COMBUSTION MODELING IN DI DIESEL ENGINES BY USING MIXING CONTROLLED MODEL

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Abstract:

The article presents a modern approach for modeling and calculating of the working process in the DI Diesel engines. The mixing controlled combustion model has been used for describing the combustion process in the combustion chamber as well as the Arrhenius type of equation provides calculation of the ignition delay. Zero dimensional thermodynamic models have been used for modeling the in-cylinder parameters. Based on the above models a Matlab code has been made. Finally different engine cylinder parameters are present calculated by the Matlab calculation code.

Keywords: diesel engine, combustion modeling, direct injection, working process, Matlab

1. Introduction

Modeling of the combustion process as part of the whole working process in the modern diesel engines is an important point of the numerical engine development. A lot of models have been presented during the years [3, 5, 6]. They could be divided into two groups: empirical and phenomenological. The simpler empirical models such as Double Wibe function [3, 5] can be used in calculating the combustion process without paying attention to the injection and evaporation of the fuel, geometrical parameters on the injection nozzle and combustion chambers etc. Usually, an accurate prediction on the heat release in the diesel combustion chamber by means of Wibe function is possible if the researcher has a lot of practical experience in real combustion analyses of the same engine or of an engine with similar mixing process. The more complicated phenomenological models are most commonly used in modern 0D programs for calculation of the combustion chamber heat release. The calculations are based on the real injection process including the injection pressure and the geometrical parameters of the nozzle and the combustion chamber as well as the fuel jet atomization and evaporation models are included in the model. Typical phenomenological diesel combustion models are mixing controlled model [2, 4, 5], stationary gas jet model, packet models, and time scale models [5].

2. Description of the mixing controlled model.

The mixing controlled combustion model was first presented by Chmela et al. [2] and further was developed by Lakshminarayanan [4]. In fact it is not fully phenomenological model because the model is based on empirical functions as well as no atomization and evaporation processes of the fuel jet. Also the variation of the pressure and temperature in combustion chamber are neglected. At the same time the injection process with all needed parameters and the mixing rate in combustion chamber are included. In an analogy with Double Wibe function the heat release rate is influenced by fuel availability and kinetic energy in combustion chamber which are represented in the model by two functions. The heat release rate in modern DI diesel engines with high pressure injection sys-
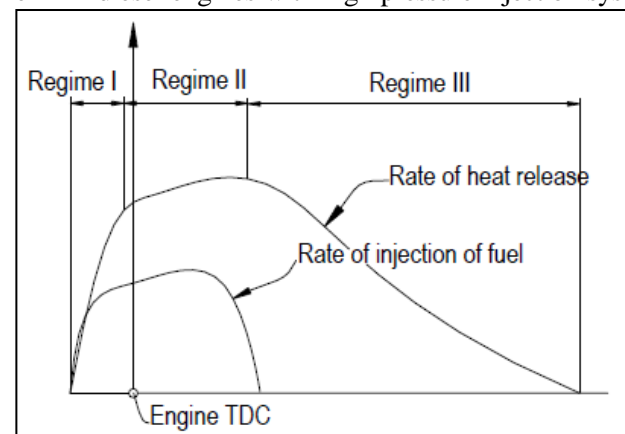

Fig.1 Typical rate of heat release in modern DI diesel engines [4]

tems has a typical shape Fig.1 [4].

It is important to note that using the high pressure injection systems reduces the ignition delay period which in turn decreases the heat release rate during the first combustion period. During the first period of combustion the rate of heat release increase depends on the rate of kinetic energy input by fuel jet. The available kinetic energy steadily increases from zero at the start of injection up to the end of the first period. The first period is finished when the fuel jet reaches the

combustion chamber walls. It immediately reduces the jet kinetic energy and the rate of heat release has almost constant value during the second period. The second period finishes at the end of injection process. During the third period the rate of heat release decreases constantly to the end of combustion process because of decrease of fuel availability and kinetic energy in the chamber.

On the basis of the assumptions above the rate of heat release can be presented in following form [4]:

$$(1) \qquad \frac{dQ}{d\varphi} = C_m . f_1 . f_2$$

Where:

$\frac{dQ}{d\varphi}$ - Rate of heat release;

C_m - Constant of the model;

$C_m = 1000 \frac{kJ}{kg} / deg$;

f_1 - function, depend on fuel availability;

f_2 - function, depend on mixing rate.

Fuel availability presented by f_1 can be easily calculated by the following equation [4]:

$$(2) \qquad f_1 = g_f - \frac{Q_f}{H_u}$$

Where:

g_f - Cumulative mass of the fuel;

$$(3) \qquad g_f = \int_{SOI}^{\varphi} \frac{dg_f}{d\varphi}$$

Q_f - Cumulative heat release;

$$(4) \qquad Q_f = \int_{SOI}^{\varphi} \frac{dQ}{d\varphi}$$

H_u - Lower calorific value of the fuel.

To calculate the function f_1 it is necessary to know the fuel injection rate. When the geometrical parameters of the injector and fuel pressure are known the fuel injection rate can be calculated by the following equation:

$$(5) \qquad \frac{dg_f}{d\varphi} = C_d . \rho_f . F_n . \sqrt{2 \frac{(p_f - p_c)}{p_f}}$$

Where:

C_d - Discharge coefficient of the nozzle holes;

$C_d = 0,8$;

ρ_f - Density of the injected fuel;

F_n - Effective area of the nozzle;

p_f - Pressure of the fuel in the nozzle;

p_c - Cylinder pressure.

The effective area of the nozzle can be calculated using the equation:

$$(6) \qquad F_n = z . \frac{\pi . d^2}{4}$$

Where:

z - Number of the holes on the nozzle;

d - Diameter of the holes on the nozzle;

Function f_2, which presents the mixing rate, depends on the density of turbulent kinetic energy [4]. Three sources of kinetic energy there are in the diesel engine combustion chamber: the fuel jet, the swirl flow and the squish flow. Previous research has shown that main kinetic energy, which is approximately 96%, inputs into by fuel injection [4]. That is the reason why we neglect the kinetic energy which in result from swirl and squish flow.

A modified $k - \varepsilon$ theory for zero dimensional combustion models has been proposed [4] to present the mixing rate in combustion chamber. In the combustion chamber the conditions are such that the mixing rate cannot be zero. For that reason the mixing rate is presented as an exponential function:

$$(7) \qquad f_2 = e^{C_r . \frac{\sqrt{k}}{\sqrt[3]{V}}}$$

Where:

k - Density of turbulent kinetic energy;

V - Instantaneous cylinder volume;

C_r - Constant for mixing rate;

$C_r = 0,002 s$

The density of the turbulent kinetic energy at any time can be calculated by the ratio of the total turbulent energy of the spray and the total mass of the air-fuel mixture in the spray [4]:

$$(8) \qquad k = \frac{E_u}{g_f . (1 + \lambda . l_0)}$$

Where:

E_u - Total turbulent kinetic energy of fuel jet;

λ - Air excess ratio in diffusion burning;

$\lambda = 1,4$;

l_0 - Air fuel ration for stoichiometric combustion of diesel;

$l_0 = 14,2$.

The rate of the total turbulent energy of the fuel get is given by following equation [4]:

$$(9) \qquad \frac{dE_u}{d\varphi} = \frac{dE_i}{d\varphi} - \frac{dE_{diss}}{d\varphi}$$

Where:

$\frac{dE_i}{d\varphi}$ - Rate of input kinetic energy;

$\frac{dE_{diss}}{d\varphi}$ - Rate of dissipation energy.

The rate of input kinetic can be presented by following equation [4]:

$$(10) \qquad \frac{dE_i}{d\varphi} = C_{turb} \frac{1}{2} . \frac{dg_f}{d\varphi} . v^2$$

The above equation can be rewritten as follows:

$$(11) \qquad \frac{dE_i}{d\varphi} = C_{turb} . 18 \rho_f . \left(\frac{n}{C_d . F_n} \right)^2 . \left(\frac{1}{\rho_f} . \frac{dg_f}{d\varphi} \right)^3$$

Where:

C_{turb} - Efficiency of conversion of kinetic energy to turbulence energy in free jet;

$C_{turb} = 0,2$;

n - Engine speed.

The rate of dissipation energy is given by following equation [4]:

$$(12) \qquad \frac{dE_{diss}}{d\varphi} = \frac{C_{diss}}{6.n}.E_u$$

Where:

C_{diss} - Dissipation constant;
$C_{diss} = 0,01s^{-1}$.

During combustion there are three regimes of heat release. At the first regime the turbulent energy depends on the kinetic energy on free jet. There is no contact of the injected fuel with the combustion chamber walls. During this period equation (11) was used for calculating the rate of turbulent energy input with fuel jet.

The penetration of the fuel jet and speed of the free jet can be calculated by following equations [4]:

$$(13) \qquad s = \sqrt{8.C_d.U_{inj}.d_e.t_{SOI}}.\sqrt[4]{\frac{T_0}{T_{surr}}}$$

$$(14) \qquad V = \sqrt{8.C_d.U_{inj}.d_e}.\frac{1}{2.\sqrt{t_{SOI}}}.\sqrt[4]{\frac{T_0}{T_{surr}}}$$

Where:

U_{inj} - Velocity of fuel jet at the exit of the nozzle;

d_e - Equivalent diameter of orifice;

$d_e = d.\sqrt{\frac{\rho_f}{\rho_0}}$;

t_{SOI} - Time after start of injection;
T_0 - Reference temperature, 294K
T_{surr} - Surrounding temperature.

When the fuel jet reaches the combustion chamber walls it reduces the kinetic energy input with the fuel because of decreasing the jet speed. During the second period we can calculate the jet penetration and the speed of wall jet by following equations [4]:

$$(15) \qquad s_{wall} = 0,75\sqrt{U_{inj}.d_e.t_{imp}}.\sqrt[4]{\frac{T_0}{T_{surr}}}$$

$$(16) \qquad V_{wall} = 0,75\sqrt{U_{inj}.d_e}.\frac{1}{2.\sqrt{t_{imp}}}.\sqrt[4]{\frac{T_0}{T_{surr}}}$$

Where:

t_{imp} - Time at witch impingement occurs.

3. Modeling of ignition delay period

A modeling of ignition delay period is necessary to determine the start of heat release in combustion chamber. The most commonly used model of ignition delay period is based on Arrhenius equation [3, 4]:

$$(17) \qquad \tau_{id} = a.q^{-k}.p^{-n}.e^{\left(\frac{E_a}{R_u.T}\right)}$$

Where:

q - Equivalence ration;
p - Cylinder pressure;

T - Cylinder temperature;
E_a - Activation energy;
R_u - Gas constant;
a, k and n - Empirical constant.

Different empirical constants have been proposed by different authors but the following equation was used for calculating the ignition delay period in this paper [1, 4, 7]:

$$(18) \qquad \tau_{id} = 3,45.p^{-1,02}.e^{\left(\frac{2100}{T}\right)}$$

Here p and T are mean value of the pressure and the temperature during the ignition delay period.

4. Algorithm of the calculation program.

Our research is focused on the combustion modeling while the gas exchange processes are neglected. The calculation starts at the intake valve close point and finishes at the exhaust valve open point. The cylinder parameters are calculated by means of zero a dimensional model, based on first wall on the thermodynamic.

A heat exchange model with cylinder walls is included in the code by using Woschni's equation for calculation of the heat exchange coefficient [8]. Fig. 2 shows the block diagram of the algorithm. The code has been made in Matlab.
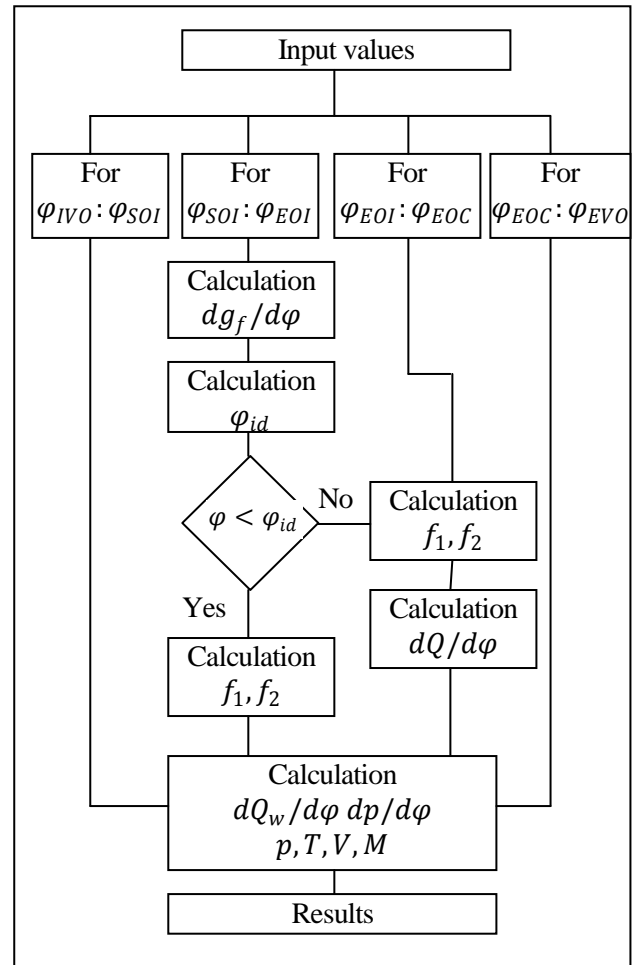

Fig.2 Block diagram of the calculation algorithm

5. Results

Research using different value of injection timing and injected fuel has been made. The engine speed was kept a constant. The geometric parameters of the engine are shown in the table below.

Cylinder bore	D=84mm
Cylinder stroke	S=86mm
Compression ratio	ε=17
Length of combustion chamber	L_{bowl}=35mm
Number of nozzle holes	7
Diameter of the holes	d=0,12mm
Boost pressure	p_{intake}=0,2MPa
Intake temperature	T_{intake}=340°K
Engine speed	n=2000min^{-1}

5.1. The results in air excess ratio $\alpha = 2$

Calculations with three different values of injection timing, constant injection pressure and injection duration has been made. The injection pressure is 100Mpa while injection duration is 22 deg by crank angle. The rate of injected fuel is shown in Fig. 3.
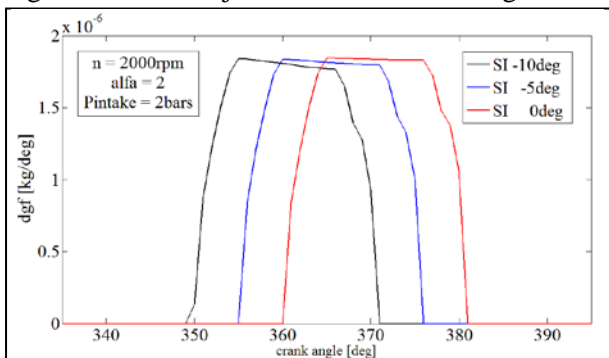
Fig.3 Calculated rate of injected fuel in different injection timing and constant air excess ratio $\alpha = 2$

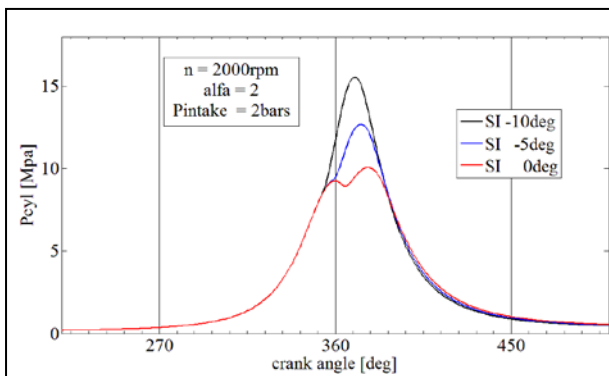The in cylinder parameters are shown in the following figures.

Fig. 4 Calculated cylinder pressure in different injection timing and constant air excess ratio $\alpha = 2$

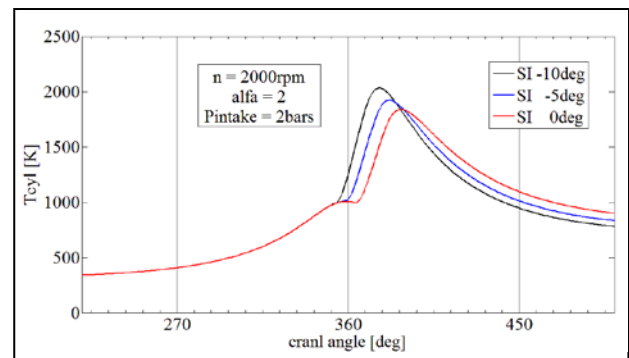The in-cylinder parameters are the results of

Fig. 5 Calculated in cylinder temperature in different injection timing and constant air excess ratio $\alpha = 2$

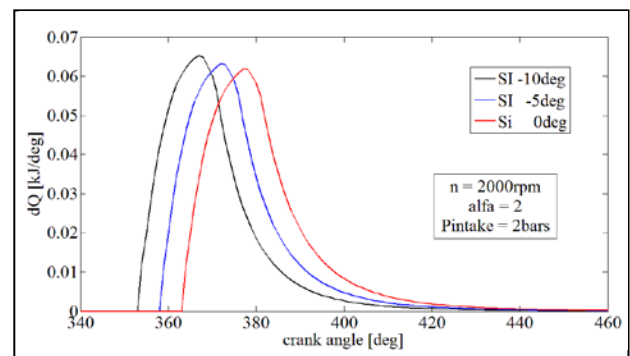different rate of heat release which can be seen in Fig.6.

Fig. 6 Calculated rate heat release in different injection timing and constant air excess ratio $\alpha = 2$

5.2. The results in air excess ratio $\alpha = 1,3$

The same study in full engine load which correspond to air excess ratio $\alpha = 1,3$, has been made. To provide such air excess ratio the injection pressure was set up to 130MPa and injection duration was increased to 28 deg by crank angle. The same engine speed, intake pressure and injection timing values have been introduced in the model. Fig. 7 shows the rate of injected fuel into the chamber.
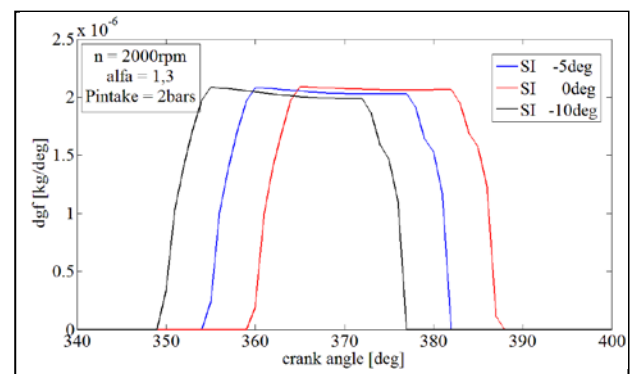

Fig. 7 Calculated rate of injected fuel in difference injection timing and air excess ratio $\alpha = 1,3$

The thermodynamic parameters in the cylinder and the rate of heat release can be seen in the figures below.
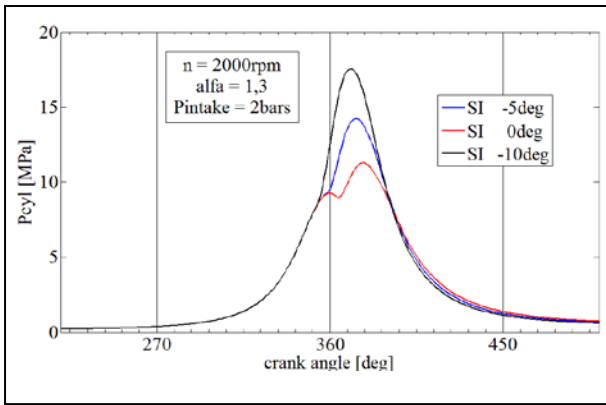
Fig. 8 Calculated cylinder pressure in different injection timing and constant air excess ratio $\alpha = 1,3$
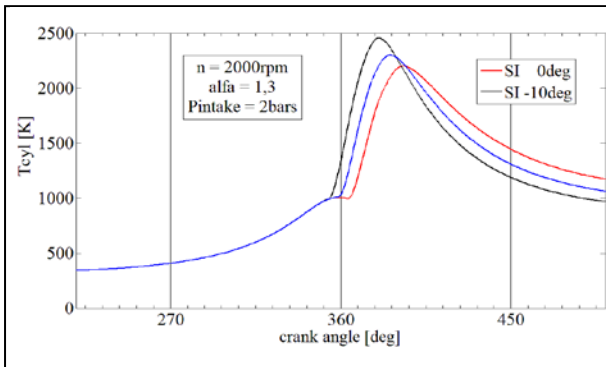

Fig. 9 Calculated in cylinder temperature in different injection timing and constant air excess ratio $\alpha = 1,3$
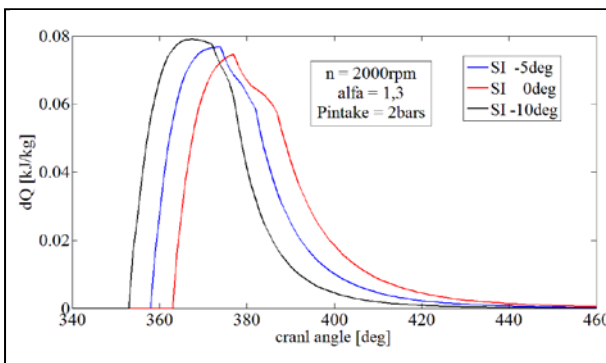

Fig. 10 Calculated rate heat release in different injection timing and constant air excess ratio $\alpha = 1,3$

The results shows the appropriateness of the mixed controlled combustion models for calculating the combustion heat release by using the geometrical parameters of the nozzle, injection timing, injection pressure and the geometrical parameters of combustion chamber. In full engine load because of the higher injection pressure and more injected fuel the fuel jet reaches the combustion chamber walls. It reduces the input kinetic energy with the fuel jet and reduces the rate of heat release which can be seen on Fig. 10.

5.3. Analysis of the rate of heat release

According equation (1) the rate of heat release is depends on two functions. The functions in relative values are shown in Fig. 11.

The results show that at the beginning of the injection the function f_2 has bigger influence on the rate of heat release than another function. When the impingement point occurs the f_2 starts to decrease rapidly to the end of the injection. After the end of injection the function decreases asymptotically. The f_1 increases with lower speed at the beginning of the combustion and has maximum value at the end of in-
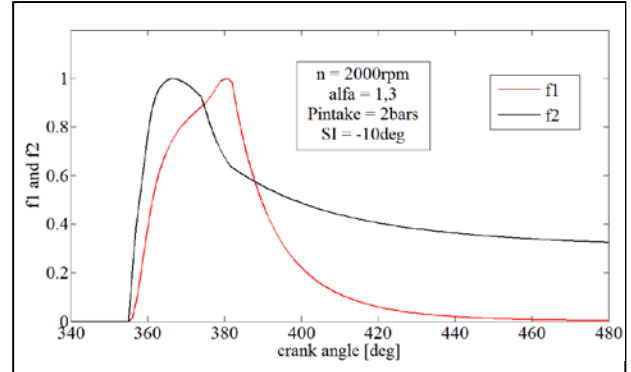

Fig. 11 Typical shape of the functions f_1 and f_2

jection. The function becomes zero at the end of the combustion process.

6. Conclusion

✓ The studies in different operation points of the engine show that a mixed controlled combustion model can be successfully used for predicting the heat release in modern diesel engines with direct injection;

✓ The influence of a great number of injection parameters (injection pressure, number and diameter of nozzle holes and injection timing) as well as geometrical parameters of the combustion chamber can be investigated by the code which has been presented here;

✓ The estimation of model accuracy can be made in additional research after comparing the calculated parameters with those which have been measured on the appropriate engine.

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Моделиране на горивния процес в дизеловите двигатели с вътрешно горене с помощта на турбулентен модел на горивния процес

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