efficiency of the open water propeller, η_0 was reached. The determined parameters of driving propellers and the assumed ranges of operational power values enabled the determination of the nominal power of the continuous operation of the engines of the main power plant.

Table 1. Main parameters of the designed vessel

No.	Unit	Value	Description	No.	Unit	Value	Description
L_{pp}	[m]	179.2	Length between pp.	v_s	[m/s]	10.55	Service speed
K_L	[m]	184.7	Length on waterline		[knt]	20.50	
B	[m]	29.5	Breadth	z	[–]	2	Number of propellers
T	[m]	7.15	Design draft	$P_{0.7R}/D$	[–]	1.2	Pitch – diameter ratio
C_B	[–]	0.65	Block coefficient	n_p	[1/min]	134	Propeller revolutions
Δ	[Mg]	25200	Mass displacement	Z	[–]	2	Number of engines
m_{DWT}	[Mg]	7471.5	Deadweight	P_{cSR}	[kW]	8400	Power of each engine

Source: Own elaboration on the basis of [1].

2. Determination of the type of main engines

Possible dual fuel engines from the most popular manufacturers on the ship market, i.e. MAN B&W 61/50DF and Wärtsilä 46DF, were considered. The criterion of choice for the dual fuel engine was its specific heat consumption and the manufacturer's declaration regarding meeting the IMO Tier III regulation. When natural gas is used, the specific heat consumption is determined as the sum of specific gas consumption (calorific value of liquefied natural gas 48,0 MJ/kg) and that of the pilot dose of oil (W_{FO} = = 42,7 MJ/kg):

$$\begin{aligned} \dot{q}_{fNG} &= \dot{q}_{NG} + \left(b_{fPFO} W_{FO}\right) = \\ &= \dot{q}_{NG} + \dot{q}_{fPFO} \ [kJ/kWh] \end{aligned} \tag{1}$$

In the case when only oil is used, for the given the specific fuel consumption listed by the manufacturer [6], the relation for the specific heat consumption is as follows:

$$\dot{q}_{fFO} = b_{fFO} W_{FO} [kJ/kWh] \tag{2}$$

Determination of specific heat consumption for the studied engines of the main power system of the designed ferry is shown in Fig 2.

Graphic representations of relations (1) and (2) indicated the most advantageous values of specific heat consumption of engines of the 51/60 DF type manufactured by MAN B&W. The function describing the specific heat consumption of the stated engine reaches its minimal value for the range of load of

85–94%. When it is supplied with oil, the range of optimum operation is narrower, i.e. 85–88% of the nominal power.

The above and the value of engine nominal power close to the maximum continuous power were decisive for choosing the engines of the 9L51/60DF type manufactured by MAN B&W as the engines of the main power.

The proposed power system takes into account the installation of a waste heat recovery system (Fig. 3) for the design parameters of secondary energy carriers, and they were identified for a chosen engine. Temperature, exhaust gas mass, and charging air fluxes presented in Fig. 2 refer to the situations when the engines are supplied with both natural gas and oil.

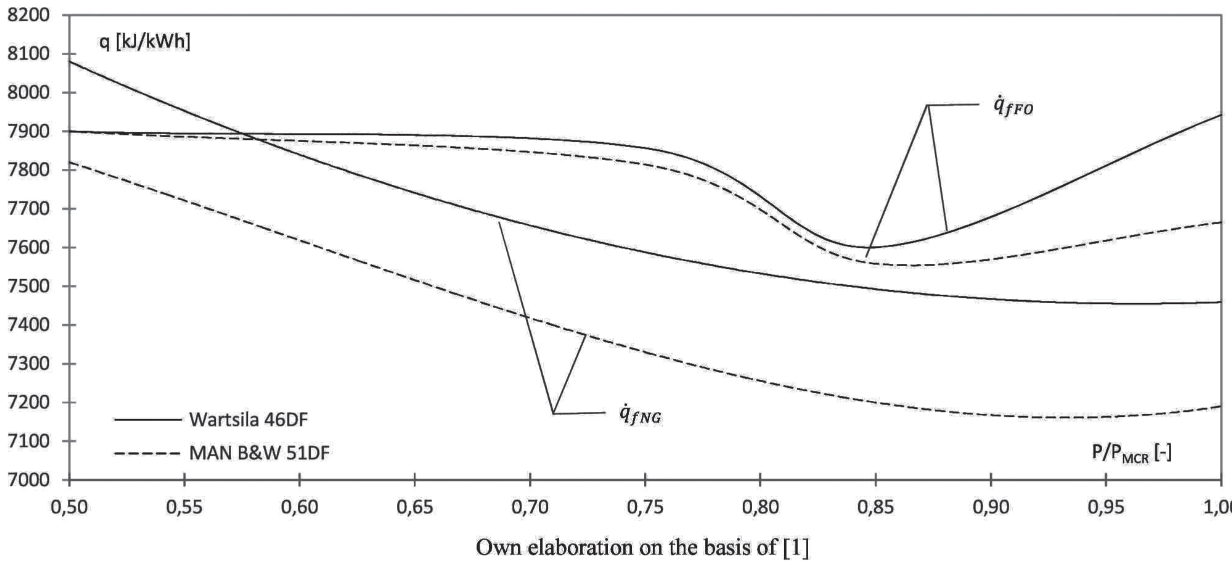

Fig. 1. Specific heat consumption for the studied dual fuel main engines

Source: Own elaboration on the basis of [1].

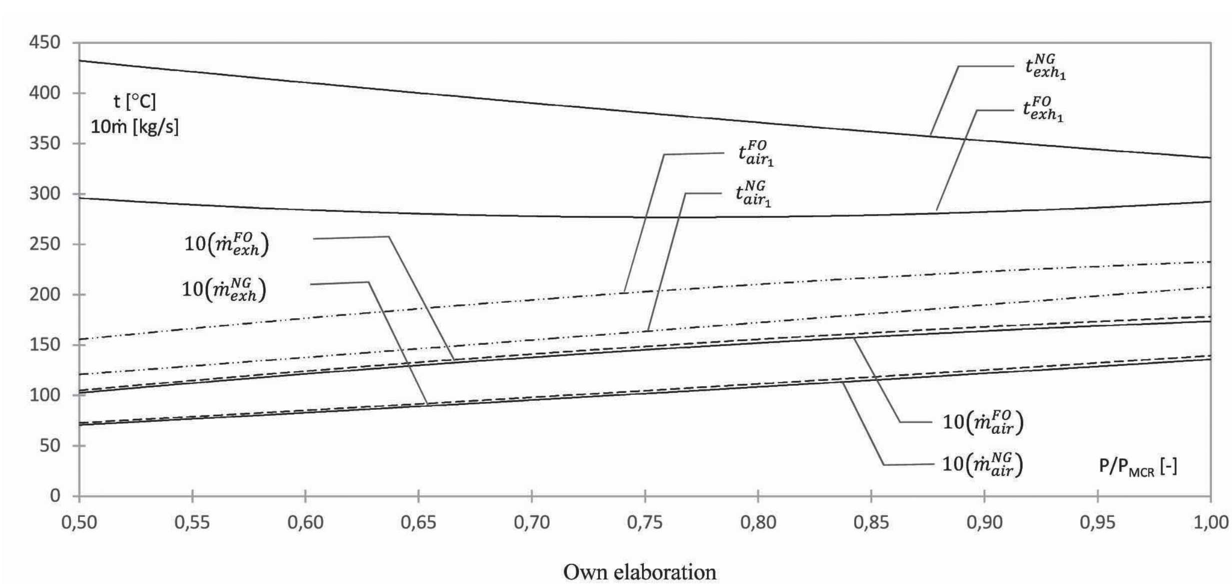

Fig. 2. Temperature and exhaust gas mass and charging air fluxes as a function of load of the 9L51/60DF type engine manufactured by MAN B&W type 9L51/60DF

Source: Own elaboration.

To exemplify the advantages of applying natural gas as fuel over using marine oil, parameters of waste heat recovery and EEDI were calculated.

3. Mathematical model of power plant with Waste Heat Recovery System

The mathematical model of the designed power plant was constructed for an operational state i.e. sea voyage. Mathematical description was confined to basic functions needed to carry out the transportation task at the expected value of EEDI.

The energy balance comprises flows of the chemical energy of fuel supplied to main and auxiliary engines:

$$\sum_{i=ME_j}^{AE_k} \dot{Q}_{Fi} = \sum_{j=1}^{2} \dot{Q}_{F\ ME_j} + \sum_{k=1}^{2} \dot{Q}_{F\ AE_k} = \\ = \sum_{j=1}^{2} \left(\dot{q}_{F\ ME_j} \cdot P_{ME_j} \right) + \sum_{k=1}^{2} \left(\dot{q}_{F\ AE_k} \cdot P_{AE_k} \right) \tag{3}$$

The total flow of energy delivered to the engines (3) is the sum of energy flows delivered to the main and auxiliary engines equivalent to the sum of multiplied specific heat consumptions and effective power. The effective power of main and auxiliary engines is described as follows:

$$\sum_{i=ME_j}^{AE_k} P_{e\ i} = \sum_{j=1}^{2} P_{e\ ME_j} + \sum_{k=1}^{2} P_{AE_k} \tag{4}$$

Assuming the efficiency of η_S propulsion transfer and determining rotational efficiency according to [5], for the known coefficients of advance K_T and torque K_Q the effective power of main engines was calculated from the relation below:

$$\sum_{j=1}^{2} P_{e\ MEj} = \frac{1}{\eta_S} \sum_{j=1}^{2} \overline{P}_{D\ j} = \frac{1}{\eta_S} \sum_{j=1}^{2} \overline{Q}_{D_j}\, \omega_{P_j} = \\ = \frac{1}{\eta_S} \frac{1}{\eta_R}\, \varrho \left\{ \sum_{j=1}^{2} \omega_{P_j} \left[K_{Q_j} (n_{P_j}{}^2 D_{P_j}{}^5) \right] \right\} \tag{5}$$

The contribution of electrical energy flow destined for servicing the engines of the main power was calculated using an empiric relation given by Marine Environmental Protection Committee [7]:

$$\sum_{k=1}^{2} P_{AE\ k} = \left[0{,}025 \left(\sum_{j=1}^{2} P_{e\ MEj} + \frac{\sum_{o=1}^{2} P_{PTI\ o}}{0{,}75} \right) \right] + 250 \tag{6}$$

Calculations of parameters of the Waste Heat Recovery System have been conducted for the proposed configuration, which consists of the following (Fig. 3): double-pressure steam-operated exhaust gas boiler – low pressure (1) and high pressure (4), charged air – boiler feeding water heat exchanger of the engine (2), high pressure steam separator (3), low pressure steam separator (5), steam turbine driving ship's synchronous generator (6), and steam condenser (7). The calculations have been conducted for the combustion products generated in the working process of MAN B&W 8L51/60DF for the load range of $\frac{P_{SR}}{P_{MCR}} = (0{,}5 \div 1{,}0)$. For the calculation process the following have been assumed:

– The high and low pressure feed water mass flow is equal to the calculated steam mass flow:
$\dot{m}_{SS}^{HP} = \dot{m}_{SH}^{HP} = \dot{m}_{FW}^{HP} = \dot{m}_{S/W}^{HP} = \dot{m}_{HP}$,
$\dot{m}_{SS}^{LP} = \dot{m}_{SH}^{LP} = \dot{m}_{FW}^{LP} = \dot{m}_{S/W}^{LP} = \dot{m}_{LP}$.
– The high pressure and low pressure boiler feed water temperature is equal to the temperature of saturated steam:
$t_{FW}^{HP} = t_{SS}^{HP}$, $t_{FW}^{LP} = t_{SS}^{LP}$.
– The exhaust gas temperature is higher than high and low pressure steam saturation temperature
$$t_{exh3} = t_{SS}^{HP} + \Delta T,\ \ t_{exh5} = t_{SS}^{LP} + \Delta T$$
where $\Delta T = 10K$.
– The high and low pressure superheated steam temperature is lower than exhaust temperature
$$t_{SH}^{HP} = t_{exh1} - \Delta T,\ t_{SH}^{LP} = t_{exh3} - \Delta T$$
where $\Delta T = 12\,K$.
– The temperature at the end of waste heat boiler (assuming 3.5% sulphur content in fuel oil) should be higher than or equal to
$$t_{exh4} \geq 145\,°C + \Delta T = 160\,°C$$
where $\Delta T = 15\,K$.
– The temperature of water in the feed water tank is constant and equals to $t_{FW} = 70°C$.

By applying the first law of thermodynamics and assuming the power of feeding pumps to be negligible and efficiency of exhaust boiler $\eta_B = 1$, the high pressure steam mass flux possible to be generated in the exhaust gas boiler was determined as follows:

$$\dot{m}_{HP} = \frac{\left[\dot{m}_{exh} \cdot c_{p_{exh}} \Big|_{t_3}^{t_1} \cdot \left(t_{exh_1} - t_{exh_3} \right) \right] +}{i_{SH}^{HP} - i_{FW}} \\ \frac{+ \left[\dot{m}_{air} \cdot c_{p_{air}} \Big|_{t_2}^{t_1} \cdot \left(t_{air_1} - t_{air_2} \right) \right]}{} \left[\frac{kg}{s} \right]. \tag{7}$$

The exhaust gas temperature was calculated behind the steam super heater using the following equation:

$$t_{exh2} = t_{exh1} - \frac{\dot{m}_{HP} \cdot \left(i_{SH}^{HP} - i_{SS}^{LP}\right)}{\dot{m}_{exh} \cdot c_{p_{exh}}\Big|_{t_2}^{t_1}} \; [°C] \tag{8}$$

The temperature of exhaust gases as it passes the HP evaporator section was calculated as follows:

$$t_{exh3} = t_{exh2} - \frac{\dot{m}_{HP} \cdot \left(i_{SS}^{HP} - i_{FW2}^{HP}\right)}{\dot{m}_{exh} \cdot c_{p_{exh}}\Big|_{t_3}^{t_2}} \; [°C] \tag{9}$$

Under the abovementioned assumption on the equality of mass fluxes of produced steam and water, the temperature of the charged air of the engine was calculated behind the HP water heater as follows:

$$t_{air2} = t_{air1} - \frac{\dot{m}_{FW}^{HP} \cdot c_{p_{FW}}\Big|_{t_2}^{t_1} \cdot \left(t_{FW} - t_{FW2}^{HP}\right)}{\dot{m}_{air} \cdot c_{p_{air}}\Big|_{t_2}^{t_1}} [°C] \tag{10}$$

Low pressure section parameters were calculated in a similar manner using equations (7) – (10).

In the proposed WHRS configuration, the steam turbine is driven by the produced superheated steam, and assuming the pressure and dryness fraction to be $x = 0.86$ at the end of the expansion process, the corresponding enthalpy in the condenser i_{con}, was estimated. The effective power of a two- stage steam turbine P_{ST} was determined by means of the following relation:

$$P_{ST} = [\dot{m}_{SH}^{HP} \cdot \left(i_{SH}^{HP} - i_{con}\right) + \dot{m}_{SH}^{LP} \cdot \left(i_{SH}^{LP} - i_{con}\right)] \cdot \eta_{I\,ST} \cdot \eta_{M\,ST} \; [kW] \tag{11}$$

Finally, the electric power $P_{ST_{el}}$ that is produced by the ship's synchronous generator driven by steam turbine was calculated using the following:

$$P_{ST_{el}} = P_{ST} \cdot \eta_G \left[kW\right] \tag{12}$$

Due to the assumption of the equal rotational speed of the shafts, $n_{ST} = n_G$, only the synchronous generator efficiency was taken into account as $n_G = 0.95$.

4. Energy Efficiency Design Index calculation for different types of marine fuel

The general mathematical relationship given by IMO for EEDI determination together with symbols presented in the mathematical model are as follows [7]:

$$EEDI = \frac{\left(\sum_{j=1}^{2} P_{e\,ME\,j} \cdot b_{f\,ME\,j} \cdot C_{F\,ME\,j}\right) + }{m_{DWT} \cdot v_s \cdot f_{jRoRo} \cdot f_{cRoPax}}$$

$$\frac{+\left(\sum_{k=1}^{2} P_{e\,AE\,k} \cdot b_{f\,AE\,k} \cdot C_{F\,AE\,k}\right) - }{} \tag{13}$$

$$\frac{-\left\{\left(\sum_{j=1}^{2} P_{STel\,j}\right)\left(\sum_{k=1}^{2} b_{f\,AE\,k} \cdot C_{F\,AE\,k}\right)\right\}}{}$$

Specific fuel consumption of auxiliary engines, at the initial design stage, was assumed according to [2] as that for half of the nominal load. The type of auxiliary engines was assumed to be the same as that of the main engines and specific fuel consumption and heat were read from [6]. Due to using dual fuel diesel engines as both the main and auxiliary engines, specific fuel consumption in relation (14) for running on LNG was determined as a specific natural gas consumption and that of a pilot dose of oil according to [7]. The value denoted as C_f was defined as a dimensionless conversion coefficient between fuel consumption and carbon dioxide emission. The coefficient reaches different values depending on the kind of applied fuel, i.e. for natural gas – C_{fNG} = = 2,750 gCO_2/g, and for the heavy fuel – C_{fHFO} = 3,114 gCO_2/g, reaching a maximum value for the distilled low -sulphur value – C_{fMDO} = 3,206 gCO_2/g [7].

In relation (14), f_{jRoRo} denotes a correction coefficient, taking into account the construction qualities of the vessel, described by the equation below:

$$f_{jRoRo} = \left\{ Fn_P{}^{\alpha} \cdot \left(\frac{L_{PP}}{B}\right)^{\beta} \cdot \left(\frac{B}{T}\right)^{\gamma} \cdot \left(\frac{L_{PP}}{\nabla^{1/3}}\right)^{\delta} \right\}^{-1} \tag{14}$$

Determination of the correction coefficient in respect to deadweight was based on relation (16), and registered tonnage was estimated using exemplary unit.

$$f_{cRoPax} = \left\{\frac{\left(m_{DWT} / GT\right)}{0,25}\right\}^{-0,8} \tag{15}$$

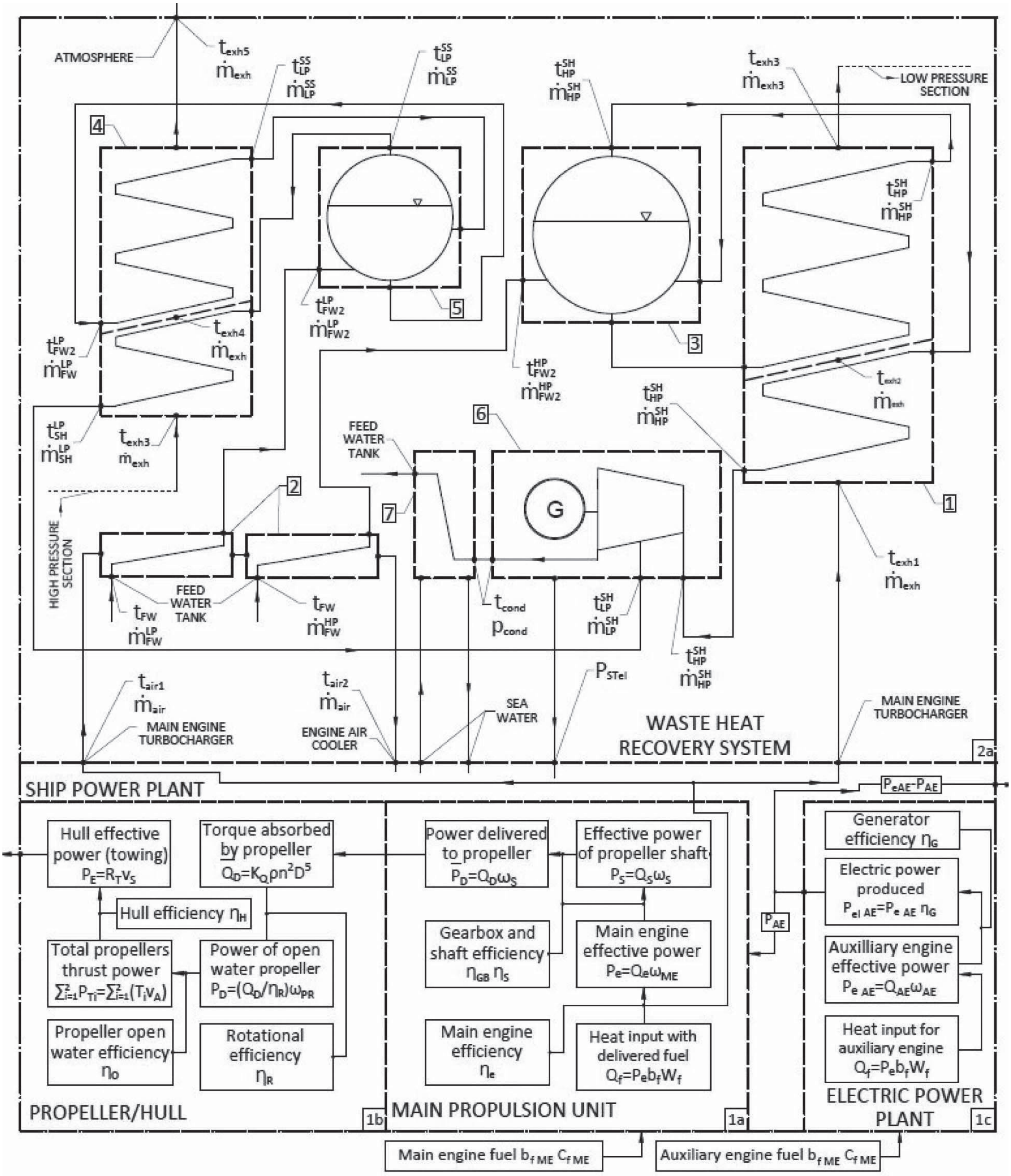

Fig. 3. Graphic representation of calculation procedures in a ship power plant

EEDI calculations were performed for three types of applied fuel:

– Using liquefied natural gas (LNG) for running dual fuel engines of the main power plant and the auxiliary engines applying the presented waste heat recovery system;
– Using heavy fuel oil (HFO) with a sulphur content over 3.5% of the fuel mass with chemical treatment of the exhaust to reduce the amount of sulphur oxides and using waste heat from the main power engines; and,
– Using marine oil with a sulphur content below 0.1% of the fuel mass applying the SCR reactor to obtain satisfactory emission of nitrogen oxides from the considered power system.

5. Presentation of results

The calculated WHRS parameters using relations (7)-(12) for the HP and the LP sections of the boiler and for engine's load range, $\frac{P_{SR}}{P_{MCR}} = 0.75$, are presented in Table 2. The presented values correspond to heat recovery from one of the two main engines.

Figure 4 presents the calculated total effective power and the power for the low and high pressure of the utilization steam turbine.

Table 2. Waste Heat Recovery System parameter calculation results

Fuel type	$\frac{P_{SR}}{P_{MCR}}$	P_{SR}	t_{exh1}	t_{exh2}	t_{air1}	t_{air2}	ΣQ_{exh}	$\dot{Q}_{air}$	$\Sigma \dot{Q}_{rec}$	$\dot{m}^{LP}_{S/W}$	$\dot{m}^{HP}_{S/W}$	P_{ST}	P_{STel}	η_{rec}
[–]	[–]	[MW]	[°C]	[°C]	[°C]	[°C]	[MW]	[MW]	[MW]	[kg/s]	[kg/s]	[MW]	[MW]	[–]
HFO	0.75	6.30	276	272	207	184	1.86	0.34	2.21	0.182	0.651	0.44	0.43	0.193
LNG	0.75	6.30	381	272	162	117	2.63	0.46	3.09	0.185	0.916	0.71	0.70	0.228

Source: Own elaboration.

Figure 4 presents the calculated total effective power and the power for the low and high pressure of the utilization steam turbine.

The graph of limiting values of EEDI together with the points determined from relation (13) for three options of the used fuel is presented in Fig 5.

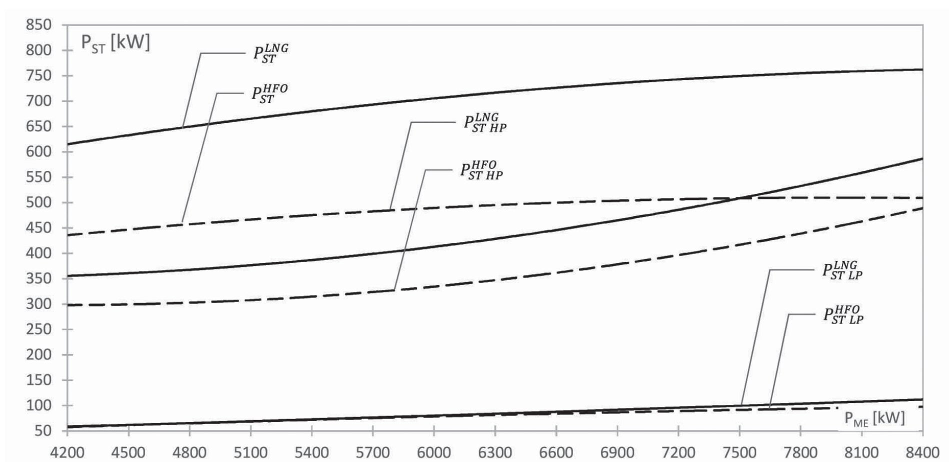

Fig. 4. Total effective power and effective power of low and high pressure utilization steam turbine operating in the waste heat recovery system
Source: Own elaboration.

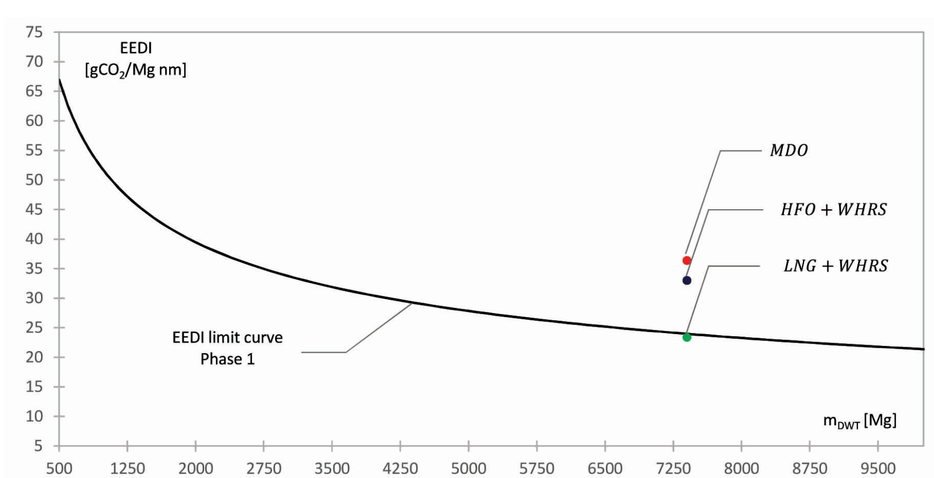

Fig. 5. A graph of EEDI limiting values determined for the studied options
Source: Own elaboration.

Conclusions

In the case when the engine is run on HFO with a sulphur content over 3.5%, on the basis of [3], it was assumed that the flow of exhaust gases would be directed towards the de-sulphuring system. This solution does not eliminate the risk of the occurrence of low temperature corrosion, which would cause a drop of enthalpy of the burning products from the main engines, but would reach a higher value for natural gas. The consequence of the above was also observed in Fig. 5. The described option meets only the regulation for SO_x emissions in the studied ECA region.

Using an SCR reactor in the exhaust system requires an appropriate temperature of exhaust for carrying out an effective process of their purification from NOx. That, according to [3], makes the application of waste heat recovery system impossible in such a configuration. Applying the low-sulphur fuel, MGO, together with the SCR reactor gives off expected emissions of SO_x and NOx, with the values of the energy efficiency design coefficient above the acceptable level (Fig. 6).

Using dual-fuel main and auxiliary engines run on natural gas together with the WHRS (Fig 3) gave off satisfactory emissions of SO_x, $NO_{x,}$ and the acceptable value of EEDI for the studied area of ship operation.

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