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Marcin ZACHAREWICZ
Ryszard ZADRĄG
Artur BOGDANOWICZ
Paweł SOCIK
Paweł WIRKOWSKI
Jakub HOŁOWNIA

Combustion Engines
Polish Scientific Society of Combustion Engines

Evaluation of the impact of the thermal state of a diesel engine on its efficiency

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The paper presents the results of model and empirical research on the influence of the thermal state of a diesel engine (oil temperature) on its indicated (thermal) efficiency. The paper contains a test plan, including a description of the test object, test equipment, and measurement points on a real object. In the following part, the results of tests carried out on a real object (laboratory single-cylinder engine) and the results of model tests obtained on the original engine model are presented. The results are presented both in tabular and graphical form. The obtained test results allowed to determine the relative value of the influence of the engine's thermal state on its efficiency for various operating conditions (load and rotational speed).

Key words: *marine diesel engine, engine efficiency, mathematical model*

1. Introduction

The issue of the influence of the thermal state of a diesel engine on its efficiency is important in the case of marine engines, both main propulsion and auxiliary engines driving marine generators. According to the operating guidelines, marine diesel engines of the main propulsion should be preheated [17, 18] to a temperature of about 323 to 333 K before starting them. This is done by heating the lubricating oil with electric heaters or with steam in the case of ships equipped with steam boilers. There are also solutions in which the oil is preheated in the main propulsion engines by means of water cooling the auxiliary engines.

The Polish Navy uses both wet and dry sump engines. For wet-sump engines, the oil heater is mounted in the engine crankcase. For engines with a dry sump (Zviezda type M 503 and M520), the heated oil is circulated by means of a pre-lubrication pump and a drain pump. There is no preheating requirement on auxiliary engines. Nevertheless, these engines, like the main engines, are installed in the ship's engine room and have an ambient temperature of about 293 K. In the case of the main engines, the heating process can be omitted in special situations, such as the ship's emergency exit to sea.

The idea of conducting research on the impact of the thermal state of a diesel engine on its efficiency is not new, but it has not been sufficiently explored in the case of powering marine reciprocating internal combustion engines.

Andrews et al. [1] showed that engine fuel consumption, for cars, has a linear dependence on ambient temperature. Over an urban drive cycle, the fuel consumption was shown to increase by 18% when the ambient temperature decreased from 304 K to 271 K. The work of Tobergte et al. [15, 17] highlighted a similar trend for three different variants of the engine (a 1200 cm^3 3-cylinder SI engine, a 1400 cm^3 4-cylinder SI engine, and an 1800 cm^3 SI engine) [11]. Kozak's research showed that the soot contamination of engine oil increases with its "mileage" [7, 8].

Burke et al. stated in their research work that an increase in the engine temperature from 323 K to 353 K reduces the engine friction by 44% because of 67% lower oil viscosity. Moreover, reduction in the emissions of nitrogen oxides (NO_x) and fuel consumption of 13.5% and 0.7% respectively have been achieved. The research investigated that hotter engine temperatures reduce ignition delay, making combustion occur earlier in the cycle, which has a positive effect on fuel consumption but a negative effect on NO_x emissions [4].

Genca and Radica in their research on the effect of heating a compression-ignition engine on its performance observed a decrease in fuel consumption and CO_2 production. The research was conducted in metropolitan and non-metropolitan conditions at low ambient temperatures [6].

Bielaczyc et al. [2] in their work point out that despite meeting the stringent standards imposed by the EU, toxic exhaust emissions, including NO_x, are still a major problem. According to the researchers, the main cause is the high viscosity of oil and friction of engine components. Therefore, they highlight the significance of the issue of conducting research on optimizing the performance of a cold-start vehicle both in terms of fuel consumption and toxic emissions.

Broatch A. et al. [3] evaluated the suitability of different biodiesel fuels, with and without additives, for cold starting DI (direct injection) diesel engines. The results have shown that the engine start-ability with pure biodiesel fuels can be deteriorated. The article also demonstrated that by using diesel/biodiesel blends the start-ability of the engine can be recovered with the additional benefit of reducing the opacity peak of the exhaust gases.

Tauzia et al. [14] show that the effects of coolant and oil temperature on engine behaviour are quite complex with several interactions :

– during the intake phase, the volumetric efficiency was modified. It affect for turbocharger (the energy available for the turbine being reduced when exhaust gas was at a lower temperature)
– turbocharger friction and heat transfer altered as well

- friction losses increased when oil or coolant temperature decreased
- the exhaust energy decreased.

Despite the requirement to heat the oil in the main engines before starting them, there are situations when the engine is started in the so-called cold state. Such a situation was recreated in the article, where empirical and model tests of the influence of the thermal state of the engine on its indicated efficiency (thermal efficiency) were carried out. In the empirical research, a stand of a single-cylinder engine driving an eddy current brake through a planetary gear was used. This stand was equipped with the equipment necessary for the tests, such as measurement of torque, crankshaft rotational speed, fuel and oil temperature, gravimetric measurement of fuel consumption, and the possibility of cylinder indication.

2. Study plan

In accordance with the research methodology, a test plan was developed, taking into account the test object, the measuring equipment used, and the measured energy parameters of the test object, i.e. in the case of the tested engine, its crankshaft rotational speed, engine torque load, oil temperature and indicated pressure. Parallel to the empirical research, model research was carried out using the author's model of a marine diesel engine with self-ignition [20, 21]. The study plan was presented in the form of an algorithm (Fig. 1).

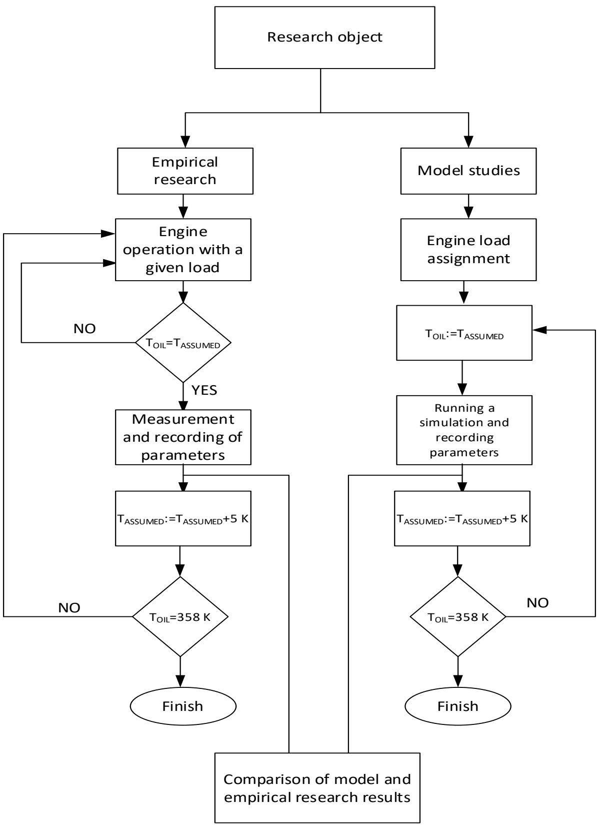

Fig. 1. An algorithm representing the study plan

Empirical and model tests were carried out for the same engine operating parameters (engine torque load and crankshaft rotational speed). In the case of the model, the thermal state of the engine itself was the input parameter (assumed the same as in the case of empirical tests). Both in the modelling and empirical studies, it was decided to measure to model) the course of the indicated pressure and the hourly fuel consumption. The courses of indicated pressure allowed to calculate the engine's indicated power. In addition, it was decided to calculate the engine power on the basis of its rotational speed and the torque registered on the eddy current brake.

It was decided to conduct tests for a fixed rotational speed of the crankshaft of 1000 and 1200 rpm, respectively, and for two load torque values using an eddy current brake (power generated by the engine of 5 and 6 kW respectively). It was assumed that the first measurement point would correspond to the oil temperature of 293 K, and each subsequent one would correspond to an increase in oil temperature by 5 K up to 353 K. During the tests, the following engine operating parameters were recorded:

- pressure indicated as a function of the angle of rotation of the crankshaft using a proprietary electronic indicator
- rotational speed of the crankshaft using a tachometric generator
- oil temperature with a PT100 type thermocouple
- hourly fuel consumption using a gravimeter
- torque on the basis of strain gauge readings installed on the eddy current brake.

The object of the research was a laboratory single-cylinder diesel engine installed at the stand at the Institute of Shipbuilding and Operation of the Polish Naval Academy. The basic engine data are grouped in Table 1, while the view of the laboratory stand is shown in Fig. 2 and Fig. 3 shows the control panel of the single-cylinder engine stand.

Table 1. Technical data of the engine used in the tests

Cylinder layout and number	horizontal, single cylinder
Piston stroke	160 mm
Cylinder diameter	135 mm
Cylinder displacement	2290 cm^3
Compression ratio	1:16
Maximum fuel consumption	215 g/kWh
Maximum torque at 1100 rpm	145 Nm
Maximum injection pressure	17 MPa
Maximum rated power	20 kW for 1500 rpm

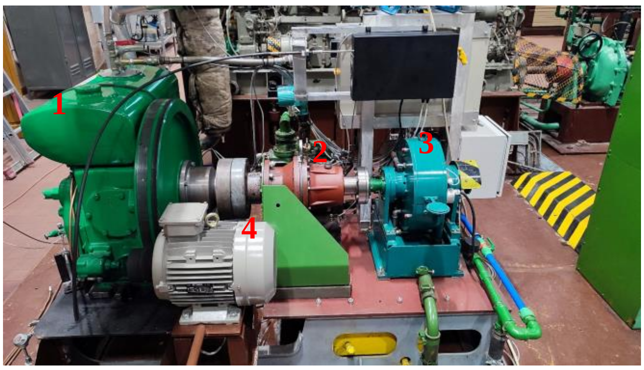

Fig. 2. Laboratory engine station: 1 – engine, 2 – planetary gear, 3 – brake, 4 – electric starter

The engine's fuel supply system consists of a fuel tank, a gravimetric system enabling measurement of fuel mass flow, a filter, a piston injection pump, a high-pressure pipe, and an injector with a pintle tip. The injector is equipped with a single spring. The fuel is injected directly into the chamber located in the piston.

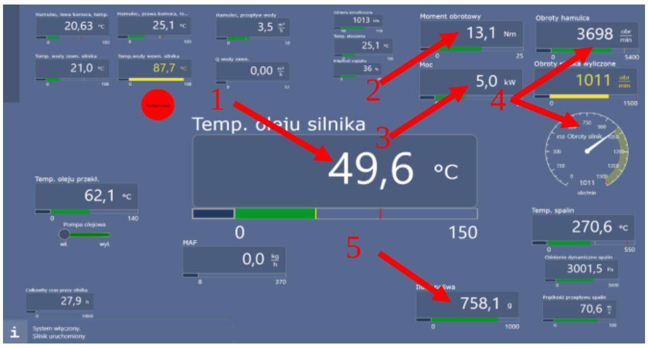

Fig. 3. Single-cylinder engine stand control panel: 1 – oil temperature, 2 – torque, 3 – engine power, 4 – rotation speed, 5 – mass of fuel in the tank

3. Empirical research

In accordance with the assumed test plan, all the mentioned parameters of engine operation were measured and recorded for given rotational speeds and loads. Measurements were recorded for oil temperatures ranging from 293 K to 353 K, respectively, with a resolution of 5 K. Engine operating parameters such as oil temperature, hourly fuel consumption, crankshaft rotational speed, and torque were recorded at a frequency of 1 Hz with a resolution of 12 bits. Indicated pressure was measured at a frequency of 10 kHz with a resolution of 12 bits. The measurement of the indicated pressure (for each measurement point) was repeated three times, and on its basis, the indicated engine power was calculated from dependence 1 [11–13, 16]:

$$N_i = V \cdot n \cdot p_i \cdot z \cdot i \tag{1}$$

where: V – engine cylinder capacity, n – average rotational speed of the crankshaft, z – number of ignitions per revolution of the crankshaft, p_i – indicated work, i – number of cylinders.

The indicated power calculated on the basis of eq. (1) was one of the input parameters to the mathematical model. In addition, the power was also calculated based on the measurement of the angular velocity of the crankshaft and the torque recorded by the sensors mounted on the eddy current brake based on the relationship:

$$N_c = \omega \cdot T \tag{2}$$

where:

$$\omega = 2 \cdot \pi \cdot n$$

Fuel consumption measurements to be calculated for the calculation of the enthalpy flux to the engine along with fuel from the relationship [20]:

$$\dot{H} = \dot{m} \cdot G_h \tag{3}$$

Having the values of the enthalpy flux supplied to the engine (eq. (3)) and the indicated power (eq. (1)), it was possible to calculate the indicated efficiency of the engine from the relation [9, 18, 19]:

$$\eta_i = \frac{N_i}{\dot{H}} \tag{4}$$

In addition, during the tests, the torque and rotational speed of the crankshaft measured on the brake made it possible to calculate the mechanical power of the engine-gearbox-brake system, and on this basis, to calculate the efficiency of the engine-gearbox-brake system based on the following relationship:

$$\eta_c = \frac{N}{\dot{H}} \tag{5}$$

In the case of the conducted tests, the fuel calorific value G_h was determined using a KL-11 type calorimeter, and the fuel mass flow was calculated using a gravimeter. On the other hand, the power value recorded on the eddy current brake takes into account all losses in the engine and in the planetary gear. Efficiency η_c can be described as the sum of indicated η_i, mechanical η_m, theoretical $\eta_{t,}$ and transmission η_{tr} efficiency.

4. Model research

Parallel to the empirical research, model research was carried out. An original mathematical model implemented in a computer program was used for model research [21]. The program is based on thermodynamic relationships and allows to conduct research for virtually any compression-ignition engine.

Input parameters for the mathematical model include, but are not limited to:

- basic technical parameters of the engine, including cylinder stroke and diameter, number of cylinders, valve opening and closing angles, injection advance angle, combustion chamber volume, etc.
- rotational speed of the crankshaft
- indicated power
- physical and chemical parameters of the fuel
- supply air parameters such as pressure, temperature, humidity, density
- addition, filling efficiency and much more are taken into account.

In accordance with the adopted test plan, it was assumed that the simulated rotational speed of the engine crankshaft would be consistent with the setting of the real engine. The indicated power was assumed to be consistent with that obtained as a result of empirical tests. The calorific value of the fuel G_h was determined by Guzma formula [10]:

$$\begin{aligned} G_h = 340 \cdot C_m + 1017 \cdot H_m + \\ + 63 \cdot N_m + 191 \cdot S_m - 106 - 25 \cdot w_m \end{aligned} \tag{6}$$

where: C_m – mass fraction of coal, H_m – mass fraction of hydrogen, N_m – mass fraction of nitrogen, S_m – mass fraction of sulfur, w_m – mass fraction of water

5. Research results

The results of both empirical and model tests were presented in the form of graphs of engine efficiency as a function of oil temperature. The efficiency of the real engine was calculated on the basis of dependence 4, while the efficiency of the modelled engine was calculated on the basis of dependence 5. In the case of empirical tests, the value of indicated power was calculated on the basis of indicated pressure measurements, and the value of the enthalpy flux was calculated on the basis of the mass flow of the fuel supplying the engine (overflow included) and its calorific value (dependence 4). In the model tests, the value of the indicated power, obtained from dependence 1, was the input parameter to the model. Based on the equations of the mathematical model, the fuel mass flow and its calorific value were calculated (dependence 6). As a consequence, it allowed us to calculate the enthalpy flux $\dot{H}$. The efficiency of the modelled engine was calculated on the basis of dependence 5. The indicated power value adopted in the equation in some cases may slightly differ from the indicated power obtained as a result of empirical tests. This is due to the fact that in the mathematical model used, the fuel consumption is calculated (in successive iterations), and on its basis the course of the indicated pressure and, consequently, the indicated engine power is determined. The condition for completing the model calculations is obtaining the value of the indicated (calculated) power not differing by more than 2% in relation to the power constituting the input parameter. In the efficiency calculations, the indicated power calculated as a result of solving the model equations was used [5].

In the case of tests conducted for the rotational speed of the crankshaft of 1200 rpm, for technical reasons, the maximum oil temperature was decided to be 343 K.

Significant discrepancies between the given engine load and the indicated power result, among others, from the design of the test stand used. The tested engine is loaded with an eddy current brake. Optimal parameters of cooperation between the engine and the brake (adaptation of rotational speed) are ensured by the planetary gear, which is a source of significant mechanical losses. The carrie out preliminary tests showed that these losses are practically constant (independent of the rotational speed of the engine crankshaft and its load). They depend only on the temperature of the oil in the transmission, which during the tests was constant and amounted to 353 K. Therefore, they are treated as a constant error (all analyses referred to the indicated power).On the basis of the empirical and modelling data the courses of indicated engine efficiency as a function of oil temperature (for the rotational speed of the crankshaft of 1000 rpm) were developed. They are presented in Fig. 4

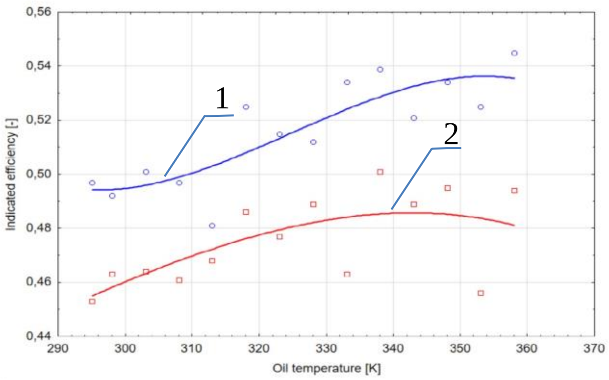

Fig 4. Courses of changes in engine efficiency as a function of oil temperature for an engine loaded with a power of 5 kW at a crankshaft speed of 1000 rpm: 1 – empirical research, 2 – model tests

– course for the engine load of 5 kW and Fig. 5 – course for the engine load of 6 kW. The waveforms obtained as a result of modeling for various loads are presented in Fig. 6, while the results of empirical tests are shown in Fig. 7.

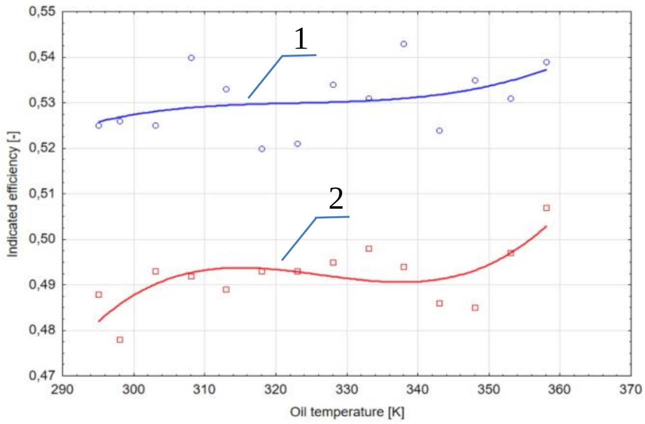

Fig 5. Courses of changes in engine efficiency as a function of oil temperature for an engine loaded with a power of 6 kW at a crankshaft speed of 1000 rpm: 1 – empirical research, 2 – model tests

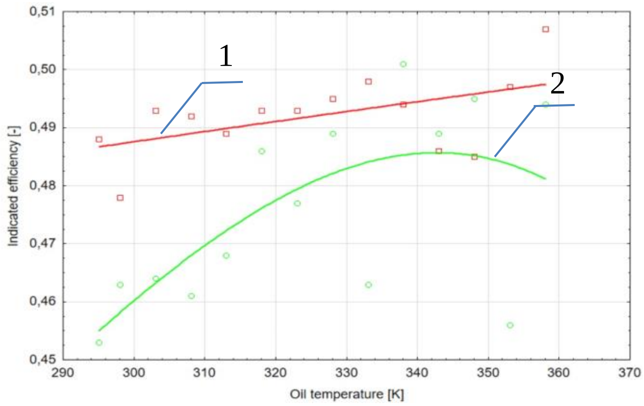

Fig. 6. Waveforms of changes in engine efficiency as a function of oil temperature obtained as a result of modelling an engine operating at 1000 rpm loaded with power: 1 – 6 kW, 2 – 5 kW

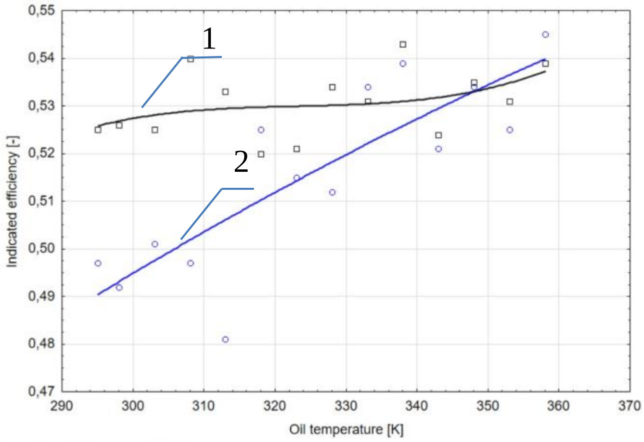

Fig. 7. Courses of changes in engine efficiency as a function of oil temperature obtained as a result of empirical tests for an engine with a rotational speed of 1000 rpm and loaded with the following power: 2 – 5 kW, 1 – 6 kW

On the basis of the empirical and modelling data, the courses of indicated engine efficiency as a function of oil temperature (for the rotational speed of the crankshaft of 1200 rpm) were developed. They are presented in Fig.: 8 – course for the engine load with a torque of 5 kW and 9 – course for the engine load with a torque of 6 kW. The waveforms obtained as a result of modelling for various loads are presented in Fig. 10, while the results of empirical tests are shown in Fig. 11.

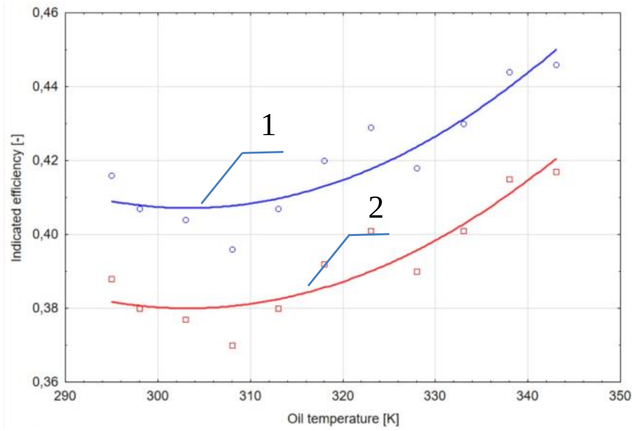

Fig. 8. Courses of engine efficiency changes as a function of oil temperature for an engine loaded with a power of 5 kW at a crankshaft speed of 1200 rpm: 1 – empirical tests, 2 – model tests

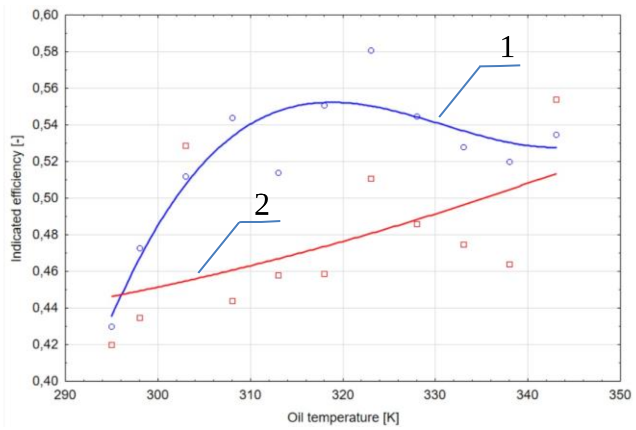

Fig. 9. Courses of changes in engine efficiency as a function of oil temperature for an engine loaded with a power of 6 kW at a crankshaft speed of 1200 rpm: 1 – empirical tests, 2 – model tests

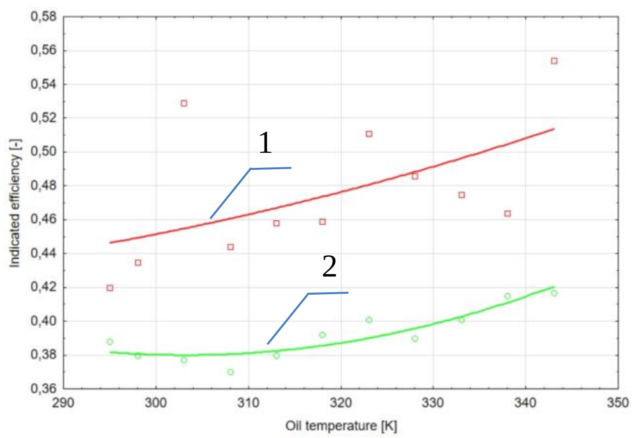

Fig. 10. Waveforms of changes in engine efficiency as a function of oil temperature obtained as a result of modelling an engine operating at 1200 rpm loaded with power: 1 – 6 kW, 2 – 5 kW

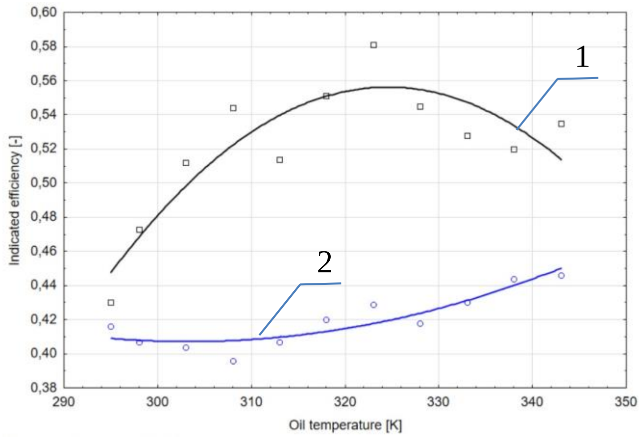

Fig. 11. Courses of changes in engine efficiency as a function of oil temperature obtained as a result of empirical tests for an engine with a rotational speed of 1200 rpm and loaded with the following power: 2 – 5 kW, 1 – 6 kW

The results obtained from the conducted tests show that the fluctuations in the indicated (thermal) efficiency are in line with the authors' expectations.

From the researchers' perspective, the relative changes in engine efficiency as a function of its thermal state (oil temperature) under different loads and crankshaft speeds are more significant than the specific shape of the efficiency curves. Additionally, it was observed that the relative increase in indicated efficiency for a warm engine compared to a "cold" engine is approximately 10%.

A comparative analysis of the results obtained from mathematical modelling and empirical studies clearly indicates that the indicated efficiency of the modelled engine is about 8% lower than that of the real engine. The nature of changes in efficiency as a function of oil temperature for model and empirical studies converges. The discrepancies between the model and empirical results are probably due to the simplifications adopted in the model regarding:

– the course of the combustion process
– heat exchange between the medium inside the cylinder and its walls
– calorific value of the fuel calculated based on dependence 6.

In addition, the discrepancies between the parameters obtained as a result of modelling and empirical research were significantly affected by the accuracy of the measuring equipment used. The approximately constant eight percent discrepancy between model and empirical tests may result, among others, from imperfections in the model, including the assumed material constants (heat conductivity coefficient) of the combustion chamber elements, incorrectly assumed engine cooling water temperature (temperature measurement cooling water on the real engine was carried out at the entrance to the radiator) and other causes that are difficult to clearly identify.

6. Summary

The results presented in the article, both from modelling and empirical research, indicate the significant influence of engine thermal parameters on its efficiency. In addition to the obvious observations that an increase in oil temperature is accompanied by an increase in indicated engine efficiency, actual values of changes in indicated efficiency as a function of oil temperature were determined. The research was conducted for four operating conditions of the engine, namely rotational speeds of 1000 and 1200 rpm, and engine loads of 5 and 6 kW.

As a result of the research, it was found that an increase in oil temperature in the range of 295 K to 358 K is accompanied by an increase in the indexed efficiency of the engine by an average of about 10%. The course of the engine efficiency value as a function of oil temperature is not a linear function and depends on very many factors including engine operating parameters such as its load and crankshaft speed. In addition, the tests carried out showed the adequacy of the mathematical model developed by the authors at the level of 8%. The nature of the obtained waveforms (both model and empirical) indicates that the discrepancies are mainly due to systematic errors. In contrast, the contribution of random error is negligible. The probable source of systematic errors is the simplifications used in the model, and the limited accuracy of calculations implemented in multiple iterations. In turn, the likely source of random errors is the limited accuracy of the measurement apparatus used.

Nomenclature

C_m	mass fraction of coal
DI	direct injection
V	combustion chamber volume
G_h	calorific value
H_m	mass fraction of hydrogen
$\dot{H}$	enthalpy flux
i	number of cylinders
$\dot{m}$	mass flux
N_C	calculated power by break
N_i	Indicated power
N_m	mass fraction of nitrogen
n	rotation speed
p_i	indicated work
S_m	mass fraction of sulfur
SI	stratified injection
T	torque
w_m	mass fraction of water
z	number of ignitions per revolution of the crankshaft
μ_c	efficiency of the engine-gear-brake system
μ_i	indicated efficiency
μ_m	mechanical efficiency
μ_t	theoretical efficiency
μ_{tr}	transmission efficiency
ω	angular velocity

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Marcin Zacharewicz, DSc. DEng. – Faculty of Mechanical and Electrical Engineering, Polish Naval Academy, Poland.
e-mail: *m.zacharewicz@amw.gdynia.pl*

Paweł Socik, MEng. – Faculty of Mechanical and Electrical Engineering, Polish Naval Academy, Poland.
e-mail: *p.socik@amw.gdynia.pl*

Ryszard Zadrąg, DSc., DEng. – Faculty of Mechanical and Electrical Engineering, Polish Naval Academy, Poland.
e-mail: *r.zadrag@amw.gdynia.pl*

Paweł Wirkowski, DEng. – Faculty of Mechanical and Electrical Engineering, Polish Naval Academy, Poland.
e-mail: *p.wirkowski@amw.gdynia.pl*

Artur Bogdanowicz, DEng. – Faculty of Mechanical and Electrical Engineering, Polish Naval Academy, Poland.
e-mail: *a.bogdanowicz@amw.gdynia.pl*

Jakub Hołownia, Eng. – student of Faculty of Mechanical and Electrical Engineering, Polish Naval Academy, Poland.
e-mail: *pjakub20001210@gmail.com*