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Thermal efficiency investigations on the self-ignition test engine fed with marine low sulfur diesel fuels

Within the article an issues of implementing the new kinds of marine diesel fuels into ships' operation was described taking into account restrictions on the permissible sulphur content introduced by the International Maritime Organization. This is a new situation for ship owners and fuel producers, which forces the necessity to carry out laboratory research tests on especially adapted engine stands. How to elaborate the method enabling quality assessment of the self-ignition engine performance, considered in three categories: energy, emission and reliability, represents the key issue of the organization of such research. In the field of energy research, it is necessary to know the thermal efficiency of the engine as the basic comparative parameter applied in diagnostic analyzes and syntheses of sequentially tested marine diesel fuels. This type of scientific research has been worked out for two years in the Department of Marine and Land Power Plants of the Gdańsk University of Technology, as a part of the statutory activities conducted in cooperation with the Regional Fund for Environmental Protection in Gdansk and the LOTOS Group oil company.

This article presents the algorithm and results of thermal efficiency calculations of the Farymann Diesel D10 test engine in the conditions of feeding with various low-sulfur marine diesel fuels: distillation and residual fuels. This parameters stands for one of ten diagnostic measures of the ranking of energy and emission quality of newly manufactured marine diesel fuels being built at the Department.

Key words: *marine diesel fuels, engine tests, Sankey diagram, thermal efficiency*

1. Introduction

In accordance with the IMO (International Maritime Organization) decision, the permissible sulfur content in marine diesel fuels is drastically reduced starting with 1 January 2020, from the current maximum of 3.5% to 0.5% per unit masses, for vessels operating in all international waters, except for the previously designated Sulfur Emission Control Area (SECA)[1] [1, 3]. At the same time, it is still possible to use high-sulfur marine diesel fuels, provided that the ship is equipped with an exhaust desulfurization system, operating in a closed system, which will guarantee reduction of sulfur oxides emission in exhaust gases of marine engines to the level 6 g/kWh[2] [1, 2].

Observing a development of new technologies allowing the reduction of sulfur content in marine diesel fuels or the reduction of the content of sulfur oxides in engines' exhaust gases, it can be concluded that they are primarily focused on the maximum reduction of production costs, which will determine their further applicability. When comparing the prices of low- and high-sulfur marine diesel fuels, it turns out that having applied traditional technologies, a difference in the purchase price of 1 tons can reach even 300 dollars, which at the daily consumption of 20–30 tons gives savings up to 10,000 dollars [6].

In the long term, eg. annual perspective, this is up to 3 million dollars (taking into account necessary downtimes in the ship's usage). Then, it is worth considering retrofitting a ship's power plant with the exhausts desulfurization system, Such investment should pay back after 2–3 years of the ship's usage. For this reason, more and more shipowners decide to take such a step, as long as the technical conditions allow it.

On the other hand, intensive technological works are being undertaken by refine-ries to lower the production costs of low-sulfur marine diesel fuels, so-called modified fuels. This type of works additionally require conducting engine tests, aimed at assessing the energy, emission and reliability effects of their application. For obvious reasons, preliminary tests should be carried out in laboratory conditions, on specially adapted and metered engine test beds, according to the methodology that allows formulation of the unambiguous assessment of their suitability for feeding real objects, i.e. full-size marine Diesel engines.

This type of research works has been conducted for two years in the Department of Marine and Land Power Plants of the Faculty of Ocean Engineering and Ship Technology at the Gdańsk University of Technology [4]. One of the key issues of this methodology is the assessment of the energy quality of the tested marine diesel fuels, which is based on the thermal efficiency of the laboratory engine. Its construction should be maximally simplified, preferably single-cylinder, which guarantees high accuracy of calculations of the transformed energy streams [4, 7, 8].

2. Thermal efficiency of the SI engine

In order to determine the calculation formula for the thermal efficiency of a single-cylinder Diesel engine, it is necessary to develop a simplified energy balance of the working process realised in its cylinder section (Fig. 1).

Within the considerations, it was assumed that the energy balance equation will be determined for the cylinder section of the engine bounded by the system boundary, which stands for the inner surface of the cylinder and there is no accumulation of the internal energy of the working medium in the cylinder (its accumulation is negligibly small). Therefore, the balance equation for the processes taking place inside the cylinder section has the same form for the steady and unsteady processes:

$$\dot{\mathrm{H}}^*_{\mathrm{exch}} + \mathrm{P_i} + \dot{\mathrm{Q}}_{\mathrm{cs}} - \dot{\mathrm{H}}^*_{\mathrm{air}} - \dot{\mathrm{Q}}_{\mathrm{fuel}} = 0 \tag{1}$$

where: $\dot{\mathrm{H}}^*_{\mathrm{exch}}$ – enthalpy flux of the exhaust discharged from the cylinder, $\mathrm{P_i}$ – indicated power of the engine, $\dot{\mathrm{Q}}_{\mathrm{cs}}$ –

[1] In these zones, from January 1, 2015, the sulfur content in the fuel may not exceed 0.1% per mass unit.
[2] The investment costs of such an installation amount to EUR 4–5 million.

heat flux transmitted by the thermodynamic medium in the cylinder section to its walls, $\dot{H}^{*}_{air}$ – enthalpy flux of the air feeding the cylinder section, $\dot{Q}_{fuel}$ – heat flux brought to the engine with the fuel feeding the cylinder section.

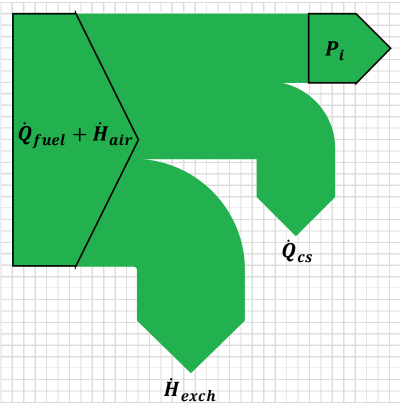

Fig. 1. Hypothetical flow diagram of energy flow in the cylinder section of a single-cylinder Diesel engine

Thermal efficiency represents the basic parameter of a Diesel engine characterizing the efficiency of its work in terms of thermal-flow approach, in steady states, i.e. when the average values of its effective torque, rotational speed and thermal state are unchanged over time. For these operating conditions, the average values of energy flows: input and output from the cylinder section also remain unchanged. In the simplest approach, thermal efficiency η_{th}, constitutes a ratio of the indicated power P_i to the energy flow brought to the engine with the feeding fuel $\dot{Q}_{fuel}$ and air $\dot{H}^{*}_{air}$ during one operation cycle:

$$\eta_{th} = \frac{P_i}{\dot{Q}_{fuel} + \dot{H}^{*}_{airpow}} \quad (2)$$

In the next steps of calculation, the following are determined:

A heat flux brought to the engine along with the fuel feeding the cylinder section:

$$\dot{Q}_{fuel} = \dot{m}_{fuel} \cdot \left(\xi_{cc} \cdot LCV + h_{fuel} \text{-} h_{fuelCC}\right) \quad (3)$$

where: $\dot{m}_{fuel}$ – a mass stream of the fuel feeding the cylinder section, determined by the weight method, from the amount of fuel m_{fuel} burnt in the engine at a given time, ξ_{cc} – a heat release coefficient in the combustion chamber, taking into account the losses of the heat brought with the fuel as a consequence of the incomplete combustion, dissociation of the exhaust gases and heat's penetration to the cylinder walls (it is estimated that ξ_{cc} takes values from 0.80 to 0.95, depending on the construction and dimensions of the combustion chamber and the value of the excess-air ratio), LCV – lower calorific value of the fuel, $h_{fuel} = c_{pfuel}(t_{fuel}) \cdot t_{fuel}$ – specific enthalpy of the fuel at the given temperature t_{fuel}, $h_{fuelCC} = c_{pfuel}(t_{CC}) \cdot t_{CC}$ – specific enthalpy of the fuel brought to the combustion chamber (CC) at the temperature inside the CC – t_{CC} (heat lost to warm the fuel to the temperature inside the combustion chamber).

In the first approximation (especially for distillation fuels, not requiring preheating before delivery to the engine's combustion chamber), the influence of the fuel enthalpy and heat release coefficient in the combustion chamber may be neglected, assuming, in a simplified form, that the heat flux brought to the engine with the fuel equals the flux of the fuel's chemical energy:

$$\dot{Q}_{fuel} = \dot{m}_{fuel} \cdot LCV \quad (4)$$

The enthalpy flux of the air feeding the cylinder section may be determined from the following relationship:

$$\dot{H}^{*}_{air} = \dot{m}_{air} \cdot c_{pair}(t^{*}_{air}) \cdot t^{*}_{air} \quad (5)$$

Due to small alterations of the air temperature at the intake of the naturally aspirated test engine, it might be assumed that the specific heat at constant pressure of the air c_{pair} is constant. In turn, the air mass stream $\dot{m}_{air}$ in equation (5) may be determined by using the mutual relations between the mass stream of the feeding fuel $\dot{m}_{fuel}$ and the mass stream of the air flowing into the working space of the cylinder section $\dot{m}_{air}$, which are determined with the excess-air-ratio λ:

$$\lambda = \frac{\dot{m}_{air}}{\dot{m}_{fuel} \cdot L_0} \quad (5)$$

where: L_0 – theoretical (minimum) air requirement for burning 1kg of fuel (constant for the given fuel type).

Hence, the mass stream of the feeding air is determined from the dependence, as follows:

$$\dot{m}_{air} = \lambda \cdot \dot{m}_{fuel} \cdot L_o \quad (6)$$

The theoretical air requirement for burning 1 kg of fuel with a known chemical composition expressed by mass contents of carbon C, hydrogen H, sulfur S and oxygen O is determined from the equation derived from the stoichiometric relations for total and complete combustion reactions (there is neither fuel nor oxygen in the exhaust) [7, 8]. After appropriate transformations, it takes the form in which L_0 is expressed in kg of air per kg of fuel:

$$L_o = \frac{1}{0{,}232} \cdot \left(\frac{8}{3}C + 8H + S\text{-}O\right) \ \frac{\text{kg air}}{\text{kg fuel}} \quad (7)$$

The above equations show that in order to calculate a thermal efficiency of the test engine, the measuring system should be designed in such a way to make possible determining: the engine's indicated power, the excess-air-ratio, temperatures of the feeding air and fuel as well as the fuel consumption in the adjusted, representative steady load condition. It is also necessary to know the basic chemical composition of the fuel and its calorific value [1].

3. Measurement results and their analysis

In order to investigate the effectiveness of the proposed method for determining a thermal efficiency of the test SI engine, experimental research was carried out on the Farymann Diesel engine of D10 type fed with six different marine diesel fuels. No adjustments were made to the engine injection system while introducing a new type of the fuel.

In the first stage of the research, the elementary chemical composition and lower calorific value of the tested fuels were determined – Table 1.

In the next stage of the research, the measurements of engine control parameters were carried out in accordance with the developed methodology – Fig. 2 [4]. As can be easily noticed from the presented scheme of recording and processing the measurement signals, in the engine tests of a new type of marine fuels, highly specialized measuring equipment was applied – stationary as well as portable, which allows precise observation of the engine's working and associated (residual) processes in steady and unsteady (transient) states.

In order to determine a thermal efficiency of the test engine, the following parameters were registered, simultaneously:

– rotational speed (angular position in ^{o}CSR) of the engine's crankshaft – n (measurement accuracy ±0.1%, sampling period – 0.5 ms),
– indicated pressure – p_c (measurement accuracy ±3%, sampling period – 15 µs, every 1^{o}CSR),
– fuel consumption – m_{fuel} (measurement accuracy ±0.2%, sampling period – 12.5 ms),
– fuel temperature – t_{fuel} (measurement accuracy ±2%, sampling period – 93.75 ms),
– consumption (by evaporation) of the cooling water – m_w (measurement accuracy ±0.2%, sampling period – 12.5 ms),
– cooling water temperature – t_w (measurement accuracy ±2%, sampling period – 93.75 ms),
– lubricating oil temperature – t_{ol} (accuracy of ±2%, sampling period – 93.75 ms),
– exhaust temperature – t_{exch} (measurement accuracy ±1%, sampling period – 0.1 ms),
– load current of the generator (armature) – I_{arm} (measurement accuracy ±1.5%, sampling period – 0.1 ms),
– voltage at the terminals of generator's armature – U_{arm} (measurement accuracy ±1.5%, sampling period – 0.1 ms),
– excess-air-ratio – λ (measurement accuracy ±1.0%, sampling period – 2 s),
– temperature of the test engine's external surfaces (thermogram) – t_M (measurement accuracy ±1.5%, sampling period – 0.1 ms),
– vibration acceleration (with SV80 converter) a_v (measurement accuracy ±0.1%, sampling period – 20 µs).

Table. 1. Basic physicochemical properties and chemical composition of the tested marine diesel fuels: distillation (PD) and residual (PP)

PARAMETER	PD1	PP2	PP3	PP4	PD5	PP6
Cetane number(dist.) / CCAI (Calculated Carbon Aromaticity Index) (resid.)	57,2	747	755	750	51	791
Density in w 15^{o}C, kg/m^3	827,1	884,5	872,7	878,7	820	885
Kinematic viscosity in 40^{o}C (dist.) / 50^{o}C (resid.), mm^2/s	2,99	308	77,83	165,30	2,37	16,48
Flashpoint temperaturę in a closed cup in ^{o}C	61,5	270	88	107	56	102
The content of carbon [%m/m]	86,26	86,10	86,14	86,12	86,63	86,54
The content of hydrogen [%m/m]	11,10	11,90	11,72	11,80	11,20	11,75
The content of nitrogen [%m/m]	0,05	0,02	0,027	0,02	0,04	0,02
The content of sulfur [%m/m]	0,09	0,01	0,028	0,01	0,0008	0,10
Lower Calorific Value [MJ/kg]	43,23	43,08	43,04	43,20	42,70	42,44

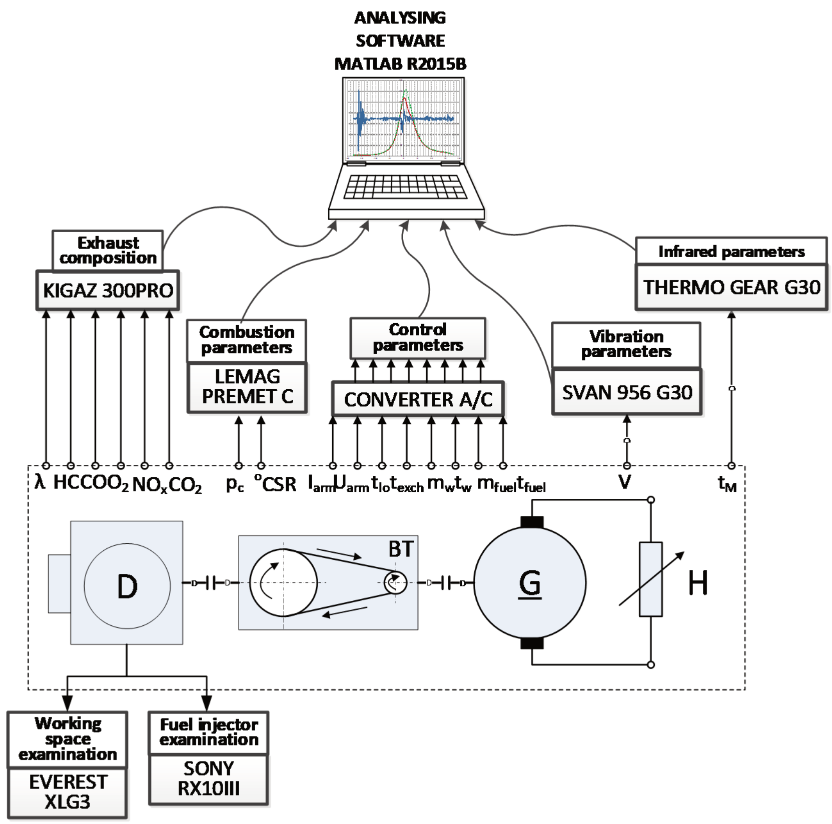

Fig. 2. Schematic diagram of the research test bed with measurement signal conditioning and recording system: D – single-cylinder Farymann Diesel D10 engine; BT – belt transmission (multiplier: transmission ratio i=0,426); G – direct current generator; H – heating system

Energy profiles' investigations of the laboratory engine fed with various types of marine diesel fuels may be carried out after reaching the determined thermal state of its structural design. It means that the cooling system "keeps up" with the receiving heat streams from the elements of the piston-cylinder group, which directly take off the heat released in the fuel combustion process worked out in the combustion chamber. The greater the temperature difference between the working medium and the walls limiting the combustion chamber, the greater the energy flow released as a result of fuel combustion is lost for warming the engine's construction elements. For this reason, before measuring the parameters of the working process, the engine should be wormed up, leading it to the state of thermal stabilization, in which the values of structural clearances and lubricating oil viscosities will be nominal. It is necessary to avoid then engine's long-term unloaded operation, at low rotational speeds, because in such conditions, the process of fuel atomisation and combustion occurs, in this case – incomplete, favouring a deposit formation in the engine's working spaces and in the exhaust passages as well as an increase in emissions of harmful and toxic chemical compounds in the engine's exhaust (especially carbon monoxide).

Within the experimental research program, whose main aim is to perform comparative analysis of the tested marine diesel fuels in terms of their energy quality, a registration of the control parameters is carried out in three/four measurement sequences (if the performed on-line preliminary analysis of the engine control parameters' measurement uncertainty, in particular, fuel consumption, does not indicate the presence of gross errors, the number of measurement sequences is limited to three), in one, always the same (reference) state of the determined engine load. Before each measurement sequence, which lasts exactly 20 minutes, the amounts of fuel and cooling water are made up to the established initial levels in the tanks. By this way, the influence of external disturbances (fixed thermal, flow, dynamic state of the engine, etc.) was minimized. Furthermore, an effectiveness of the observations of the engine functioning increases (the smallest parametrical anomalies are captured), as well as high repeatability of the recorded measurement results is achieved. Their mathematical processing is carried out according to the computational algorithm presented in the second point of the article. Final results of thermal efficiency calculations of the test engine are summarized in Table 2.

However, their graphical interpretation, in the form of an extended Sankey diagram, made for the whole propulsion unit fed with one of the tested fuels, is presented in Fig. 3 [4]. It includes, in addition to the heat flux emitted by the engine's hull to the surrounding $\dot{Q}_{sur}$, and the heat flux through the water in the water in the tank $\dot{Q}_w$, also the mechanical losses in the propulsion unit P_m, the engine's effective power Pe, which is equal to the generator's propulsion power P_{GP} and a stream of the residual heat $\dot{Q}_r$, which stands for energy losses not included in the balance (e.g. a fluxes of acoustic and mechanical vibration energy from the thermal-flow and mechanical system of the entire propulsion unit: engine-multiplier-generator).

Table 2. Calculated values of the engine's thermal efficiency characterizing the energy quality of the tested marine diesel fuels

Marine diesel fuel	PD1	PP2	PP3	PP4	PD5	PP6	Range
η_{th} [%]	58.75	58.58	59.06	58.80	61.79	60.07	3.31

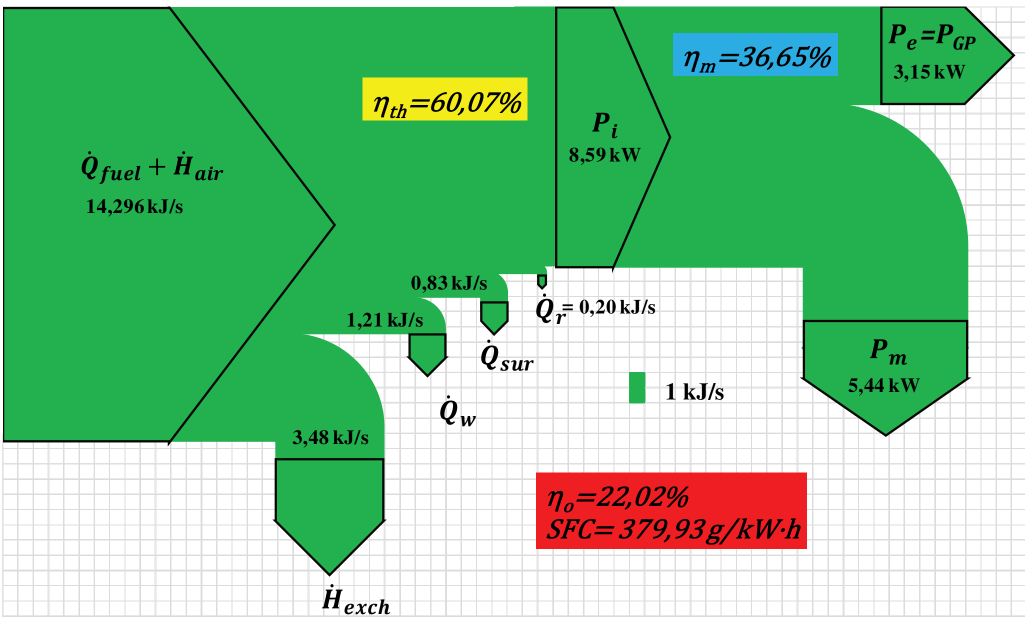

Fig. 3. A stream graph of the energy flow within the Farymann Diesel D10 engine propulsion unit fed with low-sulfur marine diesel fuel PP6

4. Remarks and final conclusions

The following general conclusions can be formulated on the basis of the results obtained:

– The proposed configuration of the measurement signals and processing system of the single-cylinder test engine makes it possible to determine its thermal efficiency, as the basic indicator of the energy quality assessment of newly implemented, modified marine diesel fuels;
– The fuel type alteration has a significant impact on a thermal efficiency of the engine. The value of its range for six tested marine fuels was 3.21%, which indicates a significant diagnostic sensitivity of this parameter.
– A higher calorific value of the fuel feed does not translate directly to the increased thermal efficiency of the engine.
– An impact of various marine diesel fuels on the energy state of the test engine, in the sense of its performance and efficiency, is a complex phenomenon. For this reason, it is necessary to significantly expand the set of diagnostic parameters, which are able, after appropriate normalisation, fulfill the role of a stimulant, destimulant and nominate, determining the energy quality assessment of the considered marine fuels. This will enable a construction of the ranking, in which they will be adjusted in the order from "the best" to " the worst" one according to the determined values of the synthetic (aggregate) variable [4, 5].

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