INVESTIGATION INTO THE EFFECT OF DIFFERENT FUELS ON IGNITION DELAY OF M-TYPE DIESEL COMBUSTION PROCESS

by

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An ignition delay is a very complex process which depends on a great number of parameters. In practice, definition of the ignition delay is based on the use of correlation expressions. However, the correlation expressions have very often limited application field. This paper presents a new correlation which has been developed during the research project on the direct injection M-type diesel engine using both the diesel and biodiesel fuel, as well as different values of a static injection timing. A dynamic start of injection, as well as the ignition delay, is defined in two ways. The first approach is based on measurement of a needle lift, while the second is based on measurement of a fuel pressure before the injector. The latter approach requires calculation of pressure signals delay through the fuel injection system and the variation of a static advance injection angle changing. The start of a combustion and the end of the ignition delay is defined on the basis of measurements of an in-cylinder pressure and its point of separation from a skip-fire pressure trace.

The developed correlation gives better prediction of the ignition delay definition for the M-type direct injection diesel engine in the case of diesel and biodiesel fuel use when compared with the classic expression by the other authors available in the literature.

Key words: *ignition delay, diesel engine, M-type, biodiesel*

Introduction

The process of a fuel auto ignition in the internal combustion (IC) diesel engines is a result of numerous close related, complex physical and chemical processes that are often called an ignition delay. According to usual definitions, the ignition delay starts with an injection of liquid phase of fuel into the working chamber of the IC engine. The processes which follows are: development of spray, dispersion, and evaporation of fuel. All of the mentioned processes represent a physical ignition delay.

The first chemical pre-ignition reactions between the fuel and oxygen start during the evaporation process of the injected fuel. The auto ignition of the air/fuel mixture in an evaporated phase occurs after the completion of the chemical ignition delay.

It is very complex to separate the physical and the chemical ignition delay. Additionally, thermodynamic boundary conditions of the process are continuously changing. Changes of the thermodynamic boundary conditions of the process are caused by the piston movement, as well as by cooling of the combustion chamber walls due to the impingement and evaporation of the injected fuel. A local equivalent air/fuel ratio is influenced by the evaporation of fuel, diffusion, and spatial movement of the working mixture. Since chemical ignition delay mostly depends on the temperature, pressure, and equivalent air/fuel ratio, a auto ignition of the injected spray depends on development of the physical parameters. At the end of the ignition delay and the first local auto ignitions, the combustion process evolves in a manner determined by the level of heterogeneity.

The main difference in a procedure of the air/fuel mixture formation in the diesel and gasoline engines, as well as in the industrial flame devices could be recognized in a fact that the temperature difference between air and fuel is greater in the case of the diesel engine. Analyzing the expression by Arhenius [1-3] one can notice that the temperature has a significant influence on processes of chemical reactions. High temperatures will have an influence on a fuel structure in the case of the fuel mixture with a hot air in the diesel engine. Molecules of the fuel, especially in the case of heterogeneous mixtures, lead to formation of peroxides with impulsive partial reactions. The process of molecules decay follows, where this decay accelerates auto ignition. From the other side, the rest of the molecules become richer with carbon which is an inert in reaction and causes a soot formation. With a release of hydrogen from a molecule, the molecule leaves without the natural catalyst because the carbon reaction is accelerated to a desired value by a final product of the hydrogen reaction but only if this reaction is happening on a molecular level. The greater increasing of the temperature during the combustion process leads to decay processes that can not be stopped, until the total breakdown of molecules. In that way, the usual character of reactions is achieved in the diesel engine, the high speed of reactions at the beginning, related with noise and finally an additional oxidation of the carbon.

In order to overcome all of disadvantages related with the classic fuel supply of the diesel engine, the M-system of the diesel fuel injection makes possible a setting of a wide and a fine film of fuel on walls of a ball shape working chamber [4]. Fuel introduced into the working chamber in this way is not forced to mix with the air on high temperatures which is the case with the other systems of the mixture formation. The fuel introduced into the working chamber with the M-system is heated up to the allowed values. In this way, oxygen from the air could be accepted by the fuel in small quantities without the oxygen hydraulic "muffle" and simultaneous preheating of the fuel molecule. Self ignition of the non-controlled large fuel quantity is avoided [4].

Selection of a small distance between the injector orifice and the walls of the combustion chamber enables only a small quantity of fuel set apart from the injected spray. A part of the fuel located on the wall of the working chamber gradual heats and evaporates. A speed of evaporation depends on a temperature between the wall and gas.

Only evaporated part of the fuel can participate in the mixture formation. The mixture formation happens very fast because of a gas phase of the fuel and a significant speed of air motion. Evaporated parts of fuels did not exposed to any or a very small decay and they create a mixture that combusts due to ignition by the other sources. The sources could be red-hot parts of carbons created by self ignition of a small fuel quantity during the initial phase.

Background

Numerous expressions for the definition of the ignition delay for the IC engines, as a function of parameters of the IC engine and a state of the working mixture in cylinder, could be found in the literature. Most of them are based on the Arrhenius law of combustion speed in the following form:

$$\tau_{\mathrm{id}} \quad Ap \quad {}^{b}\mathrm{e}^{\frac{E_{\mathrm{A}}}{\tilde{R}T}} \tag{1}$$

where A and b are coefficients and E_{A} the activation energy.

Disadvantage of the expression (1) could be recognized in an inadequate interpretation of the ignition delay and the expression could be used only for definition of the chemical ignition delay. In order to determinate the whole ignition delay, it is necessary to incorporate definition of the physical ignition delay. Expressions integral the physical and chemical ignition delays have a more complex form than expression (1). Also, a problem of temperature (T) that should be used in expression (1) exists. Taking into consideration the literature sources, the first group of authors thinks that the gas temperature at the fuel injection should be used, while the second group uses the average temperature in the working chamber during the ignition delay. Similar opinions exist in the case of the pressure in the chamber.

Majority of the expressions for the definition of the ignition delay were based on selection of the appropriate values of a pressure and a temperature related with the state of engine, for example, a pressure and a temperature at start of the injection process, p_{fi} and T_{fi}, respectively. Additionally, it could be declared that an influence of the engine load, defined by the equivalence air/fuel ratio, during compression stroke is defined using only state of air T_{ic} and p_{ic} at the intake valve closing (indirectly by coefficient of residual gasses), while speed loads has an influence directly on T_{fi} and p_{fi} [5]. Selection of p and T on this way could introduce a certain error in the correlation expressions.

Taking into consideration the fact that all correlation expressions presented in the literature are semi empirical and achieved by the measurements on certain engines, on a certain speed and load regimes, using a certain fuel type, a question regarding its universality for application in other cases could be raised.

This paper analyzes frequently used semi empirical expressions from the literature [6-11]. The intention is to check application of these expressions on the diesel engine with the M-system of the fuel injection in the case of the diesel and biodiesel fuel use. Analyzed correlations are systematically presented in tab. 1. Subsequently, the experimental results are used to propose a modified expression for the M-system engine operating with diesel or biodiesel fuel.

The approach to development of a new correlation expression is based on the fact that the period of the ignition delay should be a function of a pressure, a temperature, properties of used fuels and engine speed regimes. The arguments for this approach are based on a physical intuition and chemical kinetic considerations.

The first part of the paper contains the basic information regarding the research object, as well as settings for the experimental researching in order to obtain reliable data for definition of the ignition delay. The second part is to obtain the correlation expression as a function of the following variables: pressure, temperature, fuel type, and engine speed. The expression for definition of the ignition delay is created based on the results of researching on the engine, as well as on the different static advance injection angle. The final part of the paper presents possibilities for the prediction period of the ignition delay using the new correlation expression and the comparison with the experimental results.

Experimental setup

The experimental investigation is carried out in the Laboratory for IC Engines on the Faculty of Mechanical Engineering, University of Maribor, Slovenia. The object of researching is the diesel engine M.A.N. of D2566MUM type. The modern equipment with several measurement chains and digital registering and processing is used during the research. Additionally, the measurement devices without possibilities for automatic registration and data acquisition are used too. The basic information regarding the IC engine and the diesel fuel injection system used during the research are presented in tabs. 2 and 3, respectively.

Table 1. Correlations for definition of period of ignition delay [6-11]

Author	Correlations
Shipinsky	$\tau_{id} \quad \frac{0.8216}{p_m^{0.386}} \exp \frac{4644}{T_m} \quad \frac{40}{CN}^{0.69}$
Sitkei	$\tau_{id} \quad 0.5 \quad 0.133 p_{ic}^{0.7} \exp \frac{3930}{T_{ic}} \quad 0.00463 p^{1.8} \exp \frac{3930}{T_{ic}}$
Sitkei (version 1)	$\tau_{id} \quad 0.5 \quad 0.0735 p_{fi}^{1.6} \exp \frac{3930}{T_{fi}}$
Sitkei (version 2)	$\tau_{id} \quad 0.5 \quad (0.0266 p_{fi}^{0.7} \quad 0.0735 p_{fi}^{1.6}) \exp \frac{3930}{T_{fi}}$
Wolfer	$\tau_{id} \quad 0.429 p_{fi}^{1.19} \exp \frac{4650}{T_{fi}}$
Wolfer (version 1)	$\tau_{id} \quad 3.52 p_{fi}^{1.022} \exp \frac{2100}{T_{fi}}$
Tolstov	$\tau_{id} \quad 0.0523(1 \quad 0.00016n) \frac{T_{fi}}{p_{fi}}^{0.5} \exp \frac{1477}{T_{fi}}$
Assanis and co-authors	$\tau_{id} \quad \frac{2.4}{p_m^{1.02} \Phi^{0.2}} \exp \frac{2100}{T_m}$

Table 2. Basic information of the M. A. N. D2566MUM IC diesel engine

Engine type	6-cylinder, in-line, aspirated, horizontal diesel engine		
Displacement	11413 dm^3	Con. rod length	244 mm
Bore	125 mm	Effective power/speed	160 kW/2200 rpm
Stroke	155 mm	Compression ratio	18:1

Table 3. Basic information of the diesel fuel injection system

Injection system	In-line diesel fuel injection pump equipped with governor and timing device		
Fuel injection pump Type	Bosch PES 6A 95 D410 LS2541	Timing device	Mechanical flyweight
– Speed maximum	1230 rpm	Injector manufacturer Type	Bosch DLLA 25 S 834
– Fuelling (fuel quantity)	131 + 2 mm^3 per cycle per cylinder	– Needle opening pressure	175 bar
– Pump plunger (diameter lift)	Φ 9.5 mm 8 mm	– Maximum of the needle lift	0.30 mm
Governor	Bosch RQ 250/1100 AB 965 DL	Fuel pipe (diameter length)	Φ Φ 1.8 900 mm

The M-type diesel engine was tested using an electrical dynamometer. Other parameters that are not closely related with the definition of the ignition delay are monitored too for verifications.

Pressure in a cylinder, pressure after the high pressure pump, and pressure before injector are measured by the piezoelectric sensors. The location of piezoelectric sensor in a head of the first cylinder is shown in fig. 1, whilst the location of a sensor for measuring pressure before the injector is shown in fig. 2. These piezoelectric sensors are connected to the amplifier Kistler, of 5007 type and a connector block NI SCXI-1301. This module enables data transfer to an 8-chanell differential amplifier module NI SCXI-1140 for sampling [12]. A needle lift was measured by an inductive sensor and its signals were registered by the data acquisition system.

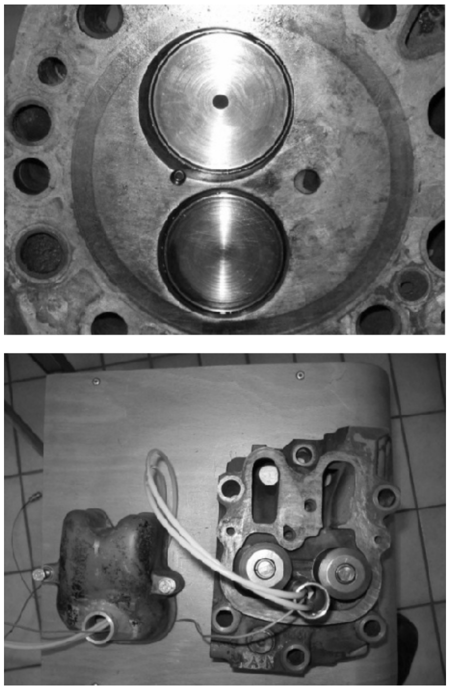

Figure 1. Piezoelectric sensor located in the cylinder head

The data acquisition system is built on the NI platform, SCXI series with the data acquisition card type NI AT-MIO-16E-2 built in a PC [12]. This card provides data processing with 16 independent analog inputs, 12-bit resolution and 500 kS/s speed sampling. A necessary software support is obtained by the computer program LabVIEW 7.0 where the whole algorithm for the data acquisition is completed. NI DAQmx is used for fast and efficient setting of parameters for each input parameters. In order to define a realistic start of an injection and combustion, additional routines necessary for post processing are developed using FORTRAN.

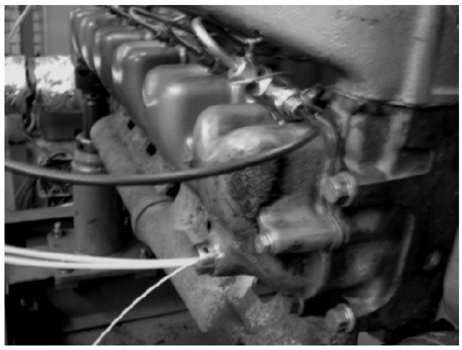

Figure 2. Location of piezoelectric sensor for pressure registration in HP line before injector

The engine test regimes are selected in the way that they enable realistic comparison of engine operations for the use of different fuels (diesel, biodiesel, and its mixtures). During the research project, a special care is dedicated to a calorific fuel value, and the main objective is to achieve almost the same effective engine power in the case of the use of different fuels. The engine speeds at full load regime are varied from the minimum (1000 rpm) to the maximum governed (2200 rpm) speed.

The results of researching, presented in the paper, are related with already mentioned regimes of the engine operation and for the static advance injection angle of 21 deg bTDC and 23 deg bTDC. The reason for selection of these values could be explained by the results of the previous research presented in [13] where the optimal regime of the engine operation in the case of biodiesel fuel and pollutants emissions, engine powers, and specific fuel consumptions are achieved for 21 deg, bTDC. According to the references of the engine manufacturer, an optimal angle of the advance injection in the case of the diesel fuel is 23 deg bTDC.

Since fuels used in this paper are diesel and biodiesel, as well as its mixtures, it is necessary to define the basic characteristics of pure fuels that are presented in tab. 4.

Table 4. Basic physical and chemical properties of diesel and biodiesel fuels

Properties	Diesel	Biodiesel
Density at 15 °C [kg/m^3]	845	865
Kinematic viscosity at 40 °C [mm^2/s]	2.5	4.3
Calorific value [MJ/kg]	42.6	37.3
Cetane number	46	>49
Contents – mass fraction of carbon – mass fraction of hydrogen – mass fraction of sulfur – mass fraction of oxygen	 0.860 0.134 0.003 –	 0.7750 0.1210 0.0001 0.1040
Stoichiometric combustion ratio	14.5	12.4
Sound velocity [m/s]	1430	1460

The HP pump is equipped with a timing device for a static advance angle. A characteristic of the timing device could be expressed as follows:

$$\varphi_{\mathrm{V}} = 0.012(n - 1200) \ \ [\mathrm{deg\,CA}] \tag{2}$$

in the case of the HP pump speed range 1200-2200 rpm.

Interesting angles that define the ignition delay (φ_{id}) are: timing device angle (φ_{V}), static advance injection angle (φ_{SA}), dynamic advance injection angle (φ_{DA}), and angle of signals delay between the HP pump and the start of injection (φ_{D}).The mentioned angles, as well as the profile of a needle lift (h) is shown in the fig. 3. Figure 3 shows how a needle lift defines a start of injection and how start of combustion is defined using a location of in-cylinder pressure trace separation from skip-fire pressure trace. The first derivation of the pressure curve in function of crankshaft angle was used to check start of combustion process.

Experimental technique for determining start of injection

In order to analyze the ignition delay it is necessary to identify a start of injection into the working chamber of the IC engine and the start of combustion defined as the end of the ignition delay.

During the research, a dynamic advance injection angle is defined in two ways. The first way is based on a known geometry of the fuel injection system, a known needle opening pressure and a moving static advance injection angle due to an action of timing device. The second way is based on using measured a needle lift. The start of combustion is defined on the basis of processing the in-cylinder pressure.

The simplest way for definition of the start of the dynamic injection is measuring needle lift. The start of the needle lift determines a start of a fuel injection.

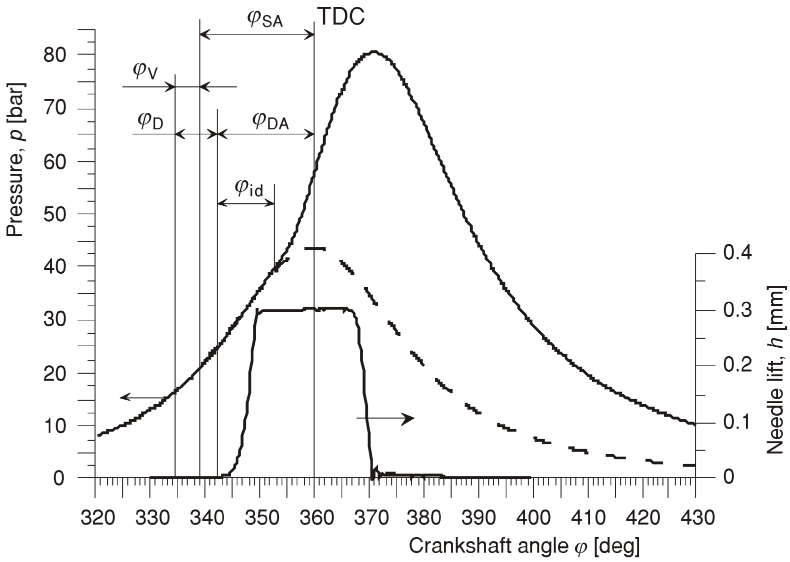

Figure 3. Influence of timing device on static advance injection angle on 1600 rpm in case of diesel fuel use

In the case that diagram of the needle lift is unknown and knowing characteristic values of the fuel injection system, based on the fig. 3, the dynamic advance injection angle could be found in the following way:

$$\varphi_{DA} = \varphi_{SA} + \varphi_V - \varphi_D \tag{3}$$

where the angle of signal delay between the high pressure pump and the start of injection (φ_D), depending on a design of the fuel injection system, could be represented like a sum of the delay of the fuel flow through the high pressure line (φ_{DP}) and a delay of the fuel flows through the in-

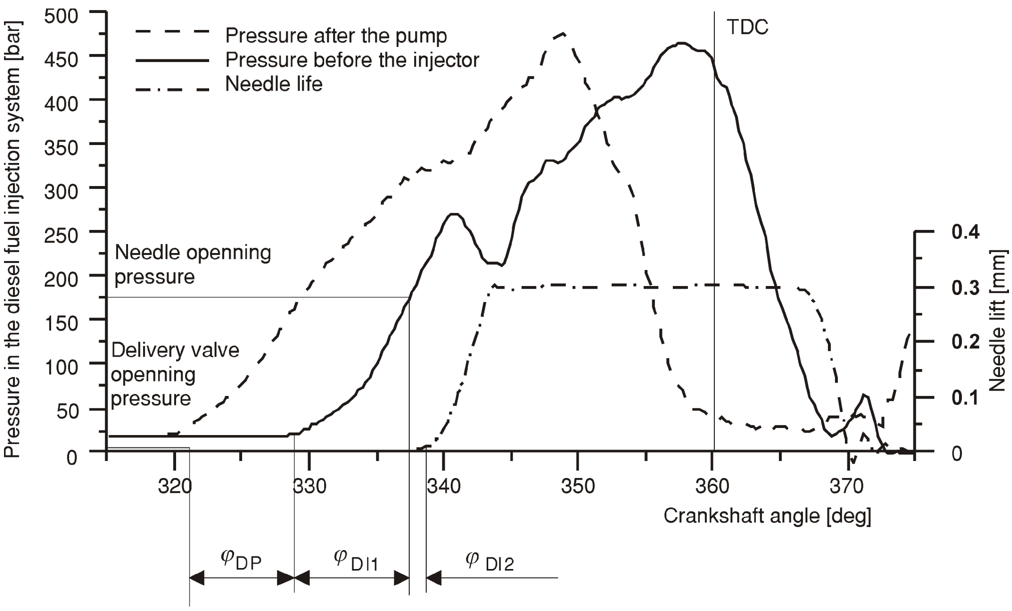

Figure 4. Pressure after pump and before injector and needle movement at 2200 rpm in case of diesel fuel use

jector (φ_{DI}), that is shown in fig. 4. The described method for determination of the start of injector needle lift was used in order to confirm experimental obtained values:

$$\varphi_D = \varphi_{DP} + \varphi_{DI} \tag{4}$$

For example, the values of angles φ_{DP}, φ_{DI1}, and φ_{DI2}, presented in fig. 4 are obtained during the measurement of a dynamic pressure at the start and the end of the high pressure pipe, as well as registering diagram of a needle lift.

Besides the angle of a signal delay between the high pressure pump and the start of injection, caused by a delay of the fuel flows through the high pressure pipe, as it shown in fig. 4, a period for setting enough high pressure in the system in order to lift a needle from its seat (φ_{DI1} in fig. 4) has an influence on the dynamic advance injection angle. Finally, an angle of signals (waves) delay through the whole system with a known length of pipe and an injector, a sound velocity and an engine speed, could be defined using the following expression:

$$\varphi_D \quad \varphi_{DP} \quad \varphi_{DI1} \quad \varphi_{DI2} \quad \frac{6n}{a}(l_p \quad l_i) \quad \varphi_{DI1} \tag{5}$$

The start of combustion as the end of period of the ignition delay is difficult to determinate precisely. There are several approaches for its determination, but every single approach brings certain unreliability. Depending on the criteria selection, the results achieved using certain methods could be different. One of the most reliable approaches for defining the start of combustion and the end of the ignition delay is to use a diagram of the in-cylinder pressure. This approach is based on a sudden gradient of an indicated pressure curve, caused by a start of combustion. Using of luminosity detectors, that detect the flame appearance, represents an alternative method for definition of the ignition delay. However, the experiences in this field show that a visible flame appearance is delayed in relation with a sudden change of an in-cylinder pressure gradient caused by combustion [6-8, 11]. Some authors [8] suggested that the start of combustion should be defined using second or even third derivative of pressure analogues techniques from the gasoline IC engines for the appearance of knocking. This phenomenon is very close to a process of self ignition in the diesel engines and for this reason the mentioned method offers possibilities for the automatic data processing.

Start of combustion was determinate on two different ways in this researching. The first way was based on using in-cylinder pressure diagrams and the skip-fire pressure traces, while second way was based on using the first derivation of pressure in function of crankshaft angle.

An example of the start of combustion definition at an engine speed of 1600 rpm and a use of the diesel fuel at a static advance injection angle of 21 deg bTDC, is shown in fig. 3.

Correlation for ignition delay

The expression for the definition period of the ignition delay at the M-type diesel engine is presented here based on experiments described above. The important parameters in the correlation expression are: pressure (p_{fi}) and temperature (T_{fi}) in a cylinder at the start of the fuel injection, an engine speed (n) and a type of used fuel defined by the cetane number (CN), according to tab. 1. In order to define an appropriate correlation expression for the ignition delay, certain numbers of other expressions presented in the literature were analyzed. The most relevant approach for this work was the approach presented in [14]. By modifications, the correlation expression convenient for the ignition delay definition in the case of the M-type diesel engine powered by different fuels is obtained [15]:

$$\varphi_{\mathrm{id}} = 1.04 \cdot 10^{-3} n p_{\mathrm{fi}}^{0.2} \mathrm{e}^{\frac{20CN - 16550}{\tilde{R} T_{\mathrm{fi}}}} \tag{6}$$

The constants in the correlation expression (6) are defined on the basis of the experimental results, as well as the use of statistical methods like minimization of least-square error in order to achieve minimum differences between the measured and the calculated results.

In order to make a validation and test possibilities of the new correlation expression, the results of calculation are compared with the results obtained by the existing expressions. Figures 5 and 6 represent the results of calculation obtained by the use of expressions by Tolstov and Assanis *et. al*. (see tab. 1), as well as the results obtained by the use of the new correlation expression in the case of the diesel and biodiesel fuels used at the static advance injection angles of 21 and 23 deg bTDC. Two different injection angles were used in this researching. Injection angle of 23 deg bTDC is recommended injection angle declared by motor manufacturer in case of using pure diesel fuel, while angle of 21 deg bTDC is a optimal solution for achieving engine characteristics like power and pollutant's emissions in case of diesel and biodiesel fuel use [13].

It could be noticed some differences (fig. 6) in start of combustion between the cases of pure diesel and biodiesel usage on higher engine speeds. Reasons for such differences could be explained by circumstances that followed the measurements and they have no physical meaning.

Although the expressions by Tolstov and Assanis are conceptually different, these expressions give very close results. However, these correla- tions give significantly different values for the ignition delay in comparison with the realistic values obtained by the experiments. Analyzing the results presented in figs. 5 and 6, the new correlation expression gives a very good prediction of the ignition delay in the case of all influenced parameters: type of used fuel, engine speed, and static advance injection angle.

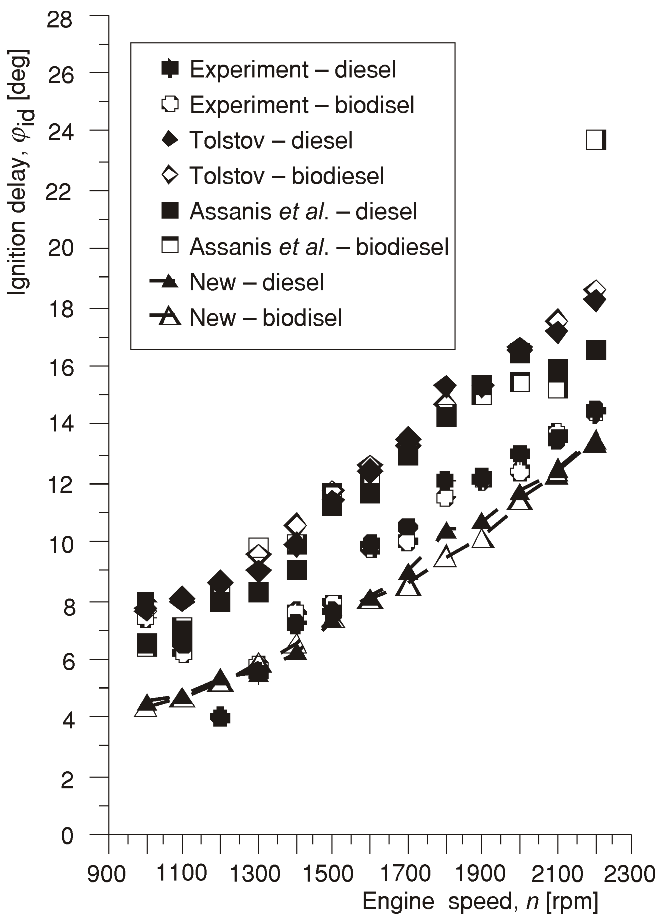

Figure 5. Comparison of ignition delay φ_{id} between diesel and biodiesel fuels in case of static advance injection angle of 23 deg bTDC

Decreasing value of the static advance injection angle from 23 deg bTDC to 21 deg bTDC, or *i. e.* with a later start of the injection, shortens the ignition delay. The reasons for this fact

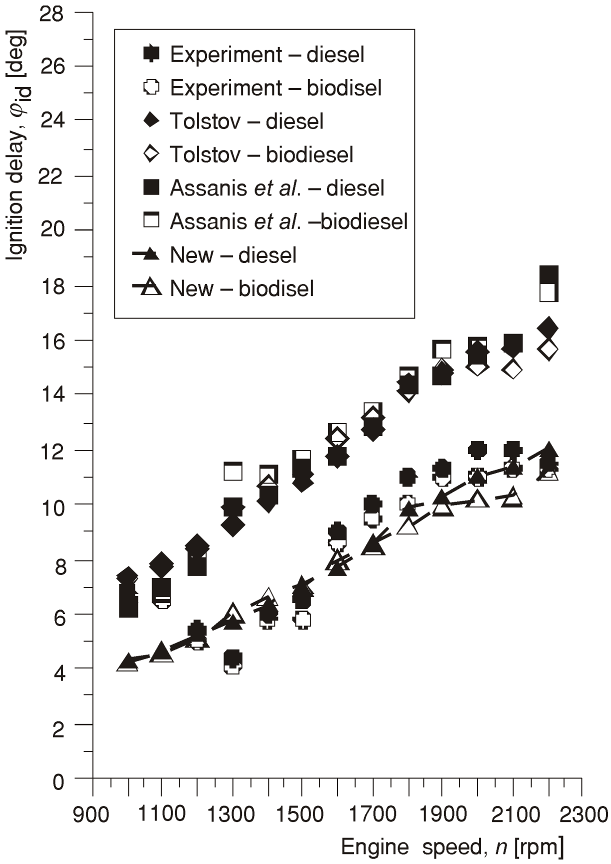

Figure 6. Comparison of ignition delay φ_{id} between diesel and biodiesel fuels in case of static advance injection angle of 21 deg bTDC

could be found in better conditions for the air/fuel mixture ignition due to higher values of the pressure and the temperature nearer to TDC. Furthermore, the analysis of obtained results show increased differences in the ignition delay between the diesel and biodiesel fuels with retard injection, especially at greater engine speeds. Due to the greater cetane number of biodiesel and with the better conditions of the air/fuel mixture ignition, difference in the combustion process in the case of the diesel and biodiesel fuels becomes more visible.

Conclusions

A new developed correlation expression for the definition of the ignition delay characterized by the direct injection of the M-type diesel engine in the case of the diesel and biodiesel fuels use is presented in this paper. Measurements of an indicated pressure, a pressure after the high pressure pump, a pressure before the injector, and a needle lift are done for the complete engine speed range of the IC engine in the case of the diesel and biodiesel fuels use and for the different values of the static advance injection angle. The time (angle) of a start of the injection (dynamic advance injection angle) is defined by a needle lift or in other words by a needle lift from its seat. Knowing all necessary design parameters of the fuel injection system, a needle opening pressure and timing device characteristics, the dynamic advance injection angle is checked by the second method presented in the paper. The compared results are in a very close agreement.

The period of the ignition delay is set as a difference between the start of injection and the start of combustion or the fuel self ignition. The point of separation of the in-cylinder pressure trace from the skip-fire pressure trace is chosen as a start of combustion. The results of calculation are based on the experimental values of the in-cylinder pressures for the requested time (angle) moments like p_{fi} and p_{CB}, while the temperatures T_{fi} and T_{CB} are defined using the state equation based on the measured values of air flow through the manifold. The new correlation expression is compared with the correlation expressions by Tolstov and Assanis et.al. It is noticed that the correlation gives more realistic results for the M-type diesel engine on different engine speeds in the case of different fuels use. By comparison of the new correlation expression with the experimental results one can notice a potential of the correlation regarding the determination of the ignition delay at the direct injection M-type diesel engine in the case of different fuels use in all engine speed regimes and different static advance injection angles.

Nomenclature

a	– sound velocity, [ms^{-1}]
CN	– cetane number of fuel, [–]
E_A	– activation energy, [$KJmol^{-1}$]
h	– needle lift, [mm]
l	– length, [mm]
n	– engine speed, [rpm]
p	– pressure, [bar]
$\tilde{R}$	– universal gas constant, [$kJkmol^{-1}K^{-1}$]
T	– temperature, [K]

Greek letters

τ	– time, [ms]
φ	– crankshaft angle, [deg]

Subscripts

CA	– crankshaft angle
CB	– combustion beginning
D	– signal delay
DA	– dynamic advance injection angle
DI	– delay of fuel flow through injector
DI1	– period of pressure rise in the injector
DI2	– period of signal delay through injector
DP	– delay of fuel flow through pipe
fi	– start of fuel injection
i	– injector
ic	– intake valve closing
id	– ignition delay
m	– average value
p	– pipe
SA	– static advance injection angle
V	– timing device

Abbreviation

bTDC	– before top dead centre
HP	– high pressure
TDC	– top dead centre

References

[1] Dobovišek, Ž., Černej, A., Combustion Processes (in Bosnian), Mechanical Engineering Faculty, University of Sarajevo, Sarajevo, 1978
[2] Joksimović Tjapkin, S., Combustion processes (in Serbian), Faculty of Technology and Metallurgy, University of Belgrade, Belgrade, 1981
[3] Kuo, K. K., Principles of Combustion, John Wiley and Sons, New York, USA, 1986
[4] Meurer, J. S., M. A. N. – M – Combustion Process (in German), M. A. N. Maschinenfabrik Augsburg, Nürnberg, Germany
[5] Dobovišek, Ž., Černej, A., Ideal and Real IC Engine Cycles (in Bosnian), Mechanical Engineering Faculty, University of Sarajevo, Sarajevo, 1976
[6] Heywood, J. B., Internal Combustion Engine Fundamentals, McGraw-Hill, International Editions Automotive Technology Series, McGraw-Hill, New York, USA, 1988
[7] Stojičić, T., Mixture Formation and Combustion Processes in the IC Engine (in Bosnian), Mechanical Engineering Faculty, University of Sarajevo, Sarajevo, 1984
[8] Assanis, D. N., *et al.*, A Predictive Ignition Delay Correlation Under Steady-State and Transient Operation of a Direct Injection Diesel Engine, *ASME Journal of Engineering for Gas Turbines and Power*, *125* (2003), 2, pp. 450-457
[9] Hiroyasu, H., Diesel Engine Combustion and Its Modeling, *Proceedings*, International Symposium COMODIA 85, Tokyo, 1985, pp. 53-75
[10] Markatos, N. C., Computer Simulation of Fluid Flow, Heat and Mass Transfer and Combustion in Reciprocating Engines, Hemisfere Publishing Corporation, New York, USA, 1987
[11] Kwon, S.-I., Arai, M., Hiroyasu, H., Effects of Cylinder Temperature and Pressure on Ignition Delay in Direct Injection Diesel Engine, *Bulletin of the M.E.S.J.*, *18* (1990), 1, pp. 3-15
[12] ***, National Instruments, The Measurement and Automation Catalog, 2004
[13] Kegl, B., Optimisation of MAN D2566MUM Diesel Engine in Case of Biodiesel Fuel Use (in Slovenian), Report fot LPP Ljubljana, Mechanical Engineering Faculty, University of Maribor, Maribor, Slovenia, MOBILIS, CIVITAS II, 6th European Frame Program, 2005
[14] Xia, Y. Q., Flanagan, R. C., Ignition Delay – A General Engine/Fuel Model, SAE technical paper 870591, 1987

[15] Bibić, Dž., Combustion Characteristics of Biodiesel and its Mixtures with Fossil Fuels in Diesel Engines, (in Bosnian), Mechanical Engineering Faculty University of Sarajevo, Sarajevo, 2007

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