

Analysis Of Various Cylinder Fin Designs For A Small-Displacement Engine Used To Power The Superefficient Prototype Vehicle Eco Arrow 3

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Abstract— The article presents the results of a simplified analysis of heat processes occurring in the engine of a prototype vehicle participating in an eco-marathon competition. Four fin configurations were examined: the original (75×75 mm), without fins, and two intermediate variants (70×70 mm and 65×65 mm). For the extreme cases, both experimental tests and numerical simulations were conducted, while for the intermediate variants only simulations were performed. The analysis considered solely the case in which airflow was not forced. The measurements enabled the determination of boundary conditions and improved the reliability of the calculations. The results demonstrated a significant influence of finning on heat accumulation, while showing no substantial cooling effect under the analysed conditions. The necessity of maintaining finning to prevent exceeding the critical temperature of 150 °C was confirmed, thereby disproving the assumption that its complete removal would be beneficial. The conclusions provide a basis for further work on optimizing engine design.

Keywords— Internal Combustion Engine, numerical simulation, passive cooling, prototype ICE vehicle for eco-marathon.

I. INTRODUCTION

During the operation of an internal combustion engine, heat is generated as a result of the combustion of the air–fuel mixture and friction. The components most exposed to the highest temperatures include the cylinders, cylinder head, valves, and pistons [1, 2]. Analysis of the heat balance indicates that only approximately 25% to 45% of the supplied energy can be converted into useful work. A significant portion of the generated heat is carried away by exhaust gases, while the remaining 25%–35% should be dissipated through the cooling system and by radiation from the external surfaces of the engine to the environment. Measurements show that the combustion temperature of the mixture fluctuates around 2000 °C [3, 4]. However, given the transient nature of such extreme temperatures, the average temperature during a single cycle is assumed to range between 800 and 1000 °C [5, 6]. It is generally accepted that the optimal operating temperature for a four-stroke spark-ignition engine is 100–150 °C, whereas exceeding 180 °C may contribute to engine damage [5, 6]. To prevent overheating and maintain engine components within safe temperature limits, cooling systems are used [1, 7–10]. Cooling should not be excessive, as it could negatively affect engine performance. Maintaining an appropriate temperature also provides natural protection of the engine against corrosive wear of the cylinder liner [11].

Many contemporary publications reveal a recurring pattern in research on cooling systems, consisting of matching boundary conditions in computational models to results obtained from experimental studies. This approach increases the reliability of numerical simulations, whose usefulness has been confirmed by recent studies [12–20].

In summary, the thermal conditions of engine operation are a crucial aspect both from the standpoint of durability and efficiency. The latter factor is particularly significant in the context of participation in fuel-economy competitions.

A. Cooling systems

The complete set of devices used for engine cooling is referred to as the cooling system [21]. A schematic classification of cooling systems is presented in the Fig. 1.

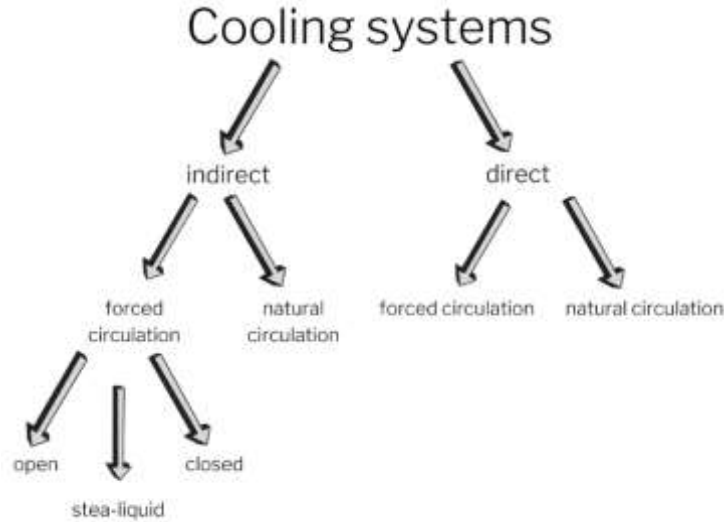


Fig. 1. Classification of cooling systems

B. Heat balance processes in an internal combustion engine

The heat balance of an engine is defined as the distribution of heat released during the combustion process into its respective components [22]. This balance makes it possible to assess the usefulness of the utilized heat, heat losses and the possibility of their reduction, the efficiency of exhaust gas energy utilization, the directions for improvements and optimization of engine performance, as well as to perform calculations of the engine's cooling system. The heat balance is determined experimentally for different engine operating conditions. In the literature, two types of heat balances for reciprocating engines can be found: the external and the internal energy (thermal) balance.

The external heat balance (thermal balance of the entire engine) is based on measurements of the mechanical energy and the thermal energy released by the engine to the environment [22]. To determine the individual components of the balance, it is necessary to measure the parameters required for their calculation. However, in practice, it is not possible to carry out all the necessary measurements. Therefore, certain values are conventionally assumed, for example, as a complement to 100%. The balance can be expressed by the following equation (1) [23].

$$\dot{Q}_o = \dot{Q}_u + S_{chl} + S_{odl} + S_{CO} + S_m + \sum S_{str} \quad (1)$$

where:

$\dot{Q}_o$ – heat flux supplied with fuel, kW;

$\dot{Q}_u$ – useful heat flux, useful power, measured, kW;

S_{chl} – cooling heat flux, cooling loss, kW;

S_{odl} – exhaust heat flux, exhaust loss, kW;

S_{CO} – incomplete combustion heat flux, incomplete combustion loss, kW;

S_m – mechanical losses, kW;

$\sum S_{str}$ – other losses, kW.

The internal energy balance represents the distribution of heat generated directly inside the cylinder [22]. It is based on a precisely determined indicator diagram and allows for

a more detailed analysis of engine performance and the influence of various factors. This balance is considerably more difficult to establish due to the necessity of using specialized measurement equipment.

C. Experimental tests

Internal combustion engines are tested on experimental stands referred to as dynamometers. For the purpose of investigating small-capacity engines, the research team constructed an inertial engine dynamometer. By reproducing the inertia of the racing vehicle through the main shaft with a rotating mass, the stand can serve as a driving conditions simulator. Furthermore, the dynamometer replicates all systems implemented in the Eco Arrow 3 vehicle. A schematic diagram of the test stand is shown in Fig. 2.

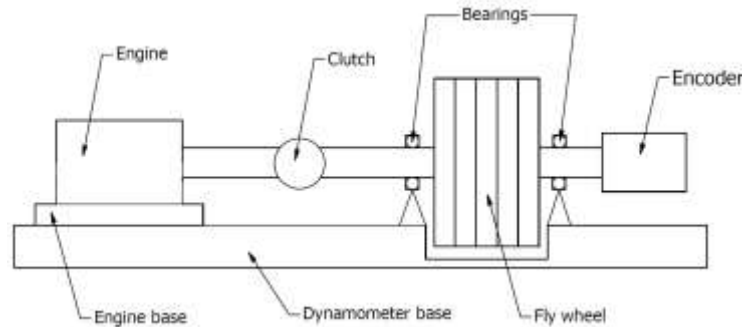


Fig. 2. Scheme of engine inertial dynamometer

D. Driving strategy during competitions and encountered challenges

The driving strategy during competitions consists of starting the engine and accelerating the vehicle to a predetermined maximum speed, after which the engine is switched off. The vehicle then continues to move using the acquired inertia until the set minimum speed is reached, at which point the engine is restarted. This cycle is repeated until the designated distance is covered.

An important aspect is to ensure optimal engine operating conditions in order to achieve the highest possible efficiency. Consequently, proper optimization of engine operating parameters has a significant impact on the race outcome.

When analysing the driving strategy itself, it becomes evident that the main issue faced by the team is maintaining the engine at its optimal operating temperature. On one hand, the engine must not overheat during operation; on the other hand, it should not cool down too quickly during idle periods.

During operation, the engine runs at high speeds—at critical moments reaching 5,000 revolutions per minute. These parameters are maintained for approximately 40 seconds. This is followed by an idle period lasting around 200 seconds (3 minutes and 20 seconds). The exact duration of these intervals depends on the characteristics of the track where the competition takes place and the time limit of the race.

II. MATERIALS AND METHODS

A. Research plan

The research was divided into two independent but thematically related stages:

- Numerical simulations in the ANSYS environment,
- Experimental tests on the inertial engine dynamometer.

These stages were concluded with a comparative analysis and the selection of the most effective cooling system configuration. In summary, the experiments were conducted according to the scheme presented in Table I.

TABLE I RESEARCH PLAN

Fin height on cylinder (h)	Type of research conducted
None	Numerical analysis and experimental tests
5 mm	Numerical analysis
10 mm	Numerical analysis
15 mm	Numerical analysis and experimental tests

B. Experimental tests

The experimental tests were carried out on engines with the original cylinder finning and on a cylinder with the fins completely removed. Considering the adopted competition driving strategy, it was crucial to ensure that the engine did not lose temperature during idle periods. Assuming that the cylinder finning serves both as a cooling system and as a heat accumulator, the study examined whether its modification could be beneficial. For this purpose, tests and a comparative analysis of these two configurations were conducted. The experiment provided data useful for numerical analysis.

The first stage of preparing the engines for testing was their modification to the configuration installed in the Eco Arrow 3 vehicle. These modifications included the removal of all covers and replacement of some systems with electronically controlled ones. Subsequently, thermocouples were installed in the engine with original finning, while in the finless version all fins were cut off. To mount the engine on the test stand, it was installed together with the starter system on an under-engine plate. The test stand with the engine prepared for experiments is shown in Fig. 3.

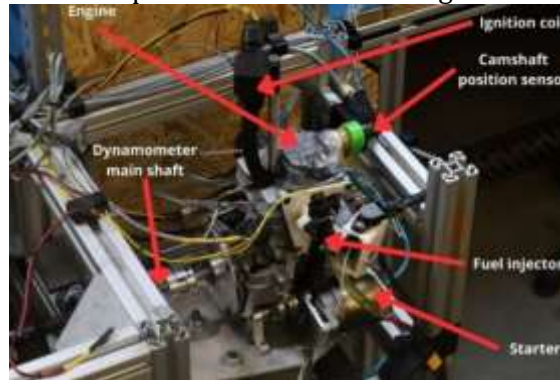


Fig. 3. Test bench

The measurement system used for recording the analysed temperatures consisted of:

- a set of six type-K thermocouples distributed on the surface of the engine body,
- a type-K thermocouple measuring ambient air (intake air) temperature,
- a Pico thermocouple interface,
- a laptop with PicoLog software, enabling the collection and recording of data transmitted by the interface.

The engine was operated for approximately 4 minutes (240 s), followed by an idle period of about 10 minutes (600 s). It should be emphasized that engine start-up and shutdown were performed manually—there was no possibility of setting precise operating times or automating engine restarts. Therefore, these values could vary slightly between individual engine cycles.

C. Numerical analysis

As the manufacturer does not provide models or detailed technical drawings of its engines, reverse engineering tools and Autodesk Inventor software were used to create 3D models. The prepared cylinder fin models are shown in Fig. 4.

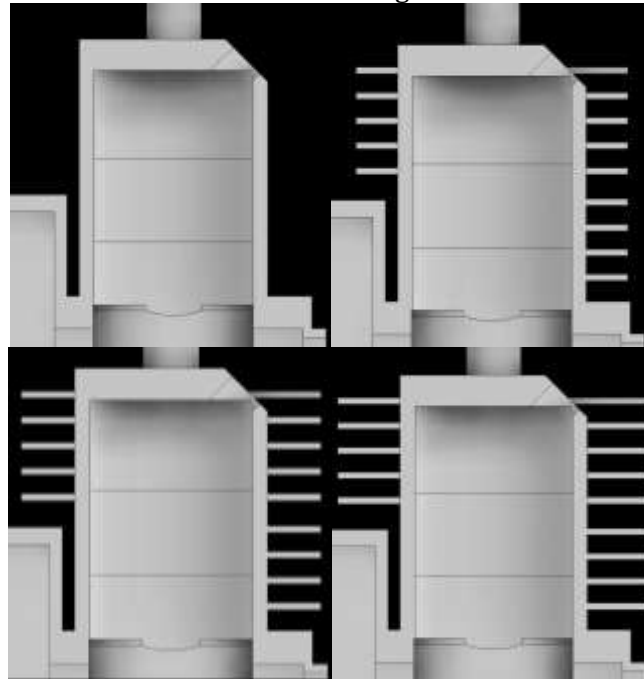


Fig. 4. 3D cylinder models

For the generated geometry, a computational mesh was created, with its parameters presented in Table II.

TABLE II MESH PARAMETERS

Number of elements, qty.				
h	Assembly	Piston	Shaft	Half of a block
15 mm	4,502,336	8,417	339,650	2,035,327
10mm	4,236,698			1,899,910
5 mm	4,018,733			1,792,797
none	3,010,718			1,289,173

The next step was defining materials used in engine components. Since the manufacturer does not provide such data, averaged values were assumed: steel (for load-bearing elements such as the crankshaft, bearings, and connecting rod) and aluminium alloy (for other components such as the crankcase and piston). The most important properties adopted in the simulations are listed in Table III.

TABLE III MATERIAL PROPERTIES ASSUMED IN THE SIMULATION

Property	Steel	Aluminium alloy
Thermal conductivity, $\frac{W}{mK}$	60,5	150

Specific heat, $\frac{J}{kgK}$	434	875
Density, $\frac{kg}{m^3}$	7850	2770

The final step in preparing the model for simulations was the definition of boundary conditions. The assumed boundary conditions are shown in Fig. 5., while their values are summarized in Table IV.

It should be noted that all numerical simulations consisted of transient heat conduction calculations in a solid material (the engine crankcase). These are simplified calculations that do not account for detailed heat transfer conditions at the walls, which could only be obtained from full thermal-fluid flow analyses, both in the surroundings and inside the engine. The boundary conditions of convective heat transfer (third kind) at the external and internal surfaces of the engine crankcase were determined based on literature data and experimental results.

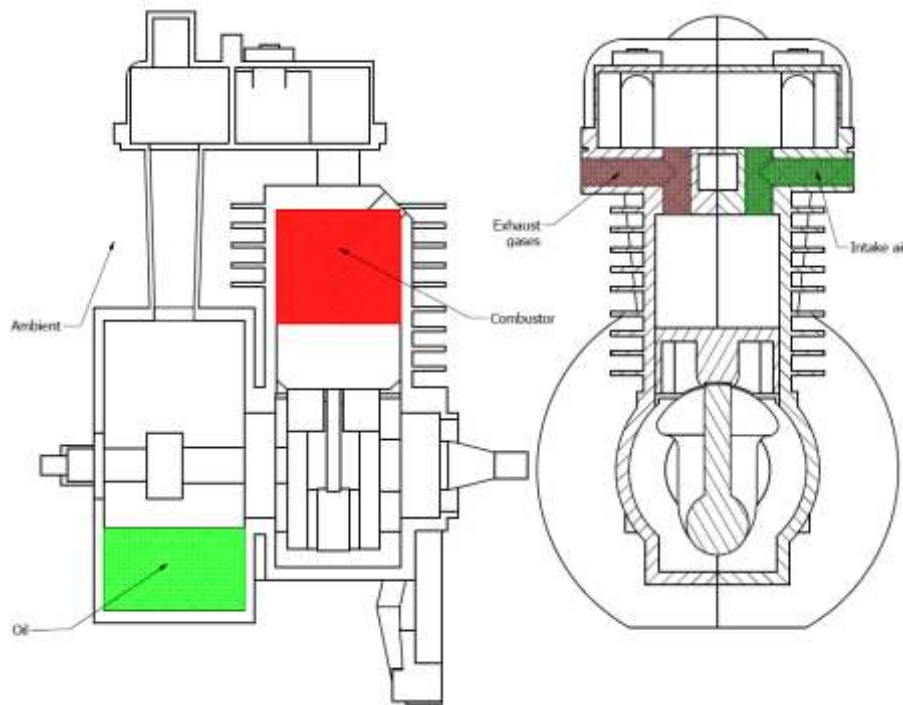


Fig. 5. Assumed boundary conditions for simulations

TABLE IV BOUNDARY CONDITIONS FOR INDIVIDUAL SIMULATIONS

Boundary condition	Running engine	Non-running engine
Ambient temperature	25 °C	
Intake air temperature	25 °C	27 °C
Heat transfer coefficient on the external engine block surface	25 $\frac{W}{m^2K}$	
Oil temperature	40 °C	80 °C
Heat transfer coefficient for oil	150 $\frac{W}{m^2K}$	100 $\frac{W}{m^2K}$
Exhaust gas temperature	370 °C	40 °C

Heat transfer coefficient for exhaust gases in the manifold	$275 \frac{W}{m^2 K}$	$5 \frac{W}{m^2 K}$
Combustor temperature	800 °C	100 °C
Heat transfer coefficient in the combustion chamber	$320 \frac{W}{m^2 K}$	$5 \frac{W}{m^2 K}$

Monitoring points were also introduced, and their positions are shown in Fig. 6.

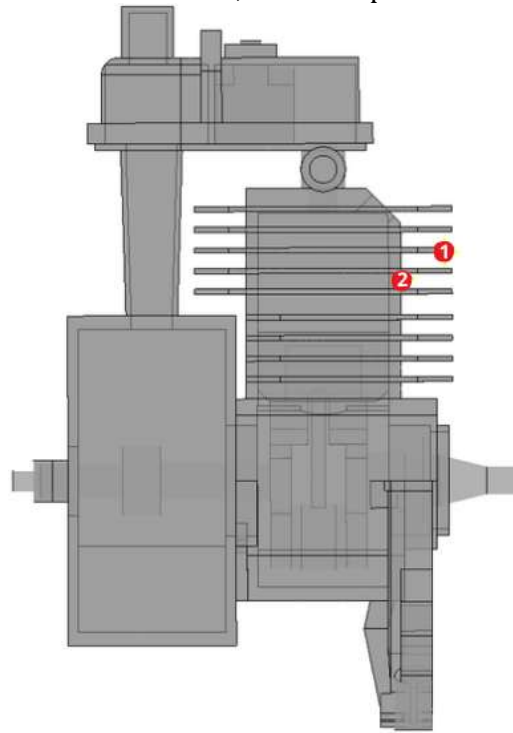


Fig. 6. Monitoring points: 1 - temperature of the tip of the finning, 2 - temperature of the external surface of the cylinder

III. RESULTS

A. Experimental tests

After completing the preparation process, the tests were carried out, and the obtained results are presented below (Fig. 7.).

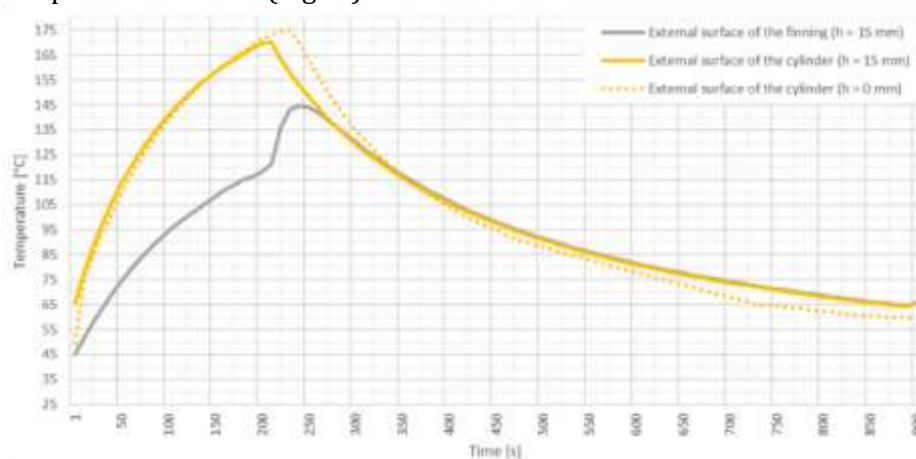


Fig. 7. Results of experiments (ambient temperature: 25 °C)

B. Numerical analysis

The following graphs (Fig. 8. - Fig. 11.) show the simulation results for the respective cylinder finning variants of the engine.

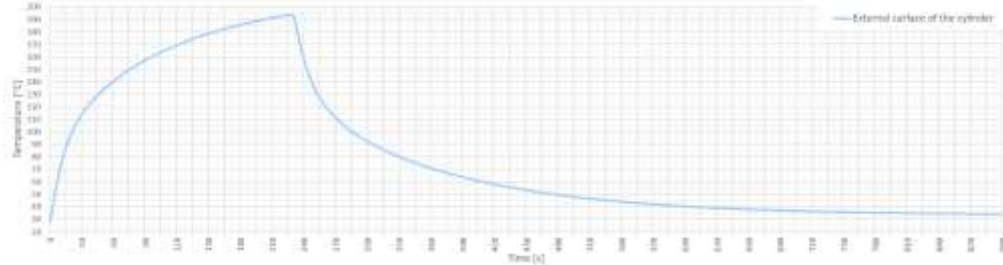


Fig. 8. Simulation results for the engine without finning, $h=0$ mm

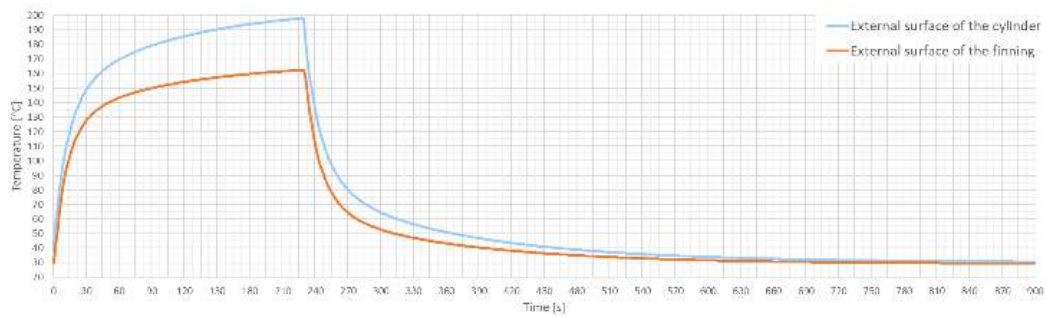


Fig. 9. Simulation results for the engine with finning, $h = 5$ mm

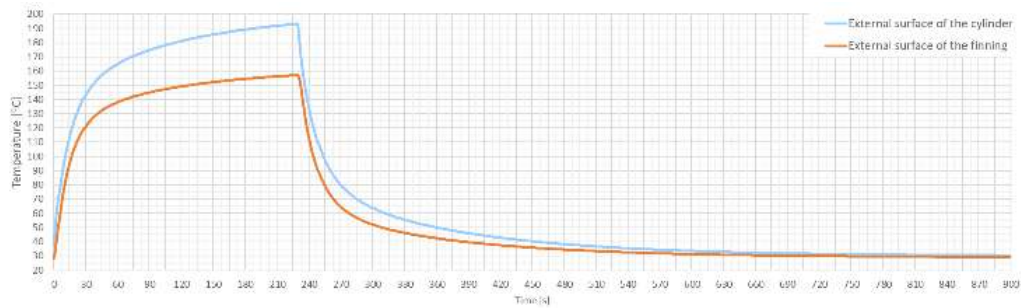


Fig. 10. Simulation results for the engine with finning, $h = 10$ mm

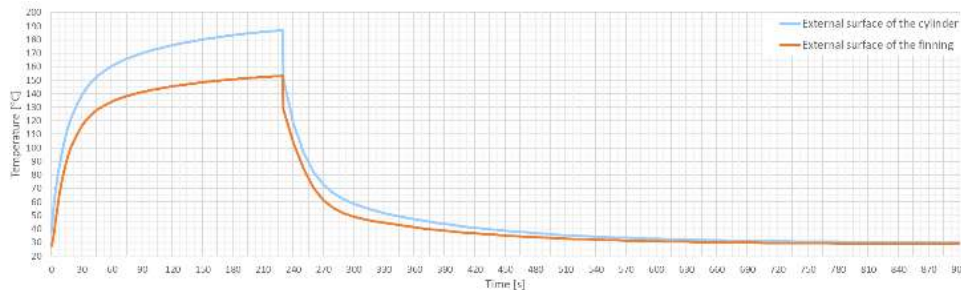


Fig. 11. Simulation results for the engine with finning, $h = 15$ mm

C. Comparison of simulation and experimental results

Percentage differences in the maximum values of each measured temperature were determined. These are presented in the tables below (Table V., Table VI.). None of the

differences exceeded 10%. Graphs comparing the temperature profiles at individual measurement points are shown in Fig. 12. – Fig. 14.

TABLE V RELATIVE DIFFERENCES FOR THE ENGINE WITH ORIGINAL CYLINDER FINNING

Measurement point	Percentage difference
Tip of finning	$\delta_w = 5,65\%$
Surface of cylinder	$\delta_w = 9,72\%$

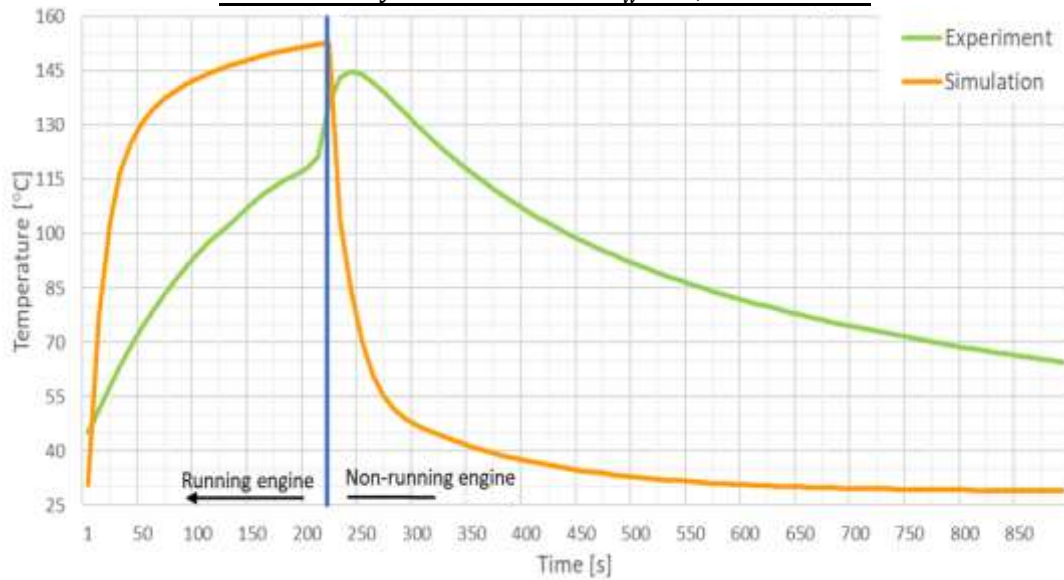


Fig. 12. Comparison of temperature curves for an engine with full finning at the tip of the fin

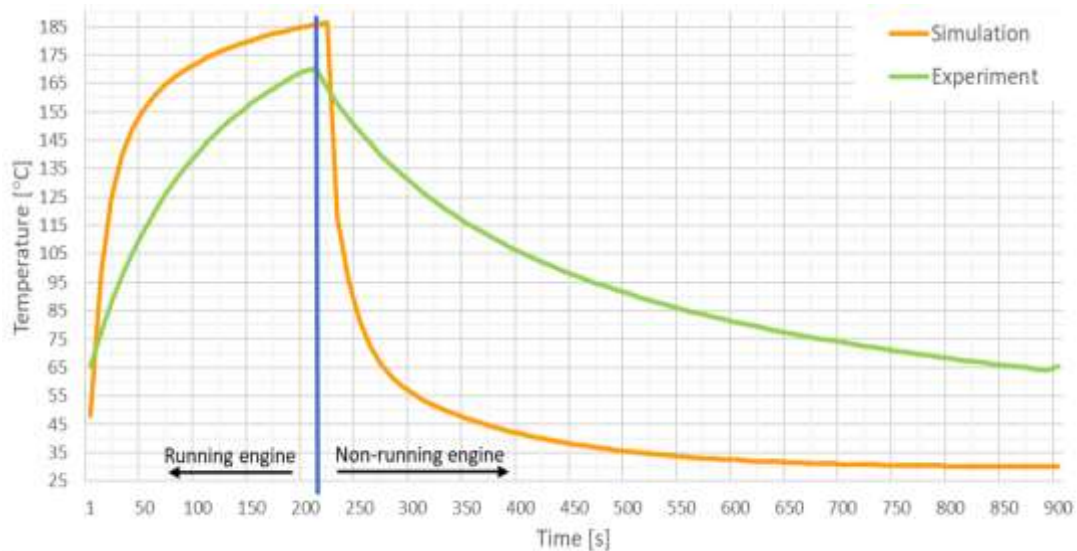


Fig. 13. Comparison of temperature curves for an engine with full finning for the external surface of the engine cylinder

TABLE VI RELATIVE DIFFERENCES FOR THE ENGINE WITHOUT CYLINDER FINNING

Measurement point	Percentage difference
Surface of cylinder	$\delta_w = 10,78\%$

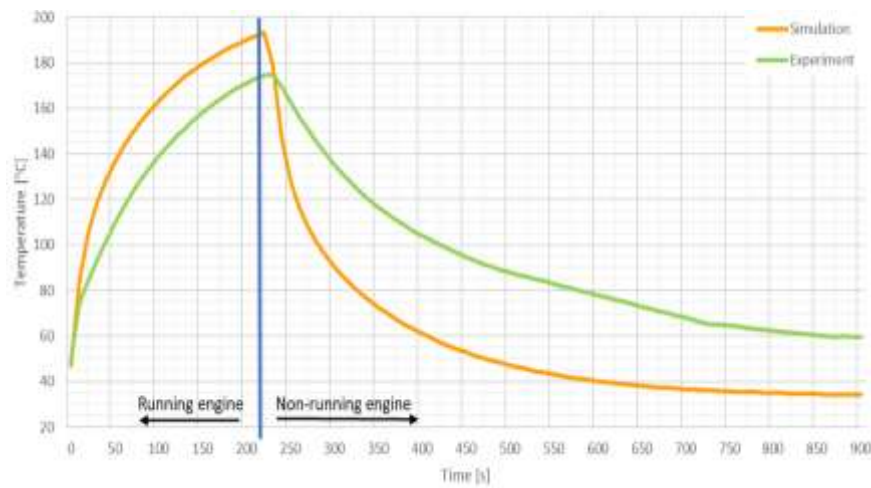


Fig. 14. Comparison of temperature curves for an engine without finning for the external surface of the engine cylinder

D. Comparative analysis of individual variants

The collected data allow for a comparative analysis of the numerical simulation results for different fin heights.

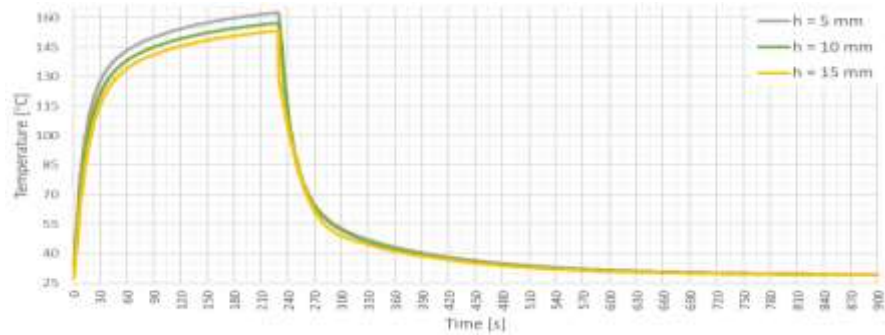
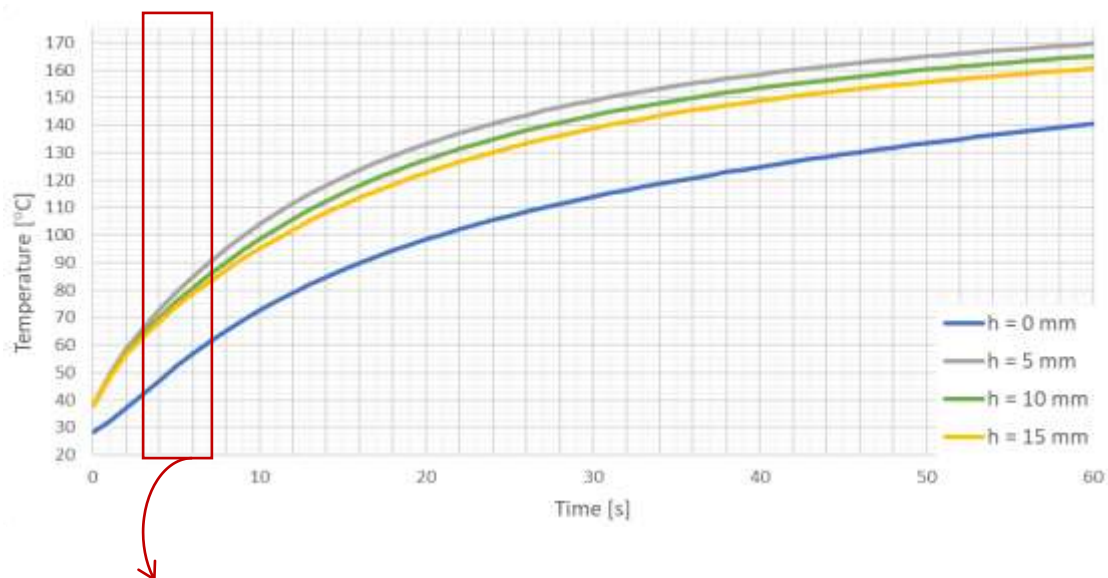


Fig. 15. Comparison of temperature curves for tip of the finning



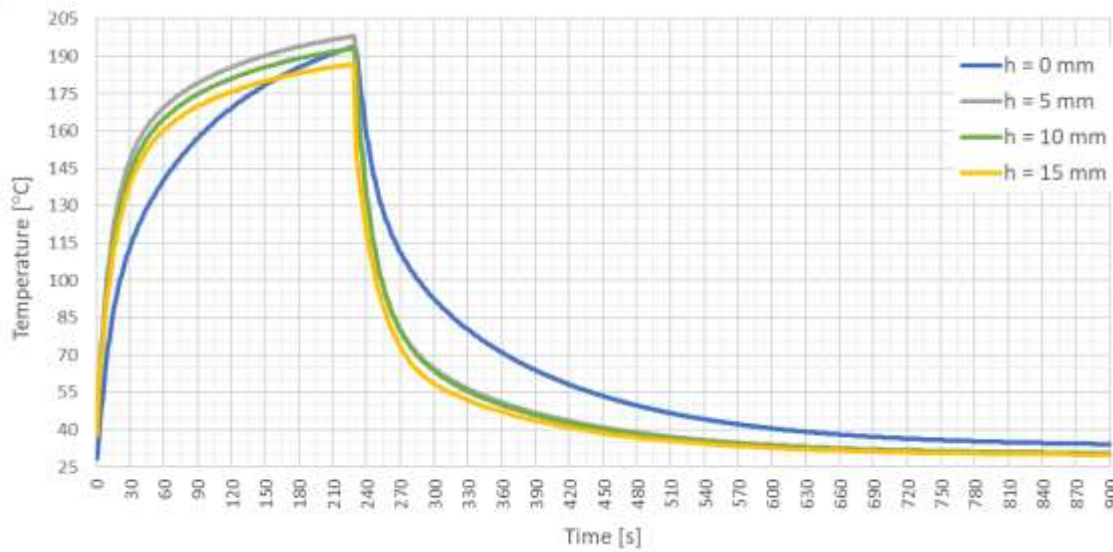


Fig. 16. Comparison of temperature curves for the cylinder (zoom for the first 60 seconds)

IV. DISCUSSION AND CONCLUSIONS

Analysis of the obtained numerical simulation results shows that the temperature profiles are similar across variants, with comparable maximum values. Significant differences are observed only for the variant without cooling fins. It is worth noting that this engine was damaged during the experimental tests, highlighting the crucial role even a minimal engine cooling system plays.

According to information obtained from the review of current knowledge, the optimal thermal operating conditions for a spark-ignition internal combustion engine range between 100 and 150 °C. Below 100 °C, the engine exhibits reduced efficiency, while above 150 °C there is a risk of damage.

Under the assumptions adopted for this study, none of the variants guarantee that the surface temperature of the cylinder remains below 150 °C during operation for the assumed 230 s. However, as mentioned, the engine is not running for such long periods during actual runs. Analysing the graphs of these temperature curves, it can be seen that 150 °C is reached by the individual engine variants with finning after the following times:

- 31 s – for an engine with a finning height of $h = 5$ mm,
- 36 s – for an engine with a finning height of $h = 10$ mm,
- 41 s – for an engine with a finning height of $h = 15$ mm.

The conducted tests and numerical analyses confirmed the hypothesis that, under the analysed conditions, the fins serve not only to remove heat from the cylinder during engine operation but also to store heat during idle periods. Considering the obtained results and taking into account that the cylinder surface temperatures are slightly overestimated by the simulation—with relative differences at the measurement point on the external cylinder wall around 10%—the variants with fin heights of 10 mm and 15 mm provide optimal thermal conditions for engine operation.

NOMENCLATURE

- h – height of the finning, mm
- δw – percentage difference, %

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