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# HEAT RELEASE CHARACTERISTIC IN INTERNAL COMBUSTION ENGINES WITH M-TYPE FUEL INJECTION PROCEDURE

*Abstract*
*The description of the combustion process in internal combustion engines is typically performed through so-called combustion parameters, such as: the maximum combustion pressure, maximum temperature of the cycle, crankshaft angle of the start of combustion, the angular interval of duration of combustion, the heat release characteristics, thee increment of the total amount of heat developed, the angular interval of combustion delay, the heat transfer etc. Due to the complexity of the combustion process in internal combustion engines to this day there are still no models that could efficiently and above all in a reliable way predict the heat release characteristics for a wide range of different types of IC engines. Phenomenological, empirical and CFD models for specific constructive characters of the engine and fuel applied are targeted explored. The paper presents a procedure for determining the heat release characteristics through approximate functions of combustion, using the example of a very specific way of preparing a mixture of fuel and air in the diesel engine, so-called M-procedure. The applied approximate function is validated through experimental test results on a specific IC engine.*
***Key words:*** *IC engine, combustion, approximate function, M-procedure*

## 1. Introduction

The essential difference in the process of creating mixtures that takes place in diesel engines, gasoline engines, and in industrial burners is in a significant temperature difference between the fuel and air to be mixed in diesel engine. It is known from the Arhenius's law [1] that temperature has a significant effect on chemical reactions. If fuel is mixed with hot air in the diesel engine, very high temperatures will affect the fuel structure itself. In this, fuel properties come to the fore, in which its molecules, especially in case of heterogeneous mixtures, tend to form peroxides with spontaneous partial reactions. This results in the process of decomposition of molecules that promote spontaneous combustion on the one hand and, on the other hand, by releasing hydrogen-rich particles, molecular residue is enriched with carbon which is a reaction inert and ultimately appears as soot in exhaust gases.

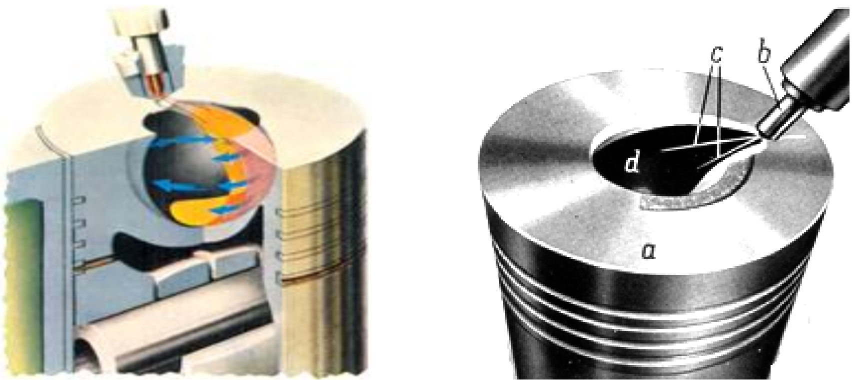
Release of hydrogen from the molecule leaves it without a natural catalyst, because the carbon reaction is accelerated to the desired extent by the end product of the hydrogen reaction, but only if it takes place at the molecular level. The more the temperature rises during the combustion, the decomposition process, which then can no longer be stopped, will lead to the complete breakdown of the molecules. This is what causes a common characteristic of diesel engine reactions: in the initial phase these are high speed reactions associated with noise, and in the end a slow post-combustion (afterburning) of inert carbon. The reasons for inadequate mixing of fuel and air in the diesel engine should be looked for within these reaction kinetic processes; a better mixing of the liquid fuel with hot combustion air will result in a greater amount of fuel subject to the described process of decomposition, which is related to spontaneous combustion. The increase in the mixing effect increases the initial reaction rate, but the amount of reaction inert carbon also increases. Through spectral analytical tests (experiments) these processes can be clearly discerned in the diesel engine. On the other hand, that same procedure can show that the combustion of gas oils in, e.g., burners is not associated with such noticeable decay of molecules. The burner operates to a very rich mixtures without soot. Slow pre-oxidation in this case considerably alters the sequence of reactions in the flame. Experimental studies found that the way creating fuel-air mixtures has a decisive impact on the character of the ignition and combustion; it is much more difficult to oxidize gas oil that consist of undecomposed fuel and air than a mixture in form of a mist consisting of fuel droplets.

Based on the above described reaction kinetic process, a series of conclusions can derived and presented in the form of recommendations for mixture formation. The fulfilment of the following three conditions can best serve to mitigate the adverse consequences of the decomposition process, which is inevitable in the spontaneous combustion [4]:

1. Limit the share of fuel in spontaneous combustion as much as possible.
   A small amount of fuel should be mixed with hot air, in the way it is normally done in diesel engines with the total amount of fuel. Finally, it is still necessary to mix the remaining fuel with air to have it burned. In order to avoid disadvantages, it is necessary to prepare the fuel before mixing it with hot air it is. Based on the experience gained from the research of flame of diesel fuel combustion, this can be achieved through the following two conditions.
2. Providing a possibility of slow pre-oxidization of fuel without overheating.
3. Evaporating fuel has to be mixed with hot air successively in small amounts, but so fast that before the spontaneous combustion begins so that the approximately stoichiometric ratio in the mixture is reached, and that it is possible to ignite it with an external source.

Through the usual process of volume mixture creation the above three conditions can not be achieved; the first condition requires only a small part of the fuel to be directly mixed with the air.

If the total amount of fuel is dispersed or comes into contact with hot surfaces, neither the first nor the second condition can be fulfilled, thus leaving no option to fulfil the third condition. To form a mixture according to the previously specified conditions, for the M-type injection procedure fuel is not injected into the air contained in the combustion chamber, which is roughly hemispherical, but injected onto the wall of the combustion chamber of the piston; actually, it is applied as a thin wide film, the figures 1 a) and b). The fuel film at engine full load has a mean thickness of about 12 microns. Such injected fuel is not forcibly mixed with air at a high temperature, as is usually the case with other fuel-air mixture procedures. In this procedure the fuel film heats up through the chamber wall only to a permissible value. Thus, oxygen from the air, so to speak thermally damped, can be accepted by fuel in small quantities without hydraulic oxygen delivery damping (as in pre chamber engines) while overheating fuel molecules. In this way, the self-ignition is off for not controlled larger amounts of fuel [2].




a) b)

Figure 1: M-type fuel injection procedure (a-piston, b-injector, c-fuel spray, d-wall of the piston had semi-hemispheric compartment)

Selecting a short distance between the nozzle orifice and the combustion chamber wall, only a small amount of fuel will separate from the injected spray which will, in contrast to the quantity of fuel applied to the wall, undergo spontaneous combustion processes that are common to the diesel engine. Part of the fuel on the wall is gradually heated and begins to evaporate; evaporation rate is dependent on the temperature difference between the wall and the gas (increasing with time) and constantly increasing gas radiation. Only that part of the fuel that evaporates can participate in the mixture creation, which takes place very quickly, on the one hand because of the fuel in the vapour state and on the other hand because of a high speed of air movement. Evaporated fuel elements either did not decompose at all or they very little did. The mixture created in this way does not reach the point of spontaneous combustion because of long induction but it burns due to other sources of ignition. Ignition sources are red-hot carbon particles created in the initial phase through spontaneous combustion of a small amount of.

Regardless that the described individual processes overlap in time, it is possible to create a scheme that results in a difference in the mixture formation procedure in conventional diesel engines from MAN M-type procedure. In the first place, the separation of the total fuel in two is noticeable in the M-type procedure. The greater part is introduced into the mixture only after evaporation, while in conventional diesel engine the fuel is introduced to the mixture prior to its evaporation. The walls of the combustion chamber, mainly the part located in the piston, has a similar task as the carburettor in the gasoline engine, and at a high pressure and a high temperature it leads to the evaporation of even those parts of the fuel of a high boiling point. Creating a mixture immediately before its combustion leaves very little time for the pressure to act, so that the knocking caused by a sudden change of pressure is avoided. The way of managing the combustion process substantially affects the shape of the heat release characteristics, so there are obvious differences in the character compared to engines with direct injection of the entire spray into the air of the combustion chamber [3]. The rest of the paper deals with the approximate function of heat release characteristics, adapted for use on engines with M-type injection procedure, using diesel or biodiesel fuel.

## 2. Experimental set-up and boundary conditions

For this experiment a 6-cylinder, four-stroke diesel engine intended for a bus unit is used. The Table 1 shows the basic data of the tested motor. Determination of relevant indicators of the IC engine was performed on the test bench in the laboratory of the Engineering Faculty of the University of Maribor in controlled conditions. The tests were performed on partial and the maximum load of the IC engines, as well as through the whole speed range.

Table 1: Basic data of the tested IC engine

| | |
|---|---|
| Engine | Natural aspirated, 4 stroke with M-type fuel injection procedure |
| Number of cylinders | 6 |
| Piston diameter and travel | 125 mm x 155 mm |
| Volume | 11,413 $dm^3$ |
| Compression ratio | 1:18 |
| Fuel injection | 23 °CA BTDC |
| Nominal power at engine speed | 160 kW at 2200 $min^{-1}$ |
| Maximal torque at engine speed | 775 Nm at 1400 $min^{-1}$ |

Common methods in for this field of testing were used to measure physical quantities, such as pressure, temperature, and flow.

Indicators of IC engines were determined for fossil and bio diesel. The used fuels met the quality criteria specified in the relevant standards and recommendations (for fossil diesel EN 590; for biodiesel EN 14214). The main characteristics of the used fuel are shown in table 2.

Table 2: Diesel and bio diesel fuel characteristics

| Fuel | Diesel | Biodiesel |
|---|---|---|
| Kinematic viscosity at 30 °C [$mm^2/s$] | 3.34 | 5.51 |
| Surface tension at 30 °C [N/m] | 0.0255 | 0.028 |
| Calorific value [kJ/kg] | 43,800 | 38,177 |
| Cetane number [-] | 45-55 | >51 |

Different physical properties of fossil and bio-diesel [4] cause some differences in the process of forming a fuel-air mixture [5]. The property differences of the observed fuels primarily affect the optimal angle of fuel injection [6], the physical processes of air- fuel mixing, and induction time [6], which ultimately has an impact on the combustion inside the IC engine, and engine power and emissions.

The researchers concluded that for achieving optimal performance of the IC engine, it is necessary to decrease the angle of fuel injection beginning [7] when using biodiesel fuel. The main characteristics of the observed fuels that contribute to this conclusion are the viscosity and cetane number in the first place. Manufacturers recommendations the optimal injection time is at 23 °CA BTDC for the IC engine fired with fossil diesel fuel, and the studies showed optimal injection timing of 21 °CA BTDC [8] for biodiesel fuel.

## 3. Mathematical modelling of the heat release characteristic

Due to the complexity of the processes that occur during combustion in the IC engine, at first glance, a complex modelling process that could describe individual processes in space and time seems appropriate and adequate. Three-dimensional calculation codes that are available today really have the potential for solving this task. However, the effort required for an accurate description of the geometry of the IC engine workspace is large, the time required for performing calculations is considerable, and their accuracy is limited due to incomplete knowledge of the pattern of individual processes. Like the other extreme, in contrast to 3D models, so-called non-dimensional or zero-dimensional models are still in use first of all because of their simple structure and short time required to obtain concrete results. Since the characteristics of heat release essentially describes the character of processes in IC engines, its knowledge is of great importance for both manufactured as well as for IC engines in design state. In manufactured engines, the heat release characteristic is usually derived from the recorded indicator diagrams, since the accuracy that can be achieved is good and satisfactory. For IC engines in design state, the heat release characteristic is very difficult to determine.

Most often, in these cases the indicator diagrams are taken from engines of similar design, or different types of heat release characteristics are assumed while analysing their impact on the process in IC engines [9]. At Figure 2 the differences between the heat release characteristics for a natural aspirated diesel engine with M-type fuel injection procedure and a turbocharged diesel engine with direct fuel injection, full load and two-speed conditions, is clearly shown.

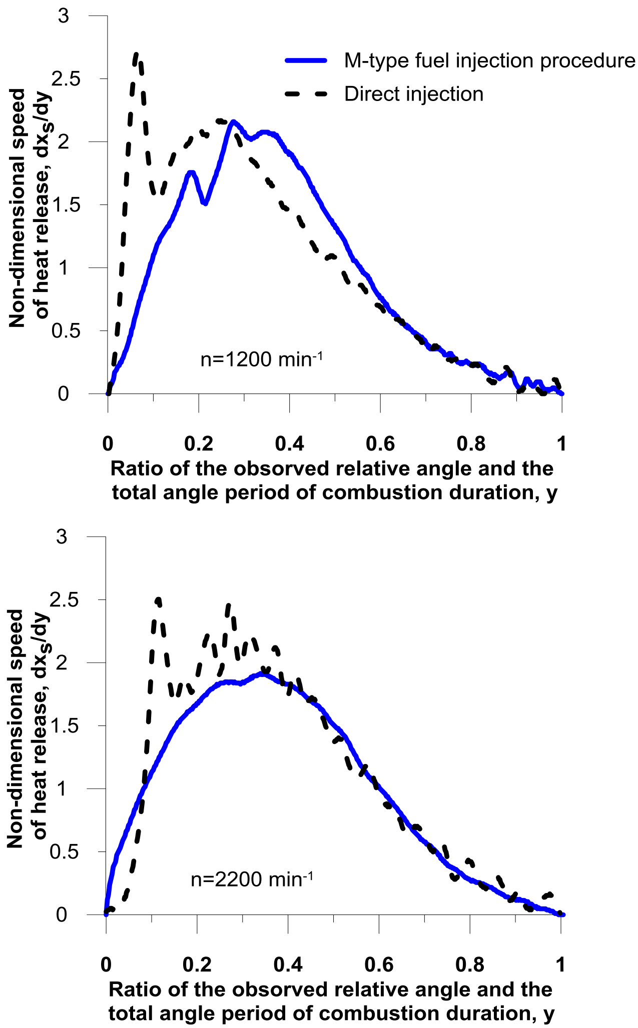



Figure 2: Heat release characteristic for a diesel engine with M-type fuel injection procedure and the direct injection diesel engine, full load

The approximated combustion function presented here is based essentially on the Marzouk-Watson's correlation expression, while retaining the basic idea of presenting the heat release characteristic by superimposing two interrelated functions. In order to obtain the best possible correlation, tests were conducted on the IC engine with different fuel injection timings (21 °CA BTDC and 23 °CA BTDC) and both for diesel and biodiesel fuel. Using statistical methods, as well as methods for fitting curves such as least squares, etc., corrections on the expressions were made to obtain useful expressions to simulate the heat release characteristic for diesel engine with M-type fuel injection procedure, taking into account the different fuels.

In diesel engine with M-type procedure the initial peak on the heat release characteristic is much less pronounced (despite the fact that a large part of the fuel is injected during the ignition delay), although the total period of combustion is roughly the same, while at higher IC engine speeds the peak generally does not appear (figure 2, on the right). Unfortunately, these two engines are not comparable from the point of injection timing and ignition delay period, and the intensity of combustion because of different engine concepts, and therefore the injection timings are different, as well as their fuel supply systems. The figure is given as an illustration to demonstrate the differences in the character of heat release characteristics. Throughout history, first used approximate analytical expressions used to describe heat release characteristics had a very simple form of a triangle, two triangles, quadrangles, etc. [10]. Using such forms to describe heat release characteristics in IC engine process simulation, pressure values, power, fuel consumption etc. can be satisfactory matched. However, where such simple forms are applied to compare pressure gradients or evaluate recorded indicator diagrams, they quickly reach their limits. The conclusion is that for the approximate functions to give satisfactory results, they must partially contain elements describing the combustion process occurrences established through macro-examination of the combustion process. The approximate functions describing heat release characteristics, which are obtained from the work on various concepts and models, essentially directly or indirectly take into account the following parameters:

- Engine operation is defined by the equivalent air ratio α and the engine speed $n$
- State in the engine at the beginning of combustion is defined by the ignition delay period $\tau_{id}$
- State of the fresh working fluid at intake to the engine or at the time the intake valve closes.

Because of the above specific heat release characteristics of the diesel engines with M-type fuel injection procedure, often used an approximate functions, such as single and double Vibe functions, could not meet the requirements for the comparison of pressure gradients and the evaluation of recorded indicator diagrams. Below follows the newly developed approximate function of the heat release characteristics, which allows a very good simulation of the process of the present IC engines using diesel and biodiesel fuel.

The new correlation expression that largely describes the diesel engine with M-type fuel injection procedure, obtained through the correction of the Marzouk-Watson's correlation reads [5]:

$$\frac{dx_s}{dy} = \beta^* \cdot f_1^*(y) + (1-\beta^*) \cdot f_2^*(y) \tag{1}$$

$$f_1^*(y) = C_1^* \cdot C_2^* \cdot y^{(C_1^*-1{,}2)} \cdot \left(0{,}9999 - y^{C_1^*}\right)^{(C_2^*-1{,}15)} \tag{2}$$

$$f_2^*(y) = C_3^* \cdot C_4^* \cdot y^{(C_4^*-0{,}3)} \cdot e^{\left(-C_3^* \cdot y^{(C_4^*+C_5^*)}\right)} \tag{3}$$

$$\beta^* = \frac{-0{,}324\ \frac{\varphi_{id}}{\alpha}}{-26{,}75 + \frac{\varphi_{id}}{\alpha}};\quad \varphi_{id}\ [°\mathrm{KV}], \propto [-] \tag{4}$$

$$\mathrm{y} = \frac{\varphi - \varphi_{CB}}{\varphi_{CD}} \tag{5}$$

While the expressions for calculating the constants are given in the following form:

$$\left.\begin{aligned}
&C_1^* = 0{,}175\ \cdot 0{,}9997^{\,\tau_{id} \cdot n} \cdot (\tau_{id} \cdot n)^{0{,}54};\quad \tau_{id}\ [\mathrm{ms}], \mathrm{n}\ [\mathrm{min}^{-1}]\\
&C_2^* = exp\left[-49{,}73 + \frac{7239}{\tau_{id} \cdot n} + 7{,}43\ ln(\tau_{id} \cdot n)\right];\quad \tau_{id}[\mathrm{ms}], \mathrm{n}\ [\mathrm{min}^{-1}]\\
&C_3^* = \frac{7{,}8}{exp\left[\frac{-(\tau_{id} \cdot n - 1207)^2}{22{,}82\ \cdot 10^5}\right]};\quad \tau_{id}\ [\mathrm{ms}], \mathrm{n}\ [\mathrm{min}^{-1}]\\
&C_4^* = \frac{1{,}52}{exp\left[\frac{-\left(\frac{\varphi_{id}}{\alpha} - 6{,}35\right)^2}{38{,}9}\right]};\quad \varphi_{id}\ [°\mathrm{KV}], \propto [-]\\
&C_5^* = \frac{CN \cdot 3{,}25}{43{,}33 + 28{,}14\ \frac{\varphi_{id}}{\alpha} - 2{,}56\left(\frac{\varphi_{id}}{\alpha}\right)^2};\quad \varphi_{id}\ [°\mathrm{KV}], \propto [-]
\end{aligned}\right\} \tag{6}$$

Compared to the Marzouk-Watson's correlation expression, constants in the exponents of the functions $f_1(y)$ and $f_2(y)$ (expressions (2) and (3)) were corrected primarily on the basis of experimental investigations. Then, using different optimization methods, constants for a given mode of operation were determined. Specifically, the Marzouk-Watson's correlation was corrected taking into account that the engine operating mode is defined with parameters such as air equivalent ratio (α), engine speed (n), period of ignition delay ( $\tau_{id}, \varphi_{id}$), because these parameters are the ones that define the current operating condition of the IC engine; air equivalent ratio (α) and engine speed (*n*) determine the current operating mode of the engine, while the ignition delay period ( $\tau_{id}, \varphi_{id}$) takes into account the state of the working fluid at the moment of closing the intake valve and the fuel injection timing. With the introduction of the new constant $C_5^*$ in the expression of the modified function $f_2^*(y)$, the impact on the heat release characteristic by type of fuel expressed through cetane number (CN) is taken into account.

## 4. Analysis of the results

With the use of the approximate function (1) heat release characteristics for diesel engine with M-type were obtained and compared with the experimental characteristics. At Figure 3 dimensionless heat release rate obtained on the basis of experiment and calculation for diesel and bio-diesel fuel, considering maximum load, and two different engine speeds with the beginning of the fuel injection at 21 °CA BTDC were compared.

As it can be seen from the figure, the proposed approximation function of non-dimensional heat release speed, based on the correlation Marzouk-Watson, accompanied by changes in the character of the curves that result from the use of different fuels, as well as various engine speeds and gives a very good agreement with experimental results, both in intensity and character in different working regimes of the engine with M type fuel injection procedure.

In order to test the new approximation function in terms of changes in the fuel injection angle, in Figure 4 the dimensionless heat release rate $dx_s/dy$ obtained based on experiments and calculations for diesel and bio-diesel fuel, considering the maximum load, and two engine speeds with the start of the fuel injection at 23 °CV BTDC are compared.

As in the previous example the new approximation function of non-dimensional heat release speed, based on the correlation Marzouk-Watson, gives a very good agreement both in intensity and character in all operating modes of the IC engine with M type fuel injection procedure, and thus confirms the sensitivity of the approximation function by use with different fuel injection angles.

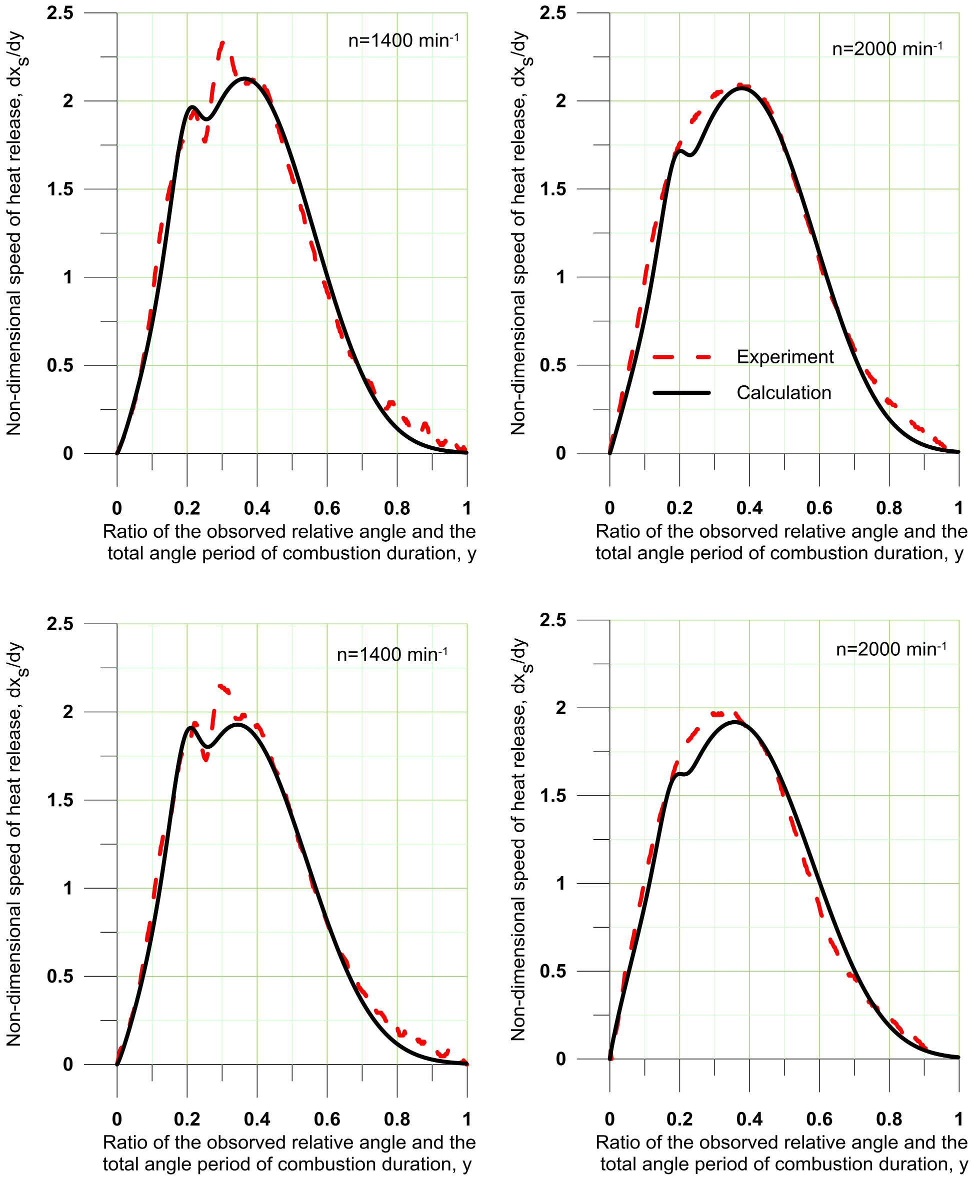



Figure 3: Non-dimensional speed of heat release $dx_S/dy$ for diesel engine with M-type fuel injection procedure using diesel and biodiesel fuel, speed 1400 min$^{-1}$ and 2000 min$^{-1}$, injection timing 21 °CA BTDC; biodiesel (above), diesel (below)

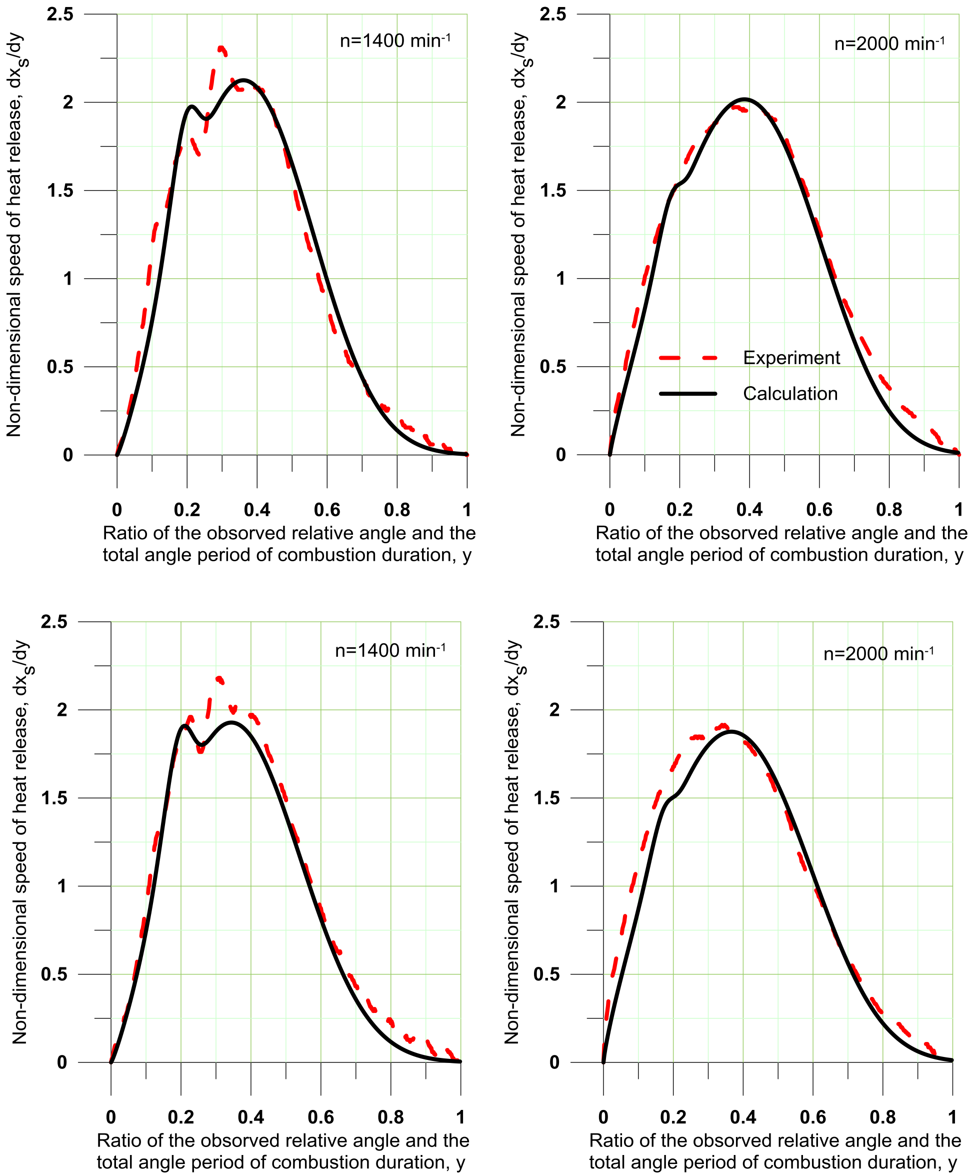



Figure 4: Non-dimensional speed of heat release $dx_s/dy$ for diesel engine with M-type fuel injection procedure using diesel and biodiesel fuel, speed 1400 min$^{-1}$ and 2000 min$^{-1}$, injection timing 23 °CA BTDC; biodiesel (above), diesel (below)

## 5. Conclusion

The paper presents research results regarding the determination of the approximate function of the heat release characteristic of an in-line, water-cooled, medium speed diesel engine with direct fuel injection via M- type fuel injection procedure. The new approximate function, which is based on the correlation expression Marzouk-Watson, via the appropriate cycle settings takes into account the changes in operation modes of the IC engine, and the application of fuels with different cetane number, enabling observation of heat release characteristics derived using diesel and biodiesel fuel. Usability of the new approximation function is tested using experimental research results. It showed very good agreement with the experimental results, both in intensity and character in all operating modes of the engine, as well as in changing the fuel injection timing.

The represented approximate function provides a tool that will allow analysis of the combustion process, using either diesel or biodiesel fuel, and making appropriate conclusions with regard to the differences in the combustion process using different fuels for engines with M-type fuel injection procedure.

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Symbols:

$n$ – *engine speed*
$C_1^*, C_2^*, C_3^*, C_4^*, C_5^*$ – *constants*
*CA – crank angle*
*y – ratio of the observed relative angle and the total angle period of combustion duration*
$\frac{dx_s}{dy}$ – *non-dimensional heat release speed*
$f_1^*(y)$ – *function*
$f_2^*(y)$ – *function*
$\alpha$ – *air equivalent ratio*
$\beta^*$ – *constant*
$\varphi$ – *crankshaft angle*
$\varphi_{CB}$ – *crank angle at start of combustion*
$\varphi_{CD}$ – *crank angle duration of combustion*
$\varphi_{id}$ – *crank angle of combustion delay*
$\tau_{id}$ – *time interval of combustion delay*
BTDC – before top dead centre

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