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QUASI-DIMENSIONAL DIESEL ENGINE MODEL WITH DIRECT CALCULATION OF CYLINDER TEMPERATURE AND PRESSURE

Vedran Mrzljak, Vladimir Medica, Ozren Bukovac

Original scientific paper
This paper describes the quasi-dimensional numerical model, implemented in previously developed 0D model. The presented model uses direct solution of equations for cylinder pressure and zone temperatures, without numerical iterations which are customary for these models. In the model there is shown a process of averaging from a set of small fuel spray packages (volumes) into big ones, which is a necessary precondition for the numerical stability. The model uses about fifty submodels. Simulations were performed in eight operating points, on four most sensitive engine cylinder operating parameters. Direct solution of temperature and pressure changes, in conjunction with the fuel spray packages averaging, represents a contribution to quasi-dimensional diesel engine process modelling.

Keywords: *combustion; diesel engine; emissions; fuel spray; quasi-dimensional (phenomenological) model*

Kvazi-dimenzijski model dizelskog motora s direktnim izračunom tlaka i temperature u cilindru

Izvorni znanstveni članak
U ovom radu prikazan je kvazi-dimenzijski numerički model, implementiran u prethodno razvijeni 0D model. Prikazani model koristi direktan izračun jednadžbi za prirast tlaka i temperatura zona u cilindru, bez upotrebe numeričkih iteracija koje su uobičajene za ove modele. U modelu je prikazan proces usrednjavanja iz seta malih paketa (volumena) mlaza goriva u veliki paket, što je nužna pretpostavka numeričke stabilnosti modela. Model koristi oko pedeset podmodela. Simulacije su provedene za četiri najosjetljivija radna parametra u cilindru motora. Direktno rješavanje jednadžbi prirasta tlaka i temperature u cilindru, u kombinaciji sa usrednjavanjem paketa mlaza goriva, predstavlja doprinos u kvazi-dimenzijskom modeliranju procesa u dizelskom motoru.

Ključne riječi: *dizelski motor; emisije; izgaranje; kvazi-dimenzijski (fenomenološki) model; mlaz goriva*

1 Introduction

Quasi-dimensional models have been developed as a compromise between 0D and CFD models for internal combustion engine simulations. On the one side, 0D numerical models assume a homogeneous mixture of gases in the cylinder, so they cannot predict engine emissions [1, 2]. On the other side, CFD models enable the most detailed simulations, but these simulations have the longest duration [3], it is not possible to achieve convergence for engine operating point and often is not known all of the input parameters and boundary conditions required for simulation, so it is necessary to assume some of them.

Quasi-dimensional model development starts from the idea of initial division of the space inside the cylinder into two zones - a zone of combustion products and fresh mixture zone [4, 5]. This kind of models can predict the engine emissions, but predictions are mostly indicative.

Progress in quasi-dimensional modelling occurs at the moment when the cylinder volume division is performed in a manner that during the fuel injection are created packages (volumes) that accompany each fuel spray, Fig. 1, and outside the fuel sprays there is a zone without fuel (zone without combustion) [6÷8]. Fuel spray packages are annular in shape, spatial creations and in the spray core they have a form of a truncated cone. As injectors can have a plurality of nozzles, separate volumes are created for each of the fuel sprays, which may be mutually identical or different. The basic assumption of these model states that between fuel spray packages is not allowed any exchange of mass and energy. The only allowed mass exchange is air entrainment from the zone without combustion into spray packages [9], when the necessary conditions are fulfilled in each spray package.

A whole range of published scientific papers is dealing with the development and application of such quasi-dimensional models. Papers [10] and [11] show the numerical analysis of such quasi-dimensional models for a direct injection diesel engine simulation and provide an overview of the engine emission calculations. Paper [12] shows the results of the quasi-dimensional model for diesel engine which uses dimethyl ether as a fuel. The authors in [13] show quasi-dimensional model results for diesel engines in severe operating conditions. Results of quasi-dimensional model which simulates different fuel types combustion is shown in the papers [14] and [15].

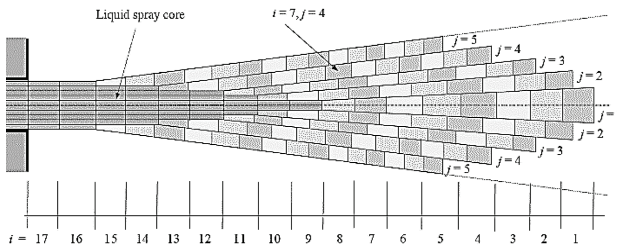

Figure 1 The fuel spray divided into packages (volumes)

At the end, these quasi-dimensional models are trying to improve modelling by using the principle that combustion products from spray packages are transferred into the zone without combustion. They are there mixed with the existing air and moved back into packages [16], [17].

In this paper the quasi-dimensional model developed in [18] is used and implemented in the existing 0D model [1]. Numerical model by its specificity and complexity is reflected in the fact that equations were developed for a direct solution of pressure and temperature changes in the cylinder, without the necessity for time consuming numerical iterations.

The changes of pressure and temperature affect every zone in the cylinder space volume, which is not known in advance, and this fact leads to a complex mathematical model.

2 Mathematical model and agglomeration process
2.1 Mathematical model

The mathematical quasi-dimensional model is based on the presumptions of the multizone model [19]. With mathematical excerpt, (details are presented in [18]), the following differential equations of pressure and temperature changes in the cylinder were obtained:

$$A_\mathrm{i} = 1 + \frac{T_\mathrm{i}}{R_\mathrm{i}}\frac{\partial R_\mathrm{i}}{\partial T_\mathrm{i}}, \tag{1}$$

$$B_\mathrm{i} = 1 - \frac{p}{R_\mathrm{i}}\frac{\partial R_\mathrm{i}}{\partial p}, \tag{2}$$

$$C_\mathrm{i} = \left(\frac{\partial u_\mathrm{i}}{\partial p}\frac{p}{B_\mathrm{i}}\frac{A_\mathrm{i}}{T_\mathrm{i}} + \frac{\partial u_\mathrm{i}}{\partial T_\mathrm{i}}\right), \tag{3}$$

$$D_\mathrm{i} = \left(\frac{u_\mathrm{i}}{m_\mathrm{i}} + \frac{1}{m_\mathrm{i}}\frac{\partial u_\mathrm{i}}{\partial p}\frac{p}{B_\mathrm{i}}\right), \tag{4}$$

$$E_\mathrm{i} = \left(\frac{p}{m_\mathrm{i}} - \frac{1}{V_\mathrm{i}}\frac{\partial u_\mathrm{i}}{\partial p}\frac{p}{B_\mathrm{i}}\right), \tag{5}$$

$$F_\mathrm{i} = \left[\frac{1}{R_\mathrm{i}}\frac{\partial u_\mathrm{i}}{\partial p}\frac{p}{B_\mathrm{i}}G_\mathrm{i} + \left(\frac{\partial u_\mathrm{i}}{\partial \lambda_\mathrm{i}}\frac{\mathrm{d}\lambda_\mathrm{i}}{\mathrm{d}\varphi} + \frac{\partial u_\mathrm{i}}{\partial Y_\mathrm{vap,i}}\frac{\mathrm{d}Y_\mathrm{vap,i}}{\mathrm{d}\varphi}\right)\right], \tag{6}$$

$$G_\mathrm{i} = \left(\frac{\partial R_\mathrm{i}}{\partial \lambda_\mathrm{i}}\frac{\mathrm{d}\lambda_\mathrm{i}}{\mathrm{d}\varphi} + \frac{\partial R_\mathrm{i}}{\partial Y_\mathrm{vap,i}}\frac{\mathrm{d}Y_\mathrm{vap,i}}{\mathrm{d}\varphi}\right), \tag{7}$$

$$H_\mathrm{i} = \frac{C_\mathrm{i}}{\left(R_\mathrm{i} + T_\mathrm{i}\frac{\partial R_\mathrm{i}}{\partial T_\mathrm{i}}\right)}, \tag{8}$$

$$K_\mathrm{1,i} = \frac{B_\mathrm{i}\,u_\mathrm{i}\,T_\mathrm{i}\frac{\partial R_\mathrm{i}}{\partial T_\mathrm{i}} + p\,u_\mathrm{i}\left(\frac{p}{R_\mathrm{i}} - 1\right)\frac{\partial R_\mathrm{i}}{\partial p}}{A_\mathrm{i}\,B_\mathrm{i}\,m_\mathrm{i}\,R_\mathrm{i}}, \tag{9}$$

$$K_\mathrm{2,i} = \frac{1}{m_\mathrm{i}\,R_\mathrm{i}\,A_\mathrm{i}}\frac{\partial u_\mathrm{i}}{\partial T_\mathrm{i}}, \tag{10}$$

$$K_\mathrm{3,i} = \frac{\left(u_\mathrm{i} + R_\mathrm{i}\,T_\mathrm{i}\right)}{m_\mathrm{i}} = \frac{h_\mathrm{i}}{m_\mathrm{i}}, \tag{11}$$

$$K_\mathrm{4,i} = \frac{\partial u_\mathrm{i}}{\partial T_\mathrm{i}} + R_\mathrm{i}\,A_\mathrm{i}, \tag{12}$$

$$S_1 = \sum_\mathrm{i}\left[\frac{\frac{1}{m_\mathrm{i}}\frac{\mathrm{d}Q_\mathrm{i}}{\mathrm{d}\varphi} - K_\mathrm{1,i}\frac{\mathrm{d}m_\mathrm{i}}{\mathrm{d}\varphi} + H_\mathrm{i}\,T_\mathrm{i}\,G_\mathrm{i} - F_\mathrm{i}}{K_\mathrm{2,i}}\right], \tag{13}$$

$$S_2 = \sum_\mathrm{i}\left(m_\mathrm{i}\,T_\mathrm{i}\frac{\partial R_\mathrm{i}}{\partial p} + \frac{\frac{E_\mathrm{i}}{p}\left(V_\mathrm{i} - m_\mathrm{i}\,T_\mathrm{i}\frac{\partial R_\mathrm{i}}{\partial p}\right)}{K_\mathrm{2,i}}\right), \tag{14}$$

$$\boxed{\frac{\mathrm{d}T_\mathrm{i}}{\mathrm{d}\varphi} = \frac{\frac{1}{m_\mathrm{i}}\frac{\mathrm{d}Q_\mathrm{i}}{\mathrm{d}\varphi} - K_\mathrm{3,i}\frac{\mathrm{d}m_\mathrm{i}}{\mathrm{d}\varphi} - F_\mathrm{i}}{K_\mathrm{4,i}} - \frac{\frac{E_\mathrm{i}}{p}\left[m_\mathrm{i}\,T_\mathrm{i}\,G_\mathrm{i} - \left(V_\mathrm{i} - m_\mathrm{i}\,T_\mathrm{i}\frac{\partial R_\mathrm{i}}{\partial p}\right)\frac{\mathrm{d}p}{\mathrm{d}\varphi}\right]}{K_\mathrm{4,i}},} \tag{15}$$

$$\boxed{\frac{\mathrm{d}p}{\mathrm{d}\varphi} = \frac{S_1 - p\frac{\mathrm{d}V_\mathrm{c}}{\mathrm{d}\varphi}}{\left(V_\mathrm{c} - S_2\right)}.} \tag{16}$$

The variables A, B, C, D, E, F, G, H, K_1, K_2, K_3 and K_4 in the equations from (1) to (14) are substitutes for differential expressions, and marks S_1, S_2 are the replacement for the sums that need to be inserted into the equation for the pressure change (16). The index i is an index for any observed volume (for each package of each fuel spray as well as for the zone without combustion).

It should be noted for the fuel spray packages that all of displayed equations are related to the thermodynamic (TD) volume of the package (volume of gases and vapours). Thermodynamic volume of the package is the geometric package volume reduced for the liquid fuel volume. The properties of liquid fuel, which is also present in each fuel spray package, are monitored by separate mathematical models, independent of the displayed one. Energy conservation equation for the liquid fuel was used to monitor the temperature of liquid fuel, which is a basic parameter for the fuel evaporation calculation. Fuel vapour in this model is considered as an ideal gas in gaseous mixture with other species.

2.2 Initial small package agglomeration into larger ones

Initial simulations performed with the present numerical model showed remarkable instability of the numerical integration process. Since the beginning of spray packages formation (the first axial row) calculations were showing unrealistic values and they quickly aborted simulation program execution.

In a series of papers related to quasi-dimensional numerical modelling, the authors have used angular integration step of 1 °CA for solving differential equations. Initially, such integration step was chosen in this model too, however, that integration step proved to be too large, so it was reduced to 0,1 °CA.

Integration step of 1 °CA for internal combustion engines with rotational speed lower than 3000 min^{-1} is too rough. Within a single integration step occurs continuous liquid jet disintegration into droplets and begins droplet heating and evaporation, respectively basic preconditions for package mixture ignition were created. Numerical integration errors in this initial phase have consequences later in the simulation. These deficiencies were observed in this model, and the integration step was reduced.

During the fuel injection, next to the nozzle, a new axial row of small spray packages was created in every integration step. Due to this fact, a very high number of spray packages will be created, and it will result in significant computer time increase.

In order to prevent occurrence of too many spray packages, the solution is found in the averaging process, shown in Fig. 2. Fig. 2(a) shows initial small packages which are formed in each integrating step of 0,1 °CA. When the cylinder process comes to a full crank angle, all content of small spray packages was transferred into a bigger one and averaged to homogenous state. In the averaging process, mass and energy conservation laws were respected, and simulation also took into account other properties, such as new Sauter mean diameter [17]. After creation of the large package, a usual numerical integration of conservation differential equations for new package was continued. In Fig. 2(b) can be seen the cylinder process arrival at full crank angle, while Fig. 2(c) shows the formation of a new large package from all small packages, created on that crank angle. Large package continues its movement through the cylinder without further averaging. Indexes along packages in Fig. 2 are radial (j) indexes of the first and the last package from each axial row. With the indexes which are related to each package (i = axial index, j = radial index), it was necessary to use an additional index k for each fuel spray when the fuel sprays are not mutually identical. In the case that fuel sprays are mutually equal, only one spray is calculated, and the results are multiplied by the total number of sprays.

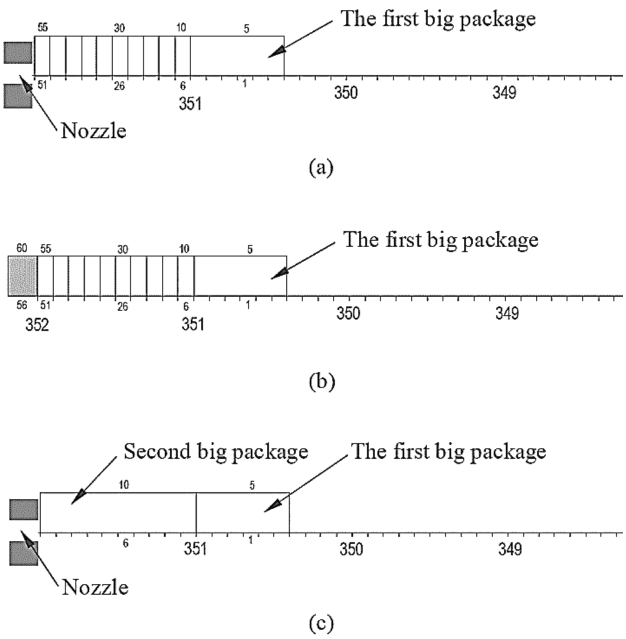

Figure 2 Averaging from small packages into big ones

In addition, in Fig. 2 to the right, in front of small spray packages can be seen the first large package created by small packages averaging at the start of fuel spray injection (6×0,1 °CA). On the left is a group of small packages which are injected in each new integration step. When the cylinder process arrived at full crank angle, all little packages are averaged into a new big one for the observed crank angle. The integration continues, and new small packages are added to the next whole crank angle.

The averaging process successfully resolves the problem of excessive packages number that accompanies each fuel spray. Also, averaging allowed the fact that the beginning and the end of fuel injection does not have to start at full crank angle, but also inside it.

3 Measurements and measuring equipment

Test engine was a high speed diesel engine with direct injection for the freight vehicle drive MAN D 0826 LOH15, Tab. 1, and the tests were carried out in the Laboratory for Internal Combustion Engines and Electromobility, at the Faculty of Mechanical Engineering, University of Ljubljana.

Table 1 Engine specifications

Displacement	6,87 l
Number of cylinders	6
Peak power	160 kW
Cylinder bore	108 mm
Stroke	125 mm
Compression ratio	18
Crank radius	62,5 mm
Length of the connecting rod	187,2 mm
Length of nozzle bore	2,3 mm
Nozzle diameter	0,23 mm
Number of nozzle holes	7
Combustion chamber	Bowl in piston

During measurements, the engine was connected to an eddy current measuring brake Zöllner B-350AC. For the measuring brake control a control system KS ADAC/Tornado was used whose manufacturer is Kristl, Seibt & Co.

Laboratory measuring sensor AVL GH12D has been used for measuring the cylinder pressure. This sensor is placed in extra hole in the cylinder head. Resulting measuring signal was led to a 4-channel amplifier AVL MicroIFEM, and from the amplifier to a 4-channel, 16-bit NI-9223 data acquisition card with a maximum sampling rate of 1 MS/s per channel. The card was inserted into the NI cDAQ-9178 housing. On the housing is also brought the TRG measuring signal (signal for the start of measurement sequence) and CAM signal (trigger signal) from the angle encoder.

Table 2 The results of selected measurement set for numerical model validation

Oper. point	Fuel consum. (kg/h)	Air consum. (kg/s)	Rotational speed (min^{-1})	Power (kW)	NO_x emission (ppm)
1	9,198	0,100764	1498	43,776	870,41
2	13,447	0,111920	1502	67,560	1222,91
3	18,040	0,126717	1502	89,319	1202,46
4	22,453	0,141457	1501	110,296	1259,51
5	14,773	0,191578	2401	56,266	391,38
6	21,815	0,224946	2402	93,958	637,24
7	28,841	0,260685	2399	126,047	773,07
8	35,364	0,293871	2399	153,189	907,28

The piston top dead centre was determined by a capacitive sensor COM Type 2653, and the position of the crankshaft was determined using angle encoder Kistler CAM UNIT Type 2613B with accuracy of 0,1 °CA.

NO_x emission was measured with Horiba OBS-2200 analyzer, by using CLD module. NO_x emissions were measured continuously during the engine testing.

4 Numerical model results

Several measurement sets were carried out and the measurement set shown in Tab. 2 was chosen for numerical model validation. Numerical model was validated in all 8 operating points. This paper presents the results for two operating points - operating points 3 and 8.

Used parameters are cylinder pressure, rate of heat release in the cylinder, cumulative heat release, and the exhaust nitrogen oxide emission.

Parameter for this numerical model calibration is the change of cylinder pressure. The reason for this selection is that the change of cylinder pressure was obtained experimentally, and the rate of heat release (shown as experiment) was calculated from the measured cylinder pressure changes by using the adjusted 0D model. The heat losses in the adjusted 0D model were calculated with the Woshni and Hohenberg numerical models. Adjusted 0D simulation model uses the linearized submodel for calculating the properties of operating substance. Quasi-dimensional model uses a different method for calculating the operating substance properties.

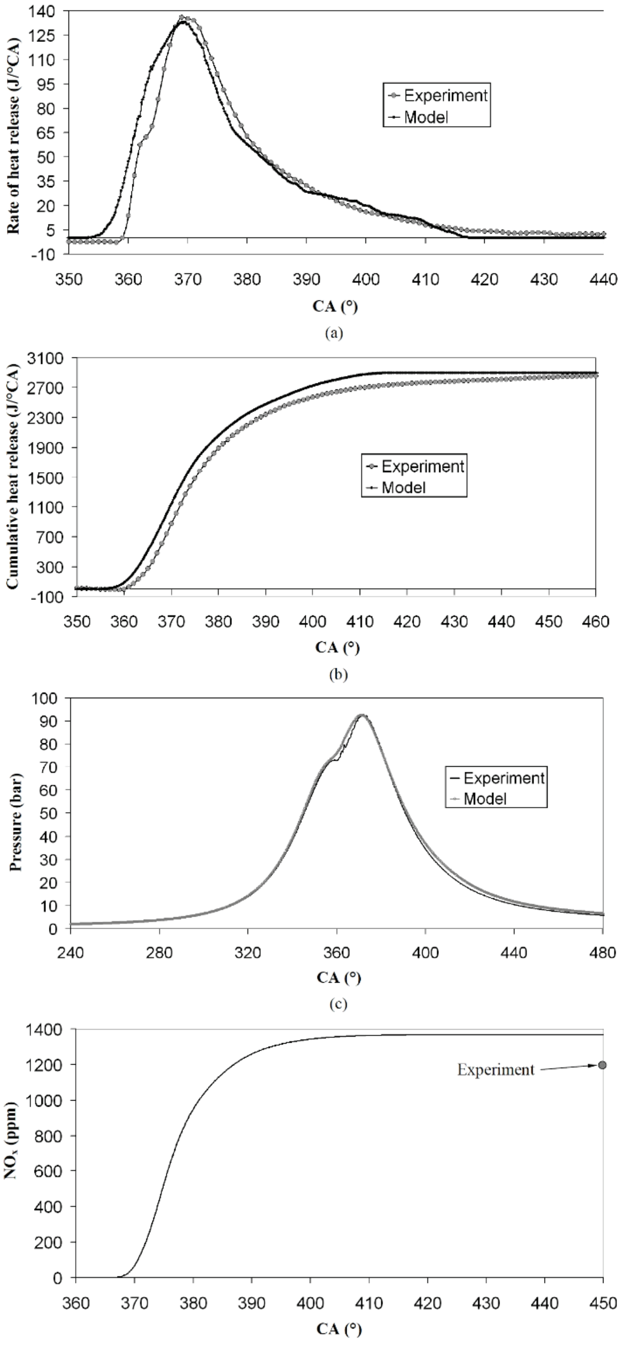

Figure 3 Rate of heat release (a), cumulative heat release (b), cylinder pressure (c) and NO_x emission (d) for operating point 3

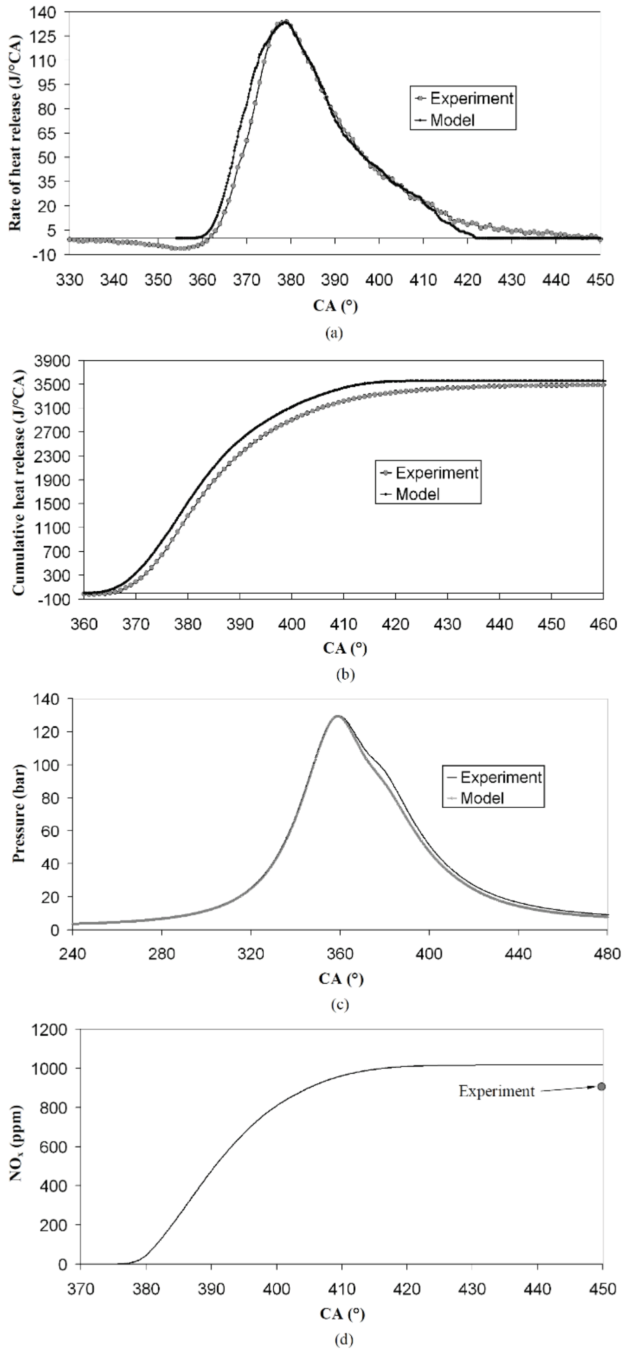

Figure 4 Rate of heat release (a), cumulative heat release (b), cylinder pressure (c) and NO_x emission (d) for operating point 8

4.1 Results for operating point 3

In this operating point deviations between simulation and experiment in the rate of heat release are evident at the process beginning. After reaching the heat release peak, the matching results trend is satisfying, Fig. 3(a).

Simulated and experimentally obtained cumulative heat release curves show a relatively good match, Fig. 3(b).

Regardless of the calibration, simulated and measured pressure curves indicate an acceptable deviation, Fig. 3(c), especially while reaching maximum pressure and at the end of the expansion.

The simulated values of nitrogen oxide, calculated by using extended Zeldovich model, Fig. 3(d), in this case, gives slightly higher results compared to measurement. The relative difference between simulated and measured values is acceptable, in this case 14 %. In this operating point measured value is shown earlier in relation to measurement (the opening of exhaust valve takes place at 482 °CA).

4.2 Results for operating point 8

The second shown simulated operating point presented a relatively good match for rate of heat release, Fig. 4(a). Apart from differences in the initial part, the only noticeable difference can be observed at the end of the cylinder process.

Simulation for the cumulative heat release shows a slightly greater deviation in relation to experiment only in the second part of the cylinder process, Fig. 4(b).

Simulation and measurements match on the pressure change curve is ideal until it reaches the maximum value. In the second part of the process simulated and measured pressures show obvious, but still acceptable deviations, Fig. 4(c).

The simulated values of nitrogen oxide, calculated by using extended Zeldovich model, Fig. 4(d), give slightly higher results compared to measurement. The relative difference between simulated and measured values is 12 %. Also in this operating point measured value is shown earlier in relation to measurement (the opening of exhaust valve takes place at 482 °CA).

5 Conclusions

This paper presents developed quasi-dimensional model implemented in 0D model with direct calculation of temperature and pressure. In numerical model has been introduced the process of agglomeration and averaging from the initial small packages to large ones that correspond to the whole crank angle. Averaging process solves the mathematical and physical problems of this numerical model.

Developed model provides satisfying prediction of measured operating parameters for the observed engine operating range.

The developed model is a good compromise that the engineers needed in engine development. Simulation results are obtained quickly and it is possible to analyse the impact of changes in certain parameters on the engine characteristics.

Numerical model presented in this paper shows some improvements compared to similar quasi-dimensional models. This model can and should be further improved by including submodel for particulate emissions.

Nomenclature

λ	air excess ratio (-)
φ	crank angle (°)
h	specific enthalpy (J/kg)
m	mass (kg)
p	pressure (Pa)
Q	amount of heat (J)
R	gas constant (J/kg·K)
T	temperature (K)
u	specific internal energy (J/kg)
V	volume (m^3)
Y_{vap}	mass fraction of fuel vapour (-)
c	cylinder
CA	crank angle
CLD	ChemiLuminescence Detector
TD	thermodynamical (volume)
ZWC	zone without combustion

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Authors' addresses

Vedran Mrzljak, Ph. D.
Faculty of Engineering, University of Rijeka
Vukovarska 58, 51000 Rijeka, Croatia
E-mail: vedran.mrzljak@riteh.hr

Vladimir Medica, Prof. D. Sc.
Faculty of Engineering, University of Rijeka
Vukovarska 58, 51000 Rijeka, Croatia
E-mail: vladimir.medica@riteh.hr

Ozren Bukovac, Ph. D.
Faculty of Engineering, University of Rijeka
Vukovarska 58, 51000 Rijeka, Croatia
E-mail: ozren.bukovac@riteh.hr