EXPERIMENTAL DETERMINATION OF DOUBLE VIBE FUNCTION PARAMETERS IN DIESEL ENGINES WITH BIODIESEL

by

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A zero-dimensional, one zone model of engine cycle for steady-state regimes of engines and a simplified procedure for indicator diagrams analysis have been developed at the Laboratory for internal combustion engines, fuels and lubricants of the Faculty of Mechanical Engineering in Kragujevac. In addition to experimental research, thermodynamic modeling of working process of diesel engine with direct injection has been presented in this paper. The simplified procedure for indicator diagrams analysis has been applied, also. The basic problem, a selection of shape parameters of double Vibe function used for modeling the engine operation process, has been solved. The influence of biodiesel fuel and engine working regimes on the start of combustion, combustion duration and shape parameter of double Vibe was determined by a least square fit of experimental heat release curve.

Key words: *biodiesel, diesel engine, double Vibe, heat release curve*

Introduction

Diesel engine and biofuels

Half of all energy and raw material sources on our planet are practically engaged in production and exploitation of vehicles. The vehicles exist due to the same resources that all living beings depend on: soil, raw materials, water, air and space. Obviously, the vehicles have significant influence on human environment; hence a special attention must be given to them [1, 2].

Transport completely depends on oil supply and it is the source of important part of greenhouse gas emission. All predictions for the future have shown that the transport will increase and that it is important to find the solutions for secure fuel supply and the possibilities for pollution reduction. One of the solutions for these problems is the utilization of alternative fuels, which had been examined as possible fuels for spark ignition (SI) engines

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at the beginning of the XX century. The first attempts to utilize ethanol were made by H. Ford, while R. Diesel predicted the use of vegetable oils.

Directive 2003/30/EC sets up guidelines for adopting biofuels as all transport fuels in the EU [3]. European Community program predicted the increase of biofuels utilization from 2% in 2005 to 5.75% in 2010, with the final goal to meet 20% in 2020. The main objectives of the Directive are to reduce emissions of carbon dioxide from transport across Europe, and to reduce the EU future reliance on external energy sources (fossil oil). Both reasons suggest the necessity of introducing bioethanol and biodiesel as motor fuels [4, 5].

Engine modeling

The main reason for the growth in engine modeling activities arises from the economic benefits; by using computer models, large savings are possible in expensive experimental work when engine modifications are being considered. Models cannot replace real engine testing but they are able to provide good estimates of performance changes resulting from possible engine modifications and thus can help in selecting the best options for further development, reducing the amount of hardware development required.

Engine modeling is a fruitful research area and, as a result, many universities have produced their own engine thermodynamic models, of varying degrees of complexity, scope and ease of use. This widespread activity probably arises from a combination of reasons, including the importance of engine modeling and its relevance to almost any aspect of engine research and development. There are also now available comprehensive "commercial" models, which have a wider, more general purpose of use with refined inputs and outputs to facilitate their use by engineers other than their developers; most of these models had their origins in university-developed models.

While the more advanced models are extremely large and complex, the basics of an engine thermodynamic model are quite straightforward and easily understood; the complexity arises later in the refinement of the calculation methods, the level of detail of sub-system representation, and the accommodation of a wide variety of alternative engine configurations and control systems.

The main goal of the paper is to determine the influence of biodiesel fuel and engine working regimes on the start of combustion, combustion duration and shape parameter of double Vibe function.

Theoretical research

Modeling of engine working cycle

A zero-dimensional, one zone (homogeneous) engine working cycle model for steady-state regimes of engines [6] has been developed at the Laboratory for internal combustion (IC) engines, fuels and lubricants of the Faculty of Mechanical Engineering in Kragujevac. The main advantage of this model is the possibility to check the results of modeling using experimental data obtained during acquisition of engine indicator diagram. In this model, real combustion process has been replaced by heat supply using Vibe function.

Modeling of the combustion process with the double Vibe function

The Vibe function is often used to approximate the actual heat release characteristics of an engine:

$$\frac{\mathrm{d}x}{\mathrm{d}\left(\dfrac{\phi}{\phi_\mathrm{z}}\right)} = C\,(m+1)\left(\frac{\phi}{\phi_\mathrm{z}}\right)^{m} \cdot e^{-C\left(\frac{\phi}{\phi_\mathrm{z}}\right)^{m+1}} \tag{1}$$

$$\mathrm{d}x = \frac{\mathrm{d}Q}{Q}$$

where:
Q – is total fuel heat input,
ϕ – is angle between initial and current time of the simple Vibe function,
ϕ_z – is duration angle of the simple Vibe function (duration of the heat release),
m – is Vibe function shape parameter,
C – is Vibe function parameter, C = 6.9 for complete combustion, and
x – is cumulative normalized heat released (mass fraction burned).

The integral of the Vibe function gives the fraction of the fuel mass that has been burned since the start of combustion:

$$x = 1 - e^{-C\left(\frac{\phi}{\phi_\mathrm{z}}\right)^{m+1}} \tag{2}$$

$$x_1 = \sigma_\mathrm{g} \cdot x = \sigma_\mathrm{g}\left[1 - e^{-C\left(\frac{\phi_1}{\phi_{\mathrm{z}1}}\right)^{m_1+1}}\right]$$

$$\frac{\mathrm{d}x_1}{\mathrm{d}\left(\dfrac{\phi_1}{\phi_{\mathrm{z}1}}\right)} = \sigma_\mathrm{g} \cdot C\,(m_1+1)\left(\frac{\phi_1}{\phi_{\mathrm{z}1}}\right)^{m_1} \cdot e^{-C\left(\frac{\phi_1}{\phi_{\mathrm{z}1}}\right)^{m_1+1}} \tag{3}$$

where:
m_1 – is the first Vibe function shape parameter,
ϕ_1 – is angle between initial and current time of the first Vibe function,
$\phi_{\mathrm{z}1}$ – is duration angle of the first Vibe function, and
σ_g – is the share of fuel mass burnt as describe by the first Vibe function.

The superposition of two Vibe (double Vibe) functions is used to approximate the measured heat release characteristics of a diesel engine with direct injection (DI) more accurately. In this case, two Vibe functions are specified, the first eg. (3) is used to model the pre-mixed combustion (explosive part – index "1") and the second eq. (4) is used to model the diffusion controlled combustion (index "2") [7, 8].

$$x_2 = (1-\sigma_g)\cdot x = (1-\sigma_g)\cdot\left[1-e^{-C\left(\frac{\phi_2}{\phi_{z2}}\right)^{m_2+1}}\right]$$

$$\frac{dx_2}{d\left(\frac{\phi_2}{\phi_{z2}}\right)} = (1-\sigma_g)\cdot C(m_2+1)\left(\frac{\phi_2}{\phi_{z2}}\right)^{m_1}\cdot e^{-C\left(\frac{\phi_2}{\phi_{z2}}\right)^{m_2+1}} \tag{4}$$

$$x = x_1 + x_2$$

$$\frac{dx}{d\alpha} = \frac{dx_1}{d\alpha} + \frac{dx_2}{d\alpha} \tag{5}$$

where:
$dx/d\alpha$ – is normalized rate of heat released,
α – is crank angle (CA),
m_2 – is the second Vibe function shape parameter,
ϕ_2 – is angle between initial and current time of the second Vibe function, and
ϕ_{z2} – is duration angle of the second Vibe function (duration of the heat release).

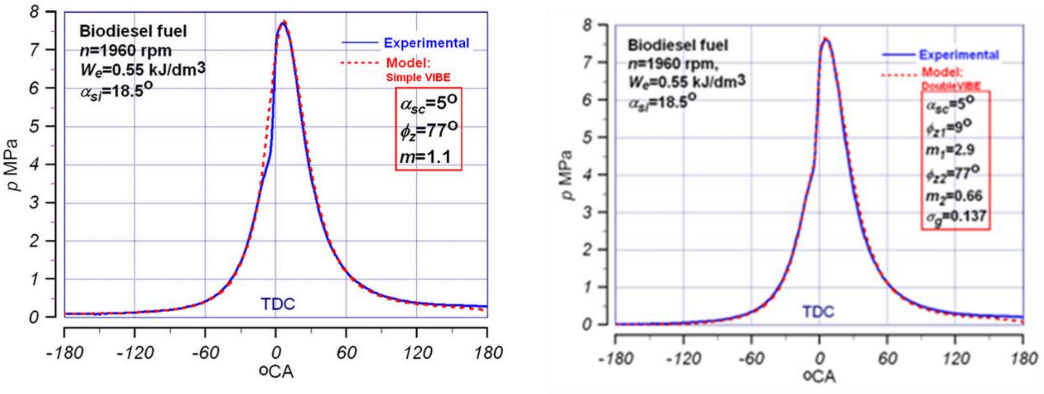
Experimental research

Acquisition of an indicator diagram

One of the most frequently used ways to obtain necessary information about the working process is recording of the cylinder pressure. Even without any calculation, the cylinder pressure record provides some information about combustion in engine cylinder, for example: peak pressure and its position, the rate of pressure rise (influencing combustion noise), *etc*. Research engineer can obtain information that is more detailed by the analysis of indicator diagram. Computation of mean indicated pressure, high-pressure and low-pressure parts of the cycle enables the determination of engine mechanical losses (after experimental determination of mean effective pressure) and gas exchange losses. The combustion process and its losses are, however, more complex, and therefore, far more sophisticated thermodynamic analysis of pressure data is required. The rate of heat released by combustion or simply "the rate of combustion" and the mean gas temperature appear as major results of that analysis [9].

The importance of the information that may be obtained from an engine in-cylinder pressure trace has been recognized since early days of engine development. Scientists and engineers have been paying great attention to indicator diagram analysis even when test instrumentation and the calculation possibilities where rather limited, before the days of digital computers and data acquisition systems. The development and application of digital data acquisition systems in engine testing have further gained the possibilities of measurement of high-speed engine variables and significantly improved the accuracy of acquired data.

Experimental verification of a theoretical model of engine working cycle is constitutive part of any modeling. Verification of mathematical model of a cycle is conducted by comparison between the results of theoretical calculations of cylinder pressure and the results of experimentally acquired cylinder pressure course. Thereat, a simple global match between calculated and experimental course of pressure is taken as model quality measure. Figure 1a shows comparison between experimental data of acquired indicator diagram and the results of a model in which simple Vibe function is applied. Figure 1b presents the results of a model in which a double Vibe function is applied. There is a better coincidence between experiment and model in figure 1b. The experiment and the model are related to DI-diesel engine with distinct pre-mixed ("explosive" – very rapid) combustion (chemically controlled) and diffusive combustion (controlled by mixing rate).



(a) (b)

Figure 1. Results of the model and experimental data of the indicator diagrams

At DI-diesel engines with distinct pre-mixed combustion and diffusive combustion, a model based on simple Vibe function cannot describe accurately enough the characteristic form of the combustion process. In that case, it is necessary to use a model based on superposition of two Vibe (double Vibe) functions – one simulating the pre-mixed part, the other simulating the diffusive part.

A measuring chain used for measurement of cylinder pressure using piezoelectric pressure transducer is depicted in figure 2. Signal from pressure transducer was lead through charge amplifier to acquisition device, AVL Indimeter 619. In addition, signal of crankshaft rotation angle and signal of top dead centre (TDC) were lead to the same device. Crankshaft rotation angle (3600 impulses per one revolution) was used as tact for A/D conversion, while the signal of TDC was used as reference value for initiation of sampling and phasing of pressure course and angle position of the crankshaft. Angle encoder, AVL 365CC, mounted directly on the crankshaft of the experimental engine, figure 3, has generated these signals.

Partial processing of the signal of engine cylinder pressure and its recording in function of crankshaft angle, were conducted by program "AVL IndiCom, Indicating Software Version 1.2" [10].

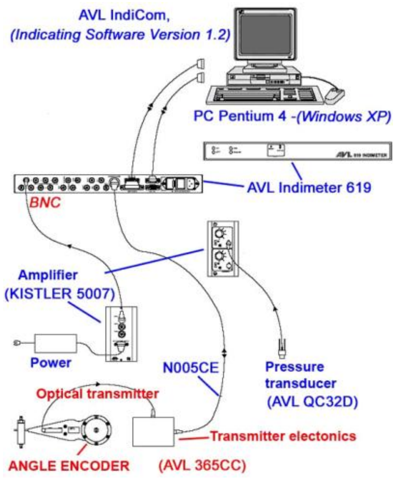


Figure 2. AVL Indicating system diagram

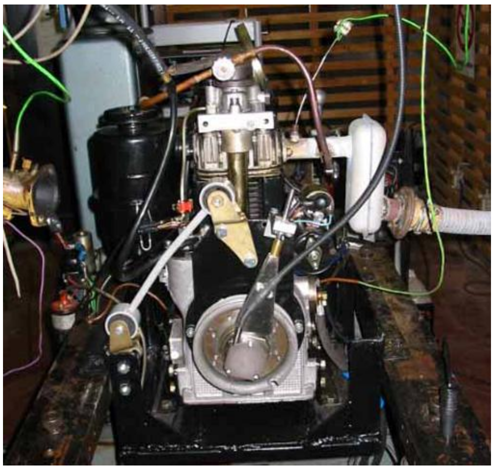


Figure 3. Test engine at test bench

The 450 was used in experiment. It is mono-cylindrical, air-cooled DI-diesel engine. Compression ratio of this engine is 17.5. Injection timing was fixed at 18.5 °CA at all regimes. Engine is loaded by hydraulic brake, SCHENK U116/2, figure 3.

Tests were done with classic diesel fuel (cetane number 52.4) and with biodiesel fuel (cetane number 53.2). Cetane numbers (CN) were determined according to our own installation and method [5], just before conducting the experiment.

Engine operating regimes were determined according to ESC (European Stationary Cycle) 13 – stage cycle and shown in figure 4. Figure 4 also shows values of air excess ratios (λ) for each regime.

Determination of heat release

Calculation rate of heat release and other relevant parameters of the working cycle, that could not be determined by "AVL IndiCom, Indicating Software Version 1.2" programme, is done using the software package developed at the Laboratory for IC engines of the Faculty of Mechanical Engineering in Kragujevac [6]. The results of these calculations for one characteristic regime of engine working with regular diesel fuel and with biodiesel fuel are shown in figure 5. The results refer to mean cylinder pressure obtained by averaging 50 cycles.

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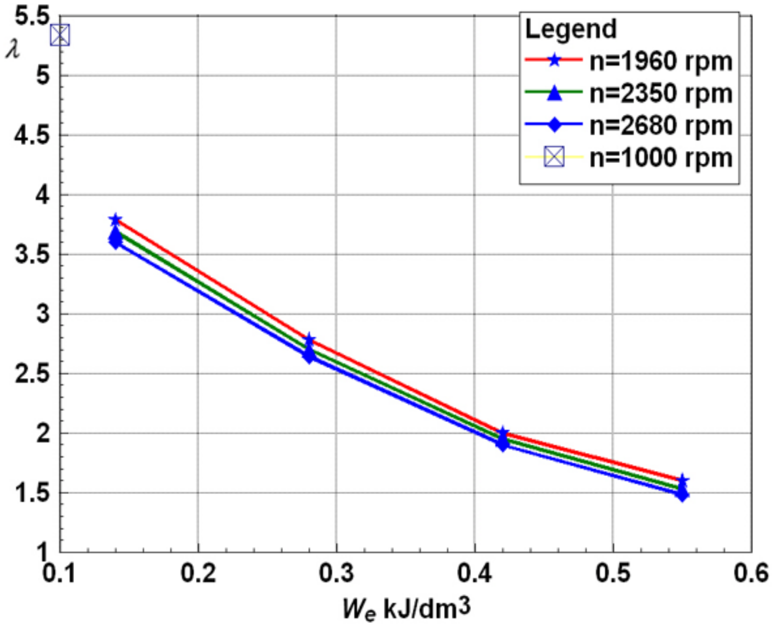

Figure 4. Working regimes of the engine and air excess ratio

Used biodiesel fuel has better characteristics of in flammability, thus a shorter ignition delay period was recorded at all regimes than for diesel fuel. The delay period is longer when the engine works on the diesel fuel. The longer delay period, when the engine works with diesel fuel, results in a higher proportion of the injected fuel remaining unburned. The large accumulation of unburned fuel during the delay period leads to a characteristic sharp peak, figure 6.

Considering that injection timing was constant at all working regimes, the start of combustion (2% of burned mass) does not depend on engine load, but only on engine speed. With the increase of engine speed, the start of combustions gets closer to TDC, due to the same value of delay of combustion. A similar case is with the centre of combustion process (50% of burned mass), only the end of combustions (96% of burned mass) moves away from TDC during load increase. Engine load increase and larger quantity of injected fuel lead to location of end of combustion moving away from TDC. Therefore, combustion is the shortest at idle speed and at the lowest engine speed, while combustion is the longest at full load and at the highest engine speed, figure 7.

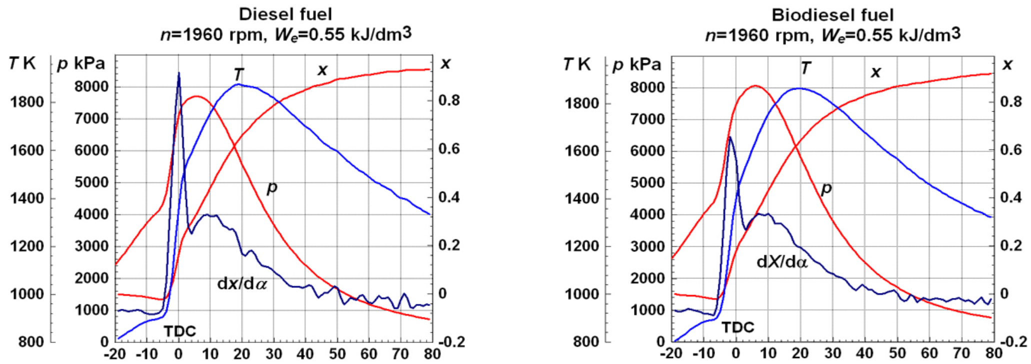

Figure 5. The results of pressure diagrams analysis: normalized rate of heat release d*x*/d*α*, cumulative heat released *x*, and mean gas temperature *T*

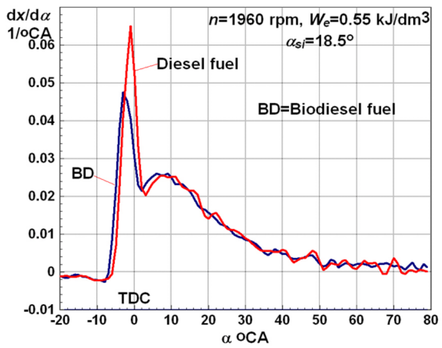

Figure 6. Actual normalized rate of heat release diagrams

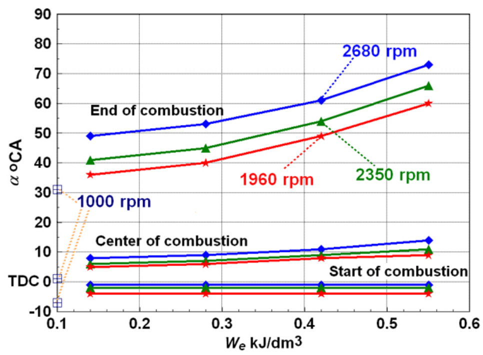

Figure 7. Start, centre and end of combustion while the engine works with biodiesel fuel

Determination of parameters of double Vibe function

After the start and the length of combustion process were determined in domain of crankshaft angle, parameters of double Vibe function may be determined. A least square fit was used in the paper, applied to experimentally determined rate of heat releases.

Figure 8 shows the approximation of two actual heat release diagrams of a DI-diesel engine by a double vibe function. Figure 8a refers to DI-diesel engine working with diesel fuel, while figure 8b refers to DI-diesel engine working with biodiesel fuel at the same working regime.

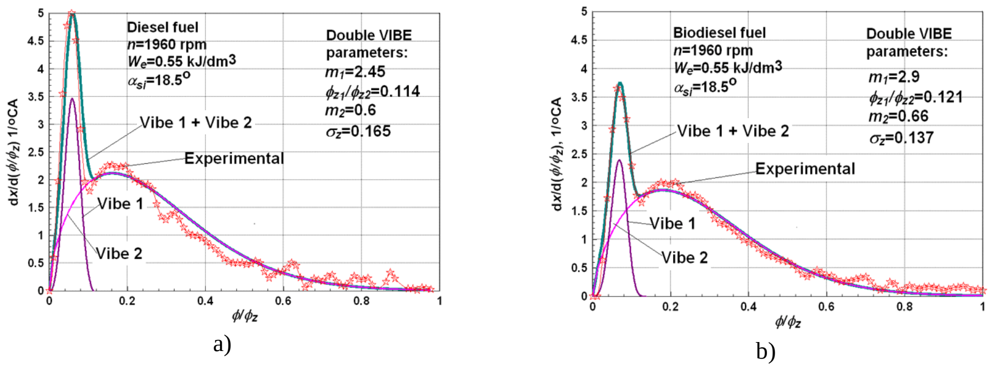

Figure 8. Approximation of two actual rate of heat release diagrams by a double Vibe function

Influence of working regimes on double Vibe parameters

By application of method presented in section *Determination of parameters of double Vibe function* to all 13 experimentally determined rates of heat releases, the parameters of double Vibe function are obtained for all 13 regimes.

Figure 9 shows the influence of engine speed and engine load while working with biodiesel fuel on the first Vibe function shape parameter. It may be concluded that engine load and engine speed do not have significant influence on the first Vibe function shape parameter. The smallest value of the first Vibe function shape parameter is recorded at idle speed and at the lowest engine speed.

Figure 10 shows the influence of engine speed and engine load while working with biodiesel on the second Vibe function shape parameter. In contrast to the previous, the second Vibe function shape parameter is substantially dependent on engine speed and engine load. At the lowest loads, there is greater influence of engine speed and the second Vibe function shape parameter decreases with the increase of engine speed. At full load, the influence of engine speed is considerably smaller and the second Vibe function shape parameter increases with the increase of engine speed. Thus, gradient of load influence decreases with the increase of engine speed.

Figure 11 shows the influence of engine speed and engine load while working with biodiesel on the relative duration of the first Vibe function. Relative duration of the first Vibe function is the largest at idle speed and it substantially decreases with the increase of engine load and the decrease of engine speed. Thereat, the influence of engine speed is greater at larger loads.

Figure 12 shows the influence of engine speed and engine load, while working with biodiesel, on the share of fuel mass burned as described by the first Vibe function. At lower loads (smaller quantity of fuel injected per cycle), during the delay of combustion (mainly dependant on CN), a relatively large quantity of fuel is injected into the cylinder (in relation to total cycle quantity). Therefore, in the first period of combustion (pre-mixed combustion), which mainly depends on chemical characteristics of fuel, proportionally larger fuel quantity burns. The influence of engine speed on this parameter is not distinct.

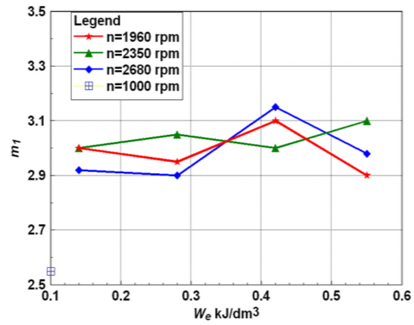

Figure 9. Influence of working regimes on the first Vibe function shape parameter

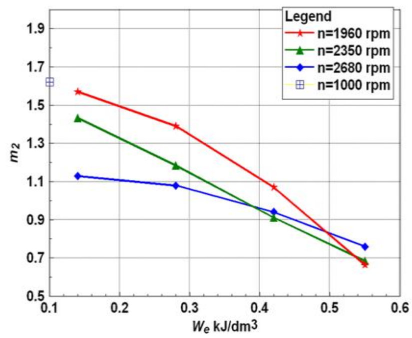

Figure 10. Influence of working regimes on the second Vibe function shape parameter

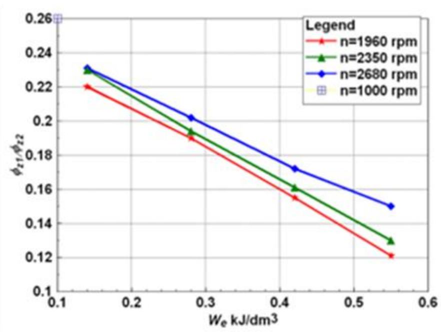

Figure 11. Influence of working regimes on duration of the first Vibe function

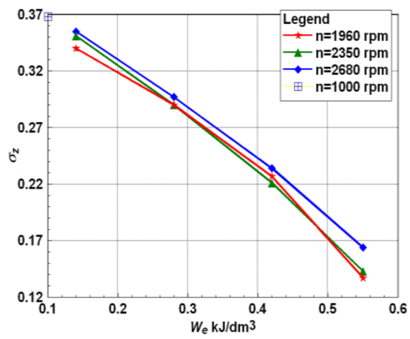

Figure 12. Influence of working regimes on the share of fuel mass burned during the first Vibe function

Conclusions

At DI-diesel engines with distinct pre-mixed combustion and diffusive combustion, a model based on simple Vibe function cannot describe accurately enough the characteristic form of the combustion process. In that case, it is necessary to use a model based on superposition of two Vibe (double Vibe) functions – one simulating the pre-mixed part, the other simulating the diffusive part.

Engine load and engine speed have not significant influence on the first Vibe function shape parameter. The smallest value of the first Vibe function shape parameter is recorded during working at idle speed and at the lowest engine speed.

The second Vibe function shape parameter essentially depends on engine speed and engine load. At lower loads, there is larger influence of engine speed and the second Vibe function shape parameter decreases with the increase of engine speed. At full loads, the influence of engine speed is considerably smaller and the second Vibe function shape parameter increases with the increase of engine speed. Thus, the gradient of load influence decreases with the increase of engine speed.

At lower loads and at idle speed, relative duration of the first Vibe function should have larger values, while at larger engine speeds – larger values of relative duration of the first Vibe function should be adopted. The influence of engine speed is larger at higher loads.

At lower loads (smaller fuel quantities), the share of fuel mass burned during the first Vibe function should have larger values, because relatively large quantity of fuel is injected and burned in the first period of pre-mixed combustion. The influence of engine speed on this parameter is not distinct.

Nomenclature

C – Vibe function parameter, C = 6.9 for complete combustion, [–]
m – Vibe function shape parameter, [–]
n – engine speed, [rpm]
p – pressure in cylinder, [Pa]
Q – total fuel heat input, [J]
T – mean gas temperature, [K]
x – cumulative normalized heat released (mass fraction burned), [–]
W_e – specific effective work of engine, [kJdm^{-3}]

Greek letters

α – crank angle, [°CA]
α_{si} – angle between start of injection and TDC (Injection timing), [°CA]
α_{sc} – angle between start of combustion and TDC, [°CA]
λ – air excess ratio, [–]
ϕ_z – duration angle, [°CA]
ϕ – angle between initial and current time, [°CA]
σ_g – the share of fuel mass burned as described by the first Vibe function, [–]

Subscripts

1 – the first Vibe function
2 – the second Vibe function

Abbreviations

BD – biodiesel
CA – crank angle degree
CN – cetane number
DI – direct injection
ESC – European stationary cycle
IC – internal combustion
PC – personal computer
SI – spark ignition
TDC – top dead centre

References

[1] Pešić, R., Davinić, A., Taranović D., Delusion of the Kyoto Protocol, Biofuels and Advance in Internal Combustion Engines**,** *Mobility & Vehicle Mechanics – International Journal for Vehicle Mechanics, Engines and Transportation Systems*, *1* (2009), 35, pp. 12-23
[2] Gruden, D., Environmental Protection in the Automotive Industry (in German), Vieweg + Teubner Verlag GmbH, Wiesbaden, Germany, 2008
[3] ***, Directive 2003/30/EC on Promotion of the Use of Biofuels or the Renewable Fuels for Transport, *Official Journal of the European Union*, 2003
[4] Stojiljković, D., *et al.*, Mixtures of Bioethanol and Gasoline as a Fuel for SI Engines**,** *Thermal Science*, *3* (2009), 13, pp. 219-228
[5] Pešić, R., Davinić, A., Veinović, S., New Engine Method for Biodiesel Cetane Number Testing, *Thermal Science*, *1* (2008), 12, pp. 125-138

[6] Pešić, R., Automobile SI Engines with Minimal Fuel Consumption, Monographic Issue of Journal *Mobility & Vehicle Mechanics*, 1994
[7] ***, User's Guide AVL BOOST Version 4.0.1, AVL List GmbH, Graz, Austria, 2003, pp. 1-270
[8] Pešić, R., *et al*., The Experimental VCR Diesel Engine and Determination of Double Vibe Function Parameters, CAR 2005, *Proceedings,* 9th International Congress on Automotive, Pitesti, Romania, 2005, pp. 1-10
[9] Tomić, M. *et al*., A Quick, Simplified Approach to the Evaluation of Combustion Rate from an Internal Combustion Engine Indicator Diagram, *Thermal Science*, *1* (2008), 12, pp. 85-102
[10] ***, Engine Instrumentation – Operating Instructions, AVL IndiCom Indicating Software Version 1.2, AVL List GmbH, Graz, Austria, 2002, pp. 1-758

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